

Course Binder

MCEG 3304 Machine Design I

Fall 2021

Prepared by Dr. Xiaobo Peng

DEPARTMENT OF MECHANICAL ENGINEERING

COLLEGE OF ENGINEERING

PRAIRIE VIEW A&M UNIVERSITY

**PRAIRIE VIEW A&M UNIVERSITY
ROY G. PERRY COLLEGE OF ENGINEERING
DEPARTMENT OF MECHANICAL ENGINEERING
MCEG 3043 MACHINE DESIGN I
FALL 2020**

I. COURSE SYLLABUS IN ABET FORMAT

II. ASSESSMENT RESULTS FOR CURRENT AND PAST THREE PERIODS
Program Outcomes Assessed in Course
Assessment Summary of Class Semester Averages for Outcomes
End of Semester Course Outcomes Assessment Reports

III. TESTS

Summary of Topics covered by Test
Original Test Questions & Master Solution/grading scheme used
(Summary of topics can be a separate page or include on the assignment itself)
Above Average (Graded original work)
Average (Graded original work)
Below Average (Graded original work)

IV. QUIZZES

Summary of Topics covered by Quiz
Original Quiz Questions & Master Solution/grading scheme used
Above Average (Graded original work)
Average (Graded original work)
Below Average (Graded original work)

V. Homework

Summary of Topics covered by Homework
Original Homework Questions & Master Solution/grading scheme used
Above Average (Graded original work)
Average (Graded original work)
Below Average (Graded original work)

VI. Project

Summary of Topics Covered by Project
Project Problem Statement and Grading Scheme
Above Average (Graded original work)
Average (Graded original work)
Below Average (Graded original work)

**One or Two Page Course Description
in ABET Format**

Course Syllabus

(ABET Self-Study 2020-2021 Template)

1. **Course number and name:** MCEG 3304 Machine Design I
2. **Credits and contact hours:** 3 credit hours; 4 contact hours
3. **Instructor's or course coordinator's name:** Dr. Xiaobo Peng
4. **Text book:** Title: Shigley's Mechanical Engineering Design
 Author: Budynas, R. G. and Nisbett, J. K.,
 Publisher, Year & Edition: McGraw-Hill Book Company, New York, 2016, 10th Edition
 - a. Other Supplemental material: None.
5. **Specific course information**
 - a. Brief description of the content of the course (catalog description): MCEG 3043 Machine Design I, 3 credit hours. Fundamentals of mechanical design methodology, design of machine elements for static and fatigue failure, design projects and classroom discussions of various design solutions.
 - b. Prerequisites or Co-requisites: CVEG2063 Mechanics of Materials
MCEG1021 Introduction to Mechanical Engineering Drawing and Design Lab I
 - c. *Indicate whether a required, elective, or selected elective (as per definition provided) course in the program: Required

- | | |
|---|--|
| <p>6. a. Specific goals for the course specific outcomes of instruction (e.g. The student will be able to explain the significance of current research about a particular topic.)
 <i>After completion of this course, the students should be able to apply the design processes to mechanical elements, using contemporary design standards and tools and have developed the following:</i></p> | <p>b. Explicitly indicate which of the student outcomes listed in Criterion 3 or any other outcomes are addressed by the course.</p> |
| <p>1. Skills for mechanical element design;</p> | <p><i>The specific goals for the course specific outcomes listed on the left column will address the following student learning outcomes:</i></p> <p>Outcome 2
 An ability to apply engineering design to produce solutions that meet specified needs with consideration of public health, safety, and welfare, as well as global, cultural, social, environmental, and economic factors.</p> |

2. An ability to analyze mechanical engineering elements;
3. Knowledge of various machine elements;
4. An ability to complete a mechanical design project.

7 Brief list of topics to be covered

- (1) Introduction/Basics
- (2) Design Process and Project Management
- (3) Material, Load and stress analysis
- (4) Deflection and stiffness
- (5) Failures resulting from static loading
- (6) Failures resulting from variable loading
- (7) The design of screws, fasteners, and connections
- (8) Rolling contact bearings
- (9) Mechanical springs

***Required** courses are required of all students in the program, **elective** courses (often referred to as open or free electives) are optional for students, and **selected elective** courses are those for which students must take one or more courses from a specified group

II ASSESSMENT RESULTS FOR CURRENT AND PAST THREE PERIODS

Program Outcomes Assessed in this course

Outcome 2: an ability to **apply engineering design to produce solutions that meet specified needs** with consideration of public health, safety, and welfare, as well as global, cultural, social, environmental, and economic factors.

Assessment Summary of Class Semester Averages for Outcomes: Fall 2019 - Fall 2021

Outcomes	2
Semester	Semester Class Average (%)
Fall 2021	82.6
Fall 2020	84.05
Fall 2019	81.3
Comments on Data	The overall performance is better than the expected average.

*Expected Semester Class Average = 70%

End of Semester Course Assessment Report

This summary contains the following Information

1. Performance Statistics

- **Total number of students in class**
- **The class average performance (stated as %) in each course outcome assessed in this class and the two previous times course was assessed.**
- **The acceptable class average for the class in each outcome area as set by your Department.**
- **Percentage of students who scored below the expected average in each outcome.**

2. Implementation Summary

- **Brief summary of plans implemented during the semester based on last assessment report**

3. Perceived Problems

- **Instructor's critical evaluation of perceived problems that affected students' performance.**

4. New Plans for Addressing Problems

- **Instructor's plans for addressing the perceived problems the next time the class is taught.**

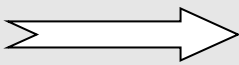
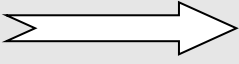
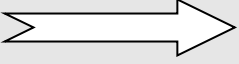
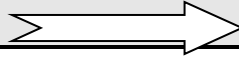
5. Overall Trend over last three assessment periods

- **Instructor's summary of overall performance trend for each outcome measured over the last three periods.**

END OF SEMESTER COURSE OUTCOME ASSESSMENT REPORT
MCEG3304-P01 MACHINE DESIGN I
Fall 2021 SEMESTER

Report Prepared by: Dr. Xiaobo Peng

Report Date: December 13, 2021

Semester	Analysis Type	Number of Students	Outcome (2)			
			Ability to apply engineering design to produce solutions			
			Expected Average	Expected Percent of students at or above average	Class Average	% Students Meeting Expected Average
Fall 2021	Direct	20	70	70	82.6	90%
Fall 2020	Direct	19	70	70	84.05	84.2%
Fall 2019	Direct	15	70	70	81.3	80%
Implementation Summary		The following activities have been implemented to improve the student learning on outcome 2: (1) A series of Tutorials on how to apply engineering design to solve engineering problems are developed and distributed to students; (2) One design project was assigned which required students to use ANSYS software to solve engineering problems; (3) Ecourses was utilized for the class to distribute the tutorials and submit projects.				
Perceived Problems		1. Students lack the abilities in applying the fundamentals to formulate the design problem and to validate the design using analysis. 2. Students showed more enthusiasm on working on the project than working on the homework.				
Plans for Addressing Problems		It is suggested to do the following: (1) More in-class training sessions are needed to give students more training on ANSYS software. (2) More out-of-class tutoring sessions are needed. (3) A tutor is needed to provide help to students.				
Overall Trend over Periods		The overall performance is better than the expected average.				

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Exam One

TOPICS COVERED IN EXAM ONE

Chapter 2. Materials

Chapter 3. Load and Stress Analysis

**MACHINE DESIGN I
FALL 2021**

**SAMPLE EXAM QUESTIONS, SOLUTION,
AND GRADING SCHEME**

Exam One
MCEG3304: Machine Design I
Fall 2021
10/11/2021
Total: 100 pts

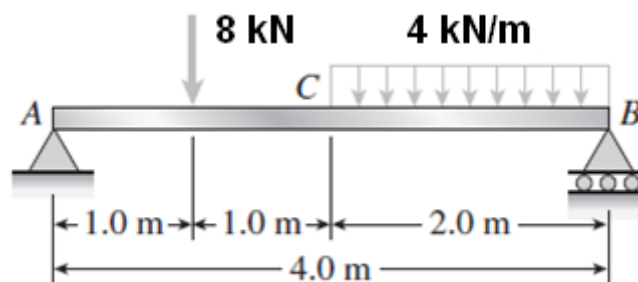
Name:

Note:

- Time: 8:00am ~ 10:00am.
- You can only use the formula provided by instructor and calculator during the exam. Using any other devices (including phone) or materials is prohibited.
- Write your answers on blank papers. Write question number and sub-question numbers, otherwise you will not receive any points on that questions.
- Return both the exam sheets and solution sheets separately.

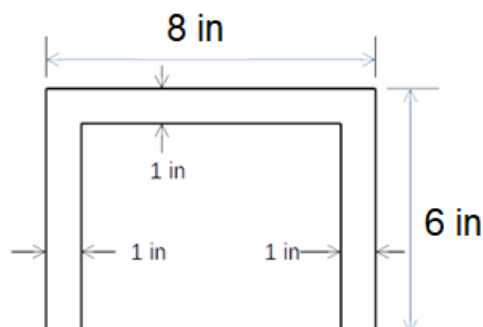
Show the detail steps of the following calculation problems.

1. (20 pts) For the beam shown
 - (1.1) (4 pts) Find the reactions at the supports A and B;
 - (1.2) (8 pts) Draw the shear-force diagram;
 - (1.3) (8 pts) Draw the bending moment diagram.



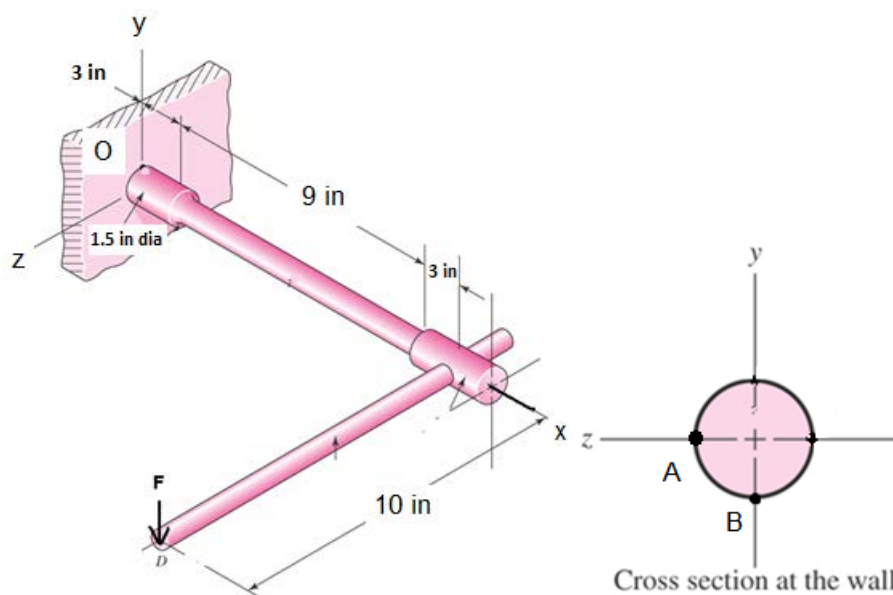
Problem 1.

2. (18 pts) The section is shown. The dimensions are in inches. Assume the neutral axis is horizontal.
 - (2.1) (8 pts) Find the distance of the neutral axis to the bottom surface.
 - (2.2) (10 pts) Find the second moment of area.



Problem 2.

3. (20 pts) For the stress states $\delta_x = 25 \text{ kpsi}$, $\delta_y = -15 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi ccw}$.
- (3.1) (12 pts) Draw a 2D-Mohr's circle diagram and label the coordinates of A, B, C, D, E, F, G points on the Mohr's circle.
- (3.2) (8 pts) Draw the stress elements of principal normal stresses and shear stresses.
4. (18 pts) For the stress states $\delta_y = 20 \text{ kpsi}$, $\delta_z = -15 \text{ kpsi}$, $\tau_{xz} = 12 \text{ kpsi ccw}$.
- (4.1) (12 pts) Calculate all three principal normal and shear stresses $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$.
- (4.2) (6 pts) Draw a complete Mohr's three-circle diagram and label the points $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$.
5. (24 pts) A cantilevered bar shown below is loaded by the force $F = 800 \text{ lbf}$. The **cross section O at the wall** is also shown with labeled points A and B.
- (5.1) (6 pts) Draw the freebody diagram and calculate all the reaction loads **at the cross section O at the wall**.
- (5.2) (12 pts) Calculate all three-dimensional stress elements of **point A** and **point B**.
- (5.3) (6 pts) Calculate the maximum principal normal and shear stresses at **point B**.



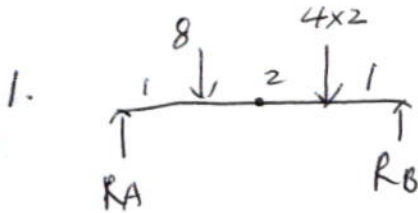
Problem 5.

MCEG3043 Machine Design I Exam One Solution Sheets

10/11/2021

Name:

(20)



(1.1) $\Sigma M_A = 0$

$$8 \times 1 + 8 \times 3 - R_B \times 4 = 0$$

$$R_B = 8$$

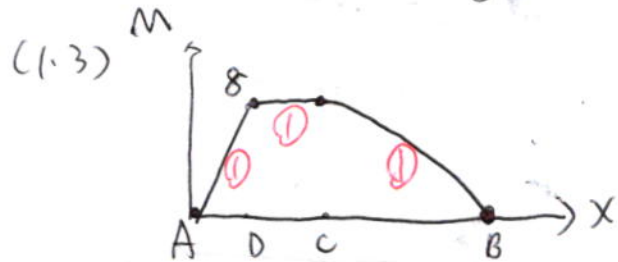
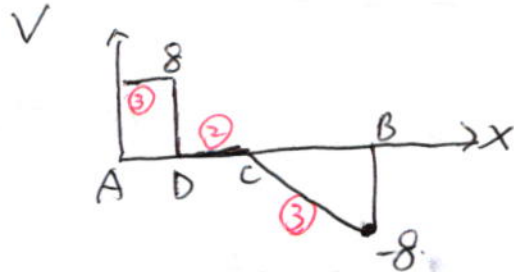
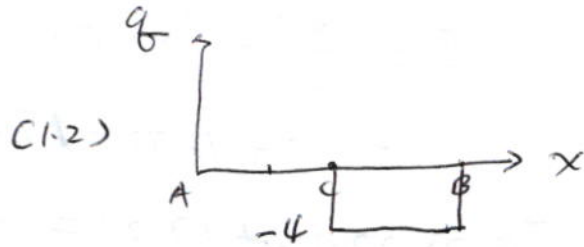
(2)

$\Sigma F_y = 0$

$$R_A + R_B - 8 - 8 = 0$$

$$R_A = 8$$

(2)



$$M_D - M_A = 8 \times 1 \Rightarrow M_D = 8$$

(2)

$$M_A = 0$$

(1)

$$M_B = 0$$

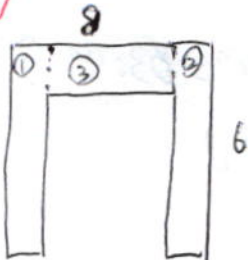
(1)

$$M_C = 8$$

(1)

(18)

2.



(2.1) $A_1 = A_2 = 6 \times 1 = 6 \text{ in}^2$ (1) $C_1 = 3 \text{ in}$ (1)

$$A_3 = 6 \text{ in}^2$$

(1)

$$C_3 = 5.5 \text{ in}$$

(1)

$$18 \times C = 6 \times 3 + 6 \times 3 + 6 \times 5.5$$

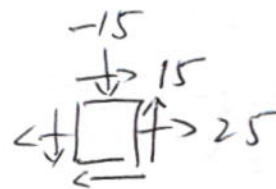
(2)

$$C = \frac{69}{18} = 3.83 \text{ in}$$

(2)

$$\begin{aligned} (2.2) \quad I_1 = I_2 &= \frac{1 \times 6^3}{12} + 6 \times (\overset{3.83}{\cancel{5.5}} - 3)^2 \\ &= 18 + \overset{4.13}{\cancel{15.375}} = \cancel{63.375} = 22.13 \quad (4) \\ I_3 &= \frac{6 \times 1^3}{12} + 6 \times (5.5 - 3.83)^2 \quad (4) \\ &= 0.5 + 16.73 = 17.23 \quad (2) \\ I &= 2 \times 22.13 + 17.23 = 61.49 \text{ m}^4 \end{aligned}$$

3. $\sigma_x = 25$, $\sigma_y = -15$, $\tau_{xy} = 15$ ccw.



(3.1) A (25, 15^{ccw})

$$B(-15, 15^{cw})$$

$C(5, 0)$

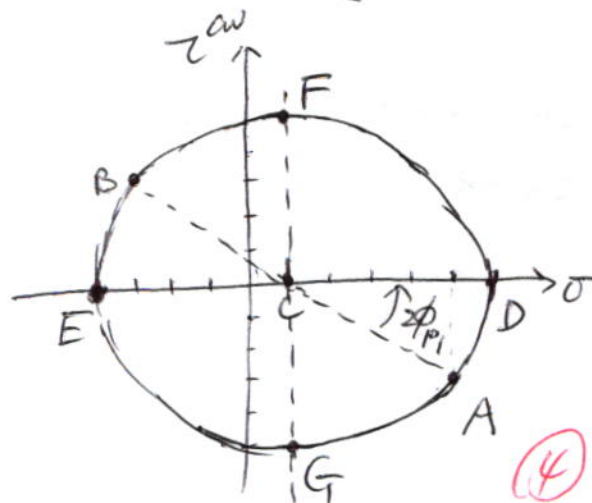
$$r = \sqrt{20^2 + 15^2} = 25$$

$$D = (30, 0)$$

$$\in (-20, 0)$$

$$F(5, 25)$$

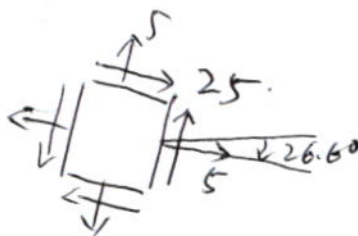
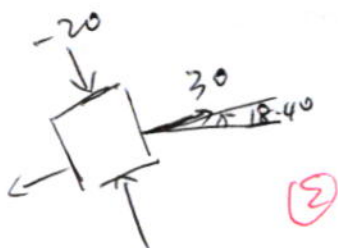
$$G(5, -25)$$



$$(3.2) \quad \phi_{pl} = \frac{1}{2} \tan^{-1} \left(\frac{2 \times 15}{25 + 15} \right) = 18.4^\circ$$

$$\phi_{s1} = 45^\circ + 18.4^\circ = 63.4^\circ \uparrow$$

$$\phi_{s1} = -26.6^\circ \downarrow \quad (2)$$



4. $\sigma_y = 20$ $\sigma_z = -15$, $\tau_{xz} = 12$ caw
 (18) (4.1) $\sigma_x = 0$, $\tau_{xy} = 0$, $\tau_{yz} = 0$. (1)

$\sigma_y = 20$ is a principal stress. (2)

$$\begin{aligned}\sigma_{A,B} &= \frac{\sigma_x + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_z}{2}\right)^2 + \tau_{xz}^2} \\ &= \frac{0 - 15}{2} \pm \sqrt{\left(\frac{0 + 15}{2}\right)^2 + 12^2} \leftarrow \boxed{14.15} \\ &= 6.65, -21.65\end{aligned}$$

(4)

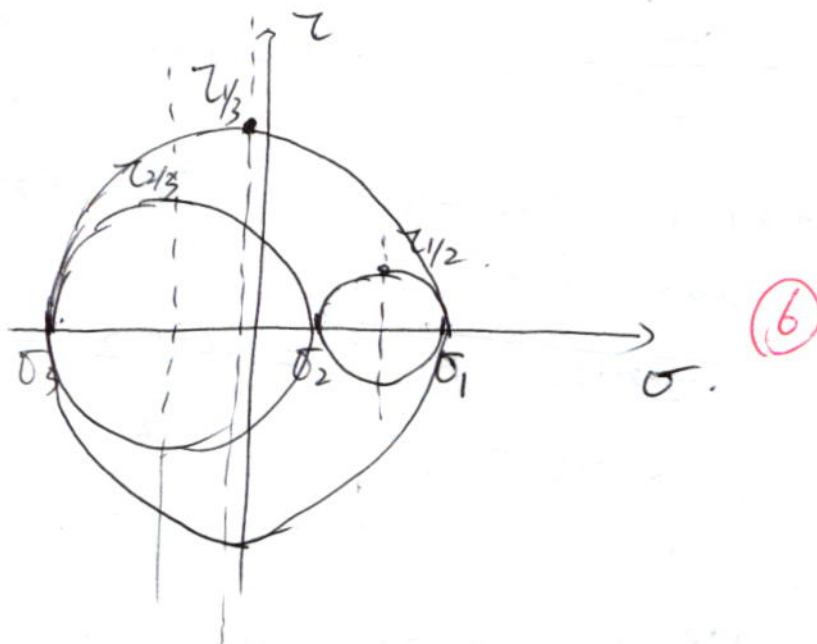
$\sigma_1 = 20$, $\sigma_2 = 6.65$, $\sigma_3 = -21.65$. (2)

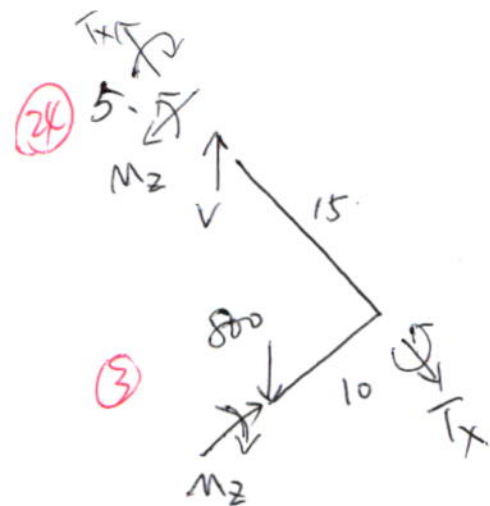
$\tau_{1/2} = \frac{20 - 6.65}{2} = 6.675$ (1)

$\tau_{2/3} = \frac{6.65 - (-21.65)}{2} = 14.15$ (1)

$\tau_{1/3} = \frac{20 - (-21.65)}{2} = 20.825$ (1)

(4.2)





(5.1) $V = 800 \text{ lbf.}$

(3) $M_z = 800 \times 15 = 12000 \text{ lbf.in}$

$T_x = 800 \times 10 = 8000 \text{ lbf.in.}$

At point A:

$V \Rightarrow \tau_{xy} = -\frac{4V}{3A} = -\frac{4 \times 800}{3 \times \frac{\pi \times 1.5^2}{4}} = -603.6 \text{ psi}$ (3)

$M_z \Rightarrow \sigma_x = 0.$ (2)

$T_x \Rightarrow \tau_{xz} = \frac{Tp}{J} = -\frac{8000 \times 0.75}{\frac{\pi \times 1.5^4}{32}} = -12072.2$ (2)

At point B:

$V \Rightarrow \tau = 0$ (2)

$M_z \Rightarrow \sigma_x = -\frac{12000 \times 0.75}{\frac{\pi \times 1.5^4}{64}} = -36216.6 \text{ psi}$ (2)

$T_x \Rightarrow \tau_{xz} = -\frac{8000 \times 0.75}{\frac{\pi \times 1.5^4}{32}} = -12072.2 \text{ psi}$ (2)

(5.3) $\sigma_x = -36216.6 \text{ psi}, \tau_{xz} = -12072.2, \sigma_y = 0, \sigma_z = 0, \tau_{xy} = 0, \tau_{yz} = 0$

$\sigma_y = 0$ is a principal stress. (1)

$\sigma_{A,B} = \frac{-36216.6 + 0}{2} \pm \sqrt{\frac{(-36216.6)^2 + (12072.2)^2}{2}} = 3655.2, -39871.8$ (3)

21763.5 psi

$\sigma_1 = 3655.2$ (1)

$\tau_{1/3} = \frac{3655.2 - (-39871.8)}{2} = 21763.5 \text{ psi}$ (1)

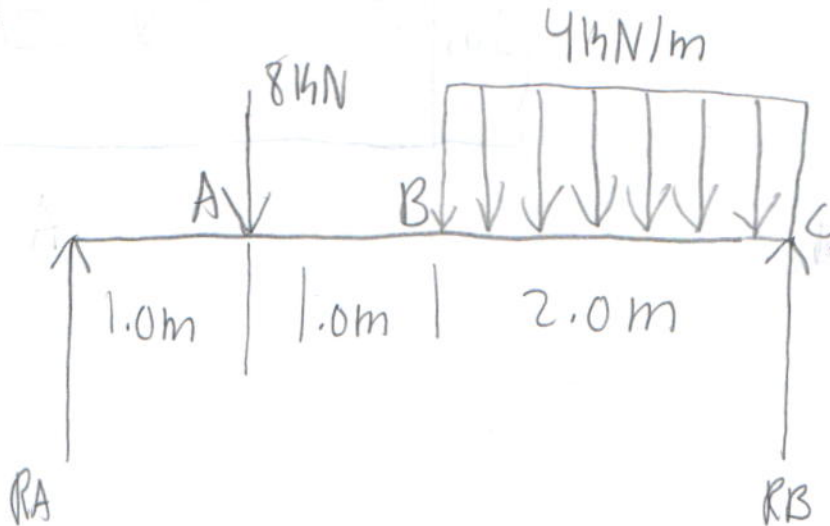
ABOVE AVERAGE

78

Name: Joseph Steele

Question 1.

(10) FBD



1.1)

$$V_C = 0$$

$$M_C = 0 \quad \checkmark$$

$$V_B = 4 \times 2 = 8 \text{ kN}$$

$$M_B = -4 \times 2 \times \left(\frac{4}{2}\right) = -16$$

$$V_A = 8 + 8 = 16 \text{ kN}$$

$$M_A = -4 \times 2 \times \left(\frac{4}{2} + 1\right) = -24$$

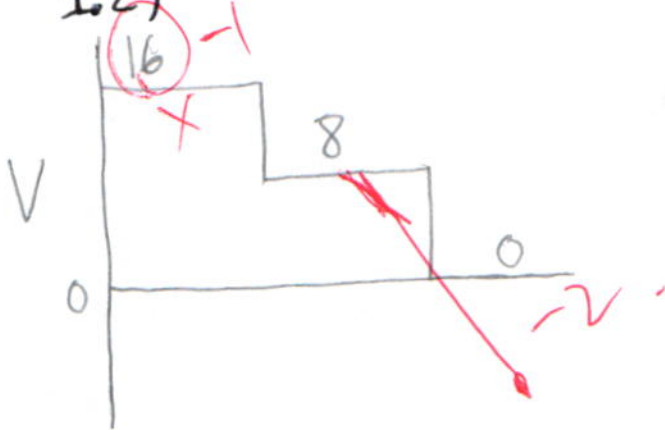
$$R_A + R_B = 16$$

$$M_A = -4 \times 2 \times \left(\frac{4}{2} + 2\right) + (-8 \times 1) = -40$$

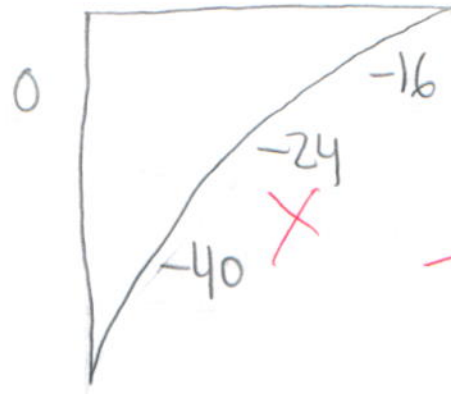
$$\sum M_A = (-8 \times 1) - 4 \times 2 \times \left(\frac{4}{2} + 2\right) + R_B(4) = 0$$

$$R_B = 10 \quad R_A = 6$$

1.2)

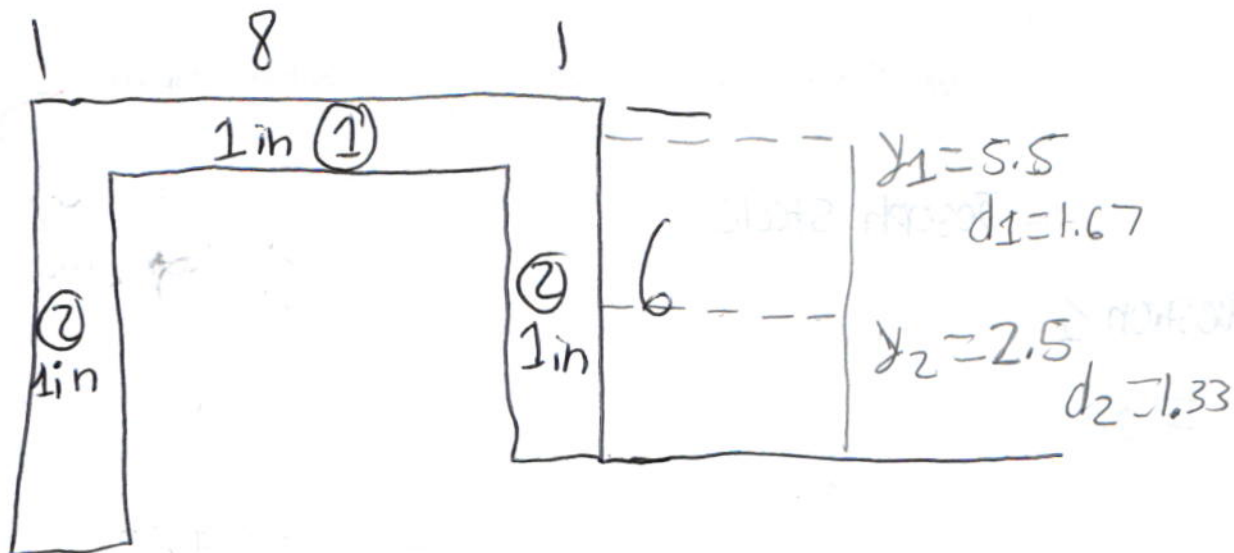


1.3)



Question 2

18
18



$$A = A_1 + A_2 + A_3$$

$$A = (8 \times 1) + (1 \times 5) + (1 \times 5) = 18 \quad \checkmark$$

2.1)

$$\bar{y} = \frac{(8 \times 1)(5.5) + (1 \times 5)(2.5) + (1 \times 5)(2.5)}{(8 \times 1) + (1 \times 5) + (1 \times 5)}$$

$$\bar{y} = 3.83 \text{ in}$$

from the bottom surface

2.2)

$$I = \frac{8 \times (1)^3}{12} + (8)(1)(1.67)^2 = 22.977$$

$$I = \frac{1 \times (5)^3}{12} + (1)(5)(1.33)^2 = 19.261$$

$$I = \frac{1 \times (5)^3}{12} + (1)(5)(1.33)^2 = 19.261$$

$$I = 61.500 \text{ in}^4$$

Question 3

(16) / 20

$\sigma_x = 25 \text{ kpsi}$ $\sigma_y = -15 \text{ kpsi}$ $\tau_{xy} = 15 \text{ kpsi CW}$

$$C = \frac{25 + (-15)}{2} = 5$$

$$R = \sqrt{\left(\frac{25 - (-15)}{2}\right)^2 + (15)^2}$$

$$= 25$$

$\sigma_1 = 25$ $\sigma_2 = -25$

$$\frac{25 + (-15)}{2} \pm \sqrt{\left(\frac{25 - (-15)}{2}\right)^2 + (15)^2}$$

$\sigma_1 = 30 \text{ kpsi}$ $\sigma_2 = -20 \text{ kpsi}$

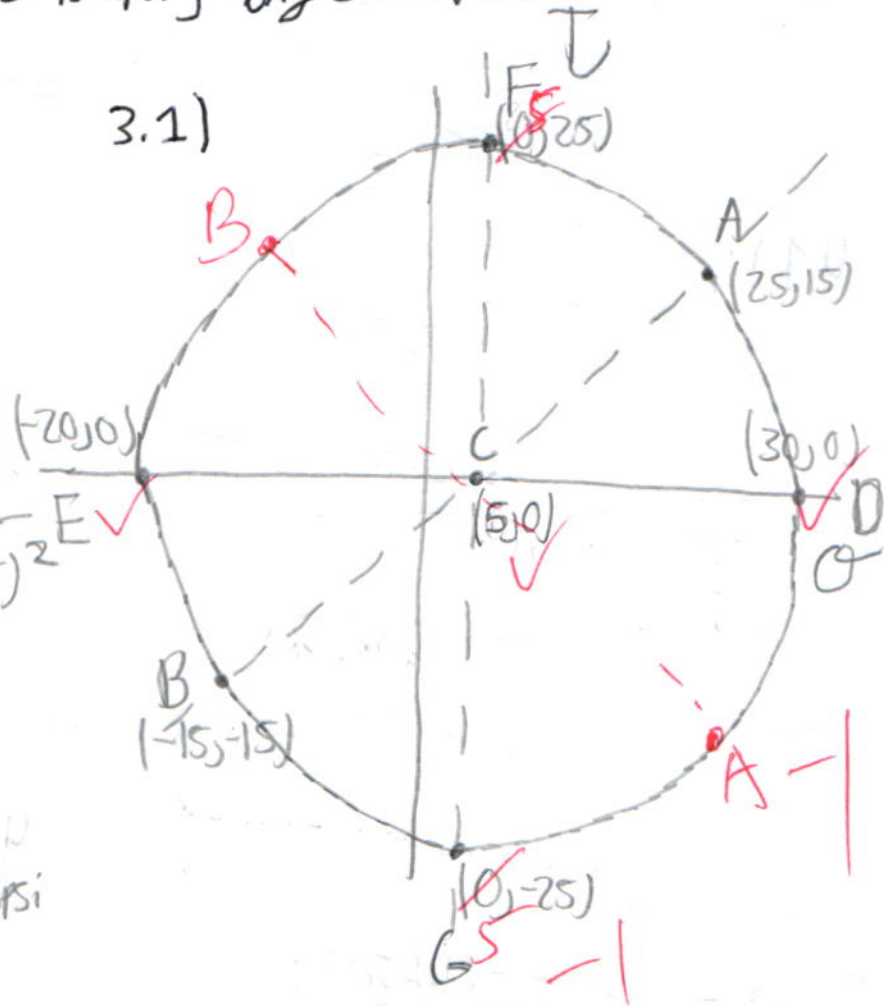
$$\frac{1}{2} \tan^{-1} \frac{2(-15)}{25 - (-15)} = 18.4349 \text{ kpsi}$$

$\theta_{p1} = 71.5651 \text{ CW}$

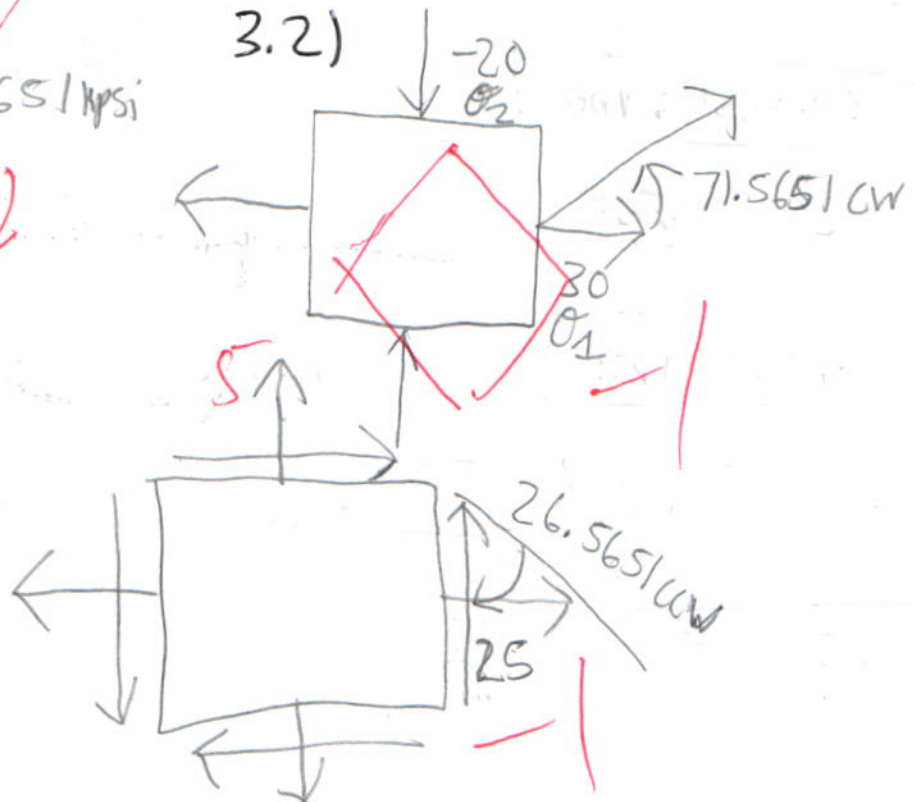
$71.5651 - 45 = 26.5651 \text{ kpsi}$

$\sigma_{s1} = 26.5651 \text{ CW}$

3.1)



3.2)



Question 4

(18) / 18

$\sigma_y = 20 \text{ kpsi}$, $\sigma_z = -15 \text{ kpsi}$, $\tau_{xz} = 12 \text{ kpsi}$ ccw

Missing elements $\sigma_x = 0$, $\tau_{xy} = 0$, $\tau_{yz} = 0$

Principle stress $\sigma_y = 20 \text{ kpsi}$

4.1)

$$C = \frac{0 + (-15)}{2} = -7.5$$

$$\frac{0 + (-15)}{2} \pm \sqrt{\left(\frac{0 - (-15)}{2}\right)^2 + (12)^2}$$

$$= 6.6509 \text{ and } -21.6509 \quad \checkmark$$

$\sigma_1 = 20 \text{ kpsi}$ $\sigma_2 = 6.6509 \text{ kpsi}$
 $\sigma_3 = -21.6509 \text{ kpsi}$

 $\checkmark$

$$\tau_{1/3} = \frac{20 - (-21.6509)}{2}$$

$= 20.82545 \text{ kpsi}$

 $\checkmark$

$$\tau_{1/2} = \frac{20 - 6.6509}{2}$$

$= 6.67455 \text{ kpsi}$

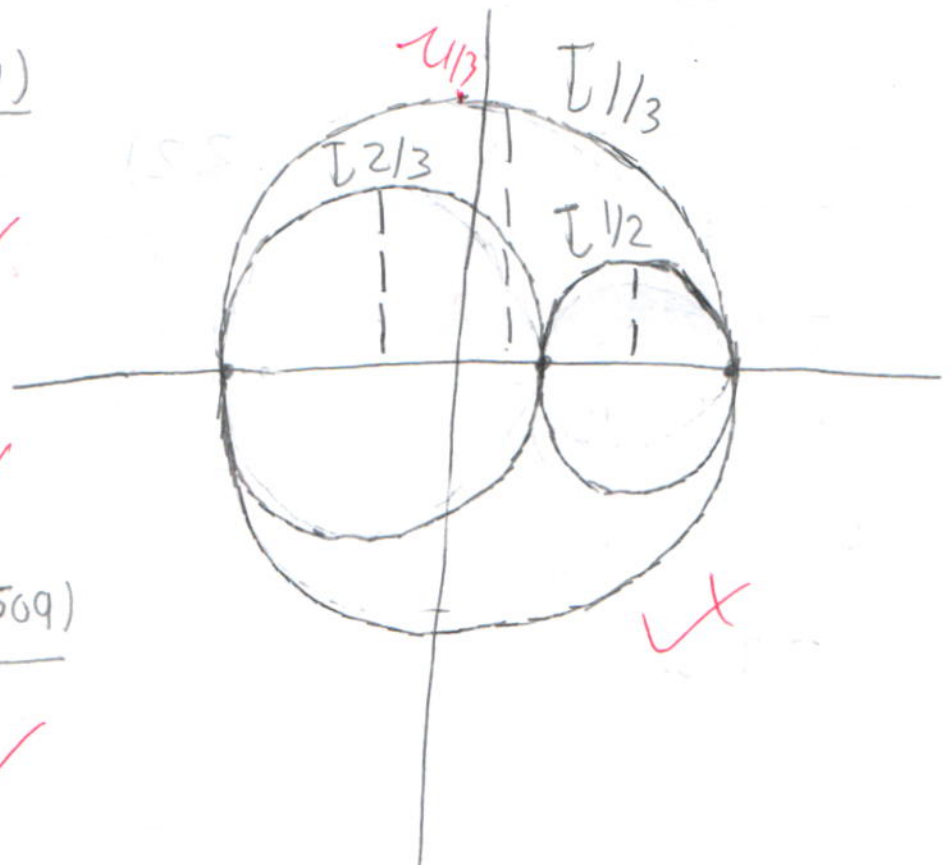
 $\checkmark$

$$\tau_{2/3} = \frac{6.6509 - (-21.6509)}{2}$$

$= 14.1509 \text{ kpsi}$

 $\checkmark$

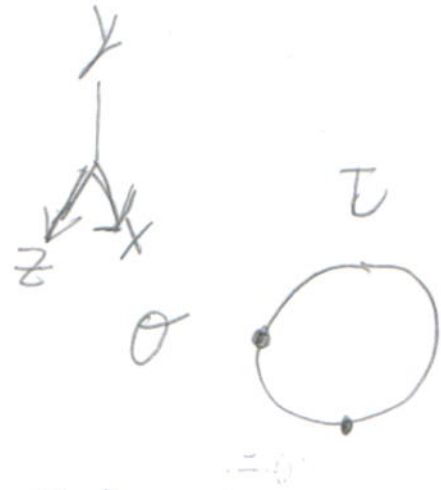
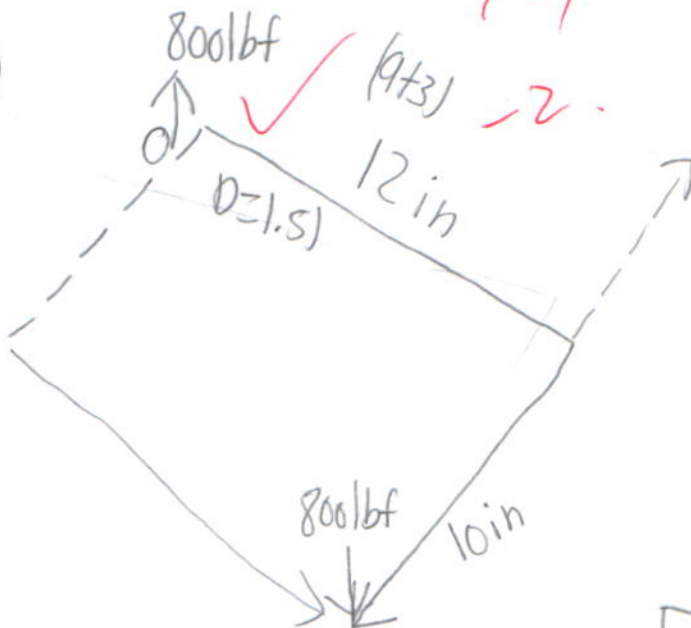
4.2)



Question 5

(16) 124

5.1)
FBD



5.3)

$$\sigma_{\text{Max}} = -4.37064$$

$$\tau_{1/3} = \frac{33.3436 - (-4.37064)}{2} = 18.85712$$

5.2)

$$\sigma_x = \frac{32 \times 9600}{\pi (1.5)^3} = 28973.2732 = 28.973 \text{ ksi}$$

$$\tau_{xz} = \frac{16 \times 8000}{\pi (1.5)^3} = 12072.19716 = 12.072 \text{ ksi}$$

$$\tau_{\text{max}} = -18.85712$$

$$\sigma_y = 0 = \text{principal stress} \quad \sigma_z = 0 \quad \tau_{xy} = 0 \quad \tau_{yz} = 0$$

$$\frac{28.973 + 0}{2} \pm \sqrt{\left(\frac{28.973 - 0}{2}\right)^2 + (12.072)^2}$$

$$= 33.3436 \text{ and } -4.37064$$

stress at A? -6.

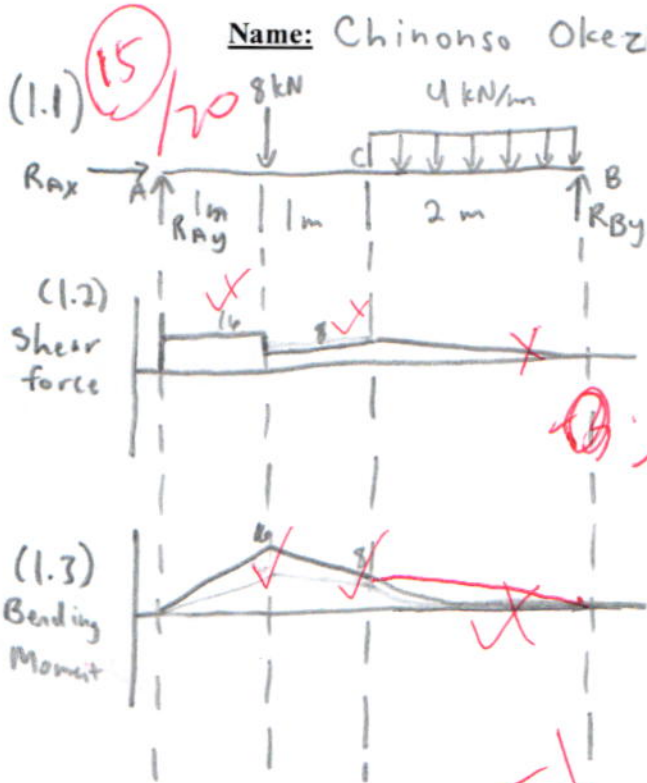
AVERAGE

MCEG3043 Machine Design I Exam One Solution Sheets

10/11/2021

64

Name: Chinonso Okezie



$$\sum F_x = 0 \rightarrow R_{Ax} = 0$$

$$\sum F_y = 0 \rightarrow R_{Ay} - 8 - (4 \times 2) + R_{By} = 0$$

$$R_{Ay} - 8 - 8 + R_{By} = 0$$

$$R_{Ay} - 16 + R_{By} = 0$$

$$\sum M_B = 0 \rightarrow (8(1)) + (4(4))R_{By} = 0$$

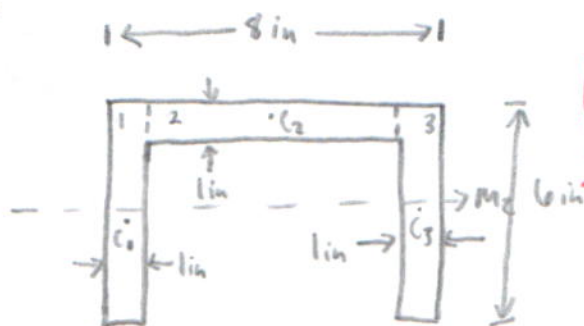
$$-8 + 16R_{By} = 0$$

$$16R_{By} = -8$$

$$R_{By} = -0.5 \text{ kN}$$

$$R_{Ay} - 16 - 0.5 = 0$$

$$R_{Ay} = 16.5$$



$$(2.1) \quad A_1 = 1 \times 6 = 6 \text{ in}^2$$

$$C_1 = 3 \text{ in}$$

$$A_2 = 6 \times 1 = 6 \text{ in}^2$$

$$C_2 = 0.5$$

$$A_3 = 1 \times 6 = 6 \text{ in}^2$$

$$C_3 = 3$$

$$(A_1 + A_2 + A_3) C = A_1 C_1 + A_2 C_2 + A_3 C_3$$

$$(6 + 6 + 6) C = 6(3) + 6(0.5) + 6(3)$$

$$18C = 18 + 3 + 18$$

$$18C = 39$$

$$C = \frac{39}{18} = 2.166666667$$

$$(2.2) \quad I = I_{ca} + Ad^2$$

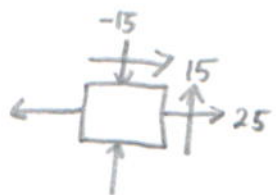
$$I_1 = \frac{1 \times 6^3}{12} + 6(3 - 2.166666667)^2 = 18 + 4.166666667 \approx 22.17 \text{ in}^4$$

$$I_2 = \frac{6 \times 1^3}{12} + 6(2.166666667 - 0.5)^2 = 0.5 + 16.66666667 \approx 17.17 \text{ in}^4$$

$$I_3 = \frac{1 \times 6^3}{12} + 6(3 - 2.166666667)^2 \approx 22.17 \text{ in}^4$$

$$I = I_1 + I_2 + I_3 = 22.17 + 17.17 + 22.17 = 61.51 \text{ in}^4$$

(3.1) $\sigma_x = 25 \text{ kpsi}$ $\sigma_y = -15 \text{ kpsi}$ $\tau_{xy} = 15 \text{ kpsi ccw}$



A(25, 15 ccw) or (25, -15)

B(-15, 15 ccw) or (-15, 15)

C(5, 0)

D(20.81, 0)

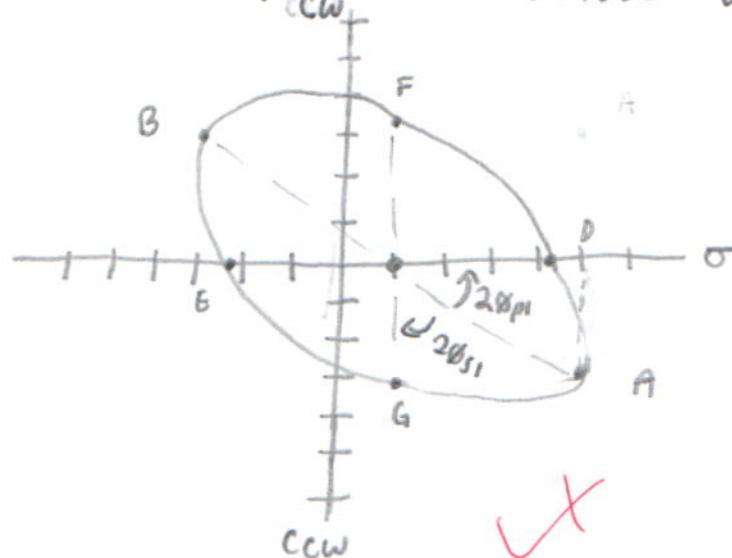
E(-10.81, 0)

F(5, 15.81)

G(5, -15.81)

$C = \frac{\sigma_x + \sigma_y}{2} = \frac{25 - 15}{2} = \frac{10}{2} = 5$

$r = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \sqrt{\left(\frac{25 - (-15)}{2}\right)^2 + 15^2} = \sqrt{\left(\frac{40}{2}\right)^2 + 225}$
 $= \sqrt{5^2 + 225} = \sqrt{25 + 225} = \sqrt{250} \approx 15.81$



A → D

$\tan 2\theta_p = \frac{2\tau_{xy}}{\sigma_x - \sigma_y} = \frac{2(15)}{25 - (-15)} = \frac{30}{40}$

$\theta_p \approx 18.435^\circ$

$\tan 2\theta_p = \left(\frac{30}{40}\right) \rightarrow 2\theta_p = \tan^{-1}\left(\frac{30}{40}\right) \approx 36.87^\circ$

$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$
 $= 5 \pm \sqrt{250} \approx 20.81, -10.81$

$\tau_1, \tau_2 = \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \pm \sqrt{250} \approx 15.81, -15.81$

A → G

$\tan 2\theta_s = 90 - 36.87 = 53.13^\circ$

$\theta_s = 26.565^\circ$



(4.1) $\sigma_y = 20 \text{ kpsi}$ $\sigma_z = -15 \text{ kpsi}$ $\tau_{xz} = 12 \text{ kpsi}$ cw

$\sigma_x = 0$

$\tau_{xy} = 0$

$\tau_{yz} = 0$

$\therefore \sigma_y = 20 \text{ kpsi}$ is the principal stress ✓

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \frac{0 + 20}{2} \pm \sqrt{\left(\frac{0 - 20}{2}\right)^2 + 0^2} = \frac{20}{2} \pm \sqrt{\left(\frac{-20}{2}\right)^2} = 10 \pm \sqrt{(-10)^2}$$

$$= 10 \pm \sqrt{100} = 10 + 10, 10 - 10 = 20, 0 \quad \text{X -4}$$

$\sigma_1 = 20$

$\sigma_2 = 20$

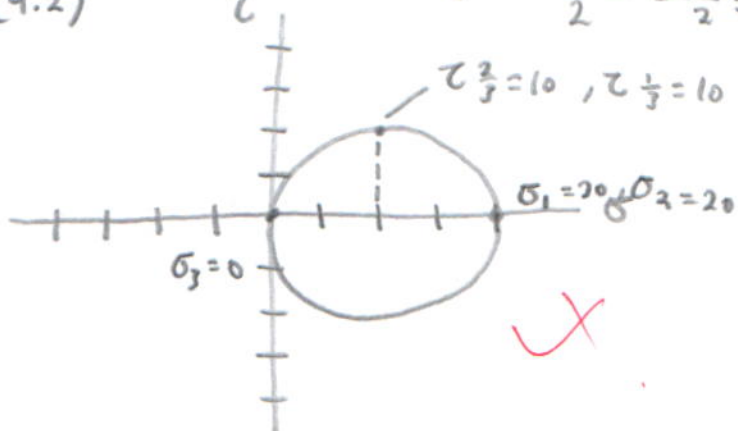
$\sigma_3 = 0$

$\tau_{\frac{1}{2}} = \frac{20 - 20}{2} = 0$

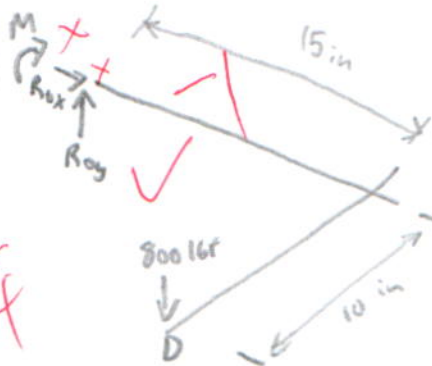
$\tau_{\frac{2}{2}} = \frac{20 - 0}{2} = \frac{20}{2} = 10$

$\tau_{\frac{1}{2}} = \frac{20 - 0}{2} = \frac{20}{2} = 10$

(4.2) $\tau_{\frac{1}{2}} = 10, \tau_{\frac{2}{2}} = 10$ ✓



(5.1)



$$V = ?$$

$$M_z = ?$$

$$T_x = ?$$

-3

$$J = \frac{\pi d^4}{32} = \frac{\pi (1.5)^4}{32} \approx 0.5$$

(5.2)

-12

(5.3)

-6

BELOW AVERAGE

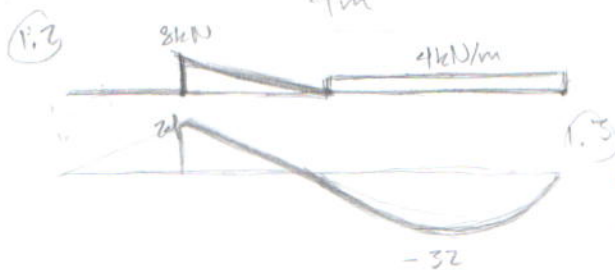
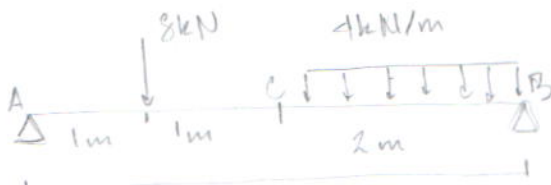
MCEG3043 Machine Design I Exam One Solution Sheets

10/11/2021

31

Name: Nehemiah Carter

10/20



$$M_A = 8(1) + 4(2)(2)$$

$$= 24 \text{ kN} \cdot \text{m} \text{ ccw}$$

$$M_B = 4(2) + 8(3)$$

$$= 32 \text{ kN} \cdot \text{m} \text{ cw}$$

1.1

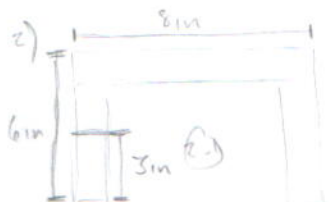
$R_A = ? - 3$

$R_B = ? - 4$

X - 8

X - 8

10/18



$$Q = \bar{y} A$$

$$= 3(6)$$

$$= 18$$

X

10/20

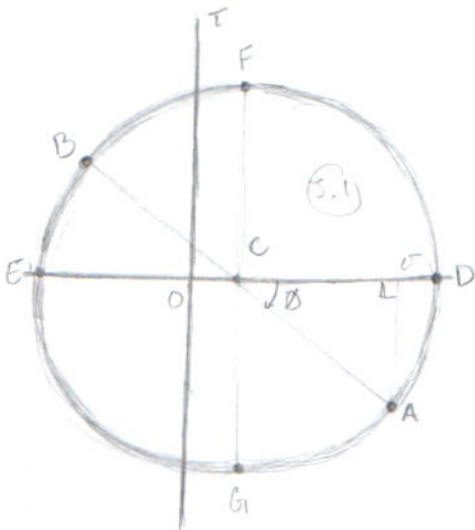
$$3) \sigma_x = 25 \text{ kpsi} \quad \sigma_y = -15 \text{ kpsi} \quad \tau_{xy} = 15 \text{ kpsi} \text{ ccw}$$

$$\phi = \frac{1}{2} \tan^{-1} \left(\frac{2\tau_{xy}}{\sigma_x - \sigma_y} \right) \rightarrow \phi = \frac{1}{2} \tan^{-1} \left(\frac{2(-15)}{25 - (-15)} \right) = -18.43^\circ$$

$$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2} \right)^2 + \tau_{xy}^2} \rightarrow \sigma_1, \sigma_2 = \frac{25 - 15}{2} \pm \sqrt{\left(\frac{25 + 15}{2} \right)^2 + 15^2}$$

$$= 5 \pm 25$$

$$= 30, -20$$



$$3.2 \quad A = (28.72, 2.91) \quad X$$

$$B = (-18.72, 12.91) \quad X$$

$$C = (5, 0) \quad \checkmark$$

$$D = (30, 0) \quad \checkmark$$

$$E = (-20, 0) \quad \checkmark$$

$$F = (5, 25) \quad \checkmark$$

$$G = (5, -25) \quad \checkmark$$

-2

(3.2)

-8

4) $\sigma_y = 20 \text{ kpsi}$ $\sigma_z = 75 \text{ kpsi}$ $\tau_{xz} = 12 \text{ kpsi}$ (ccw)

(17)
18

$\sigma_x = 0 \text{ kpsi}$ $\tau_{xy} = 0 \text{ kpsi}$ $\tau_{yz} = 0 \text{ kpsi}$ ✓

$$\sigma_{1,2} = \frac{\sigma_x + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_z}{2}\right)^2 + \tau_{xz}^2}$$

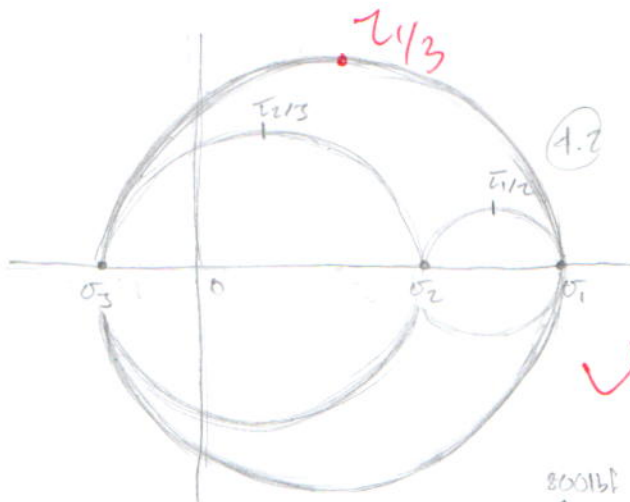
$$= 1.25 \pm 12.06$$

$$= 13.31, -10.81$$

$$\tau_{12} = \frac{20 - 13.31}{2} = 3.34$$

$$\tau_{23} = \frac{13 + 10.81}{2} = 12.06$$

$$\tau_{13} = \frac{20 + 10.81}{2} = 15.41$$



(1.1)

$\sigma_1 = 20 \text{ kpsi}$, $\sigma_2 = 13.31 \text{ kpsi}$

$\sigma_3 = 75 \text{ kpsi}$, $\tau_{12} = 3.34 \text{ kpsi}$

$\tau_{23} = 12.06 \text{ kpsi}$, $\tau_{13} = 15.41 \text{ kpsi}$

(5.2)

$$\sigma_m = \frac{M_c}{I}$$

$$= \frac{(M_y^2 + M_z^2)^{1/2} (d/2)}{\pi d^4 / 64}$$

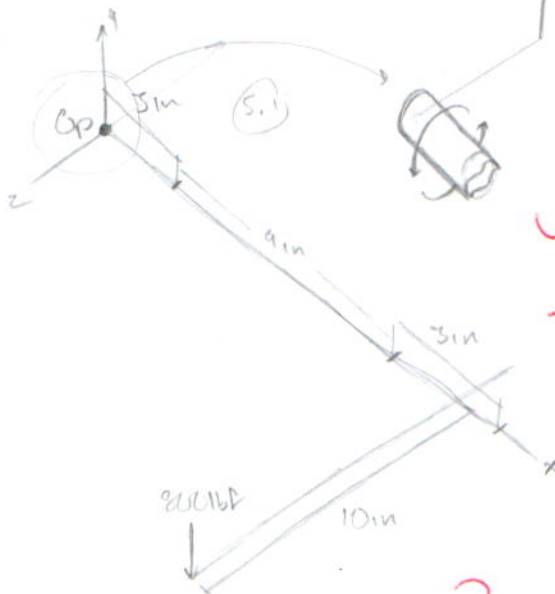
$$= \frac{(M_y^2 + (\frac{M_z^2}{1.5})^{1/2})^{1/2} (1.5/2)}{\pi (1.5)^4 / 64}$$

$= 24.26 \text{ lbf}$

$\tau_{max} = \frac{dV}{3A} = 0.114 \text{ lbf}$

5)

(3)
24



$M_z = ?$

$\tau_x = ?$

-2

(5.3)

-6

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Exam Two

TOPICS COVERED IN EXAM TWO

Chapter 4. Deflection

Chapter 5. Failure Theory

**MACHINE DESIGN I
FALL 2021**

**SAMPLE EXAM QUESTIONS, SOLUTION,
AND GRADING SCHEME**

Exam Two
MCEG3304: Machine Design I
Fall 2021
11/3/2021
Total: 100 pts

Name:

Note:

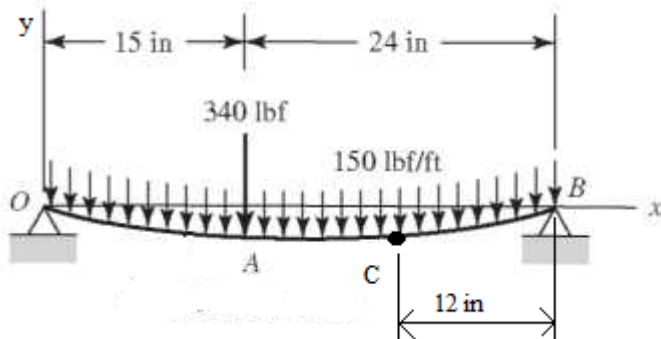
- Time: 8:00am ~ 10:00 am.
- You can only use the formula provided by instructor and calculator during the exam. Using any other devices (including phone) or materials is prohibited.
- Write your answers on blank papers. Write question number and sub-question numbers, otherwise you will not receive any points on that questions.
- Return both the exam sheets and solution sheets separately.

1. (12 pts)

(1) A 4-in-diameter steel rod ($E=30$ Mpsi) is **40 in** long and loaded by **12000 lbf** normal force. What is the spring rate for tension? What is the deformation?

(2) A 2-in-diameter steel rod ($G=11.5$ Mpsi) is **40 in** long and loaded by **15000 lbf-in** torque. What is the spring rate for tension? What is the deformation?

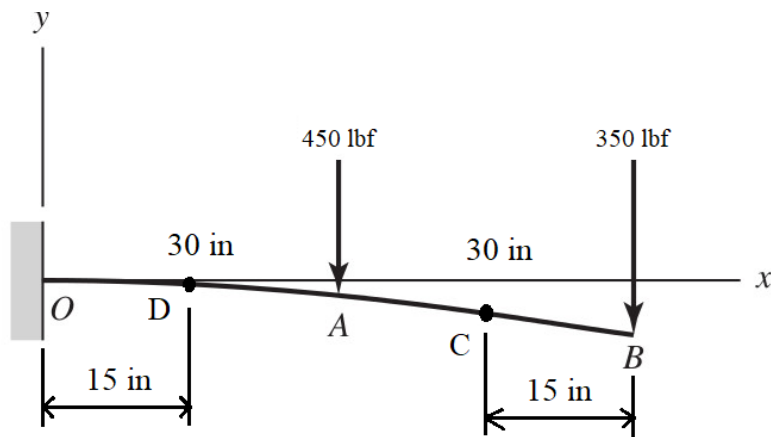
2. (14 pts) The circular beam is shown below. Assume $E = 30$ Mpsi. If the deflection at C will not exceed 0.15 in, what does the minimum diameter of the shaft need to be?



3. (20 pts) The cantilever circular beam ($d = 2$ in) is loaded by two forces shown below. Assume $E = 30$ Mpsi.

(3.1) Find the deflection at C.

(3.2) Find the deflection at D.



4. (18 pts) For a bar of AISI 1030 hot-rolled steel ($S_y = 37.5$ kpsi). The stress states

$$\sigma_x = -15 \text{ kpsi}, \sigma_y = 20 \text{ kpsi}, \tau_{xy} = -12 \text{ kpsi}$$

(4.1) (12 pts) Using the maximum-shear-stress theory to determine the safety factor.

(4.2) (6 pts) Using distortion-energy theory to determine the safety factor.

5. (12 pts) For a brittle bar ($S_{ut} = 60$ kpsi, $S_{uc} = 50$ kpsi), using the maximum-normal-stress theory determine the safety factor for the stress states $\sigma_y = 70$ kpsi, $\sigma_z = -80$ kpsi, $\tau_{xz} = 30$ kpsi.

6. (24 pts) A crank has yield strength $S_y = 50$ kpsi. It is loaded by the forces $F = 800$ lbf as shown in the figure.

(1) (6 pts) Draw the freebody diagram and calculate all the reaction loads at the cross section at the wall.

(2) (12 pts) Calculate all three-dimensional stress elements of point A and point B.

(3) (6 pts) Compute factors of safety, based upon the distortion-energy theory for point A.

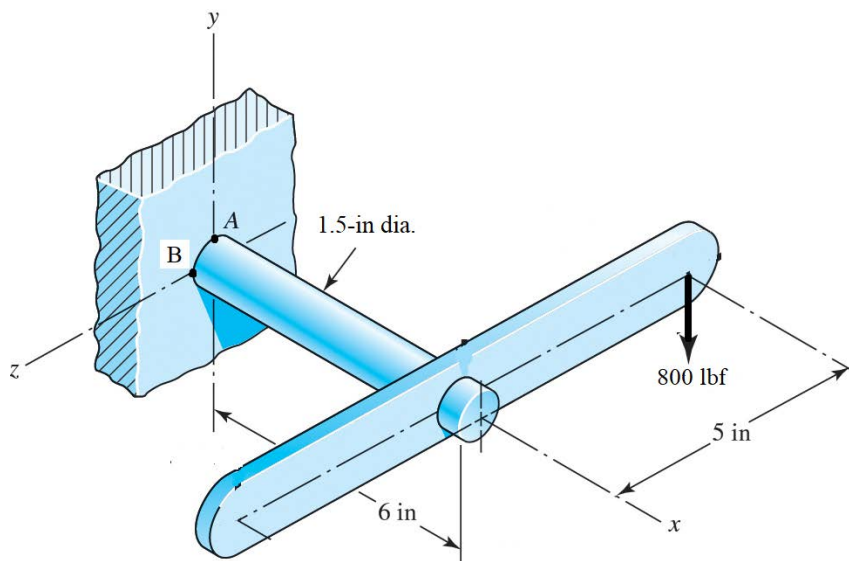
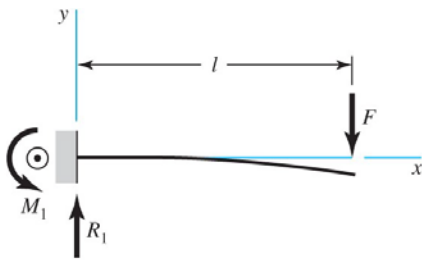


Table A-9

1 Cantilever—end load



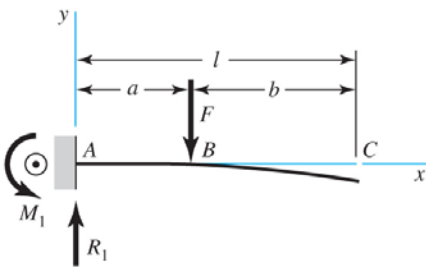
$$R_1 = V = F \quad M_1 = Fl$$

$$M = F(x - l)$$

$$y = \frac{Fx^2}{6EI}(x - 3l)$$

$$y_{\max} = -\frac{Fl^3}{3EI}$$

2 Cantilever—intermediate load



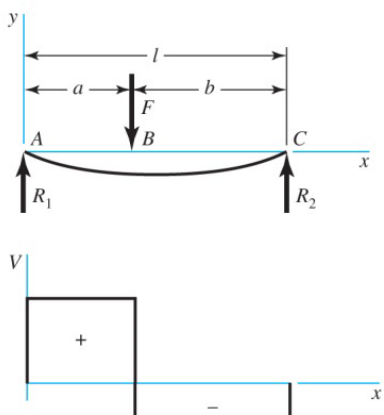
$$R_1 = V = F \quad M_1 = Fa$$

$$M_{AB} = F(x - a) \quad M_{BC} = 0$$

$$y_{AB} = \frac{Fx^2}{6EI}(x - 3a)$$

$$y_{BC} = \frac{Fa^2}{6EI}(a - 3x)$$

6 Simple supports—intermediate load



$$R_1 = \frac{Fb}{l} \quad R_2 = \frac{Fa}{l}$$

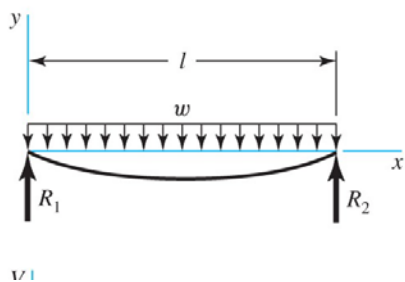
$$V_{AB} = R_1 \quad V_{BC} = -R_2$$

$$M_{AB} = \frac{Fbx}{l} \quad M_{BC} = \frac{Fa}{l}(l - x)$$

$$y_{AB} = \frac{Fbx}{6EI}(x^2 + b^2 - l^2)$$

$$y_{BC} = \frac{Fa(l - x)}{6EI}(x^2 + a^2 - 2lx)$$

7 Simple supports—uniform load



$$R_1 = R_2 = \frac{wl}{2} \quad V = \frac{wl}{2} - wx$$

$$M = \frac{wx}{2}(l - x)$$

$$y = \frac{wx}{24EI}(2lx^2 - x^3 - l^3)$$

MCEG3304 Machine Design I Exam Two Solution Sheets

11/3/2021

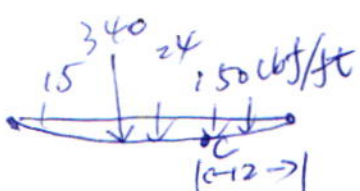
Name:

1. (1) $k = \frac{AE}{l} = \frac{\frac{\pi \times 4^2}{4} \times 30 \times 10^6}{40} = 9.42 \times 10^6 \text{ lbf/in}$ (3)

(12) $\delta = \frac{Fl}{AE} = \frac{12000 \times 40}{\frac{\pi \times 4^2}{4} \times 30 \times 10^6} = 0.001273 \text{ in}$ (3)

(2) $k = \frac{GJ}{l} = \frac{11.5 \times 10^6 \times \frac{\pi \times 2^4}{32}}{40} = 0.4516 \times 10^6 \text{ lbf/in}$ (3)

$\theta = \frac{Tl}{GJ} = \frac{15000 \times 40}{11.5 \times 10^6 \times \frac{\pi \times 2^4}{32}} = 0.033215$ (3)

2. (14)  $y_c = \frac{Fa(l-x)}{6EI} (x^2 + a^2 - 2lx) + \frac{wx}{24EI} (2lx^2 - x^3 - l^3)$ (2)

$\alpha = 15 \quad l = 39 \text{ in}, F = 340 \quad w = 150/12 = 12.5 \text{ lbf/in}$

$y_c = -0.15$ (2)

$$\frac{340 \times 15 \times (39 - 27)}{6 \times 30 \times 10^6 \times I \times 39} (27^2 + 15^2 - 2 \times 39 \times 27) + \frac{12.5 \times 27}{24 \times 30 \times 10^6 \cdot I} (2 \times 39 \times 27^2 - 27^3 - 39^3) = -0.15$$

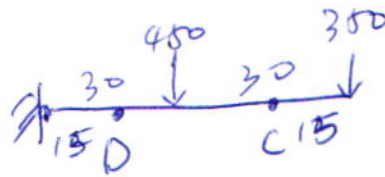
(1) $I = \frac{-0.1020421202}{-0.15} = 0.13614$

(2) $D = \sqrt[4]{\frac{0.13614 \times 64}{\pi}} = 1.29 \text{ in}$

3.
(20)

$$d = 2 \text{ in.}$$

$$I = \frac{\pi d^4}{64} =$$



(3.1)

$$C \quad (X_c = 45 \text{ in}) \quad (2)$$

$$y_1 = \frac{F a^2}{6EI} (a - 3x) = \frac{450 \times 30^2 \times (30 - 3 \times 45)}{6 \times 30 \times 10^6 \times \frac{\pi \times 2^4}{64}} = -0.3 \text{ in} \quad (2)$$

$$F = 450 \text{ lb} \quad a = 30 \text{ in}$$

$$y_2 = \frac{F x^2 (x - 3l)}{6EI} = \frac{350 \times 45^2 \times (45 - 3 \times 60)}{6 \times 30 \times 10^6 \times \frac{\pi \times 2^4}{64}} = -0.677 \text{ in} \quad (2)$$

$$F = 350 \quad l = 60 \text{ in} \quad (1)$$

$$y_c = -0.3 - 0.677 = -0.977 \text{ in} \quad (2)$$

$$(3.2) \quad D \quad (X_D = 15 \text{ in}) \quad (2)$$

$$y_1 = \frac{F x^2}{6EI} (x - 3a) = \frac{450 \times 15^2 \times (15 - 3 \times 30)}{6 \times 30 \times 10^6 \times \frac{\pi \times 2^4}{64}} = -0.0537 \text{ in} \quad (2)$$

$$F = 450, \quad a = 30 \text{ in.} \quad (1)$$

$$y_2 = \frac{F x^2}{6EI} (x - 3l) = \frac{350 \times 15^2 \times (15 - 3 \times 60)}{6 \times 30 \times 10^6 \times \frac{\pi \times 2^4}{64}} = -0.0919 \text{ in} \quad (2)$$

$$F = 350 \quad l = 60 \quad (1)$$

$$y_D = -0.0537 - 0.0919 = -0.1456 \text{ in} \quad (2)$$

4. (18) $S_y = 37.5$ $\sigma_x = -15$, $\sigma_y = 20$, $\tau_{xy} = -12$

(4.1) MSS. $\sigma_2 = 0$ is a principal stress

(12) $\sigma_{A,B} = \frac{-15+20}{2} \pm \sqrt{\left(\frac{-15-20}{2}\right)^2 + (-12)^2} = 23.72, -18.72$ (5)

$\sigma_1 = 23.72$, $\sigma_2 = 0$, $\sigma_3 = -18.72$ (3)

$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = 21.22$ (2)

$n = \frac{0.5 \times S_y}{\tau_{max}} = \frac{0.5 \times 37.5}{21.22} = 0.88$ (2)

(4.2) DE $\sigma^1 = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$
 (6) $= ((-15)^2 - (-15 \times 20) + 20^2 + 3 \times (-12)^2)^{1/2}$
 $= 36.84$ (4)

$n = \frac{S_y}{\sigma^1} = \frac{37.5}{36.84} = 1.02$ (2)

5. $S_{ut} = 60$ $S_{uc} = 50$ $\sigma_y = 70$ $\sigma_z = -80$, $\tau_{xz} = 30$

(12) MNS: $\sigma_y = 70$ is a principal stress

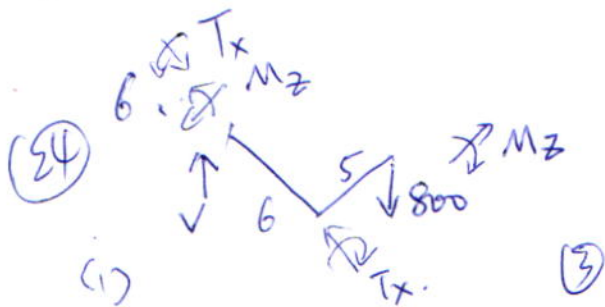
$\sigma_{A,B} = \frac{\sigma_x + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_z}{2}\right)^2 + \tau_{xz}^2} = \frac{0 + 80}{2} \pm \sqrt{\left(\frac{0 + 80}{2}\right)^2 + (30)^2}$
 $= 81, -11$ (4)

$\sigma_1 = 70$, $\sigma_2 = +10$, $\sigma_3 = -90$ (3)

$n_1 = \frac{S_{ut}}{\sigma_1} = \frac{60}{70} = 0.86$ (2)

$n_2 = \frac{-S_{uc}}{\sigma_3} = \frac{-50}{-90} = 0.56$ (2)

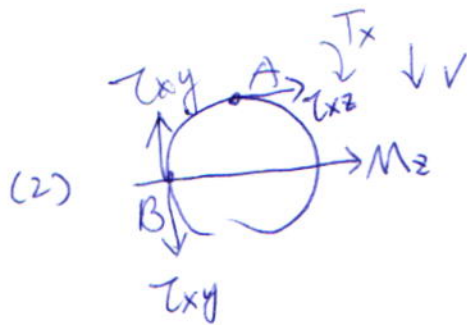
$\therefore n = 0.56$ (1)



(1) $V = 800 \text{ lbf}$

$M_z = 800 \times 6 = 4800 \text{ lbf}\cdot\text{in}$

$T_x = 800 \times 5 = 4000 \text{ lbf}\cdot\text{in}$ (3)



A: (2) $V = 800 \Rightarrow \tau = 0$

(2) $M_z \Rightarrow \sigma_x = \frac{M_y}{I_z} = \frac{4800 \times 0.75}{\frac{\pi \times 1.5^4}{64}} = 14486.6 \text{ psi}$

(2) $T_x \Rightarrow \tau_{xz} = -\frac{Tl}{J} = -\frac{4000 \times 0.75}{\frac{\pi \times 1.5^4}{32}} = 6036.1$

B: $V \Rightarrow \tau_{xy} = -\frac{4V}{3A} = -\frac{4 \times 800}{3 \times \pi \times 0.75^2} = -603.6 \text{ psi}$ (2)

$M \Rightarrow \sigma_x = 0$

$T_x \Rightarrow \tau_{xy} = \frac{Tl}{J} = \frac{4000 \times 0.75}{\frac{\pi \times 1.5^4}{32}} = 6036.1 \text{ psi}$ (2)

(3) DE: $\sigma_x = 14486.6, \sigma_y = 0, \sigma_z = 0$

$\tau_{xy} = 0, \tau_{yz} = 0, \tau_{xz} = -6036.1$

$\sigma' = \frac{1}{\sqrt{2}} [(14486.6)^2 + (14486.6)^2 + 6 \times 6036.1^2]^{\frac{1}{2}}$

$= 17865.2 \text{ psi}$ (4)

$n = \frac{50000}{17865.2} = 2.8$ (2)

ABOVE AVERAGE

MCEG3304 Machine Design I Exam Two Solution Sheets

11/3/2021

Name: Dwayne Smith

83

1. $k = \frac{AE}{L} = \frac{(\pi(2)^2)(30) \times 10^6}{40} = \boxed{9.425 \times 10^6}$ ✓

(8) $\frac{M}{EI} = \frac{12000 \times 40}{(30 \times 10^6 \frac{\pi(2)^4}{64})} = \boxed{1273.24}$ ✓ -1

2. $k = \frac{GJ}{L} = \frac{(11.5 \times 10^6)(\frac{\pi(2)^4}{32})}{40} = \boxed{0.452}$ ✓

$\frac{M}{EI} = \frac{15000}{30 \times 10^6 (\frac{\pi(2)^4}{64})} = \boxed{636.62}$ ✓ -3

3. d = 2 in

2.) Table A-4-6 (14)/14

$$y_{BC} = \frac{F_u(l-x)}{6EI l} (x^2 + a^2 - 2lx)$$

$$0.15 = \frac{340(15)(39-27)}{6(30)I(39)} (27^2 + (15)^2 - 2(39)(27)) = -0.06645$$

Table A-4-7

$$y = \frac{wx}{24EI} (2lx^2 - x^3 - l^3)$$

$$0.15 = \frac{150(27)}{24(30)I} (2(39)(27)^2 - 27^3 - 39^3)$$

$$I = -0.064$$

$$\frac{\pi d^4}{64} = -0.135$$

$$d = 1.28 \text{ in}$$

20 / 20 $d = 2 \text{ in}$

3.) Table A-9-1

3.1) $y = \frac{Fx^2}{6EI} (x - 3a)$ ✓

$= \frac{350 (45)^2}{6(30) \frac{\pi(2)^4}{64}} (45 - 3(60))$

$y_1 = -0.6768$ ✓

Table A-9-2

$y_2 = \frac{Fa^2}{6EI} (u - 3x)$ ✓

$= \frac{450 (30)^2}{6(30) \frac{\pi(2)^4}{64}} (30 - 3(45))$

$y_2 = -0.3008$ ✓

$y_1 + y_2 = y_c$

$-0.6768 + -0.3008$

$y_c = \boxed{-0.9776 \text{ in}}$ ✓

3.2 | Table A-9-1

$$y = \frac{Fx^2}{6EI} (x - 3l)$$

$$y_{10} = \frac{350(15)^2}{6(30) \frac{\pi(2)^4}{64}} (15 - 60(3))$$

$$= -0.09141$$

Table A-9-2

$$y_{AB} = \frac{Fx^2}{6EI} (x - 3l)$$

$$= \frac{450(15)^2}{6(30) \frac{\pi(2)^4}{64}} (15 - 3(30))$$

$$y_{20} = -0.041778$$

$$y_{10} + y_{20} = y_D$$

$$y_D = -0.09141 + -0.041778$$

$$= \boxed{-0.13368 \text{ in}}$$

4) $\sigma_x = -15$ $\sigma_y = 20$ $\tau_{xy} = -12$ $S_y = 37.5$
 $\sigma_z = 0$ $\tau_{xz} = 0$ $\tau_{yz} = 0$

4.1) $\sigma_{1,2} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$
 $= \frac{-15 + 20}{2} \pm \sqrt{\left(\frac{-15 - 20}{2}\right)^2 + (12)^2}$
 $= 2.5 \pm 21.219$

$\sigma_1 = 23.719$ $\sigma_2 = 0$ $\sigma_3 = -18.719$ ✓

$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} \Rightarrow \frac{23.719 + 18.719}{2} = 21.219$ ✓

$n = \frac{0.5 (37.5)}{21.219} = 0.8836$ ✓

4.2) $\sigma' = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$
 $= (15^2 - (-15)(20) + 20^2 + 3(12)^2)^{1/2}$

$\sigma' = 36.837$ ✓

$n = \frac{37.5}{36.837} = 1.018$ ✓

5 | 11/12

$$\sigma_y = 70 \text{ kpsi} \quad \sigma_z = -80 \text{ kpsi} \quad \tau_{xz} = 30 \text{ kpsi}$$

$$\sigma_x = 0 \quad \tau_{xy} = 0 \quad \tau_{yz} = 0$$

$$\sigma_{1,2} = \frac{0 - 80}{2} \pm \sqrt{\left(\frac{80}{2}\right)^2 + (30)^2}$$

$$= -40 \pm 50$$

$$\sigma_1 = 70 \quad \sigma_2 = 10 \quad \sigma_3 = -90 \quad \checkmark$$

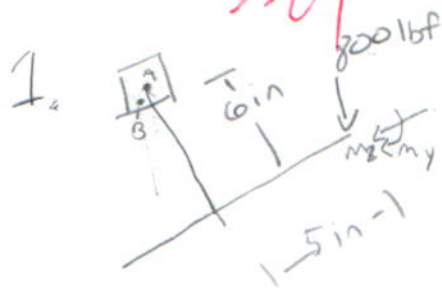
$$n_1 = \frac{60}{70} = \boxed{0.857} \quad \checkmark$$

$$n_2 = \frac{-50}{-90} = \boxed{0.556} \quad \checkmark$$

$$n = 0.56 \quad -1$$

6

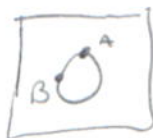
12/24



$A \begin{matrix} m_y \\ m_x \end{matrix}$

$$V = 800 \text{ lbf} \quad \checkmark$$

$$m = 800 \times 6 = 4800 \quad \checkmark$$



$B \begin{matrix} m_z \\ m_x \end{matrix}$

$$B = 800 \times 5 = 4000 \quad \checkmark$$

2.

$$\sigma = - \frac{m_y}{I} \quad A$$

$$\sigma_1 =$$

$$\sigma_3 = \frac{800 \times 6 \times 0.75}{\frac{\pi (1.5)^4}{64}} = 14.35 \text{ Kpsi} \quad \checkmark$$

$$\tau = ?$$

stress at B

0.0

-4

$$\tau_{max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{4V}{3A} = \frac{4(800)}{3 \frac{\pi (1.5)^2}{4}} = 603.61$$

$$3) \sigma' = \frac{1}{\sqrt{2}} \left((\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right)^{1/2}$$

$$n = \frac{s_y}{\sigma'}$$

-6.

AVERAGE

MCEG3304 Machine Design I Exam Two Solution Sheets

11/3/2021

68

Name: TY Dupre

1.1) 4 in = d E = 30 mpsi. (30 x 10⁶) 40 in = L 12 coils normal force
 Spring rate $k = \frac{AE}{L} = k = \frac{(a)(30 \times 10^6)}{L}$ $A = \frac{\pi d^2}{4}$
 $G = 12.567$

Spring rate $k = \frac{(12.567)(30 \times 10^6)}{40} = 9,424,500$ ✓

Deformation $\frac{M}{EL} = \frac{d^2 y}{dx^2}$ $\delta = \frac{FL}{AE} \rightarrow 3$

1.2) G = 11.5 mpsi 40 in long 2 in = d
 $k = \frac{GJ}{L} = \frac{(11.5 \times 10^6)(J)}{40 \text{ in}}$ $J = \frac{\pi d^4}{32}$ $J = \frac{\pi 2^4}{32} = 1.57$

Spring rate $k = \frac{(11.5 \times 10^6)(1.57)}{40} = 451,375$ ✓

Deformation $\theta = \left(\frac{T L}{G J} \right) = \frac{15000(40)}{(11.5 \times 10^6)(1.57)} = 0.033$ ✓

2 deflection at $x \leq 0.15$ in

$$x = 27 \text{ in} - 2$$

(12)
14 $\frac{Wx}{24EI} (2x^2 - x^3 - l^3) + \frac{Fa(1-x)}{6EI} (x^2 + a^2 - 2lx) = 0.15$ in

for uniform load $W = 150 \text{ lbf/ft} = \frac{150}{12} = 12.5 \text{ lbf/in}$ ✓

$$\frac{12.5(27)}{24(30 \times 10^6)I} (2(27)^2 - 27^3 - 39^3) + \frac{340(15)(39-27)}{6(30 \times 10^6)I} (27^2 + 15^2 - 2(39)(27))$$

$$\frac{150}{24(30 \times 10^6)I} (-49815) + \frac{137700}{234(30 \times 10^6)I} (-567)$$

$$\frac{-0.0104}{I} + \frac{-0.0111}{I} = 0.15 \Rightarrow -\frac{0.0215}{I} = 0.15 \Rightarrow I = 6.98$$

$$d = \sqrt[4]{\frac{64 \times I}{\pi}} = \sqrt[4]{\frac{64 \times 6.98}{\pi}} = \boxed{3.45 \text{ in diameter}}$$

✓

$$E = 30 \text{ MPsi} = 30 \times 10^6 \text{ Psi}$$

$$\text{deflection at C and D } d = 2$$

3)

and load at B

$$y_{\max} = -\frac{FL^3}{3EI}$$

$$I = \frac{\pi d^4}{64} \quad \frac{\pi (2)^4}{64} = I = 0.785$$

$$\frac{350 (60)^3}{6 (30 \times 10^6) (0.785)} = -0.535 \text{ in at B}$$

$$y_{BC} = \frac{Fa^2}{6EL} (a - 3x)$$

$$\frac{450 (30)^2}{6 (30 \times 10^6) (0.785)} (30 - 3(45)) = -0.3009$$

$$\text{Deflection at C} = -0.3009 + -0.535 = \boxed{-0.836 \text{ in}}$$

Deflection at D

$$y_{AD} = \frac{F(x)^2}{6EL} (x - 3a)$$

$$\frac{450 (15)^2}{6 (30 \times 10^6) (0.785)} (15 - 3(30)) = -0.054$$

$$\text{Deflection at D} = -0.054 + -0.535 = \boxed{-0.589 \text{ in}}$$

4) $S_y = 37.5 \text{ kPsi}$ (1.8)/1.8

$\sigma_x = -15 \text{ kPsi}, \sigma_y = 20 \text{ kPsi}, \tau_{xy} = -12 \text{ kPsi}$

$\sigma_z = 0 \quad \tau_{xz} = 0 \quad \tau_{zy} = 0$

$\sigma_z = 0 \leftarrow \text{Principal stress}$

$$\sigma_{1,3} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

↓

$$\frac{-15 + 20}{2} \pm \sqrt{\left(\frac{-15 - 20}{2}\right)^2 + (-12)^2} \Rightarrow 2.5 \pm 21.22 \Rightarrow \underline{-18.72}, \underline{23.72}$$

$\sigma_1 = 23.72, \sigma_2 = 0 \quad \sigma_3 = -18.72$

$m_{T_{max}} = \frac{23.72 - (-18.72)}{2} = 21.22$ ✓

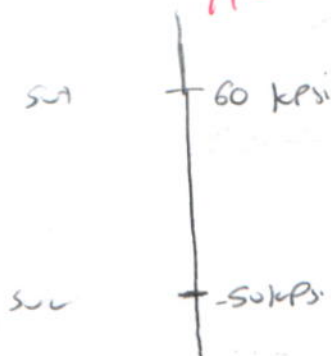
$m_{ss} = \rho = \frac{0.5 S_y}{T_{max}} = \frac{0.5(37.5)}{21.22} = \boxed{0.884}$ ✓

de $\frac{1}{\sqrt{2}} \left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right]^{1/2}$

$$\frac{1}{\sqrt{2}} \left[(-15 - 20)^2 + (20 - 0)^2 + (0 - (-15))^2 + 6((-12)^2 + (0)^2 + (0)^2) \right]^{1/2} = 36.837 = \sigma'$$

$\rho = \frac{S_y}{\sigma'} = \frac{37.5}{36.837} = \boxed{1.018}$ ✓

5) $\sigma_{ut} = 60 \text{ kpsi}$ $\sigma_{uc} = 50 \text{ kpsi}$



$\sigma_y = 70$ $\sigma_z = -80$ $\tau_{xz} = 30 \text{ kpsi}$

$\sigma_x = 0$ $\tau_{xy} = 0$ $\tau_{yz} = 0$

$\sigma_y = 70$ is Principle

$\sigma_1, \sigma_2 = \frac{\sigma_x + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_z}{2}\right)^2 + \tau_{xz}^2}$ We use xz instead of xy

$\frac{0 + (-80)}{2} \pm \sqrt{\left(\frac{0 - (-80)}{2}\right)^2 + 30^2}$ ✓

$-40 \pm \sqrt{(40^2) + 30^2} \Rightarrow -20 \pm 50 \Rightarrow 30, -70$ ✓ -!

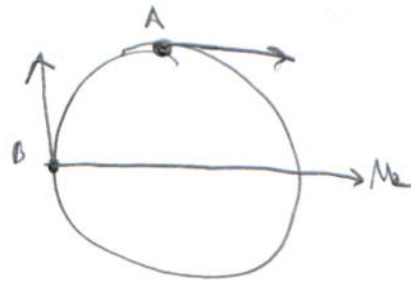
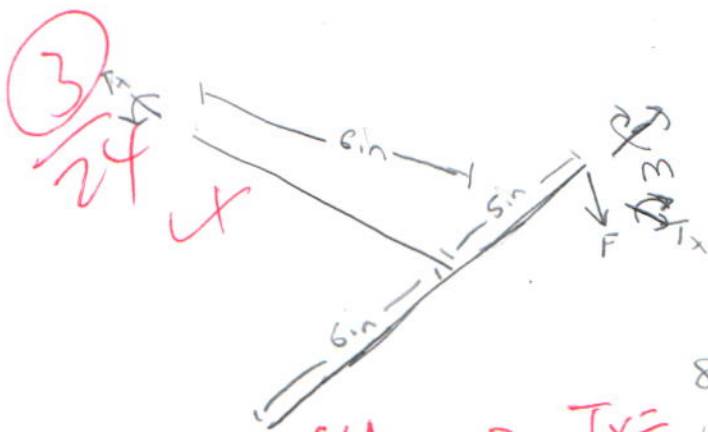
$\sigma_1 = 70$ $\sigma_2 = 30$ $\sigma_3 = -70$

$n_1 = \frac{\sigma_{ut}}{\sigma_1} = \frac{60}{70} = \boxed{0.857}$ ✓

$n_2 = \frac{-\sigma_{uc}}{\sigma_3} = \frac{-50}{-70} = \boxed{0.714}$ ✓

$n = ?$ -!

(c) $S_y = 50 \text{ kpsi}$ $F = 800 \text{ lbf}$



(1) $-3 \quad T_x = \frac{800 \times 5 \text{ in}}{4000 \text{ lbf}} = 1 \text{ lbf} \quad \checkmark$

(2) $\sigma_m = \frac{m_c}{B} \frac{(M_c)}{I} = \frac{(M_y^2 + M_z^2)^{1/2} (d/4)}{\pi d^4 / 64} = \frac{32}{\pi d^3} (M_y^2 + M_z^2)^{1/2}$

$I = \frac{\pi (1.5)^4}{64} = I = 0.249$

Normal stress for beam bends $\sigma = \frac{M_y}{I}$

✓ -12

(3) -6

BELOW AVERAGE

MCEG3304 Machine Design I Exam Two Solution Sheets

11/3/2021

47

Name: Emanuel Santiago Sanchez

1

(6) ①
12

$$E = 30 \text{ MPsi}$$

$$l = 40 \text{ in}$$

$$F = 12000 \text{ lbf}$$

$$D = 4 \text{ in}$$

$$K = \frac{AE}{l}$$

$$A_c = \frac{\pi d^2}{4}$$

$$\sigma = \frac{F}{A_c} = ? - 3$$

$$K = \frac{\pi d^2 (30 \times 10^6)}{4(40)}$$

$$= \frac{\pi (4)^2 (30 \times 10^6)}{160} = 9.4248 \times 10^6 \text{ psi} \cdot \text{in}$$

✓

②

$$G = 11.5 \text{ MPsi}$$

$$l = 40 \text{ in}$$

$$T = 15000 \text{ lbf} \cdot \text{in}$$

$$d = 2 \text{ in}$$

$$\theta = \frac{Tl}{GJ}$$

$$J = \frac{\pi d^4}{32}$$

$$K = \frac{GJ}{l}$$

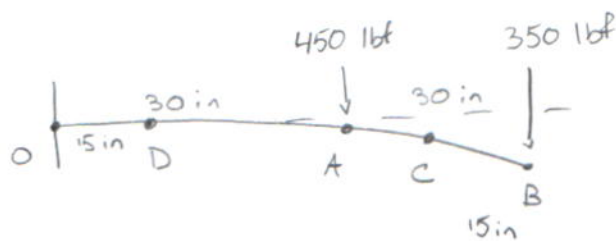
$$K = \frac{\pi d^4 (G)}{32l}$$

$$= \frac{\pi (2)^4 (11.5 \times 10^6)}{32(40)} = 4.516 \times 10^7 \text{ psi} \cdot \text{in}^3$$

✓

θ = ? - 3

3
④



① Deflection @ C

$$y_1 = \frac{FL^3}{3EI} \quad \times -2$$

In ques,
I used $y_{max} = y_c$
formula...

$$y_2 = \frac{Fx^2}{6EI} (a - 3x) \quad \checkmark$$

@ $x = L$

$$y_2 = \frac{FL^2}{6EI} (a - 3L)$$

$x = 45 \text{ in}$

$$y = y_1 + y_2 = \frac{FL^3}{3EI} + \frac{FL^2}{6EI} (a - 3L)$$

$$y = \frac{450(45)^3 + 450(45)^2(15 - 3(45))}{3(30 \times 10^6)(\frac{\pi d^4}{64})} \quad \times -1$$

$$y = \underline{\hspace{2cm}} \text{ in.} \quad -1$$

$$y_1 \Rightarrow y_2 \Rightarrow y = y_1 + y_2$$

$$D = 2 \text{ in.}$$

$$E = 30 \text{ Mpsi}$$

$$\frac{\pi d^4}{64}$$

(2)

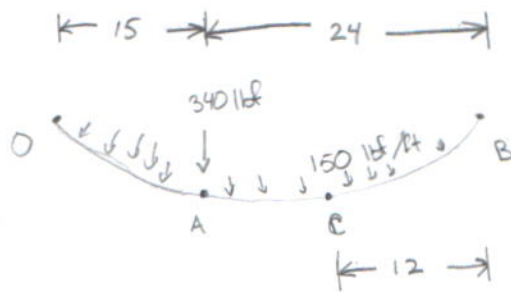
-10

$$I = \frac{\pi d^4}{64}$$

2

①

4/14



$$E = 30 \text{ MPsi}$$

$$F = 340 \text{ lbf}$$

$$y_1 = \frac{Fbx}{6EIL} (x^2 + b^2 - l^2)$$

$$y_2 = \frac{Fa(L-x)}{6EIL} (x^2 + a^2 - 2Lx)$$

$$x = 27 \text{ in} - 2$$

$$y = y_1 + y_2$$

$$.15 = \frac{Fbx}{6EIL} (x^2 + b^2 - l^2) + \frac{Fa(L-x)}{6EIL} (x^2 + a^2 - 2Lx)$$

$$\text{Solve for } d \text{ from } I = \frac{\pi d^4}{64}$$

$$d^4 = \underline{\hspace{2cm}}$$

$$* d = \underline{\hspace{2cm}}$$

4

(18) / 18

$$S_y = 37.5$$

$$\sigma_x = -15, \sigma_y = 20, \tau_{xy} = -12 \text{ kpsi}$$

$$\sigma_z = 0, \tau_{zx} = 0, \tau_{zy} = 0$$

①

$$\begin{aligned}\sigma_{A,B} &= \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} \\ &= \frac{-15 + 20}{2} \pm \sqrt{\left(\frac{-15 - 20}{2}\right)^2 + 12^2} \\ &= 2.5 \pm \sqrt{450.25} \\ &= 2.5 \pm 21.22\end{aligned}$$

$$\sigma_{A,B} = 23.72, -18.72$$

$$\sigma_1 = 23.72, \sigma_2 = 0, \sigma_3 = -18.72$$

$$n = \frac{.5(S_y)}{Z_{max}} = \frac{.5(37.5)}{21.22} = .88$$

$$n = .88$$

$$\begin{aligned}Z_{max} &= \frac{\sigma_1 - \sigma_3}{2} \\ &= \frac{23.72 - (-18.72)}{2} \\ &= \frac{42.44}{2} = 21.22\end{aligned}$$

②

$$\begin{aligned}\sigma' &= \frac{1}{\sqrt{2}} \left[(-15 - 20)^2 + (20 - 0)^2 + (15)^2 + 6(12^2 + 0 + 0) \right]^{1/2} \\ &= \frac{1}{\sqrt{2}} \left[(-35)^2 + 20^2 + 15^2 + 6(12)^2 \right]^{1/2} \\ &= \frac{\sqrt{2714}}{\sqrt{2}} = \frac{52.096}{\sqrt{2}} = 36.84\end{aligned}$$

$$n = \frac{S_y}{\sigma'} = \frac{37.5}{36.84} = 1.02$$

$$n = 1.02$$

5

$$S_{ut} = 60 \text{ kpsi}$$

MNS

$$S_{uc} = 50 \text{ kpsi}$$

$$\sigma_y = 70 \checkmark, \sigma_z = -80, \tau_{xz} = 30 \text{ kpsi}$$

$$\sigma_x = 0, \tau_{xy} = 0, \tau_{yz} = 0$$

$$\sigma_{A,B} = \frac{70 - 80}{2} \pm \sqrt{\left(\frac{70 + 80}{2}\right)^2 + 30^2}$$

$$= -5 \pm \sqrt{5625 + 900}$$

$$= -5 \pm 80.78 \quad \checkmark \quad -2$$

$$\sigma_{A,B} = 75.78, -85.78$$

$$\sigma_1 = 75.78 \quad \sigma_2 = 70 \quad \sigma_3 = -85.78 \quad \Rightarrow \text{kpsi}$$

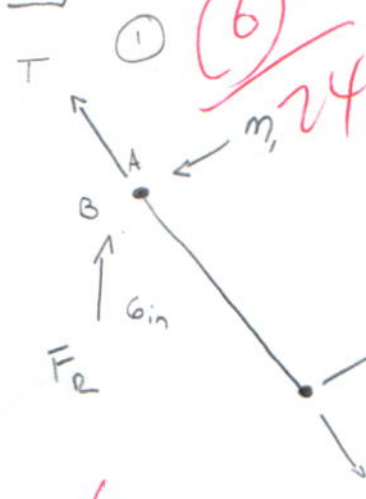
$$n_1 = \frac{60}{75.78} = .79 \quad \Rightarrow \quad n_{ut} = .79$$

$$n_2 = \frac{-(50)}{-85.78} = .58 \quad \Rightarrow \quad n_{uc} = .58$$

Failure

$$n = 0.58 \quad -1.$$

6) $S_y = 50$, $F = 800 \text{ lbf}$ $d = 1.5$



$\checkmark$ ~~FF~~ = 800 lbf
 $T = 5 \times 800 = 4000$

$M = 800 \times 6$

$I = \frac{32}{\pi d^3}$

$\sigma = -\frac{M_y}{I}$

$z_{xy} =$

② $\sigma = -\frac{\pi d^3 (800)}{32} = -265.07 \text{ lbf} \cdot \text{in}^3$

Find $\sigma_x, \sigma_y, z_{xy}$

$z_{xy} =$ _____

③ DE @ A

$\sigma' = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3z_{xy}^2)^{1/2}$

$\sigma' =$ _____

$n = \frac{S_y}{\sigma'} = \frac{50}{\sigma'} \neq$ _____ -6.

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Final Exam

TOPICS COVERED

Chapter 8. Design of Bolts

Chapter 11. Design of Bearings

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUESTIONS, SOLUTION,
AND GRADING SCHEME**

Final Exam
MCEG33043 Machine Design I
Fall 2021
12/3/2021
Total: 100 pts

Name:

Note:

- Time: 9:30am ~ 11:30 am.
- You can only use the formula provided by instructor and calculator during the exam. Using any other devices (including phone) or materials is prohibited.
- Write your answers on blank papers. Write question number and sub-question numbers, otherwise you will not receive any points on that questions.
- Return both the exam sheets and solution sheets separately.

Show the detail steps of the following calculation problems 1 - 5.

1. (16 pts) An M 14 × 2 hex-head bolt with a regular hexagonal nut and a 14R washer under each of bolt head and the nut is used to clamp together two 18-mm steel plates.

(1.1) (7 pts) Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

(1.2) (9 pts) Determine the bolt stiffness k_b , the bolt material is carbon steel ($E=207\text{GPa}$).

2. (21 pts) A **2.5-in Magnesium plate ($E = 6.5 \text{ Mpsi}$)** and a **4.5-in Copper plate ($E = 17.2 \text{ Mpsi}$)** are compressed with one bolt and one regular hexagonal nut. The bolt is **3/4 in-16 UNF**. Determine the stiffness of the members k_m .

3. (21 pts) A total of N bolt are used to clamp two member plates. A total external load of 65 kips is applied to the entire joint. The bolts are preloaded to 75 percent of the proof load. Assume the load is equally distributed to all the bolts. The stiffness of each bolt is $k_b = 3 \text{ Mlbf/in}$ and the stiffness of the member is $k_m = 7 \text{ Mlbf/in}$.

Bolt properties: 7/16 in - 14 UNC, $E= 30 \text{ Mpsi}$, Proof strength, $S_p = 120 \text{ kpsi}$.

(3.1) (8 pts) Find the minimum number of bolts required for yielding safety factor of 1.15.

(3.2) (5 pts) With the number of bolts obtained in (1), determine the realized safety factor of joint separation.

(3.3) (8 pts) With the number of bolts obtained in (1), determine the resultant bolt load F_b and resultant load on members applied by one bolt F_m .

4. (18 pts) An **02-series Cylindrical Roller Bearing** is desired a life of 30000 hours at 350 rev/min with a radial load of 15 kN. The catalog rating life L_R is 1×10^6 revolutions.

(4.1) (9 pts) Select the smallest bore size from Table 11-3 that can satisfy this condition.

(4.2) (9 pts) With the bearing selected from (1), estimate the desired bearing life in hours at 300 rev/min if the radial load is changed to 20 kN.

5. (24 pts) (22 pts) An **02-series Deep-Groove ball bearing** with a 40-mm bore (see Tables 11-1 and 11-2 for specifications) has an axial load of 4 kN and a radial load of 7 kN applied. The outer ring rotates at 800 rev/min. The catalog rating life is 1 million revolutions.

(5.1) (16 pts) What is the equivalent radial load F_e ?

(5.2) (8 pts) Estimate the life of bearing in hours?

Table 8-1

Diameters and Areas of
Coarse-Pitch and Fine-
Pitch Metric Threads.*

Nominal Major Diameter d mm	Coarse-Pitch Series			Fine-Pitch Series		
	Pitch p mm	Tensile- Stress Area A_r mm ²	Minor- Diameter Area A_r mm ²	Pitch p mm	Tensile- Stress Area A_r mm ²	Minor- Diameter Area A_r mm ²
1.6	0.35	1.27	1.07			
2	0.40	2.07	1.79			
2.5	0.45	3.39	2.98			
3	0.5	5.03	4.47			
3.5	0.6	6.78	6.00			
4	0.7	8.78	7.75			
5	0.8	14.2	12.7			
6	1	20.1	17.9			
8	1.25	36.6	32.8	1	39.2	36.0
10	1.5	58.0	52.3	1.25	61.2	56.3
12	1.75	84.3	76.3	1.25	92.1	86.0
14	2	115	104	1.5	125	116
16	2	157	144	1.5	167	157
20	2.5	245	225	1.5	272	259

Table 8-2

Diameters and Area of Unified Screw Threads UNC and UNF*

Size Designation	Nominal Major Diameter in	Coarse Series—UNC			Fine Series—UNF		
		Threads per Inch N	Tensile- Stress Area A_r in ²	Minor- Diameter Area A_r in ²	Threads per Inch N	Tensile- Stress Area A_r in ²	Minor- Diameter Area A_r in ²
0	0.0600				80	0.001 80	0.001 51
1	0.0730	64	0.002 63	0.002 18	72	0.002 78	0.002 37
2	0.0860	56	0.003 70	0.003 10	64	0.003 94	0.003 39
3	0.0990	48	0.004 87	0.004 06	56	0.005 23	0.004 51
4	0.1120	40	0.006 04	0.004 96	48	0.006 61	0.005 66
5	0.1250	40	0.007 96	0.006 72	44	0.008 80	0.007 16
6	0.1380	32	0.009 09	0.007 45	40	0.010 15	0.008 74
8	0.1640	32	0.014 0	0.011 96	36	0.014 74	0.012 85
10	0.1900	24	0.017 5	0.014 50	32	0.020 0	0.017 5
12	0.2160	24	0.024 2	0.020 6	28	0.025 8	0.022 6
$\frac{1}{4}$	0.2500	20	0.031 8	0.026 9	28	0.036 4	0.032 6
$\frac{5}{16}$	0.3125	18	0.052 4	0.045 4	24	0.058 0	0.052 4
$\frac{3}{8}$	0.3750	16	0.077 5	0.067 8	24	0.087 8	0.080 9
$\frac{7}{16}$	0.4375	14	0.106 3	0.093 3	20	0.118 7	0.109 0
$\frac{1}{2}$	0.5000	13	0.141 9	0.125 7	20	0.159 9	0.148 6
$\frac{9}{16}$	0.5625	12	0.182	0.162	18	0.203	0.189
$\frac{5}{8}$	0.6250	11	0.226	0.202	18	0.256	0.240
$\frac{3}{4}$	0.7500	10	0.334	0.302	16	0.373	0.351
$\frac{7}{8}$	0.8750	9	0.462	0.419	14	0.509	0.480
1	1.0000	8	0.606	0.551	12	0.663	0.625
$1\frac{1}{4}$	1.2500	7	0.969	0.890	12	1.073	1.024
$1\frac{1}{2}$	1.5000	6	1.405	1.294	12	1.581	1.521

Table A-31

Dimensions of
Hexagonal Nuts

Nominal Size, in	Width <i>W</i>	Height <i>H</i>		
		Regular Hexagonal	Thick or Slotted	JAM
M5	8	4.7	5.1	2.7
M6	10	5.2	5.7	3.2
M8	13	6.8	7.5	4.0
M10	16	8.4	9.3	5.0
M12	18	10.8	12.0	6.0
M14	21	12.8	14.1	7.0
M16	24	14.8	16.4	8.0
M20	30	18.0	20.3	10.0
M24	36	21.5	23.9	12.0
M30	46	25.6	28.6	15.0
M36	55	31.0	34.7	18.0

Table 11-2

Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width, mm	Fillet	Shoulder		Load Ratings, kN			
			Radius, mm	Diameter, mm		Deep Groove		Angular Contact	
				d_s	d_H	C_{10}	C_0	C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5

Table 11-3

Dimensions and Basic Load Ratings for Cylindrical Roller Bearings

02-Series					03-Series			
Bore, mm	OD, mm	Width, mm	Load Rating, kN		OD, mm	Width, mm	Load Rating, kN	
			C_{10}	C_0			C_{10}	C_0
25	52	15	16.8	8.8	62	17	28.6	15.0
30	62	16	22.4	12.0	72	19	36.9	20.0
35	72	17	31.9	17.6	80	21	44.6	27.1
40	80	18	41.8	24.0	90	23	56.1	32.5
45	85	19	44.0	25.5	100	25	72.1	45.4
50	90	20	45.7	27.5	110	27	88.0	52.0
55	100	21	56.1	34.0	120	29	102	67.2
60	110	22	64.4	43.1	130	31	123	76.5
65	120	23	76.5	51.2	140	33	138	85.0
70	125	24	79.2	51.2	150	35	151	102
75	130	25	93.1	63.2	160	37	183	125
80	140	26	106	69.4	170	39	190	125
85	150	28	119	78.3	180	41	212	149
90	160	30	142	100	190	43	242	160
95	170	32	165	112	200	45	264	189
100	180	34	183	125	215	47	303	220
110	200	38	229	167	240	50	391	304
120	215	40	260	183	260	55	457	340
130	230	40	270	193	280	58	539	408
140	250	42	319	240	300	62	682	454
150	270	45	446	260	320	65	781	502

MCEG3304 Machine Design I Final Exam Solution Sheets

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Name:

MCEG 3304 Machine Design I Final Exam Solution.

1. (16) M14x2 bolt, 14R washer.

(1.1) (7)

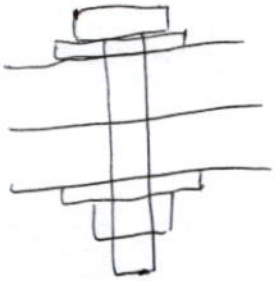
$$l = 2 \times t_w + t_1 + t_2 = 3.5 + 3.5 + 18 + 18 = 43 \text{ mm} \quad (2)$$

$$t_w = 3.5 \text{ mm} \quad (\text{Table A-33}) \quad (1)$$

$$H_{\text{nut}} = 12.8 \text{ mm} \quad (\text{Table A-31}) \quad (1)$$

$$L > l + H = 43 + 12.8 = 55.8 \text{ mm} \quad (2)$$

$$L = 60 \text{ mm} \quad (1)$$



(1.2) (9)

$$L_T = 2d + 6 = 2 \times 14 + 6 = 34 \text{ mm} \quad (1)$$

$$l_d = L - L_T = 60 - 34 = 26 \text{ mm} \quad (1)$$

$$l_t = l - l_d = 43 - 26 = 17 \text{ mm} \quad (1)$$

$$E = 207 \text{ GPa} \quad (1)$$

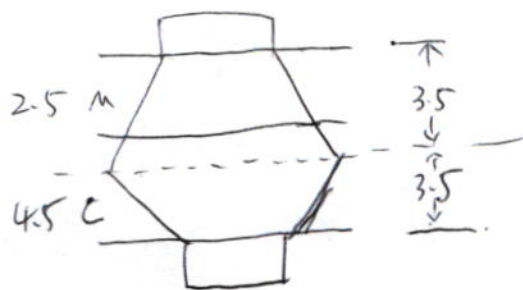
$$A_t = 115 \text{ mm}^2 \quad (\text{Table 8-1}) \quad (1)$$

$$A_d = \frac{\pi d^2}{4} = \frac{\pi \times 14^2}{4} = 153.94 \text{ mm}^2 \quad (1)$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.94 \times 115 \times 207}{153.94 \times 17 + 115 \times 26} \quad (2)$$

$$= 653.57 \times 10^6 \text{ N/m} \quad (2)$$

2. (21) $\frac{3}{4}$ in -16 UNF bolt.



Frustum one: Magnesium

$$E_1 = 6.5 \text{ Mpsi} \quad (1) \quad t_1 = 2.5 \text{ in} \quad (1)$$

$$d = 0.75 \text{ in}$$

$$D_1 = 1.5d = 1.5 \times 0.75 = 1.125 \text{ in} \quad (1)$$

$$k_1 = \frac{0.5774 \times \pi \times 6.5 \times 0.75}{\ln \frac{(1.155 \times 2.5 + 1.125 - 0.75)(1.125 + 0.75)}{(1.155 \times 2.5 + 1.125 + 0.75)(1.125 - 0.75)}} \quad (2)$$

$$= 7.18 \text{ Mlbf/in} \quad (1)$$

Frustum Two: Copper

$$E_2 = 17.2 \text{ Mpsi} \quad (1) \quad t_2 = 3.5 - 2.5 = 1 \text{ in} \quad (1) \quad d = 0.75 \text{ in}$$

$$D_2 = D_1 + 2 \times t_1 \times \tan 30^\circ = 1.125 + 2 \times 2.5 \times \tan 30^\circ = 4.01 \text{ in} \quad (2)$$

$$k_2 = \frac{0.5774 \times \pi \times 17.2 \times 0.75}{\ln \frac{(1.155 \times 1 + 4.01 - 0.75)(4.01 + 0.75)}{(1.155 \times 1 + 4.01 + 0.75)(4.01 - 0.75)}} \quad (2)$$

(1)

$$\downarrow 1.089846$$

Frustum 3: Copper

$$E_3 = 17.2 \text{ Mpsi} \quad (1) \quad t_3 = 3.5 \text{ in} \quad (1) \quad d = 0.75 \text{ in} \quad D_3 = D_1 = 1.125 \text{ in} \quad (1)$$

$$k_3 = \frac{0.5774 \times \pi \times 17.2 \times 0.75}{\ln \frac{(1.155 \times 3.5 + 1.125 - 0.75)(1.125 + 0.75)}{(1.155 \times 3.5 + 1.125 + 0.75)(1.125 - 0.75)}} \quad (2)$$

(1)

$$\downarrow 3.8$$

$$k_m = \left(\frac{1}{7.18} + \frac{1}{271.98} + \frac{1}{17.53} \right)^{-1} = 5 \text{ Mlbf/in} \quad (1)$$

(1)

3. $P_{total} = 65 \text{ kip}$ $F_i = 0.75 F_p$ $k_b = 3 \text{ Mlb/in}$ $k_m = 7 \text{ Mlb/in}$
 bolt $7/16 \text{ in} - 14 \text{ UNC}$ $E = 30 \text{ Mpsi}$ $S_p = 120 \text{ kpsi}$

(1) $n_p = 1.5$

(8) $\frac{S_p A_t}{C P + F_i} = 1.5$ (2)

$C = \frac{k_b}{k_b + k_m} = \frac{3}{3+7} = 0.3$ (1) $A_t = 0.1063$ (Table 8-2) (1)

$F_i = 0.75 \times S_p A_t = 0.75 \times 120 \times 0.1063 = 9.567 \text{ kip}$ (2)

$\frac{120 \times 0.1063}{0.3 \times P + 9.567} = 1.5 \Rightarrow P = \left(\frac{120 \times 0.1063}{1.5} - 9.567 \right) \div 0.3$
 $= 5.084 \text{ kip}$ (1)

$N = \frac{P_{total}}{P} = \frac{65}{5.084} = 12.78$ $N = 13$ (1)

(2) $P = \frac{P_{total}}{N} = \frac{65}{13} = 5 \text{ kip}$ (2)

(5) $n_o = \frac{F_i}{P(1-C)} = \frac{9.567}{5 \times (1-0.3)} = 2.73$ (1)

(3) $F_b = C P + F_i = 0.3 \times 5 + 9.567 = 11.067 \text{ kip}$ (1)

(8) $F_m = (1-C) P - F_i = (1-0.3) \times 5 - 9.567 = -6.067 \text{ kip}$ (1)

4. $n_D = 350 \text{ rev/min}$ $L_D = 30000 \text{ hr.}$ $F_D = 15 \text{ kN.}$

(18) $L_R = 1 \times 10^6 \text{ rev}$ roller bearing $\alpha = ~~7/10~~ 10/3$

(4.1) $C_{10} = L_R^{1/\alpha} = F_D \cdot (n_D \cdot 60 \cdot L_D)^{0.3}$ (3)

(9) $C_{10} = \frac{15 \times (60 \times 30000 \times 350)^{0.3}}{(1 \times 10^6)^{0.3}}$ (2)

$= 103.73 \text{ kN}$ (2)

From Table 11-3, select 80-mm bore roller bearing ($C_{10} = 106 \text{ kN}$) (2)

(4.2) $C_{10} = 106 \text{ kN}$ $n_D = 300 \text{ rev/min}$ $F_D = 20 \text{ kN.}$

(9) $106 \times (1 \times 10^6)^{0.3} = 20 \times (300 \times 60 \times L_D)^{0.3}$ (3)

$L_D = \frac{106^{10/3} \times 1 \times 10^6}{20^{10/3} \times 300 \times 60} = 14,420.5 \text{ hrs}$ (2)

(3)

5. $F_a = 4 \text{ kN}$ $F_r = 7 \text{ kN}$ $V = 1.2$ ①
 (24) $n_D = 800 \text{ rev/min}$ $L_R = 1 \times 10^6 \text{ rev.}$ 40-mm ~~roller~~ ball bearing.

(5.1) $C_0 = 16.6 \text{ kN}$ (Table 11-2) ①

(16) $F_a/C_0 = 4/16.6 = 0.241$ ①

F_a/C_0	e	$\frac{e-0.34}{0.38-0.34} = \frac{0.241-0.17}{0.28-0.17}$	③
0.17	0.34		
0.241	e	$e = 0.366$	
0.28	0.38		

$F_a/VF_r = 4/1.2 \times 7 = 0.476 > e$ ②
 ①

$X_2 = 0.56$

F_a/C_0	Y_2	$\frac{Y_2-1.31}{1.15-1.31} = \frac{0.241-0.17}{0.28-0.17}$	③
0.17	1.31		
0.241	Y_2	$Y_2 = 1.207$	
0.28	1.15		

$F_e = X_2 V F_r + Y_2 F_a = 0.56 \times 1.2 \times 7 + 1.207 \times 4 = 9.532 \text{ kN}$ ③ ①

(5.2) $C_{10} \cdot L_R^{1/3} = F_e (60 \cdot L_0 \cdot n_D)^{1/3}$ $C_{10} = 30.7 \text{ kN}$ ①
 (8) $L_0 = \frac{C_{10}^3 \cdot L_R}{F_e^3 \times 60 \times n_D} = \frac{30.7^3 \times (1 \times 10^6)}{9.532^3 \times 60 \times 800}$ ④ ②

$= 696 \text{ hrs.}$ ①

ABOVE AVERAGE

MCEG3304 Machine Design I Final Exam Solution Sheets

12/3/2021

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94

Name: Joseph Steele

① M14x2 14R washer

$h=12.8$

① 1.1
16

$$l = t_1 + t_2 + 2w \quad \text{washer thickness} = 3.5$$

$$l = 18 + 18 + 2 \times (3.5) = 43 \checkmark$$

$$43 + 12.8 \checkmark$$

$$L > 60 \text{ mm} \checkmark$$

1.2

$$l_b LT = 2d + 6 \text{ mm}$$

$$2(14) + 6 = 34 \text{ mm} \checkmark$$

$$l_d = L - LT = 26 \text{ mm} \checkmark$$

$$l_t = l - l_d = 43 - 26 = 17 \text{ mm} \checkmark$$

$$A_t = 115 \text{ mm}^2$$

$$A_d = \frac{\pi d^2}{4} = \frac{\pi (14)^2}{4} = 153.93 \text{ mm}^2 \checkmark$$

$$l_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.93 \times 115 \times 207}{(153.93 \times 17) + (115 \times 26)}$$

$$653.545 \text{ MN/m} \checkmark$$

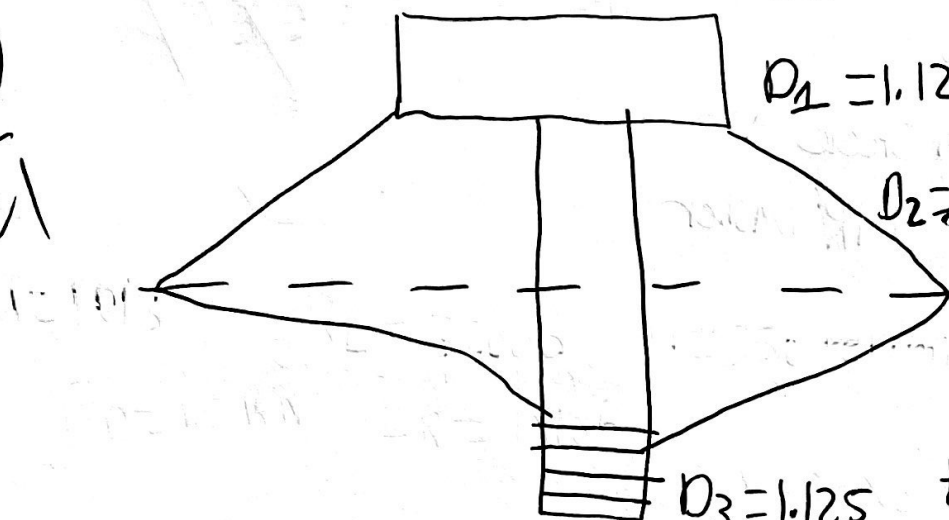
② 2.5 in Mg 4.5 in Cu 3/4 in-16 UNF

$$l = 2.5 + 4.5 = 7$$

$$D = 1.5 \times (0.75) = 1.125$$

$$\frac{7}{2} = 3.5$$

(19)
21



$$D_1 = 1.125 \quad t_1 = 3.5 \text{ in}$$

$$D_2 = 1.125 + 2(3.5 - 2.5 \tan 30^\circ) = 6.3211$$

$$l = 7$$

$$t_2 = 3.5 - 2.5 = 1$$

$$D_3 = 1.125 \quad t_3 = 4.5 \text{ in}$$

$$k = \frac{0.5774 \pi E d}{l_n \frac{(1.155 t + D - d)(D + d)}{(1.155 t + D + d)(D - d)}}$$

$$\left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = k_m$$

$$k_1 = \frac{(0.5774 \pi \times 6.5 \times 0.75)}{l_n \frac{(1.155 \times 3.5 + 1.125 - 0.75)(1.125 + 0.75)}{(1.155 \times 3.5 + 1.125 + 0.75)(1.125 - 0.75)}}$$

$$= 6.7140 \text{ Mlb/in}$$

$$k_2 = \frac{(0.5774 \pi \times 6.5 \times 0.75)}{l_n \frac{(1.155 \times 1 + 6.3211 - 0.75)(6.3211 + 0.75)}{(1.155 \times 1 + 6.3211 + 0.75)(6.3211 - 0.75)}}$$

$$= 238.311 \text{ Mlb/in}$$

$$k_3 = \frac{(0.5774 \pi \times 17.2 \times 0.75)}{l_n \frac{(1.155 \times 4.5 + 1.125 - 0.75)(1.125 + 0.75)}{(1.155 \times 4.5 + 1.125 + 0.75)(1.125 - 0.75)}}$$

$$= 17.067 \text{ Mlb/in}$$

$$k_m = \left(\frac{1}{6.7140} + \frac{1}{238.311} + \frac{1}{17.067} \right)^{-1} = 4.722 \text{ Mlb/in}$$

3.

$$P_{total} = 65 \text{ kips} \quad l_b = 3 \quad l_m = 7$$

$$7/16 \text{ in } - 14 \text{ UNC} \quad S_p = 120 \quad E = 30$$

$$A_t = 0.1063 \text{ in}^2$$

21

3.1

$$F_i = 0.75 \times 120 \times 0.1063$$

$$F_i = 0.75 \times A_t \times S_p$$

$$= 9.567$$

$$C = \frac{l_b}{l_b + l_m} = \frac{3}{3 + 7} = 0.3 \text{ Mbf/in}$$

Yielding safety factor

$$n_p = \frac{S_p A_t}{C P + F_i}$$

$$n_p = 1.15 \quad 1.15 = \frac{120 \times 0.1063}{0.3 \times 65 + 9.567}$$

$$F_i = \frac{1.1 S_p A_t C P}{n_p S_p A_t C P - C P}$$

13 bolts

$$N = 12.785$$

$$N = 13 \text{ bolts}$$

3.2

$$n_o = \frac{F_i}{P(1-C)}$$

$$N = \frac{65}{1.5} = 43.33$$

$$\frac{65}{13} = 5 = P$$

$$= \frac{9.567}{1.285(1-0.3)} = 2.733 \text{ safe}$$

3.3

$$P_b = \frac{l_b P}{l_b + l_m} = \frac{3 \times 5285}{3 + 7} = 1.5$$

$$P_b + F_i = F_b$$

$$1.5 + 9.567 = 11.067$$

$$P_m = (1-c)P = (1-0.3) \times 5 = 3.5$$

$$F_m = P_m - F_i$$

$$3.5 - 5 = -1.5$$

4.18
18
4.1

$$a = 10/3 \quad R_L = 30000 \quad n = 350 \text{ rev/min}$$

$$F_D = 15 \text{ kN} \quad L_R = 1 \times 10^6$$

$$C_{10} = 15 \text{ kN} \left(\frac{30000 \times 350 \times 60}{10^6} \right)^{1/3.333} = 103.747$$

80 mm bore is the smallest bore size

4.2

80 mm bore $n = 300 \text{ rev/min} \quad F_D = 20 \text{ kN}$

$$C_{10} = 106$$

$$106 = 20 \text{ kN} \left(\frac{R_L \times 300 \times 60}{10^6} \right)^{1/3.333}$$

$$5.3 = \left(\frac{R_L \times 300 \times 60}{10^6} \right)$$

$$259.4247 = R_L \times 0.018$$

$$R_L = 14386.92778 \text{ hrs}$$

⑤

② 4
40 mm bore

$n = 800 \text{ rev/min}$

$$4 \text{ kN} = F_a = F_r = 7 \text{ kN}$$

S.1

$$\frac{F_a}{V} \cdot e = \frac{4}{24} = 0.166 \quad \text{between } 0.110 \text{ and } 0.17$$

$$C_{10} = 41.8 \quad C_0 = 24.0 \times -1$$

$$V = 1.2$$

$$X_3 = 0.17$$

$$X_2 = 0.166$$

$$X_1 = 0.110$$

$$Y_3 = 0.34$$

$$Y_1 = 0.30$$

$$Y_2 = \frac{(X_2 - X_1)(Y_3 - Y_1)}{(X_3 - X_1)} + Y_1$$

$$e = 0.337$$

$$F_a / V F_r$$

$$\frac{4}{1.2(7)} = 0.47617 e \checkmark$$

$$F_e = X_i V F_r + Y_i F_a$$

$$(0.56 \times 1.2 \times 7 + 1.23 \times 4) = \boxed{9.624} \checkmark$$

S.2

$$C_{10} = 41.8$$

$$a = 3$$

$$41.8 = 9.624 \left(\frac{RL \times 800 \times 60}{10^6} \right)^{1/3}$$

$$4.343 = (RL \times 0.048)^{1/3}$$

$$81.916 = RL \times 0.048$$

$$\boxed{1706.58 \text{ hrs}} \checkmark$$

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Name:

Shawn Houston

① (14) // 6

- M14 x 2
- 14 R > 3.5 A-33
- 18 mm steel
- 18 mm steel



①.1

$$L = 18 + 18 + 3.5 = 39.5 \text{ mm} = L \quad \checkmark$$

$$L > L + t \Rightarrow 39.5 + 12.8 = 52.3 \text{ mm} = L \quad \checkmark$$

$$L_T = 2d + L \Rightarrow 2(14) + L = 34 \text{ mm} = L_T \quad \checkmark$$

$$d = 14 \text{ mm} \quad \checkmark$$

$$L_d = L - L_T = 52.3 - 34 \Rightarrow L_d = 18.3 \text{ mm} \quad \checkmark$$

$$L_t = L - L_d \Rightarrow 39.5 - 18.3 \Rightarrow L_t = 21.2 \text{ mm} \quad \checkmark$$

①.2

$$A_d = \frac{\pi d^2}{4} \Rightarrow \frac{\pi (14)^2}{4} \Rightarrow A_d = 153.94 \text{ mm}^2 \quad \checkmark$$

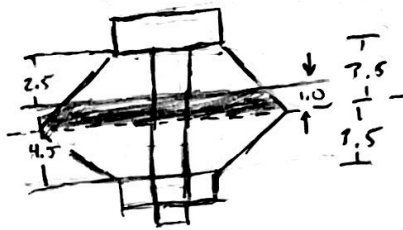
$$A_t = 115 \text{ mm}^2 \quad \checkmark \text{ table 8-1}$$

$$E = 207 \text{ GPa} \quad \checkmark$$

$$k_b = \frac{A_d A_t E}{A_d L_t + A_t L_d} \Rightarrow \frac{(153.94)(115)(207)}{(153.94)(21.2) + (115)(18.3)}$$

$$k_b = 682.46 \times 10^6 \text{ N/m} \quad \checkmark$$

② $t_1 = 2.5 \text{ in}$
 $E = 6.5 \text{ Mrsi}$ } Magnesium
 $t_3 = 4.5 \text{ in}$
 $E = 17.2 \text{ Mrsi}$ } Copper
 $3/4 \text{ in} - 14 \text{ UHF}$



$K_m = ?$

$d = 3/4 \text{ in}$

$D_1 = 1.5(3/4) = 1.125 \text{ in} = D_3 \checkmark$

$K_1 = \frac{0.5774 \pi (6.5) (3/4)}{\ln \left(\frac{(1.155(2.5) + (1.125) - (3/4))(1.125 + (3/4))}{(1.155(2.5) + (1.125) + (3/4))(1.125 - (3/4))} \right)}$

$K_1 = 7.183 \text{ mlb/in} \checkmark$

$D_2 = D_3 + 2t_3 \tan(30^\circ) \Rightarrow (1.125) + 2(4.5) \tan(30^\circ)$
 $t_2 = 1.0 \text{ in} \checkmark$

$K_2 = 6.32 \text{ mlb/in}$

$K_3 = \frac{0.5774 \pi (17.2) (3/4)}{\ln \left(\frac{(1.155(4.5) + (1.125) - (3/4))(1.125 + 3/4)}{(1.155(4.5) + (1.125) + (3/4))(1.125 - 3/4)} \right)}$

$K_3 = 17.07 \text{ mlb/in} \checkmark$

$\therefore \frac{1}{K_m} = \frac{1}{7.183} + \frac{1}{6.32} + \frac{1}{17.07}$

$K_m \cdot \frac{1}{K_m} = 0.356 \text{ in}$

$l = 0.356 \text{ km} \checkmark$

$K_m = 2.809 \text{ mlb/in}$

③ $P_{TOTAL} = 65 \text{ kips}$

75%

$k_b = 3 \text{ M lb/in}$

$k_m = 7 \text{ M lb/in}$

$H_{LR} = 14 \text{ UNC}$

$E = 30 \text{ Mpsi}$

$S_p = 120 \text{ ksi}$

$A_t = 0.1063 \text{ in}^2 \quad \star 8-2$

③.1

$n_p = \frac{S_p A_t}{C P + F_i} \quad \star n_p = 1.15$

$C = \frac{k_b}{k_b + k_m} \Rightarrow \frac{3}{3+7} \Rightarrow C = 0.3 \quad \checkmark$

$F_i = 0.75 F_p \Rightarrow 0.75 (12.756) \Rightarrow F_i = 9.567 \quad \checkmark$

$F_p = A_t S_p \Rightarrow (0.1063)(120) \Rightarrow F_p = 12.756 \quad \checkmark$

$1.15 = \frac{(120)(0.1063)}{(0.3)P + (9.567)} \Rightarrow 1.15(0.3P + 9.567) = (120)(0.1063)$

$0.345P + 11.00205 = 12.756$

$0.345P = 1.75395$

$P = 5.084 \text{ kips} \quad \checkmark$

$P = \frac{P_{TOTAL}}{N}$

$N = \frac{P_{TOTAL}}{P} \Rightarrow \frac{65}{(5.084)} \quad \checkmark$

$N = 12.79 \approx \boxed{13 = N} \quad \checkmark$

③.2

$n_o = \frac{F_i}{P(1-C)}$

$P = \frac{65}{13} = 5 = P \quad \checkmark$

$n_o = \frac{9.567}{5(1-0.3)} \Rightarrow \boxed{n_o = 2.733} \quad \checkmark$

③.3

$P_b = \frac{k_b P}{k_b + k_m} \Rightarrow \frac{(3)(5)}{(3) + (7)} = 1.5 = P_b$

$F_b = P_b + F_i \Rightarrow 1.5 + 9.567 \Rightarrow F_b = 11.067 \text{ M lb/in} \quad \checkmark$

$F_m = (1-C)P - F_i \Rightarrow (1-0.3)5 - (9.567) \Rightarrow F_m = -6.067 \quad \checkmark$

12
18

④

$$L_D = 30000 \text{ hrs}$$

$$n_D = 350 \text{ rev/min} = n_R$$

$$F_D = 15 \text{ kN}$$

$$L_R = 1 \times 10^6 \text{ rev}$$

Roller bearing: $a = 10/3$

-2

④.1

$$C_{10} = F_D \left(\frac{L_D}{L_R} \right)^{1/a} \Rightarrow (15) \left(\frac{30000}{1 \times 10^6} \right)^{1/(10/3)}$$

$$C_{10} = F_D \left(\frac{L_D n_D 60}{L_R n_R 60} \right)^{1/a}$$

$$C_{10} = 2.624 \text{ kN} > 16.8 \text{ kN}$$

$$C_{10} = 5.239 \text{ kN} > 16.8 \text{ kN}$$

$$\boxed{\text{Bore} = 25 \text{ mm}}$$

X — |

same

④.2

$$16.8 = (20) \left(\frac{L_1 (200) 60}{(1 \times 10^6) (200) 60} \right)^{1/(10/3)}$$

$$16.8 = 0.317 L_1^{3/10}$$

$$L_1^{3/10} = 54.194$$

$$\boxed{L_1 = 602333.97 \text{ hours}}$$

— |

⑤

bore = 40 mm

$F_r = 4 \text{ kN}$

$F_a = 7 \text{ kN}$

$n_k = 800 \text{ rev/min}$

$C_{10} = 1 \times 10^6$

$C_{10} = 30.7 \checkmark$

$C_0 = 14.6 \checkmark$

$V = 1.2 \checkmark$

$a = 3$

⑤.1

$F_a/C_0 = ?^{-1}$
 $e = ? \rightarrow 3$

$\frac{F_a}{V F_r} = \frac{4}{1.2(7)} = 0.476 \Delta e$
 $\checkmark$
 $i = 2$

$F_e = X_i V F_r + Y_i F_a$

$= 0.56(1.2)(7) + (1)(4)$

$F_e = 8.704 \text{ kN}$

$Y_2 = ? \rightarrow 3$

⑤.2

1200000

$1 \times 10^6 = F_0 \left(\frac{L_0}{L_e} \right)^{\frac{1}{3}}$

X

-8

BELOW AVERAGE

MCEG3304 Machine Design I Final Exam Solution Sheets

12/3/2021

Total: 100 pts

58

Name: Joel Lewis

(16)
16

1.) M14 x 2 hex head, 14R washer, 18mm steel plates

$$l = (18 \cdot 2) + (3.5 \cdot 2) = 43 \checkmark$$

(1.1) $H = 12.8 \checkmark$

$$L > l + H = 43 + 12.8 = 55.8 \approx 60 \text{ mm} \checkmark$$

(1.2) $l_d = L - L_T = 26 \checkmark$ $L_T = 2d + 6 = 34 \checkmark$

$$l_t = l - l_d = 17 \checkmark$$

$$A_d = \frac{\pi d^2}{4} = 153.94 \checkmark$$

$$A_t = 115 \text{ mm}^2 \checkmark$$

$$d = 14 \text{ mm}$$

$$E = 207 \text{ GPa}$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.94 \cdot 115 \cdot 207}{(153.94 \cdot 17) + (115 \cdot 26)}$$

$$k_b = 653.57 \text{ MN/m} \checkmark$$

2.) 2.5 in, magnesium plate $E = 6.5 \text{ mpsi}$

4.5 in, copper plate $E = 17.2 \text{ mpsi}$, $\frac{3}{4} \text{ in-16 UNF}$



$$4.12 \times 10^6$$

$$k_1 =$$

$$E = 17.2 \checkmark$$

$$d = \frac{3}{4} \checkmark$$

$$l_1 = 1.5 \cdot d = 1.125 \checkmark$$

$$t_1 = \frac{3}{4} \checkmark$$

$$k_1 = \frac{0.5774 \pi E d}{\ln \left(\frac{(1.155 + \frac{3}{4}) + (1.125 - \frac{3}{4})(1.125 + \frac{3}{4})}{(1.155 - \frac{3}{4}) + (1.125 + \frac{3}{4})(1.125 - \frac{3}{4})} \right)}$$

$$14.88 \times 10^6 = k_2 =$$

$$E = 17.2 \checkmark$$

$$d = \frac{3}{4} \checkmark$$

$$l_2 = 1.5 \cdot d = 1.125 \checkmark$$

$$t_2 = \frac{3}{4} \checkmark$$

$$k_2 = \frac{0.5774 \pi E d}{\ln \left(\frac{(1.155 + 2.75) + (1.589 - \frac{3}{4})(1.589 + \frac{3}{4})}{(1.155 - 2.75) + (1.589 + \frac{3}{4})(1.589 - \frac{3}{4})} \right)}$$

$$k = \frac{0.5774 \pi E d}{\ln \left(\frac{(1.155 + D - d)(D + d)}{(1.155 + D + d)(D - d)} \right)}$$

$$l = 2.5 + 4.5 = 7$$

$$l_2 = 3.5$$

$$d = \frac{3}{4}$$

$$3.44 \times 10^6 = k_3 =$$

$$E = 6.5 \checkmark$$

$$d = \frac{3}{4} \checkmark$$

$$l_3 = 1.5 \cdot d = 1.125 \checkmark$$

$$t_3 = \frac{3}{4} \checkmark$$

$$k_3 = \frac{0.5774 \pi E d}{\ln \left(\frac{(1.155 + 2.75) + (1.125 - \frac{3}{4})(1.125 + \frac{3}{4})}{(1.155 - 2.75) + (1.125 + \frac{3}{4})(1.125 - \frac{3}{4})} \right)}$$

$$k_m = (k_1 + k_2 + k_3)^{-1}$$

$$k_m = (27.44)^{-1}$$

$$k_m = 0.036$$

3.) 3.)

Given: $p = 65 \text{ mpsi}$

$$A_t = 0.1063 \quad \checkmark$$

$$F_i = 0.75 S_p \cdot A_t = 0.75 \cdot 120 \cdot 0.1063$$

$$F_i = 9.567 \quad \checkmark$$

$$k_b = 3 \text{ Mlb/in}$$

$$k_m = 7 \text{ Mlb/in}$$

$$C = \frac{k_b}{k_b + k_m} = \frac{3}{10} \quad \checkmark$$

$$\text{Bolt} = \frac{7}{16} \text{ in} - 14 \text{ UNC}$$

$$S_p = 120$$

$$E = 30$$

(3.1)

$$n_p = \frac{S_p A_t}{C \cdot p + F_i} \quad N = \frac{\tau \cdot p}{S_p \cdot A_t - F_i} \quad n_p^0 = \frac{\frac{3}{10} \cdot 65}{120 \cdot 0.1063 - 9.567} \approx 6$$

(3.2)

$$N = \frac{(1-C)p}{F_i} = \frac{(1-\frac{3}{10})65}{9.567} \approx 5$$

$$n_0 = \frac{(1-C)p}{F_i} - 1$$

(3.3)

$$F_b = \tau \cdot p \cdot F_i = (\frac{3}{10} \cdot 65) + 9.567 = 29.067$$

$$F_m = (1-\tau)p \cdot F_i = (1-\frac{3}{10})65 - 9.567 = 35.933$$

4.) O2-series, 3000 hrs @ 350 rev/min, $w = 15 \text{ kW}$, $L_R = 10^6$

6) given: $L_D = 30,000$

$$N = 350 \text{ rev/min} \quad F_D = \frac{60 L_D \cdot N}{L_R} = 630$$

$$w = 15 \text{ kW}$$

$$L_R = 1 \times 10^6$$

$$\alpha = \frac{10}{3}$$

(4.1)

$$C_{10} = \left(\frac{L_D}{L_R} \right)^{\frac{1}{\alpha}} = \left(\frac{30,000}{1 \times 10^6} \right)^{\frac{1}{10/3}} = 220$$

C_{10} Bore ± 130

(4.2)

$$w = 20$$

$$N = 350$$

5.) Given: 40 mm bar

5/24

$$W = 4 \text{ kN}$$

$$W = 7 \text{ kN}$$

$$N = 800$$

$$L_R = 10^6$$

$$F_D = \frac{WLD \cdot N}{L_R} =$$

F_R

$$F_a / C_o = ? -1$$

$$e = ? -3$$

(5.1)

$$F_e = X_1 V F_R + Y_1 F_a$$

$$O_D = 90$$

$$C_{10} = 30.7$$

$$C_0 = 16.6$$

$$V = 1.2$$

$$\frac{F_a}{V F_R} ? -2$$

$$F_e = 1 \cdot 1.2 \cdot 4 + (0 + 0.014) = 13.2$$

X
-2

$$Y_2 = ? -3$$

(5.2)

$$L_D = 7000 \text{ hrs}$$

X
-8

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 1

TOPICS COVERED IN QUIZ 1

1. Free-body Diagram

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME FOR QUIZ 1**

MCEG3304: Machine Design I

Fall 2021

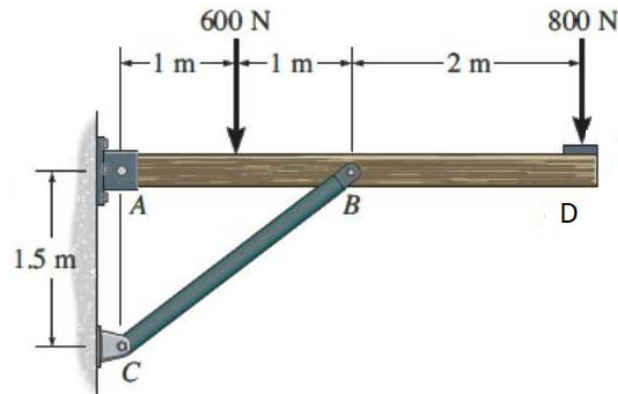
Quiz #1

Total points: 30

Date: 9/6/2021

Name:

1. Sketch the free-body diagram of element AD and BC in the figure. Compute the magnitude and direction of each force on AD and BC.



MCEG3304: Machine Design I

Fall 2021

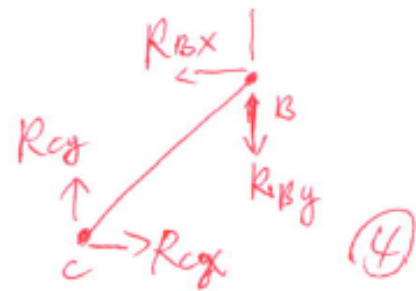
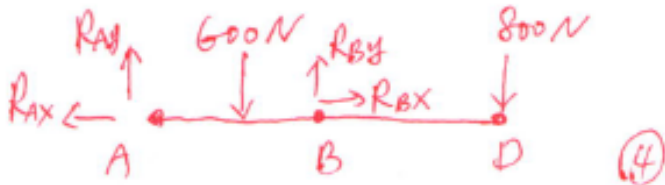
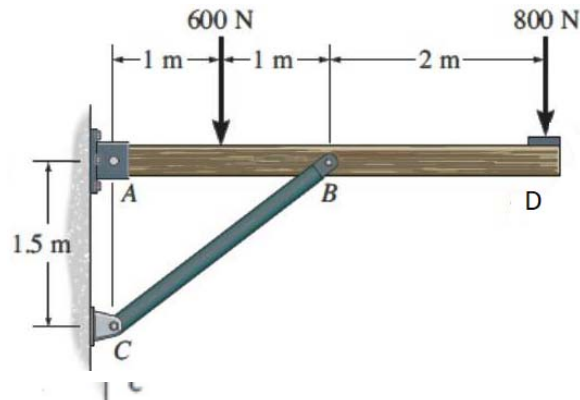
Quiz #1

Total points: 30

Date: 9/6/2021

Name:

1. Sketch the free-body diagram of element AD and BC in the figure. Compute the magnitude and direction of each force on AD and BC.



$$\sum M_A = 0$$

$$-600 \times 1 - 800 \times 4 + R_{By} \times 2 = 0$$

$$R_{By} = 1900 \text{ N} \quad (5)$$

$$R_{Ax} = R_{Bx} = 2533.3 \text{ N} \quad (3)$$

$$\sum F_y = 0$$

$$R_{Ay} + R_{By} - 600 - 800 = 0$$

$$R_{Ay} = -500 \text{ N} \quad (4)$$

$$\sum M_C = 0$$

$$-R_{By} \times 2 + R_{Bx} \times 1.5 = 0$$

$$R_{Bx} = 2533.3 \text{ N} \quad (4)$$

$$R_{Cy} = R_{By} = 1900 \quad (3)$$

$$R_{Cx} = R_{Bx} = 2533.3 \text{ N} \quad (3)$$

ABOVE AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #1

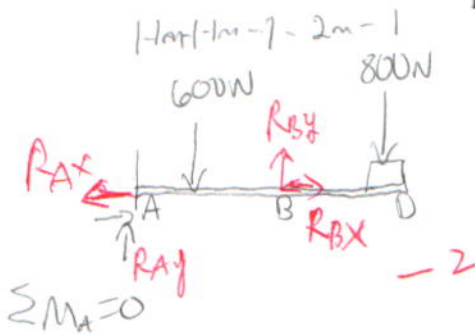
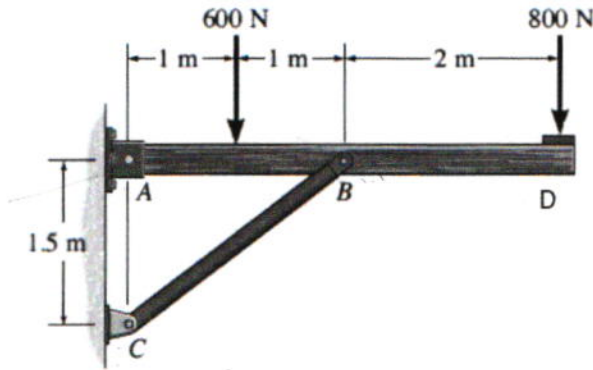
Total points: 30

Date: 9/6/2021

Name: Dwayne Smith

20/30

1. Sketch the free-body diagram of element AD and BC in the figure. Compute the magnitude and direction of each force on AD and BC.



$$-600(1) + B(2) - 800(4) = 0$$

$$B_y = 1400 \text{ N } \uparrow$$

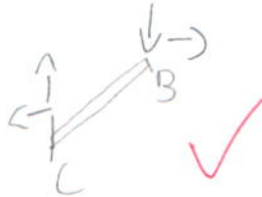
$$\Sigma F_y = 0$$

$$1400 - 600 - 800 + A_y = 0$$

$$A_y = 500 \text{ N } \downarrow$$

$$R_{Ax} = ?$$

-3



$$C_y = B_y$$

$$C_y = 1400 \text{ N } \checkmark$$

$$\Sigma F_x = 0$$

$$C_x - 1400(1.5) = 0$$

$$C_x = 2850 \text{ N } \times -2$$

$$R_{Bx} = ?$$

-3

AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #1

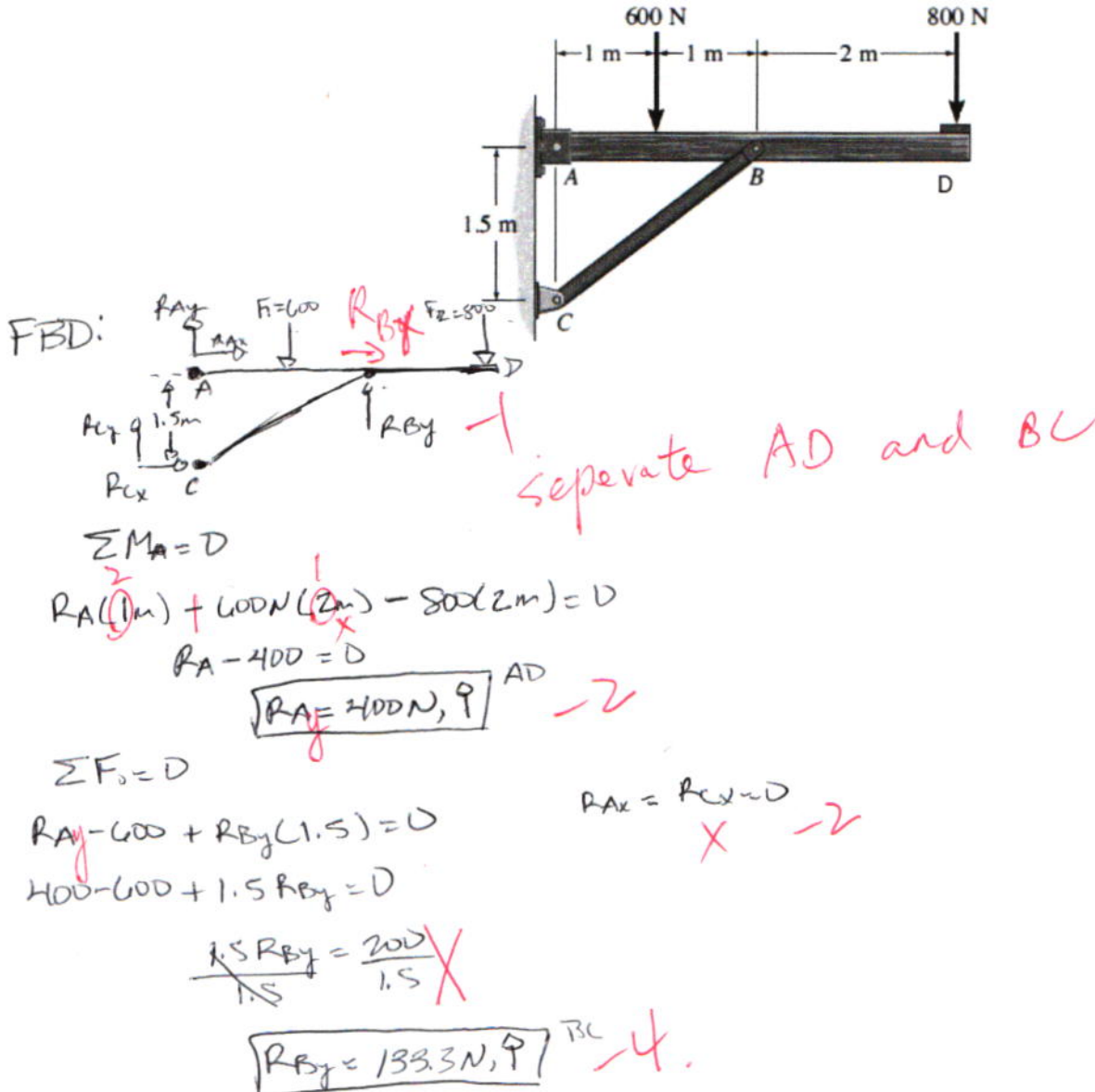
Total points: 30

Date: 9/6/2021

Name: Garrett Warren

11/30

1. Sketch the free-body diagram of element AD and BC in the figure. Compute the magnitude and direction of each force on AD and BC.



BELOW AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #1

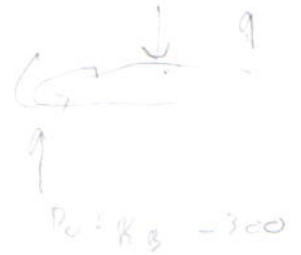
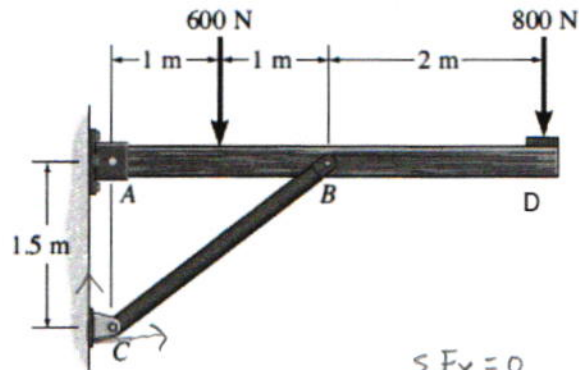
Total points: 30

Date: 9/6/2021

Name: Shumirac

Score: 8/30

1. Sketch the free-body diagram of element AD and BC in the figure. Compute the magnitude and direction of each force on AD and BC.

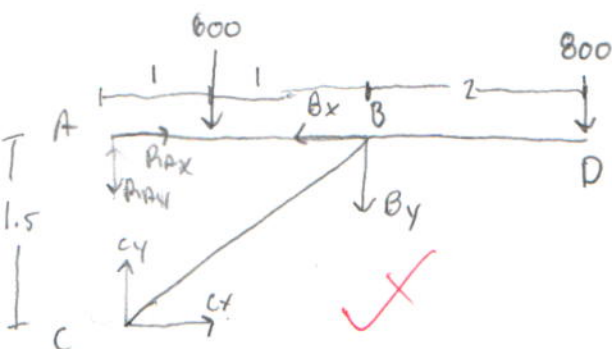
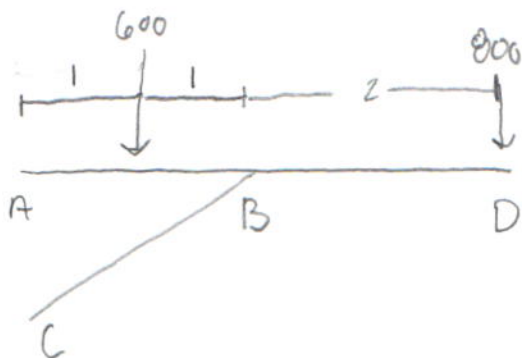


$$\sum F_y = 0$$

$$\sum F_y = 0$$

$$-600(1) = 800(4)$$

$$= -3800 \text{ on AD}$$



$$\sum M_A = 0$$

Separate AD and BC.

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 2

TOPICS COVERED IN QUIZ 2

2. Shear Force and Bending Moment Diagram

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME FOR QUIZ 1**

MCEG3304: Machine Design I

Fall 2021

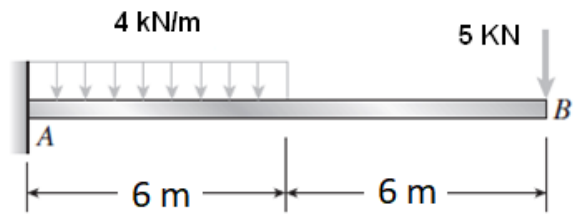
Quiz #2

Total points: 30

Date: 9/13/2021

Name:

1. For the cantilever beam shown
 - (1) Draw free-body diagram of AB and find the reactions at A
 - (2) Draw the shear-force diagram
 - (3) Draw the bending moment diagram



MCEG3043: Machine Design I

Fall 2021

Quiz #2

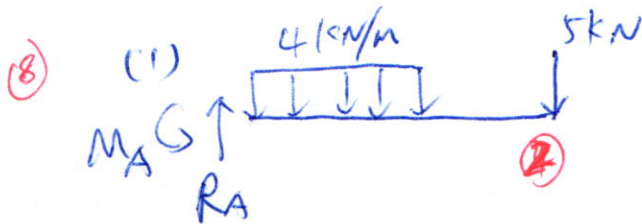
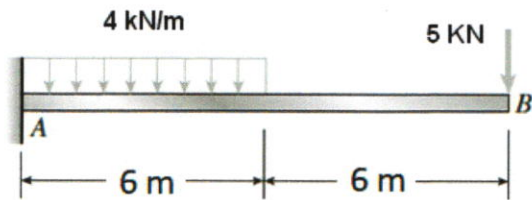
Total points: 30

Name:

Date: 9/13/2021

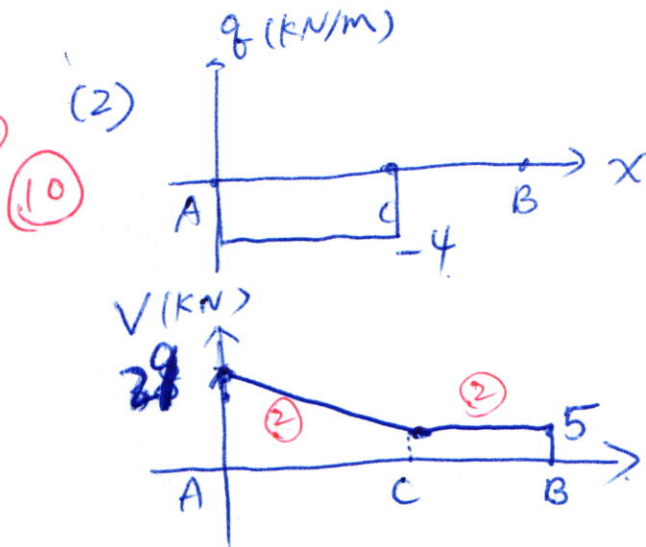
1. For the cantilever beam shown

- (1) Draw free-body diagram of AB and find the reactions at A
- (2) Draw the shear-force diagram
- (3) Draw the bending moment diagram



$$\sum F_y = 0 \quad R_A = 4 \times 6 + 5 = 29 \text{ kN} \quad (2)$$

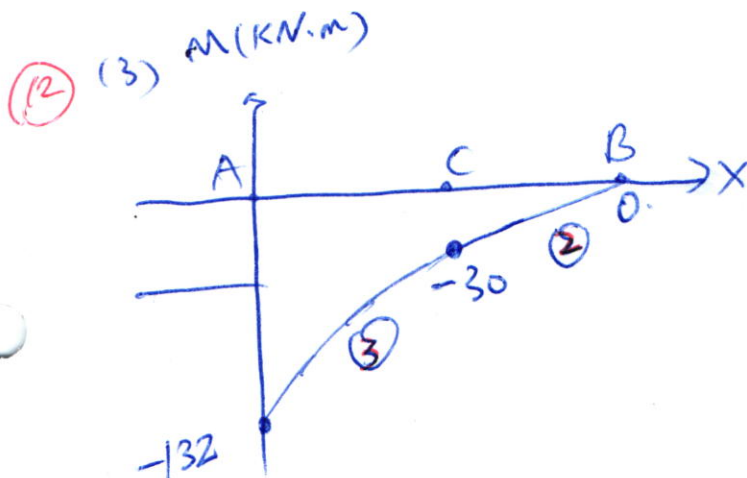
$$\sum M_A = 0 \quad M_A = 4 \times 6 \times 3 + 5 \times 12 = 132 \text{ kN}\cdot\text{m} \quad (4)$$



$$V_A = 29 \text{ kN} \quad (2)$$

$$V_C - V_A = -4 \times 6 \Rightarrow V_C = 5 \text{ kN} \quad (2)$$

$$V_B = 5 \text{ kN} \quad (2)$$



$$M_A = -132 \text{ kN}\cdot\text{m} \quad (2)$$

$$M_C - M_A = \frac{1}{2} \times 24 \times 6 + 5 \times 6$$

$$M_C = -30 \text{ kN}\cdot\text{m} \quad (3)$$

$$M_B = 0 \quad (2)$$

ABOVE AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #2

Total points: 30

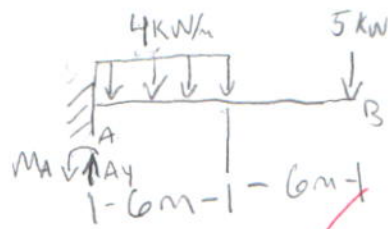
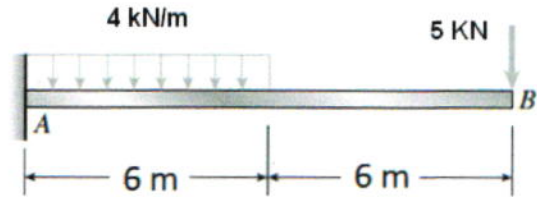
Date: 9/13/2021

Name: DWayne Smith

26/130

1. For the cantilever beam shown

- (1) Draw free-body diagram of AB and find the reactions at A
- (2) Draw the shear-force diagram
- (3) Draw the bending moment diagram



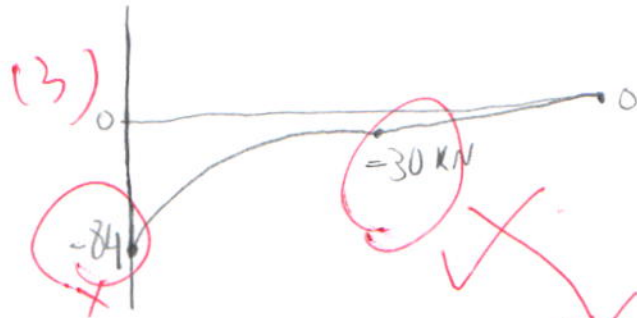
$$M_A = -(4 \times 6) \times 3 - 5 \times 12$$

$$M_A = -84 \text{ kN} \cdot \text{m} - 60 \text{ kN} \cdot \text{m} = -144 \text{ kN} \cdot \text{m}$$

$$\sum F_y = 0$$

$$A_y - (4 \times 6) - 5 = 0$$

$$A_y = 29 \text{ kN}$$



$$-132$$

why?

-3

AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #2

Total points: 30

Date: 9/13/2021

Name: Jun Newton

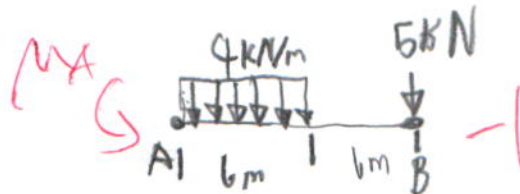
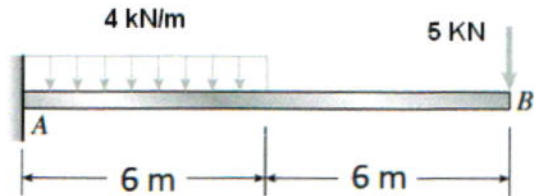
6 / 30

1. For the cantilever beam shown

(1) Draw free-body diagram of AB and find the reactions at A

(2) Draw the shear-force diagram

(3) Draw the bending moment diagram



$$R_A = R_B$$

$$R_A = ?$$

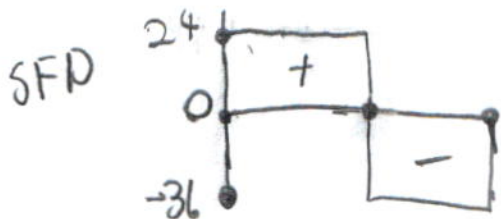
$$-2$$

$$\sum M_A = (4 \frac{\text{kN}}{\text{m}} \times 6 \text{m}) \times 3 + (5 \text{ kN} \times 12 \text{m})$$

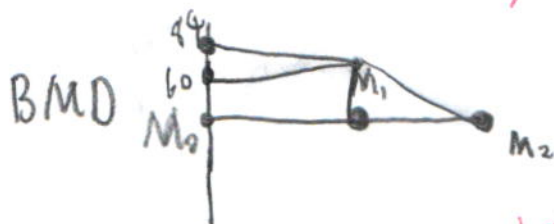
$$= 84 \frac{\text{kN}}{\text{m}}$$

$$-132$$

✓



✓ -10



$$M_0 = 84$$

$$M_1 = 64 - (4 \frac{\text{kN}}{\text{m}} \cdot 6 \text{m})$$

$$M_2 = 0$$

✓

-1

BELOW AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #2

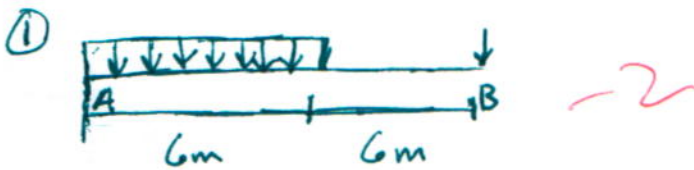
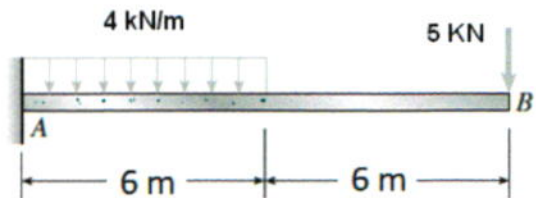
Total points: 30

Date: 9/13/2021

Name: Myles Woodfork

2/30

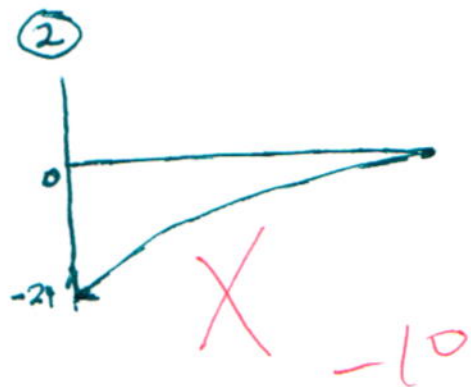
1. For the cantilever beam shown
 - (1) Draw free-body diagram of AB and find the reactions at A
 - (2) Draw the shear-force diagram
 - (3) Draw the bending moment diagram



$$0 = R_A - (4 \times 6) - 5$$

$$-R_A = -24$$

$$R_A = 24$$



③

$$M_A = ? - 4$$



-12

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 3

TOPICS COVERED IN QUIZ 3

Mohr's Circle

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

MCEG3304: Machine Design I
Fall 2021

Quiz #3

Total points: 30

Date: 9/22/2021

Name:

For the plane stress states $\sigma_x = -10\text{ksi}$, $\sigma_y = 20\text{ksi}$, $\tau_{xy} = 15\text{ksi}$ cw., find

- (1) Draw a Mohr's circle, label the points A, B, C, D, E, F, G with their coordinates.**
- (2) Find what are the principal normal stresses and what are the principle shear stresses.**
- (3) Draw the stress elements of principal normal stresses and shear stresses.**
- (4) On a separate coordinate, draw a complete Mohr's three-circle diagram and label the value of $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{2/3}$.**

MCEG3043: Machine Design I

Fall 2021

Quiz #3

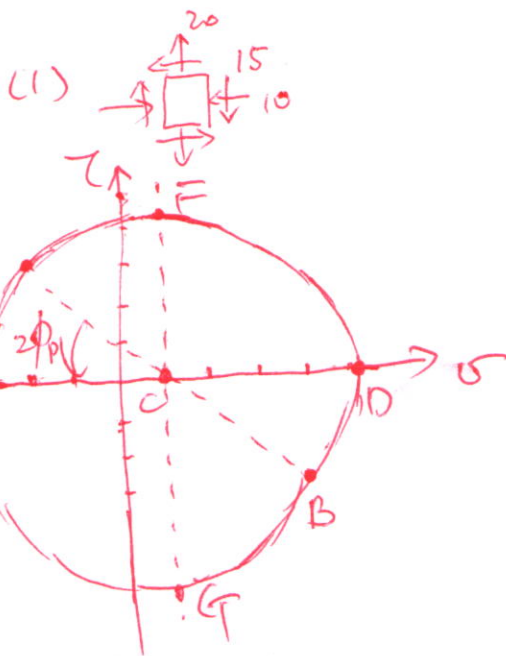
Total points: 30

Date: 9/22/2021

Name:

For the plane stress states $\sigma_x = -10 \text{ kpsi}$, $\sigma_y = 20 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi cw.}$, find

- (1) Draw a Mohr's circle, label the points A, B, C, D, E, F, G with their coordinates.
- (2) Find what are the principal normal stresses and what are the principle shear stresses.
- (3) Draw the stress elements of principal normal stresses and shear stresses.
- (4) On a separate coordinate, draw a complete Mohr's three-circle diagram and label the value of $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{2/3}$.



$$A(-10, 15^{cw})$$

$$B(20, 15^{cw})$$

$$C(5, 0)$$

$$R = \sqrt{15^2 + 15^2} = 21.2$$

$$D(26.2, 0)$$

$$E(-16.2, 0)$$

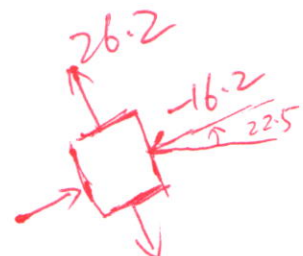
$$F(5, 21.2)$$

$$G(5, -21.2)$$

- (2) Principal normal stresses: $\sigma_1 = 26.2$, $\sigma_2 = -16.2$
 Principal shear stresses: $\tau_1 = 21.2$, $\tau_2 = -21.2$

$$(3) \phi_{p1} = \frac{1}{2} \tan^{-1} \left(\frac{2\tau_{xy}}{\sigma_x - \sigma_y} \right) = \frac{1}{2} \tan^{-1} \left(\frac{2(-15)}{-10 - 20} \right) = 22.5^\circ \quad A \rightarrow E$$

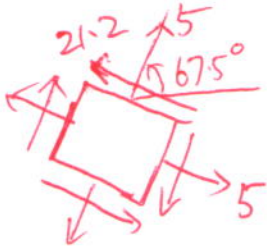
~~$\phi_{s1} = 45^\circ$~~ or $\phi_{p1} = -67.5^\circ \quad A \rightarrow D$
 $\phi_{p2} = \dots$



(3)

$$\phi_{s1} = 45^\circ + \phi_{p1} = 67.5^\circ \uparrow A \rightarrow G$$

$$\text{or } \phi_{s1} = 45^\circ - 67.5^\circ = -22.5^\circ \downarrow A \rightarrow F$$



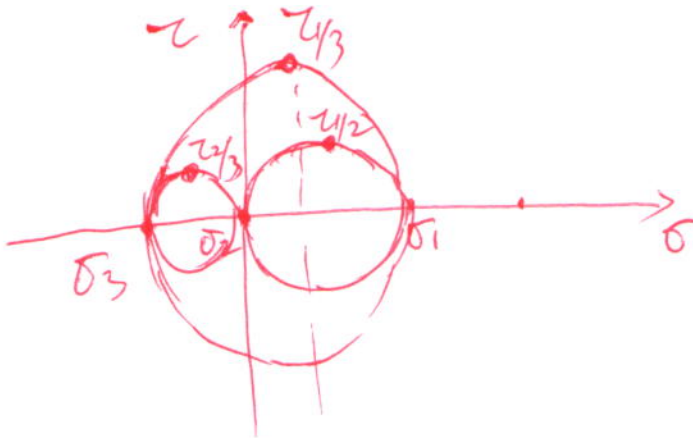
(4)

$$\sigma_1 = 26.2 \quad \sigma_2 = 0, \quad \sigma_3 = -16.2$$

$$\tau_{1/2} = \frac{26.2 - 0}{2} = 13.1$$

$$\tau_{2/3} = \frac{0 - (-16.2)}{2} = 8.1$$

$$\tau_{1/3} = \frac{26.2 - (-16.2)}{2} = 21.2$$



ABOVE AVERAGE

MCEG3304: Machine Design I
Fall 2021

Quiz #3
Total points: 30

26/30

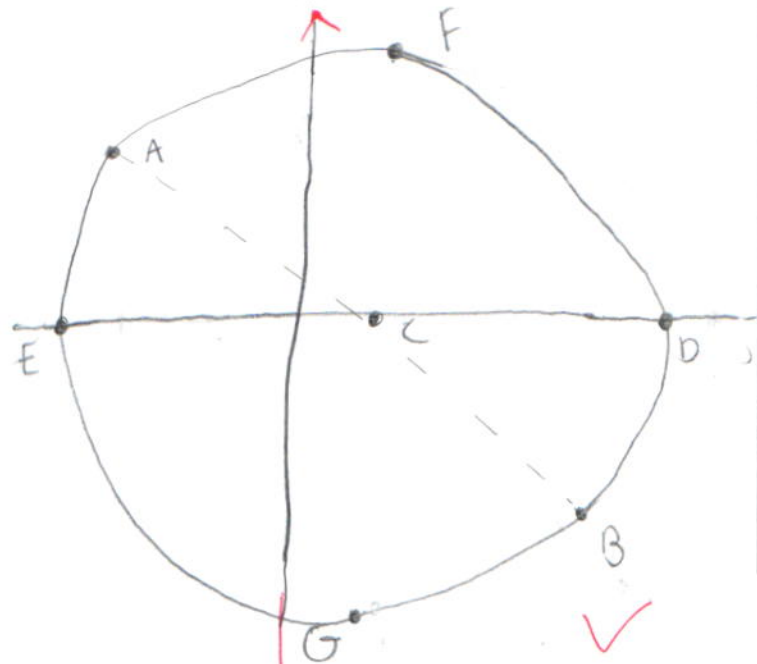
Date: 9/22/2021

Name: Dwayne Smith

For the plane stress states $\sigma_x = -10 \text{ kpsi}$, $\sigma_y = 20 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi cw.}$, find

- (1) Draw a Mohr's circle, label the points A, B, C, D, E, F, G with their coordinates.
- (2) Find what are the principal normal stresses and what are the principle shear stresses.
- (3) Draw the stress elements of principal normal stresses and shear stresses.
- (4) On a separate coordinate, draw a complete Mohr's three-circle diagram and label the value of $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{2/3}$.

- 1) A (-10, 15)
B (20, -15) ✓
C (5, 0) ✓
F = 21.21 ✓
D = (26.21, 0) ✓
E = (-16.21, 0) ✓
F = (5, 21.21) ✓
G = (5, -21.21) ✓



$$(2) \sigma_{1,2} = \frac{-10 + 20}{2} \pm \sqrt{\left(\frac{-10 - 20}{2}\right)^2 + 15^2}$$

$$\sigma_1 = 26.21, \sigma_2 = -16.21$$

$$\tau_{1,2} = \pm \sqrt{\left(\frac{-10 - 20}{2}\right)^2 + 15^2}$$

$$= +21.21, -21.21$$

2)

$$\sigma_1 = \frac{-10+20}{2} + \sqrt{\left(\frac{-10-20}{2}\right)^2 + 15^2} = 26.21$$

$$\sigma_2 = \frac{-10+20}{2} - \sqrt{\left(\frac{-10-20}{2}\right)^2 + 15^2} = -16.21$$

$$\tan 2\phi_p = \frac{2(15)}{-10-20} = 22.5^\circ \quad \checkmark$$

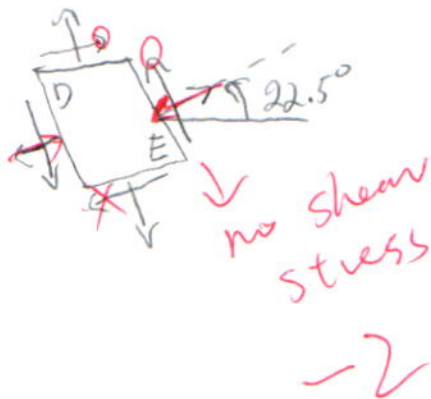
$$\tau_1 = +\sqrt{\left(\frac{-10-20}{2}\right)^2 + 15^2} = 21.21$$

$$\tau_2 = -\sqrt{\left(\frac{-10-20}{2}\right)^2 + 15^2} = -21.21$$

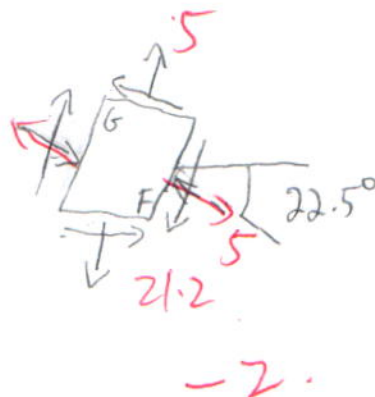
$$\tan 2\phi_s = -\frac{(-10-20)}{2(-15)} = -22.5^\circ$$

3)

Normal stress
element



Shear stress



AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #3

Total points: 30

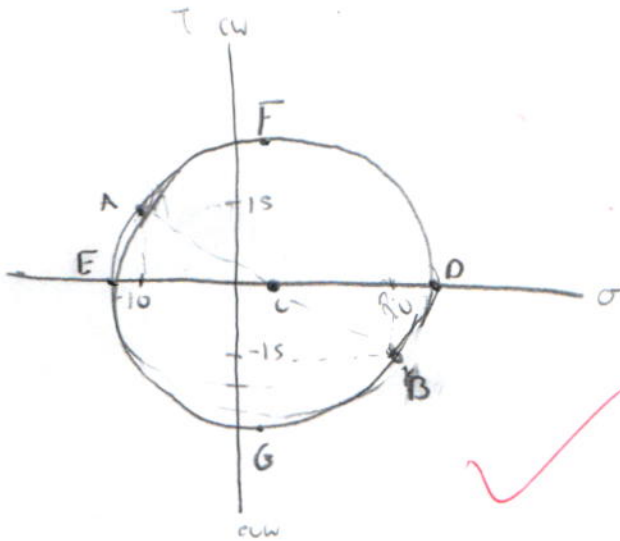
Date: 9/22/2021

Name: Ty Dupre

For the plane stress states $\sigma_x = -10 \text{ kpsi}$, $\sigma_y = 20 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi cw.}$, find

- (1) Draw a Mohr's circle, label the points A, B, C, D, E, F, G with their coordinates.
- (2) Find what are the principal normal stresses and what are the principle shear stresses.
- (3) Draw the stress elements of principal normal stresses and shear stresses.
- (4) On a separate coordinate, draw a complete Mohr's three-circle diagram and label the value of $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{2/3}$.

1)



$A(-10, 15)$ ✓ $C(5, 0)$ ✓
 $B(20, -15)$ ✓ $D = (26.21, 0)$ ✓
 $E(-16.21, 0)$ ✓ $F = (5, 21.21)$ ✓
 $G(5, -21.21)$ ✓

$$15^2 + 15^2 = C^2 \quad C = 21.21 \text{ or}$$

$$D = 5 + 21.21 = 26.21$$

$$E = 5 - 21.21$$

2)

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{-10 + 20}{2} + \sqrt{\left(\frac{-10 + 20}{2}\right)^2 + (15)^2} = 26.21 = \text{max normal stress}$$

$$\sigma_2 = \frac{-10 + 20}{2} - \sqrt{\left(\frac{-10 + 20}{2}\right)^2 + (15)^2} = -16.21 = \text{minimum normal stress}$$

(Double check with equation)



Principal shear

$$\tau_1, \tau_2 = \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\tau_1 = \sqrt{\left(\frac{-10 - 20}{2}\right)^2 + 15^2} = 21.21 \text{ - max shear stress}$$

$$\tau_2 = -\sqrt{\left(\frac{-10 - 20}{2}\right)^2 + 15^2} = -21.21 \text{ min. shear stress}$$

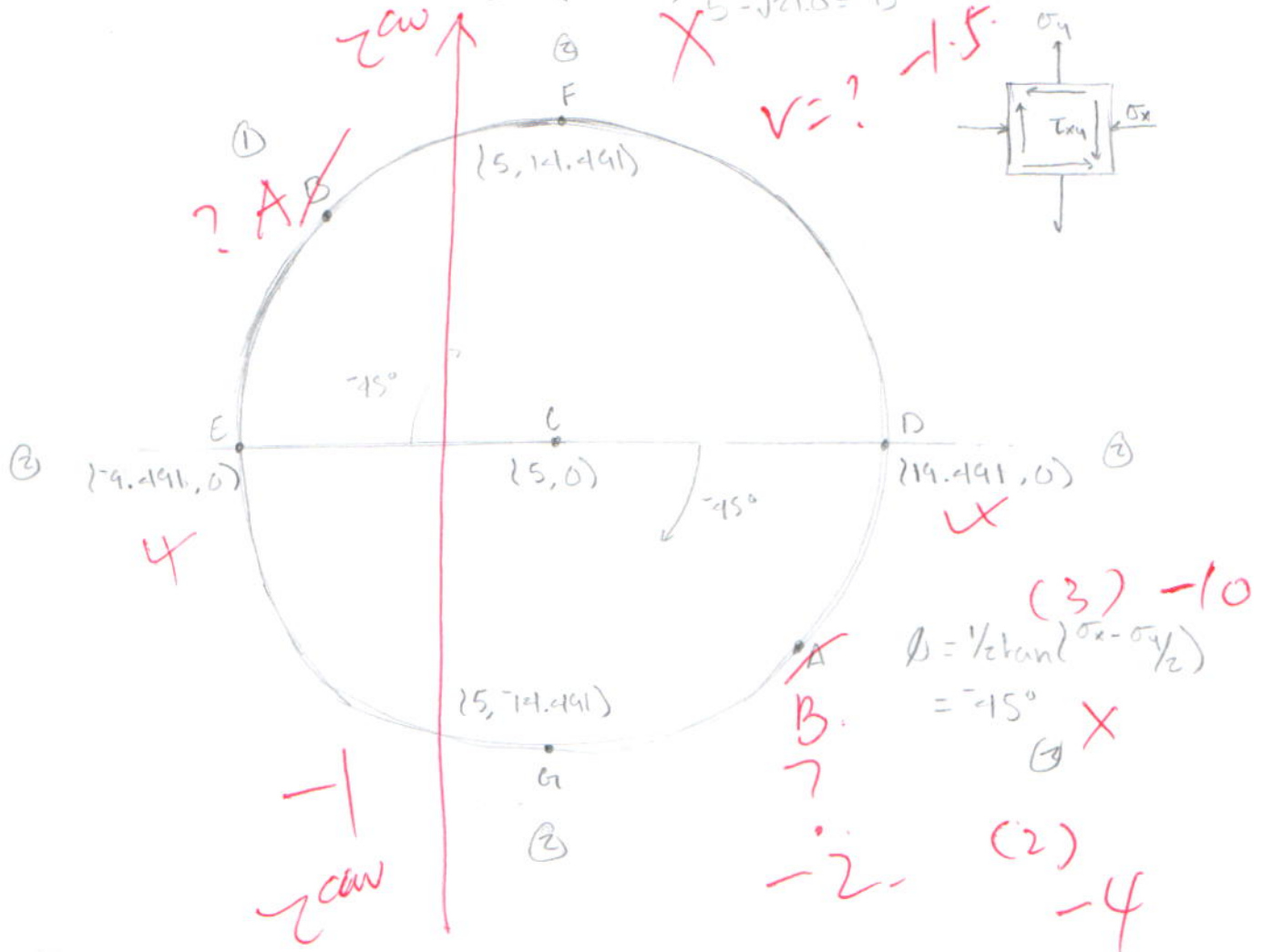
3) stress element

$$\tan^{-1} \frac{\sigma_1 - \sigma_2}{2\tau_{xy}} = \tan^{-1} \frac{26.21 - (-16.21)}{2(15)} = 18.43^\circ$$

Back on front page to draw

10

BELOW AVERAGE



**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 4

TOPICS COVERED IN QUIZ 4

Mohr's three-circle diagram

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

MCEG3304: Machine Design I
Fall 2021

Quiz #4

Total points: 30

Date: 09/27/2021

Name:

1. For the stress states $\sigma_y = 20\text{kpsi}$, $\sigma_z = -15\text{kpsi}$, $\tau_{xy} = 15\text{kpsi}$ cw., find

(1) All three principal stresses (Do not use the cubic equation (3-15)).

(2) Three principal shear stresses.

(3) Draw a complete Mohr's three-circle diagram and label all points of interest

$\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$.

MCEG3304: Machine Design I
Fall 2021

Quiz #4

Total points: 30

Date: 09/27/2021

Name:

1. For the stress states $\sigma_y = 20 \text{ kpsi}$, $\sigma_z = -15 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi}$ cw., find

(1) All three principal stresses (Do not use the cubic equation (3-15)).

(2) Three principal shear stresses.

(3) Draw a complete Mohr's three-circle diagram and label all points of interest

$\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$.

(1) $\sigma_x = 0$,
 $\sigma_z = -15$ is a principal stress. (2)

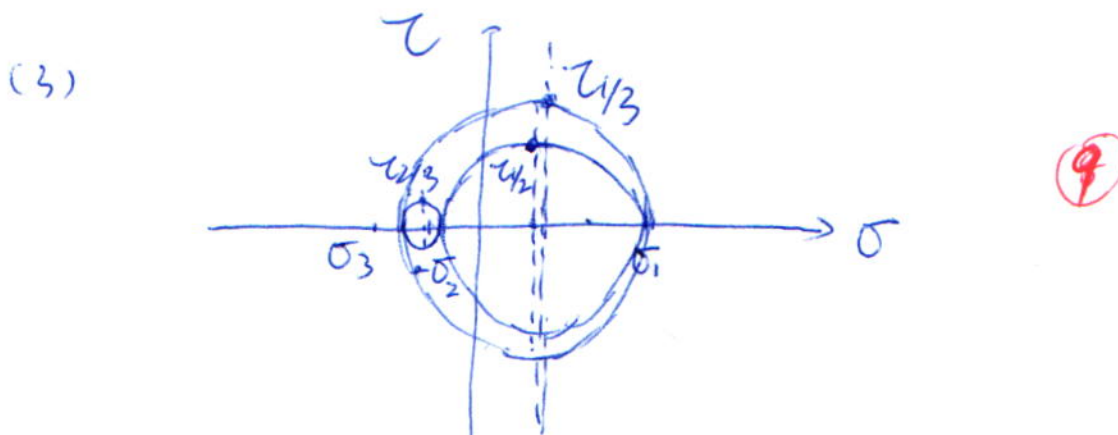
$$\sigma_{A/B} = \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$
$$= \frac{0 + 20}{2} \pm \sqrt{\left(\frac{0 - 20}{2}\right)^2 + 15^2} = 28, -8 \quad (8)$$

$\sigma_1 = 28, \sigma_2 = -8, \sigma_3 = -15$ (2)

(2) $\tau_{1/2} = \frac{28 - (-8)}{2} = 18$ (3)

$$\tau_{2/3} = \frac{-8 - (-15)}{2} = 3.5 \quad (3)$$

$$\tau_{1/3} = \frac{28 - (-15)}{2} = 21.5 \quad (3)$$



ABOVE AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #4

Total points: 30

Date: 09/27/2021

Name:

safiatai campalige

30/30

1. For the stress states $\sigma_y = 20 \text{ kpsi}$, $\sigma_z = -15 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi}$ cw., find

(1) All three principal stresses (Do not use the cubic equation (3-15)).

(2) Three principal shear stresses.

(3) Draw a complete Mohr's three-circle diagram and label all points of interest

 $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$.

1) $\sigma_y = 20 \text{ kpsi}$; $\sigma_z = -15 \text{ kpsi}$; $\tau_{xy} = 15 \text{ kpsi}$ cw
 $\sigma_x = 0 \text{ kpsi}$; $\tau_{yz} = 0$; $\tau_{zx} = 0$
because the shear stress on "z" surface are 0,
the principal stresses.

$$\begin{aligned}\sigma_{1/2} &= \frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} \\ &= \frac{0 + 20 \text{ kpsi}}{2} \pm \sqrt{\left(\frac{0 - 20}{2}\right)^2 + (-15)^2} \\ &= 10 \pm \sqrt{100 + 225}\end{aligned}$$

$$= 28.027 \text{ kpsi}, -8.027 \text{ kpsi}$$

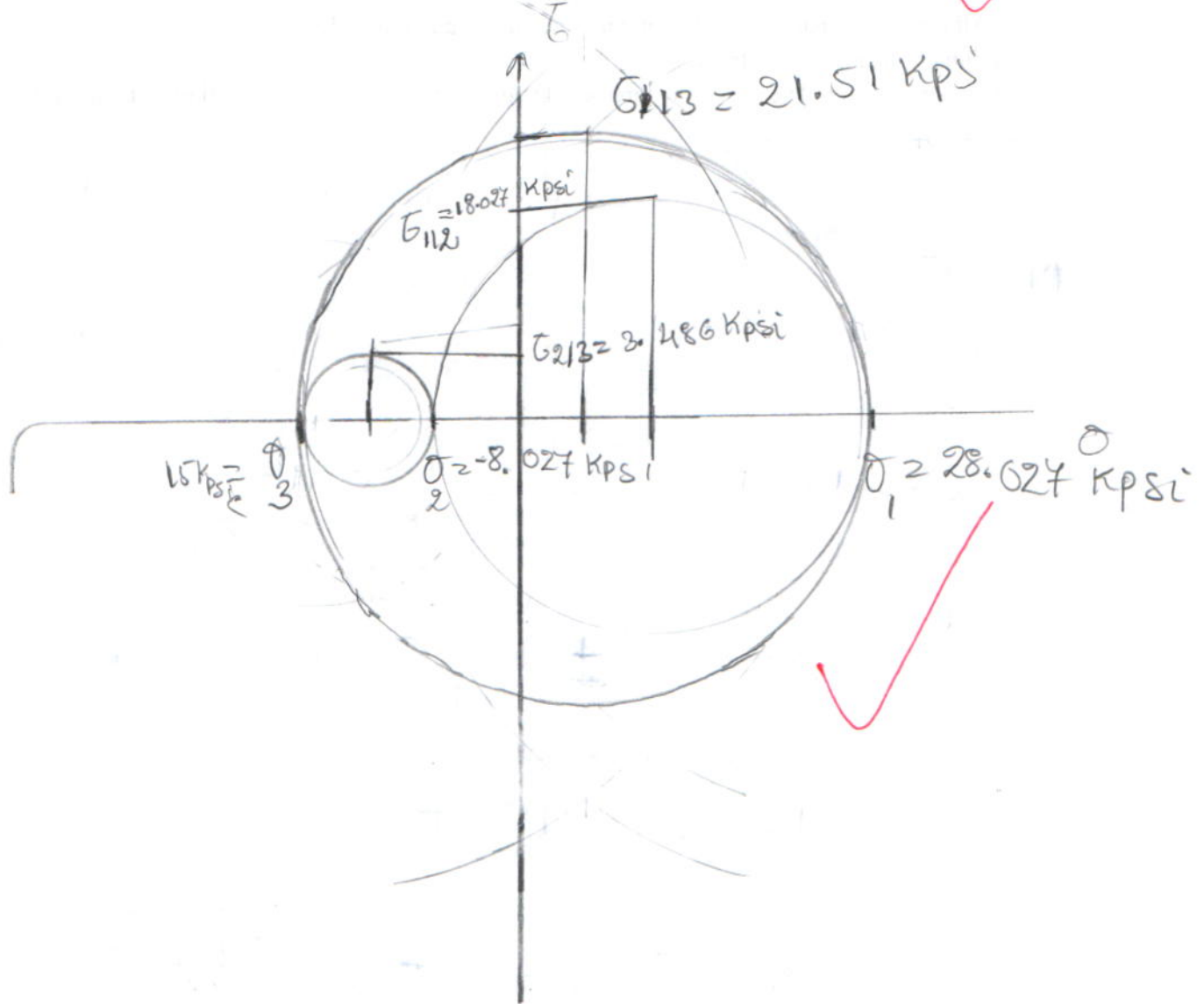
$$\text{so } \sigma_1 = 28.027 \text{ kpsi}; \sigma_2 = -8.027 \text{ kpsi}; \sigma_3 = -15 \text{ kpsi}$$

$$2') \quad \sigma_{1/2} = \frac{\sigma_1 - \sigma_3}{2} = \frac{28.027 + 8.027}{2} = 18.027 \text{ Kpsi}$$

$$\sigma_{2/3} = \frac{\sigma_2 - \sigma_3}{2} = \frac{-8.027 + 15}{2} = 3.486 \text{ Kpsi}$$

$$\sigma_{1/3} = \frac{\sigma_1 - \sigma_3}{2} = \frac{28.027 + 15}{2} = 21.51 \text{ Kpsi (max)}$$

3')



AVERAGE

MCEG3304: Machine Design I
Fall 2021

Quiz #4
Total points: 30

Date: 09/27/2021

Name: Christopher
Muelle

26/30

1. For the stress states $\sigma_y = 20 \text{ kpsi}$, $\sigma_z = -15 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi}$ cw., find

- (1) All three principal stresses (Do not use the cubic equation (3-15)).
- (2) Three principal shear stresses.
- (3) Draw a complete Mohr's three-circle diagram and label all points of interest $\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$.

1) $\sigma_y = 20$ $\sigma_z = -15$ $\tau_{xy} = -15$

$\sigma_x = 0$ $\sigma_y = 0$ $\sigma_z = 0$

Principal stress

$\sigma_z = -15$ ✓

$$\frac{\sigma_x - \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} = \frac{0 - 20}{2} \pm \sqrt{\left(\frac{0 - 20}{2}\right)^2 + (15)^2}$$

2) $\sigma_1 = 28.02$

$\sigma_2 = 0$

$\sigma_3 = -8.02$

$\sigma_2 = -8$

$\sigma_3 = -15$

$\frac{20 - 0}{2} = 10$

$10 + 18.02 = 28.02$

$10 - 18.02 = -8.02$

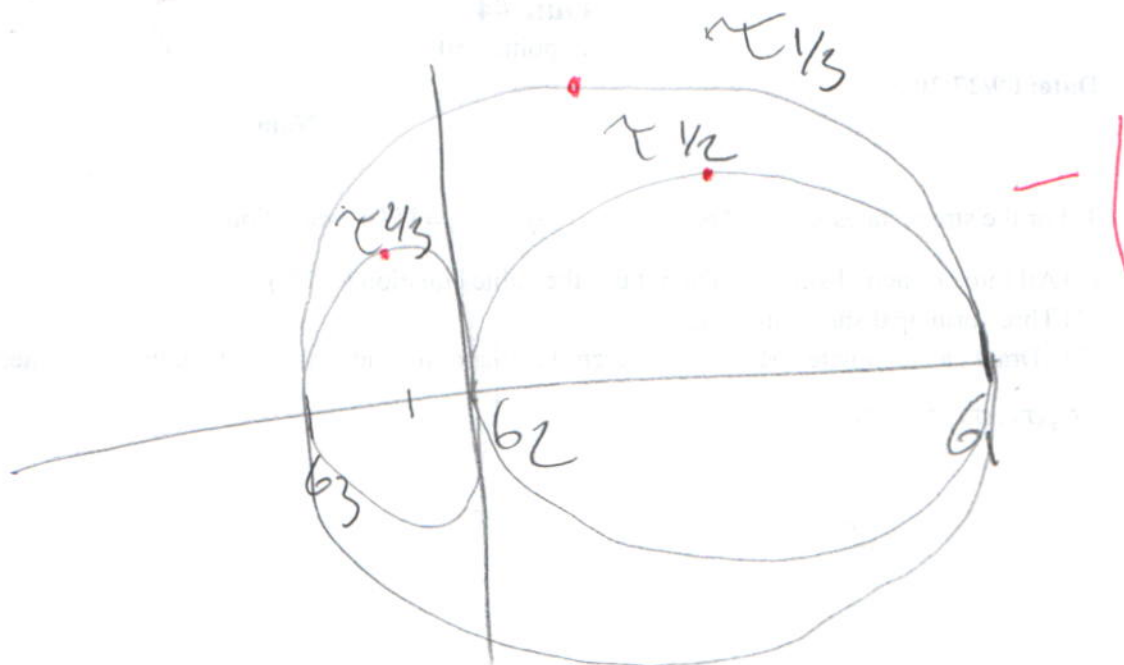
$\tau_{1/2} = \frac{18.02 - 0}{2} = 9.01$

$\tau_{1/3} = \frac{28.02 - (-8.02)}{2} = 18.02$

$\tau_{2/3} = \frac{0 - (-8.02)}{2} = 4.01$

→ BACK

3)



BELOW AVERAGE

MCEG3304: Machine Design I
Fall 2021

Quiz #4

Total points: 30

Date: 09/27/2021

Name: Chris Nylander

19/30

1. For the stress states $\sigma_x = 20 \text{ kpsi}$, $\sigma_z = -15 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi cw.}$, find

- (1) All three principal stresses (Do not use the cubic equation (3-15)).
- (2) Three principal shear stresses.
- (3) Draw a complete Mohr's three-circle diagram and label all points of interest

$\sigma_1, \sigma_2, \sigma_3, \tau_{1/2}, \tau_{2/3}, \tau_{1/3}$

$\sigma_x = 0$, $\sigma_y = 20 \text{ kpsi}$, $\sigma_z = -15 \text{ kpsi}$, $\tau_{xy} = 15 \text{ kpsi cw.}$

$\sigma_{A, B} =$

① $\sigma_1 = 28.027$

$\sigma_2 = -8.027$

$\sigma_3 = 0$

② $\tau_{1/2} = 18.027$

$\tau_{2/3} = -4.0135$

$\tau_{1/3} = 14.0135$

$$\frac{0+20}{2} \pm \sqrt{\left(\frac{0-20}{2}\right)^2 + (15)^2}$$

$$10 \pm \sqrt{(-10)^2 + (15)^2}$$

$$10 + 18.027, 10 - 18.027$$

$\sigma_1 = 28.027$, $\sigma_2 = -8.027$

$\sigma_3 = 0$

② $\tau_{1/2} = \frac{\sigma_1 - \sigma_2}{2} = \frac{28.027 - (-8.027)}{2} = \frac{36.054}{2} = 18.027$

$\tau_{2/3} = \frac{\sigma_2 - \sigma_3}{2} = \frac{-8.027 - 0}{2} = \frac{-8.027}{2} = -4.0135$

$\tau_{1/3} = \frac{\sigma_1 - \sigma_3}{2} = \frac{28.027 - 0}{2} = \frac{28.027}{2} = 14.0135$

A (0, 15) cw

B (20, 15) cw

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 5

TOPICS COVERED IN QUIZ 5

Bending normal stress

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

MCEG3304: Machine Design I
Fall 2021

Quiz #5

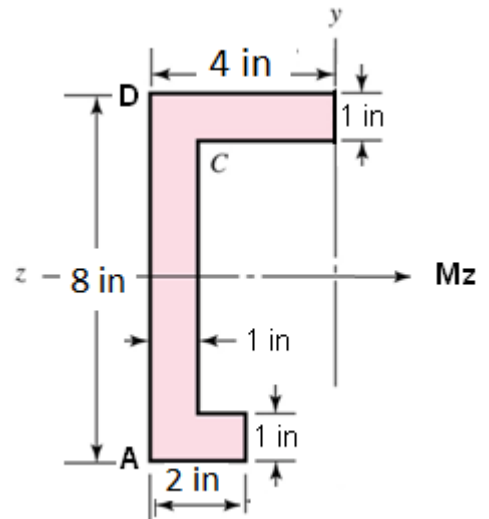
Total points: 30

Date: 10/04/2021

Name:

For the section shown, the bending moment is shown in the figure and $M_z = 100 \text{ kip in.}$

- (1) Find the distance from the mass center to the bottom surface.
- (2) Calculate the second moment of area I of the section.
- (3) Determine the bending normal stresses at top D and bottom A.



MCEG3304: Machine Design I

Fall 2021

Quiz #5

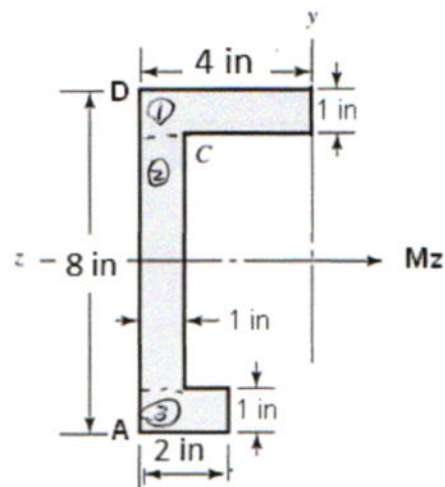
Total points: 30

Date: 10/04/2021

Name:

For the section shown, the bending moment is shown in the figure and $M_z = 100 \text{ kip in}$.

- (1) Find the distance from the mass center to the bottom surface.
- (2) Calculate the second moment of area I of the section.
- (3) Determine the bending normal stresses at top D and bottom A.



$$(1) A_1 = 4 \times 1 = 4 \quad C_1 = 7.5 \text{ in}$$

$$A_2 = 6 \times 1 = 6 \quad C_2 = 4 \text{ in}$$

$$(6) A_3 = 2 \times 1 = 2 \quad C_3 = 0.5 \text{ in}$$

$$(4+6+2) \times C = 4 \times 7.5 + 6 \times 4 + 2 \times 0.5$$

$$(2) C = 4.58 \text{ in}$$

$$(2) (4) I_1 = \frac{4 \times 1^3}{12} + 4 \times (7.5 - 4.58)^2 = 34.44 \text{ in}^4$$

$$(4) I_2 = \frac{1 \times 6^3}{12} + 6 \times (4 - 4.58)^2 = 20 \text{ in}^4$$

$$(4) I_3 = \frac{2 \times 1^3}{12} + 2 \times (4.58 - 0.5)^2 = 33.46 \text{ in}^4$$

$$(2) I = 34.44 + 20 + 33.46 = 87.9 \text{ in}^4$$

$$(3) (4) \sigma_D = \frac{100 \times (8 - 4.58)}{87.9} = 3.89 \text{ kpsi}$$

$$(4) \sigma_A = - \frac{100 \times 4.58}{87.9} = -5.21 \text{ kpsi}$$

ABOVE AVERAGE

MCEG3304: Machine Design I
Fall 2021

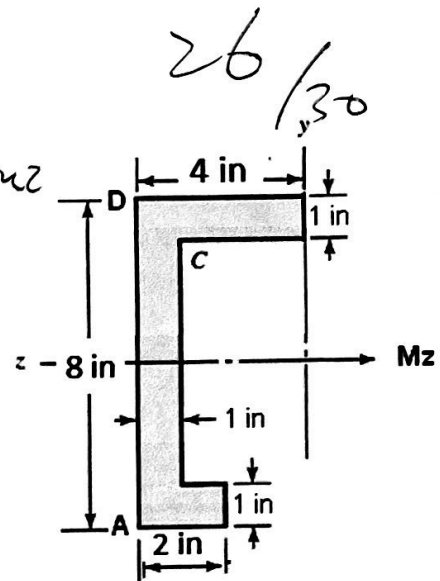
Quiz #5
Total points: 30

Date: 10/04/2021

Name: Harold Rodriguez

For the section shown, the bending moment is shown in the figure and $M_z = 100$ kip in.

- (1) Find the distance from the mass center to the bottom surface.
- (2) Calculate the second moment of area I of the section.
- (3) Determine the bending normal stresses at top D and bottom A.



①

$$\bar{y} = A_1 y_1 + A_2 y_2 + A_3 y_3$$

$$A_3 = 4 \times 1 = 4 \text{ in}^2 \quad \bar{y}_3 = 1 + 3 + 0.5 = 4.5 \text{ in}$$

$$A_2 = 6 \times 1 = 6 \text{ in}^2 \quad \bar{y}_2 = 1 + 2 = 3 \text{ in}$$

$$A_1 = 2 \times 1 = 2 \text{ in}^2 \quad \bar{y}_1 = 1 = 0.5 \text{ in}$$

$$\bar{y} = 4 \times 4.5 + 6 \times 3 + 2 \times 0.5$$

$$\bar{y} = 3.11 \text{ in}$$

③ $M_z = 100 \text{ kip} \cdot \text{in}$ $\frac{M}{I} = \frac{\sigma}{y}$

$$\sigma = \frac{100 \text{ kip} \cdot \text{in}}{27.86} \times (8 - 3.11) \text{ in}$$

$$\sigma_{\text{top}} = 17.55 \text{ kip/in}^2$$

②

$$\frac{2 \times 1^3}{12} + 2 \times 1 \times (3.11 - 0.5)^2 + \frac{6 \times 1^3}{12}$$

$$+ 6 \times 1 \times (3.11 - 2)^2 + \frac{4 \times 1^3}{12} \times 4 \times 1$$

$$(4.5 - 3.11)^2$$

$$0.25 + 13.62 + 2.25 + 3.69 + 3.33 + 7.72$$

$$I = 27.86 \text{ in}^4$$

$$\sigma_{\text{bottom}} = \frac{100 \text{ kip} \cdot \text{in}}{27.86} \times (3.11) \text{ in}$$

$$\sigma_{\text{bottom}} = 11.16 \text{ kip/in}^2$$

AVERAGE

MCEG3304: Machine Design I

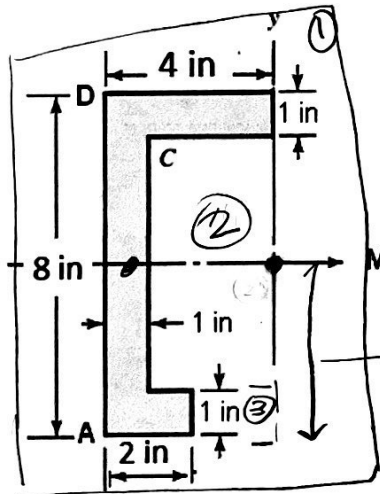
Fall 2021

Quiz #5

Total points: 30

Date: 10/04/2021

Name: Gabriela Garay



For the section shown, the bending moment is shown in the figure and $M_z = 100$ kip in.

- (1) Find the distance from the mass center to the bottom surface.
- (2) Calculate the second moment of area I of the section.
- (3) Determine the bending normal stresses at top D and bottom A.

1.) Mass of center from bottom

$$A_1 = 4 \times 8 = 32$$

$$A_2 = \frac{3}{2} \times 6 = 12$$

$$A_3 = 2 \times 1 = 2$$

$$32 - 12 - 2 = 18 \rightarrow A$$

$$\bar{y} = \frac{(32)(4) + 12(\frac{6}{2}) + 2(\frac{1}{2})}{18} = \frac{113}{18} \approx 6.27 \text{ in} \rightarrow \text{bottom}$$

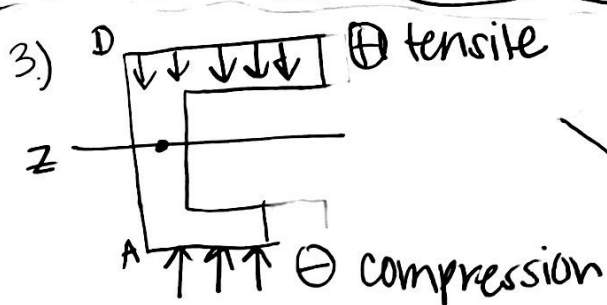
$$8 - 6.27 = 1.73 \rightarrow \text{top}$$

$$2.) I_z = I_{ca} + A d^2$$

$$I_z = \frac{bh^3}{12} + A d^2$$

$$= \left(\frac{(4)(8)^3}{12} \right) - \left(\frac{(\frac{3}{2})(6)^3}{12} \right) - \left(\frac{(2)(1)^3}{12} \right) + (18)(6.27)^2$$

$$134.5 + 18(6.27)^2 = 247.36 \text{ in}^4$$



BELOW AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #5

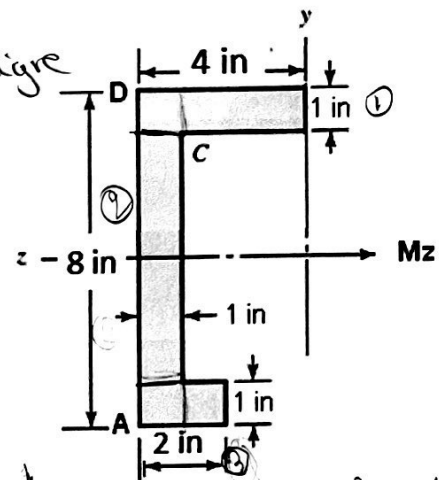
Total points: 30

Date: 10/04/2021

Name: *Safiatou Zampaligre*

For the section shown, the bending moment is shown in the figure and $M_z = 100$ kip in.

- (1) Find the distance from the mass center to the bottom surface.
- (2) Calculate the second moment of area I of the section.
- (3) Determine the bending normal stresses at top D and bottom A.



$$\begin{aligned}
 1) \quad A_1 &= 4 \times 1 \text{ in}^2 = 4 \text{ in}^2 \checkmark \\
 A_2 &= 1 \text{ in} \times 8 \text{ in} = 8 \text{ in}^2 \checkmark \\
 A_3 &= 2 \times 1 \text{ in}^2 = 2 \text{ in}^2 \checkmark
 \end{aligned}$$

$$\begin{aligned}
 A_T &= 4 \text{ in}^2 + 8 \text{ in}^2 + 2 \text{ in}^2 \\
 &= 14 \text{ in}^2 \checkmark
 \end{aligned}$$

$$\begin{aligned}
 \bar{y} &= \frac{A_1 \times y_1 + A_2 \times y_2 + A_3 \times y_3}{A_T} \\
 &= \frac{4 \times 4 + 8 \times 8 + 2 \times 10}{14} \\
 &= 3.57 \text{ in}
 \end{aligned}$$

-3

$$\begin{aligned}
 2) \quad I_1 &= \frac{1}{12} b \cdot h^3 + A d^2 \\
 &= \frac{1}{12} \times 4 \times 1^3 - 2 \\
 &= 0.33 \text{ in}^4
 \end{aligned}$$

$$\begin{aligned}
 I_2 &= \frac{1}{12} \times 1 \times 8^3 + A d^2 \\
 &= 18 \text{ in}^4 - 2
 \end{aligned}$$

$$\begin{aligned}
 I_3 &= \frac{1}{12} \times 2 \times 1^3 + A d^2 \\
 &= 0.16 \text{ in}^4 - 2
 \end{aligned}$$

$$\begin{aligned}
 I &= I_1 + I_2 + I_3 \\
 &= 0.33 + 18 + 0.16 \\
 &= 18.49 \text{ in}^4
 \end{aligned}$$

4

$$\begin{aligned}
 3) \quad \sigma &= \frac{M \cdot y}{I} \\
 &= \frac{100 \cdot y}{18.49}
 \end{aligned}$$

-8

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 6

TOPICS COVERED IN QUIZ 6

Deflection

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

Name:

MCEG3043: Machine Design I

Fall 2021

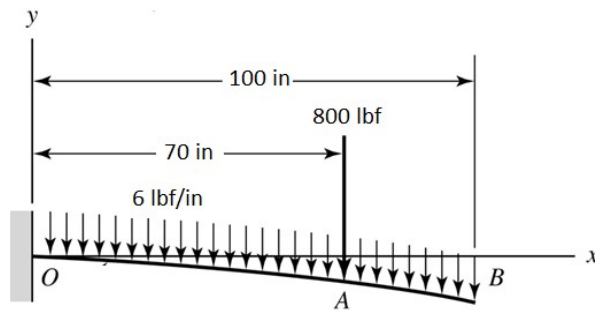
Quiz #6

Total points: 30

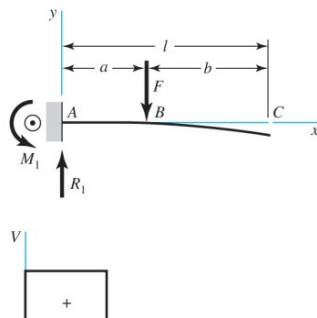
Date: 10/20/2021

The cantilever circular shaft is loaded as shown in the figure. The material is carbon steel ($E=30 \text{ Mpsi}$).

- (1) If the diameter of the shaft is 2 inch, what's the deflection at B?
- (2) If the deflection at midspan will not exceed 0.5 in, what does the minimum diameter of the shaft need to be?

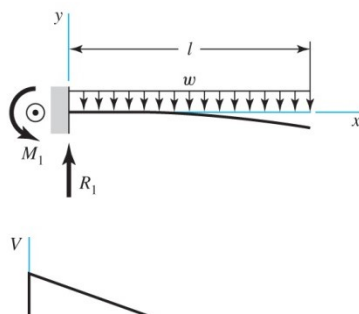


2 Cantilever—intermediate load



$$\begin{aligned}
 R_1 &= V = F & M_1 &= Fa \\
 M_{AB} &= F(x - a) & M_{BC} &= 0 \\
 y_{AB} &= \frac{Fx^2}{6EI}(x - 3a) \\
 y_{BC} &= \frac{Fa^2}{6EI}(a - 3x) \\
 y_{\max} &= \frac{Fa^2}{6EI}(a - 3l)
 \end{aligned}$$

3 Cantilever—uniform load



$$\begin{aligned}
 R_1 &= wl & M_1 &= \frac{wl^2}{2} \\
 V &= w(l - x) & M &= -\frac{w}{2}(l - x)^2 \\
 y &= \frac{wx^2}{24EI}(4lx - x^2 - 6l^2) \\
 y_{\max} &= -\frac{wl^4}{8EI}
 \end{aligned}$$

Name:

MCEG3043: Machine Design I

Fall 2021

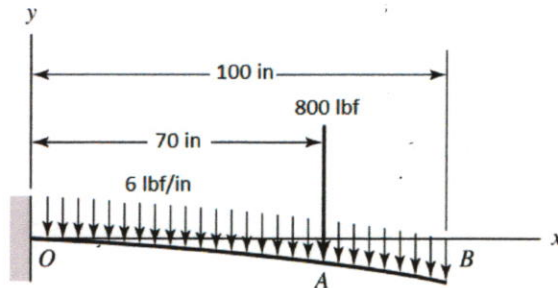
Quiz #6

Total points: 30

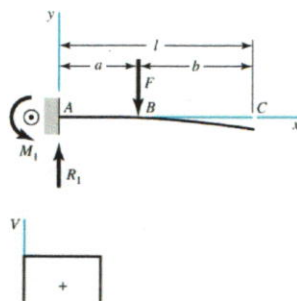
Date: 10/20/2021

The cantilever circular shaft is loaded as shown in the figure. The material is carbon steel ($E=30$ Mpsi).

- (1) If the diameter of the shaft is 2 inch, what's the deflection at B?
- (2) If the deflection at midspan will not exceed 0.5 in, what does the minimum diameter of the shaft need to be?



2 Cantilever—intermediate load



$$R_1 = V = F \quad M_1 = Fa$$

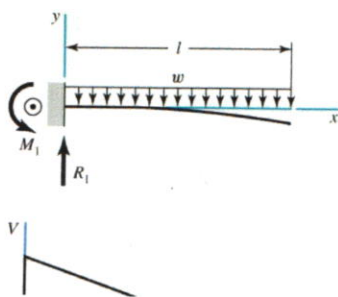
$$M_{AB} = F(x - a) \quad M_{BC} = 0$$

$$y_{AB} = \frac{Fx^2}{6EI}(x - 3a)$$

$$y_{BC} = \frac{Fa^2}{6EI}(a - 3x)$$

$$y_{\max} = \frac{Fa^2}{6EI}(a - 3l)$$

3 Cantilever—uniform load



$$R_1 = wl \quad M_1 = \frac{wl^2}{2}$$

$$V = w(l - x) \quad M = -\frac{w}{2}(l - x)^2$$

$$y = \frac{wx^2}{24EI}(4lx - x^2 - 6l^2)$$

$$y_{\max} = -\frac{wl^4}{8EI}$$

Name:

$$(1) \quad ① \quad I = \frac{\pi d^4}{64} = \frac{\pi \times 2^4}{64} = 0.7854 \text{ in}^4$$

$$② \quad X_B = 100 \text{ in}$$

$$⑤ \quad y_1 = \frac{F a^2}{6 E I} (a - 3x) = \frac{800 \times 70^2 \times (70 - 3 \times 100)}{6 \times 30 \times 10^6 \times 0.7854} = -\cancel{6.378} \text{ in}$$

$$F = 800 \text{ lbf}, a = 70 \text{ in}, E = 30 \text{ Mpsi}$$

$$⑥ \quad y_2 = \frac{W x^2}{24 E I} (4 l x - x^2 - 6 l^2) = \frac{6 \times 100^2 \times (4 \times 100 \times 100 - 100^2 - 6 \times 100^2)}{24 \times 30 \times 10^6 \times 0.7854} = -3.18$$

$$W = 6 \text{ lbf/in}, E = 30 \text{ Mpsi}, l = 100 \text{ in}$$

$$⑦ \quad y = -6.378 - 3.183 = -9.561 \text{ in}$$

$$(2) \quad X = 50 \text{ in} \quad ②$$

$$④ \quad \frac{F x^2}{6 E I} (x - 3a) + \frac{W x^2}{24 E I} (4 l x - x^2 - 6 l^2) = -0.5 \quad ②$$

$$F = 800, a = 70 \text{ in} \quad W = 6 \text{ lbf/in} \quad l = 100 \text{ in}$$

$$I = \left[\frac{800 \times 50^2}{6 \times 30 \times 10^6 \times \cancel{0.7854}} \times (50 - 3 \times 70) + \frac{6 \times 50^2 (4 \times 100 \times 50 - 50^2 - 6 \times 100^2)}{24 \times 30 \times 10^6 \times \cancel{0.7854}} \right] / -0.5$$

$$① \quad = \cancel{5.326} - 1.778$$

$$I = \frac{\pi d^4}{64} = \cancel{0.7854} \times 5.326$$

$$② \quad D = \sqrt[4]{\frac{64 \times \cancel{0.7854} \times 5.326}{\pi}} = \cancel{3.428} \text{ in}$$

$$3.227$$

ABOVE AVERAGE

Name: Emanuel Santiago

MCEG3043: Machine Design I

Fall 2021

Quiz #6

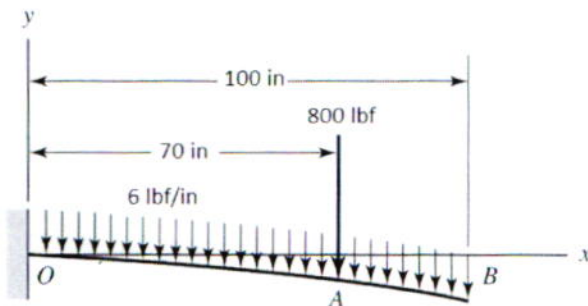
Total points: 30

28/30

Date: 10/20/2021

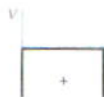
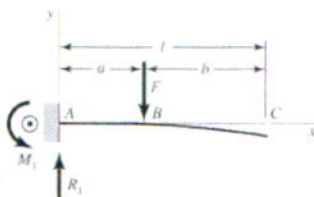
The cantilever circular shaft is loaded as shown in the figure. The material is carbon steel ($E=30$ Mpsi).

- (1) If the diameter of the shaft is 2 inch, what's the deflection at B?
- (2) If the deflection at midspan will not exceed 0.5 in, what does the minimum diameter of the shaft need to be?



$$I = \frac{\pi d^4}{64}$$

2 Cantilever—intermediate load



$$R_1 = V = F \quad M_1 = Fa$$

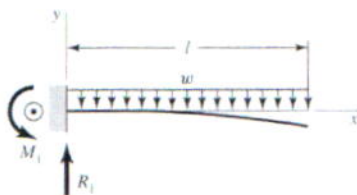
$$M_{AB} = F(x - a) \quad M_{BC} = 0$$

$$y_{AB} = \frac{Fx^2}{6EI}(x - 3a)$$

$$y_{BC} = \frac{Fa^2}{6EI}(a - 3x)$$

$$y_{max} = \frac{Fa^2}{6EI}(a - 3l) \quad \Rightarrow \quad \checkmark_1$$

3 Cantilever—uniform load



$$R_1 = wl \quad M_1 = \frac{wl^2}{2}$$

$$V = w(l - x) \quad M = -\frac{w}{2}(l - x)^2$$

$$y = \frac{wx^2}{24EI}(4lx - x^2 - 6l^2) \quad \Rightarrow \quad \checkmark_2$$

$$y_{max} = -\frac{wl^4}{8EI}$$

Name: Emanuel Santiago Sanchez

$$1) \quad v_1 = \frac{F a^2}{6EI} (a - 3L) = \frac{F a^2}{3EI} (a - 3L)$$

$$v_2 = \frac{w x^2}{24EI} (4Lx - x^2 - 6L^2)$$

$$\text{at } x = L$$

$$v_2 = \frac{w x^2}{24EI} (4L^2 - L^2 - 6L^2)$$

$$v_2 = \frac{w x^2}{24EI} (-3L^2) = -\frac{w L^4}{8EI}$$

$$v_B = v_1 + v_2$$

$$= \frac{F a^2}{3EI} (a - 3L) - \frac{w L^4}{8EI}$$

$$= \frac{800 (70)^2}{3 (30 \times 10^6) \pi/64 (2^4)} (70 - 300) - \frac{6 (100)^4}{8 (30 \times 10^6) \pi/64 (2^4)}$$

$$= -15.94 \text{ in}$$

2)

$$V_1 = \frac{Fx^2}{6EI} (x - 3a)$$

$$\text{@ } x = L/2$$

$$V_1 = \frac{FL^2}{24EI} (L/2 - 3a)$$

$$V_2 = \frac{Wx^2}{24EI} (4Lx - x^2 - 6L^2)$$

$$\text{@ } x = L/2$$

$$V_2 = \frac{WL^2}{96EI} (2L^2 - \frac{L^2}{4} - 6L^2)$$

$$= " " (-4L^2 - \frac{L^2}{4})$$

$$= " " (-\frac{17L^2}{4})$$

$$= - \frac{17WL^4}{384EI}$$

$$V_m = V_1 + V_2 = .5$$

$$= \frac{FL^2}{24EI} (L/2 - 3a) - \frac{17WL^4}{384EI}$$

$$.5 = \frac{800(100)^2}{24(30 \times 10^6)(\frac{7}{64}d^4)} (50 - 210) - \frac{17(6)(100)^4}{384(30 \times 10^6)(\frac{7}{64}d^4)}$$

$$.5 = \frac{-36.21}{d^4} - \frac{18.04}{d^4} = \frac{-54.25}{d^4}$$

$$d^4 = \frac{54.25}{.5} = 108.5$$

$$d = 3.23 \text{ in}$$

AVERAGE

Name: Joseph Steele

MCEG3043: Machine Design I
Fall 2021

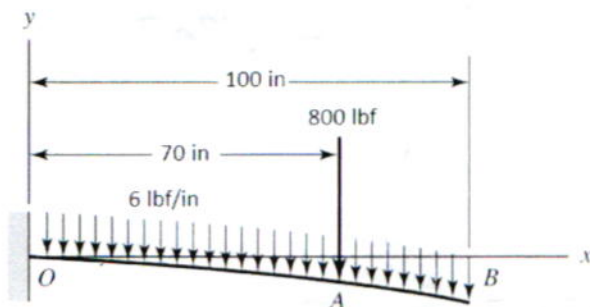
Quiz #6
Total points: 30

20/30

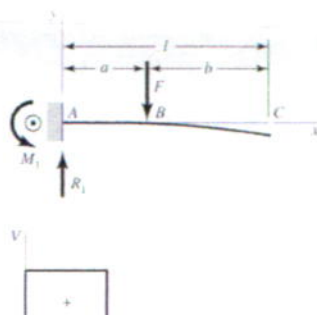
Date: 10/20/2021

The cantilever circular shaft is loaded as shown in the figure. The material is carbon steel ($E=30$ Mpsi).

- (1) If the diameter of the shaft is 2 inch, what's the deflection at B?
- (2) If the deflection at midspan will not exceed 0.5 in, what does the minimum diameter of the shaft need to be?



2 Cantilever—intermediate load



$$R_1 = V = F \quad M_1 = Fa$$

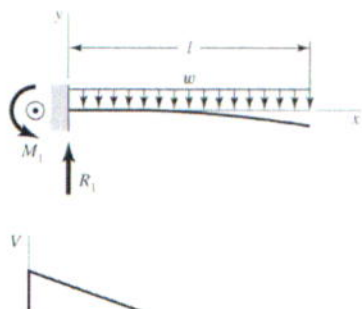
$$M_{AB} = F(x - a) \quad M_{BC} = 0$$

$$y_{AB} = \frac{Fx^2}{6EI}(x - 3a)$$

$$y_{BC} = \frac{Fa^2}{6EI}(a - 3x)$$

$$y_{max} = \frac{Fa^2}{6EI}(a - 3l)$$

3 Cantilever—uniform load



$$R_1 = wl \quad M_1 = \frac{wl^2}{2}$$

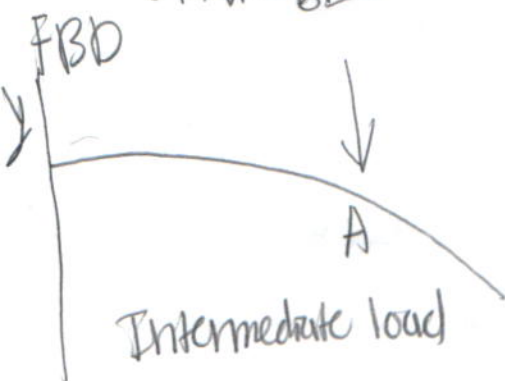
$$V = w(l - x) \quad M = -\frac{w}{2}(l - x)^2$$

$$y = \frac{wx^2}{24EI}(4lx - x^2 - 6l^2)$$

$$y_{max} = -\frac{wl^4}{8EI}$$

①

$$y_{max} = \frac{Fa^2}{6EI}(a-3l)$$



$$F = 800 \text{ lb}$$

$$a = 70 \text{ in}$$

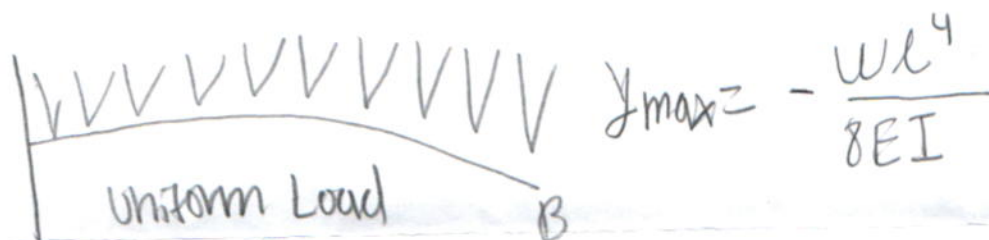
$$E = 30 \times 10^6$$

$$d = 2 \text{ in} = 0.1666 \text{ in}$$

$$I = \frac{\pi d^4}{64} = 0.785 \text{ in}^4$$

$$l = 100 \text{ in}$$

$$\frac{800(70)^2}{6(30 \times 10^6)(0.785)}(70 - 3(100)) = -6.3807$$



$$-\frac{6(100)^4}{8(30 \times 10^6)(0.785)} = -3.1847$$

Total @ B

$$y_{total} = y_A + y_B = -9.5654 \text{ in}$$

Name:

② Superstition at Maximum 0.5 inches

$x = 50 \text{ in}$
 $- 2$

$$d = \sqrt[4]{\frac{64}{\pi} I \frac{1}{8E_{yB}} \left(-\frac{Wl^4}{8EI} + \frac{Fa^2}{6EI} (a-3l) \right)}$$

$$I = \frac{1}{8(30 \times 10^6)(-0.5)} \left(-6(100)^4 + \frac{8(800 \times 70^2)}{6(6 \times 30 \times 10^6)(-0.5)} (70 - 3 \times 100) \right)$$

$$I = 5 \text{ inch}$$

$$I = \frac{\pi d^4}{64}$$

$$d = \sqrt[4]{\frac{64}{\pi} (I)}$$

$$d = \sqrt[4]{\frac{64}{\pi} (5)}$$

$$\text{Minimum diameter} = \boxed{3.176 \text{ in}}$$

BELOW AVERAGE

Name:

Harold Rodriguez

Harold Rodriguez

MCEG3043: Machine Design I

Fall 2021

Quiz #6

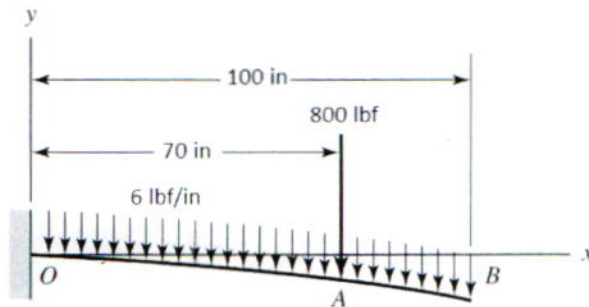
Total points: 30

16/30

Date: 10/20/2021

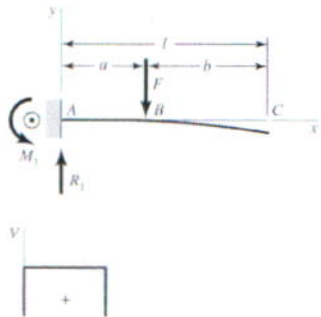
The cantilever circular shaft is loaded as shown in the figure. The material is carbon steel ($E=30$ Mpsi).

- (1) If the diameter of the shaft is 2 inch, what's the deflection at B?
- (2) If the deflection at midspan will not exceed 0.5 in, what does the minimum diameter of the shaft need to be?



$$\textcircled{1} y_{\max} = \frac{Fa^2}{6EI} (a-3l)$$

2 Cantilever—intermediate load



$$R_1 = V = F \quad M_1 = Fa$$

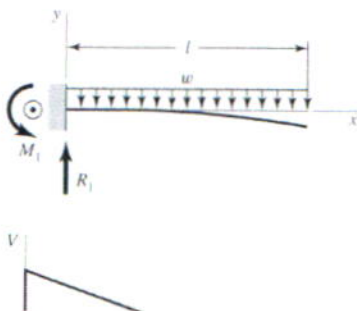
$$M_{AB} = F(x - a) \quad M_{BC} = 0$$

$$y_{AB} = \frac{Fx^2}{6EI} (x - 3a)$$

$$y_{BC} = \frac{Fa^2}{6EI} (a - 3x)$$

$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

3 Cantilever—uniform load



$$R_1 = wl \quad M_1 = \frac{wl^2}{2}$$

$$V = w(l - x) \quad M = -\frac{w}{2}(l - x)^2$$

$$y = \frac{wx^2}{24EI} (4lx - x^2 - 6l^2)$$

$$y_{\max} = -\frac{wl^4}{8EI}$$

Harold Rodriguez

Name: Harold Rodriguez

$$F = 800 \text{ lbs}$$
$$D = 2 \text{ in}$$
$$a = 70$$

$$W = 6 \text{ lb/in}$$
$$L = 100 \text{ in}$$
$$E = 30 \times 10^6 \text{ psi}$$

$$I = \frac{\pi(D^4)}{64}$$
$$I = \frac{\pi(2^4)}{64} = .785 \text{ in}^4$$

$$\textcircled{1} y_{\text{max}} = \frac{Fa^2}{6EI} (a-3L) - \frac{WL^4}{8EI}$$

$$= \frac{800 \times 70^2}{6 \times 30 \times 10^6 \times .785} (70 - 3(100)) - \frac{6 \times 100^4}{8 \times 30 \times 10^6 \times .785}$$

$$= -6.38 - 3.185$$

$$y_{\text{max}} = -9.565 \text{ in}$$

Deflection at B



$$\textcircled{2} y = \frac{Wx^2}{24EI} (4lx - x^2 - 6l^2)$$

$$y = \frac{6 \times 50^2}{24 \times 30 \times 10^6 \times .785} (4(100 \times 50) - 50^2 - 6(100)^2)$$

$$y_{\text{min}} = -6.255$$

$$D_{\text{min}} = -9.565 - -6.255$$

$$D_{\text{min}} = -3.31 \text{ in}$$

~~don't know~~

d is not given.

-14

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 7

TOPICS COVERED IN QUIZ 7

Failure Theory from Static Loading

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

MCEG3304: Machine Design I
Fall 2021

Quiz #7

Total points: 30

Date: 10/27/2021

Name:

A steel bar has yield strength of 120MPa. The stress states: $\sigma_x = 100MPa$, $\sigma_y = -50MPa$, $\tau_{yz} = 40MPa$.

- (1) Use maximum-shear-stress theory to determine the safety factor.
- (2) Use distortion-energy theory to determine the safety factor.

MCEG3043: Machine Design I

Fall 2021

Quiz #7

Total points: 30

Date: 10/27/2021

Name:

A steel bar has yield strength of 120MPa. The stress states: $\sigma_x = 100\text{MPa}$, $\sigma_y = -50\text{MPa}$, $\tau_{yz} = 40\text{MPa}$.

- (1) Use maximum-shear-stress theory to determine the safety factor.
- (2) Use distortion-energy theory to determine the safety factor.

(1) $\sigma_x = 100$ is one principal stress. (2)

$$\begin{aligned}\sigma_{A,B} &= \frac{\sigma_y + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_y - \sigma_z}{2}\right)^2 + \tau_{yz}^2} \\ &= \frac{-50}{2} \pm \sqrt{\left(\frac{-50}{2}\right)^2 + 40^2} \\ &= 22.17, -72.17 \quad (10)\end{aligned}$$

$$\therefore \sigma_1 = 100, \sigma_2 = 22.17, \sigma_3 = -72.17 \quad (4)$$

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{100 + 72.17}{2} = 86. \quad (2)$$

$$n = \frac{0.5 S_y}{\tau_{\max}} = \frac{0.5 \times 120}{86} = 0.698 \quad (2)$$

$$\begin{aligned}(2) \quad \sigma' &= \frac{1}{\sqrt{2}} \left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right]^{\frac{1}{2}} \\ &= \frac{1}{\sqrt{2}} \left[(100 + 50)^2 + (-50)^2 + (-100)^2 + 6(40^2) \right]^{\frac{1}{2}} \\ &= 149.3 \quad (8)\end{aligned}$$

$$n = \frac{S_y}{\sigma'} = \frac{120}{149.3} = 0.804 \quad (2)$$

ABOVE AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #7

Total points: 30

30/30

Date: 10/27/2021

Name: Temitayo Ojo

A steel bar has yield strength of 120MPa. The stress states: $\sigma_x = 100\text{MPa}$, $\sigma_y = -50\text{MPa}$, $\tau_{yz} = 40\text{MPa}$.

(1) Use maximum-shear-stress theory to determine the safety factor.

(2) Use distortion-energy theory to determine the safety factor.

$$\sigma_x = 100\text{MPa}, \sigma_y = -50\text{MPa}, \tau_{yz} = 40\text{MPa}.$$

$$\sigma_z = 0, \tau_{zx} = 0, \tau_{xy} = 0$$

$$\sigma_{A,B} = \frac{\sigma_y + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_y - \sigma_z}{2}\right)^2 + \tau_{yz}^2}$$

$$= \frac{-50 + 0}{2} \pm \sqrt{\left(\frac{-50 - 0}{2}\right)^2 + 40^2}$$

$$= 22.17, -72.17$$

$$\sigma_1 = 100, \sigma_2 = 22.17, \sigma_3 = -72.17$$

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{100 - (-72.17)}{2} = 86.085$$

①

MSS theory:

$$n = \frac{0.5(120)}{86.085} = 0.697$$

② DE theory

$$n = \frac{S_y}{\sigma_1}$$

$$\sigma_1 = \frac{1}{\sqrt{2}} \left((100 - (-50))^2 + (-50 - 0)^2 + (0 - 100)^2 + 6(0^2 + 40^2 + 0^2) \right)^{1/2}$$
$$= 149.33$$

$$n = \frac{120}{149.33} = 0.804$$

AVERAGE

MCEG3304: Machine Design I
Fall 2021

Safiatou
Zampaligre

Quiz #7

Total points: 30

Date: 10/27/2021

Name:

25/30

A steel bar has yield strength of 120MPa. The stress states: $\sigma_x = 100\text{MPa}$, $\sigma_y = -50\text{MPa}$, $\tau_{yz} = 40\text{MPa}$.

(1) Use maximum-shear-stress theory to determine the safety factor.

(2) Use distortion-energy theory to determine the safety factor.

$$\sigma_z = 0; \tau_{xy} = \tau_{yz} = 0$$

$$\sigma_{1/3} = \frac{\sigma_y + \sigma_z}{2} \pm \sqrt{\left(\frac{\sigma_y - \sigma_z}{2}\right)^2 + \tau_{yz}^2}$$

$$= \frac{-50 + 0}{2} \pm \sqrt{\left(\frac{-50 - 0}{2}\right)^2 + (40)^2}$$

$$= -25 \pm \sqrt{2225}$$

$$= 22.17 \text{ MPa}, -72.17 \text{ MPa} \checkmark$$

so $\sigma_1 = 100$
 $\sigma_1 = 100$
 $\sigma_2 = 22.17 \text{ MPa}$
 $\sigma_3 = -72.17 \text{ MPa}$ ~~4~~

1) Maximum shear-stress theory

$$n = \frac{s_y}{\sigma_1 - \sigma_3} = \frac{120}{22.17 + 72.17} = 1.27 > 1 \Rightarrow \text{safe}$$

$$n = 1.27 > 1 \Rightarrow \text{safe} \checkmark$$

②

85.44

$$n = \frac{s_y}{\sigma_y}$$

$$\sigma' = \frac{1}{\sqrt{2}} \left[\left((\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 \right) + 6 \left(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2 \right) \right]^{1/2}$$

$$\sigma' = \frac{1}{\sqrt{2}} \left[(100 + 50)^2 + (-50 - 0)^2 + (0 - 100)^2 + 6(0^2 + 40^2 + 0^2) \right]^{1/2}$$

$$\sigma' = 85.44 \text{ MPa}$$

$$n = \frac{120}{85.44} = 1.40 > 1 \rightarrow \text{Safe}$$

BELOW AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #7

Total points: 30

Date: 10/27/2021

Name: Gabriela Garay

A steel bar has yield strength of 120MPa. The stress

states: $\sigma_x = 100\text{MPa}$, $\sigma_y = -50\text{MPa}$, $\tau_{yz} = 40\text{MPa}$. $\sigma_z = 0$, $\tau_{zx} = 0$, $\tau_{xy} = 0$

(1) Use maximum-shear-stress theory to determine the safety factor.

(2) Use distortion-energy theory to determine the safety factor.

$$\frac{\sigma_x + \sigma_y}{2} \pm \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2} =$$

$$\frac{100 + (-50)}{2} \pm \sqrt{\left(\frac{100 - (-50)}{2}\right)^2 + 0^2} = 100, -50$$

$$\therefore \sigma_1 = 100, \sigma_2 = 0, \sigma_3 = -50$$

$$\tau_{\max} = \frac{\sigma_1 - \sigma_2}{2} \geq \frac{s_y}{2} \text{ or } \sigma_1 - \sigma_3 \geq s_y$$

$$\therefore n = \frac{0.5 \cdot s_y}{\tau_{\max}} = n = \frac{(0.5)(120)}{\frac{(\sigma_1 - \sigma_3)}{2}} = \frac{(0.5)(120)}{(100 - (-50))} = 0.4 < 1, \text{ fail}$$

(2)

$$\sigma = \left(\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2 \right)^{1/2}$$

$$\sigma = \left(100^2 - 100 \cdot (-50) + (-50)^2 + 3(40)^2 \right)^{1/2}$$

$$\sigma = \frac{1}{\sqrt{2}} \left[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2) \right]^{1/2}$$

$$\downarrow \left[100 - (-50) \right]^2 + (-50 - 100)^2 + (100 - 100)^2 + 6(-40^2 + 0^2 + 0^2) \right]^{1/2}$$

$$\sigma = \sqrt{45679.09} = 213.72 ?$$

$$n = \frac{s_y}{\sigma} = \frac{120}{213.72} = 0.56 > 1, \text{ fail}$$

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 8

TOPICS COVERED IN QUIZ 8

Chapter 8. Bolt Design

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

MCEG3304: Machine Design I
Fall 2021

Quiz #8

Total points: 30

Assigned date: 11/17/2021

Due date: 11/22/2021, 8AM

Name:

A 3-in Aluminum plate and a 2-in cast-iron plate are compressed with one bolt and nut. The bolt is 7/8 in-9 UNC.

- (1) Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.
- (2) Determine the bolt stiffness.
- (3) Determine the stiffness of the members.

Date: 11/17/2021

Name:

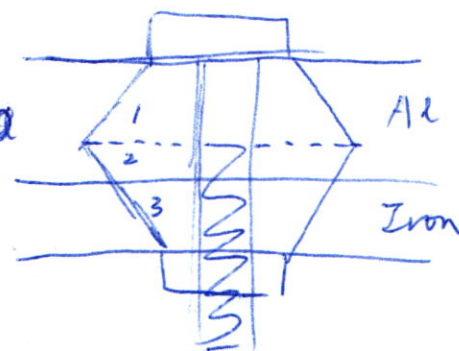
A 3-in Aluminum plate and a 2-in cast-iron plate are compressed with one bolt and nut. The bolt is 7/8 in-9 UNC.

- (1) Determine a suitable length for the bolt, rounded up to the nearest 1/4 in.
- (2) Determine the bolt stiffness.
- (3) Determine the stiffness of the members.

$$(1) \quad L > 3 + 2 + l_{\text{nut}} = 3 + 2 + \frac{3}{4} = 5.75 \text{ in} = \text{6 in}$$

$$l_{\text{nut}} = \frac{3}{4} \left(\frac{29}{32}, \frac{31}{84} \right) \text{ (Table A-31)}$$

$$L = 6 \text{ in}$$



$$(2) \quad l = 3 + 2 = 5 \text{ in}$$

$$l_T = 2d + \frac{1}{4} = 2 \times \frac{7}{8} + \frac{1}{4} = 2 \text{ in}$$

$$l_d = L - l_T = 6 - 2 = 4 \text{ in}$$

$$l_t = l - l_d = 5 - 4 = 1 \text{ in}$$

$$A_t = 0.462 \text{ in}^2$$

$$A_d = \pi \left(\frac{7}{8} \right)^2 / 4 = 0.601 \text{ in}^2$$

$$E = 30 \text{ Mpsi}$$

$$k_b = \frac{A_t A_d E}{A_t l_d + A_d l_t} = \frac{0.462 \times 0.601 \times 30}{0.462 \times 4 + 0.601 \times 1} = 3.4 \text{ Mlbf/in}$$

(3) Frustum one (Al)

(18) $E_1 = 10.4 \text{ Mpsi}$ ① $t_1 = 2.5 \text{ in}$ ① $D_1 = 1.5 \times d = 1.5 \times \frac{7}{8} = 1.3125 \text{ in}$ ①

$$k_1 = \frac{0.5774 \times \pi \times 10.4 \times 0.875}{\ln \frac{(1.155 \times 2.5 + 1.3125 - 0.875)(1.3125 + 0.875)}{(1.155 \times 2.5 + 1.3125 + 0.875)(1.3125 - 0.875)}} = 13.9 \text{ Mlbf/in.}$$

3.275862069

Frustum Two (Al)

$E_2 = 10.4 \text{ Mpsi}$ ① $t_2 = 0.5 \text{ in}$ ① $D_2 = 1.3125 + 2 \times 2 \times \tan 30^\circ = 3.6219 \text{ in}$ ②

$$k_2 = \frac{0.5774 \times \pi \times 10.4 \times 0.875}{\ln \frac{(1.155 \times 0.5 + 3.6219 - 0.875)(3.6219 + 0.875)}{(1.155 \times 0.5 + 3.6219 + 0.875)(3.6219 - 0.875)}} = 235.84 \text{ Mlbf/in.}$$

1.0725

Frustum Three (Cast-iron)

$E_3 = 14.5 \text{ Mpsi}$ ① $t_3 = 2 \text{ in}$ ① $D_3 = 1.5 \times 0.875 = 1.3125 \text{ in}$ ①

$$k_3 = \frac{0.5774 \times \pi \times 14.5 \times 0.875}{\ln \frac{(1.155 \times 2 + 1.3125 - 0.875)(1.3125 + 0.875)}{(1.155 \times 2 + 1.3125 + 0.875)(1.3125 - 0.875)}} = 20.61 \text{ Mlbf/in.}$$

3.054474708

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{13.9} + \frac{1}{235.84} + \frac{1}{20.61} \right)^{-1} = 7.99 \text{ Mlbf/in.}$$

ABOVE AVERAGE

29/30

A 3-in Aluminum plate and a 2-in cast-iron plate are compressed with one bolt and nut. The bolt is 7/8 in-9 UNC.

- (1) Determine a suitable length for the bolt, rounded up to the nearest 1/4 in.
- (2) Determine the bolt stiffness.
- (3) Determine the stiffness of the members.

1) Determine a suitable length

bolt = 7/8 in - 9 UNC

$d = 7/8$ in

$$l = t_1 + t_2$$

$$= 3 + 2$$

$$l = 5 \text{ in}$$

$$H = \frac{3}{4} \text{ from table A-31}$$

$$L = l + H$$

$$5 + \frac{3}{4} = 5.75 \text{ in} = 6 \text{ in} \checkmark$$

2) Determine the bolt stiffness

$$L = 2 \times d + \frac{1}{4}$$

$$= 2 \times \frac{7}{8} + \frac{1}{4}$$

$$= \frac{8}{4} = 2 \text{ in} \checkmark$$

$$(A_t) = 0.462 \text{ in}^2 \text{ from table 8-2}$$

$$A_d = \frac{\pi}{4} (d^2) = \frac{\pi}{4} \times \left(\frac{7}{8}\right)^2 = 0.601 \text{ in}^2 \checkmark$$

→ find length of unthreaded part

$$l_d = L - l_t = 6 - 2$$

$$l_d = 4 \text{ in} \quad \checkmark$$

→ The length of threaded part

$$l_t = l - l_d$$

$$= 5 - 4 \quad \checkmark$$

$$= 1 \text{ in}$$

$$E = 30 \text{ Mpsi} \quad -$$

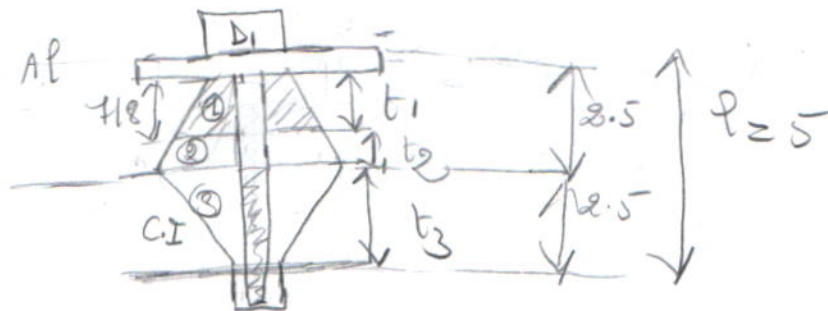
young's modulus of elasticity of aluminum (E) = ~~$10.4 \times 10^6 \text{ psi}$~~

young's modulus of elasticity of cast iron (E) = ~~$14.5 \times 10^6 \text{ psi}$~~

$$K_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.601 \text{ in}^2 \times 0.462 \text{ in}^2 \times 10.4 \times 10^6}{0.601 \times (1) + 0.462 \times 4 \text{ in}} =$$

$$K_b = 1.18 \text{ Mlb/in} \quad \checkmark$$

3) determine the member st



aluminum plate

$$t_2 = 2.5 \text{ in}$$

$$d_1 = 7/8$$

$$A_1 = 1.5 \times 7/8 = 1.3125 \text{ in}^2$$

$$E = 10.4 \times 10^6 \text{ psi}$$

$$\frac{L}{2} = 2.5$$

$$O_1 = 3 - 2.5 = 0.5 \text{ in}$$

$$t_1 =$$

$$k_1 = \frac{0.5774 \pi (E) d_1}{\ln \left(\frac{1.155 t_1 + D_1 - d_1 (D_1 + d_1)}{1.155 t_2 + D + d_1 (D - d_1)} \right)}$$

$$k_1 = \frac{0.5774 \pi \times 10.4 \times 10^6 \times 0.875}{\ln \left(\frac{1.155 \times 2.5 + 1.3125 - 0.875 (1.3125 + 0.875)}{1.155 \times 2.5 + 1.3125 + 0.875 (1.3125 - 0.875)} \right)} = 13.9182 \text{ MPa/in}$$

$$k_1 = 13.918 \text{ MPa/in}$$

$$t_2 = A - \frac{\phi}{2} \text{ in} = 3 - \left(\frac{5}{2}\right) = 0.5 \text{ in}$$

$$D_2 = 1.5 \times d + 2 (\phi - h) \tan 30^\circ = 1.5 \times 718 + 2(5 - 3) \tan 30^\circ = 3.621 \text{ in}$$

$$k_2 = \frac{0.5774 \pi (E) d}{\ln \left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)} \right)}$$

$$= \frac{0.5774 \pi \times 10.4 \times 10^6 \times 0.875}{\ln \left(\frac{(1.155 \times 0.5 + 3.621 - 0.875)(3.621 + 0.875)}{(1.155 \times 0.5 + 3.621 + 0.875)(3.621 - 0.875)} \right)}$$

$$= 235.814 \text{ MPa/in}$$

$$t_3 = 2 \text{ in}; \quad d = 718, \quad D_3 = 1.3125, \quad E = 14.5 \times 10^6 \text{ psi}$$

$$k_3 = \frac{0.5774 \pi (E) d}{\ln \left(\frac{1.155(t_3) + D_3 - d (D_3 + d)}{1.155(t_3) + D_3 + d (D_3 - d)} \right)}$$

$$= \frac{0.5774 \times \pi \times 14.5 \times 10^6 \times 0.875}{\ln \left(\frac{1.155 \times 2 + 1.3125 - 0.875 (1.3121 + 0.875)}{1.155 \times 2 + 1.3125 + 0.875 (1.3121 - 0.875)} \right)}$$

$$= 20.603 \text{ nPa/in}$$

$$k_{rm} = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1}$$

$$= \left(\frac{1}{13.918} + \frac{1}{2351814} + \frac{1}{20.603} \right)^{-1}$$

$$k_{rm} = 8.023 \text{ nPa/in}$$

AVERAGE

MCEG3304: Machine Design I
Fall 2021

Quiz #8

Total points: 30

Assigned date: 11/17/2021

Due date: 11/22/2021, 8AM

Name: Terintayo Ojo

26/30

A 3-in Aluminum plate and a 2-in cast-iron plate are compressed with one bolt and nut. The bolt is 7/8 in-9 UNC.

- (1) Determine a suitable length for the bolt, rounded up to the nearest 1/4 in.
- (2) Determine the bolt stiffness.
- (3) Determine the stiffness of the members.

① $A = \text{Aluminum plate thickness}$

$B = \text{Cast iron plate thickness}$

$L \geq l + H$

$l = 3\text{ in} + 2\text{ in} = 5\text{ in}$ ✓

Nut Height = 0.75 (Table A-31) ✓

$L \geq 5 + 0.75 = 5.75\text{ in}$

$L_T = 2(0.875\text{ in}) + \frac{1}{4} = 2\text{ in}$ ✓

$L = 6\text{ in} - 1$

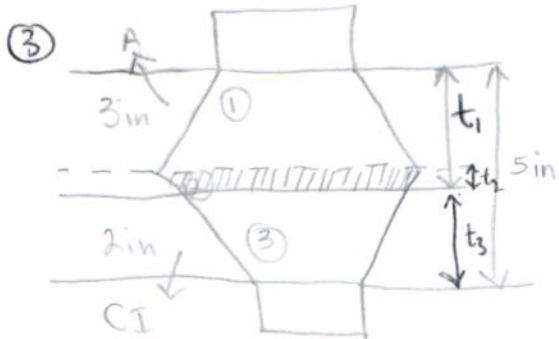
$l_d = L - L_T = 5.75\text{ in} - 2\text{ in} = 3.75\text{ in}$ ✓

$l_t = l - l_d = 5\text{ in} - 3.75\text{ in} = 1.25\text{ in}$ ✓

② $A_d = \frac{\pi \times d^2}{4} = \frac{\pi \times (7/8)^2}{4} = 0.6013\text{ in}^2$ ✓

$A_t = 0.462\text{ in}^2$ (Table 8-2) ✓

$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.6013 \times 0.462 \times 30 \times 10^6}{(0.6013)(1.25) + (0.462)(3.75)} = 3354910.884\text{ Mlb/in}$ ✓



$l = 5\text{ in}$

$l_2 = \frac{5}{2} = 2.5\text{ in}$

$E = 10.4\text{ mpsi}$ ✓

$d = 0.875\text{ in}$ ✓

$D_1 = 1.5 \times d = 1.3125$ ✓

$t_1 = 3\text{ in}$ ✓

$k_1 = 0.5774 \pi \times 10.4 \times 10^6 \times 0.875$

$\ln \frac{[1.155(\frac{3}{2}) + 1.3125 - 0.875](1.3125 + 0.875)}{[1.155(\frac{3}{2}) + 1.3125 + 0.875](1.3125 - 0.875)}$

$= 1.33233 \times 10^7\text{ lbf/in} = 13.3233 \times 10^6\text{ lbf/in}$ ✓

Frustum 2

$$E = 10.4 \text{ mpsi} \quad d = 0.875 \text{ in}$$

$$D_2 = D_1 + t_1 \times \tan 30^\circ \times 2 = 1.3125 + 3 \times \tan 30^\circ \times 2 = 4.7766 \text{ in}$$

$$t_2 = \cancel{t_2} - t_1 = \cancel{2.5 \text{ in}} - \cancel{3} = \cancel{0.5}$$

$$t_1 - \frac{t_2}{2} = 3 \text{ in} - 2.5 \text{ in} = 0.5 \text{ in}$$

$$k_2 = 0.5774 \pi (10.4 \times 10^6) (0.875)$$

$$\ln \frac{(1.155)(0.5) + (4.7766) - (0.875)(4.7766 + 0.875)}{(1.155)(0.5) + (4.7766 + (0.875)(4.7766 - 0.875))}$$

$$= 4.05157 \times 10^8 \text{ lbf/in} = 405.157 \times 10^6 \text{ lbf/in}$$

Frustum 3

$$E = 14.5 \text{ mpsi}$$

$$d = 0.875 \text{ in}$$

$$D_3 = 1.3125$$

$$t_3 = 2.5 \text{ in}$$

$$k_3 = 0.5774 \pi (14.5 \times 10^6) (0.875)$$

$$\ln \frac{(1.155)(2.5) + (1.3125) - (0.875)(1.3125 + 0.875)}{(1.155)(2.5) + (1.3125) + (0.875)(1.3125 - 0.875)}$$

$$= 19.3957 \times 10^6 \text{ lbf/in}$$

$$\frac{1}{k_m} = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)$$

$$k_m = \left(\frac{1}{13.3233 \times 10^6} + \frac{1}{405.157 \times 10^6} + \frac{1}{19.3957 \times 10^6} \right)^{-1}$$

$$k_m = 7.74698 \times 10^6 \text{ lbf/in}$$

BELOW AVERAGE

MCEG3304: Machine Design I
Fall 2021

Quiz #8

Total points: 30

Assigned date: 11/17/2021

Due date: 11/22/2021, 8AM

Name:

Shamirai Jola

A 3-in Aluminum plate and a 2-in cast-iron plate are compressed with one bolt and nut. The bolt is 7/8 in-9 UNC.

- (1) Determine a suitable length for the bolt, rounded up to the nearest 1/4 in.
- (2) Determine the bolt stiffness. k_b
- (3) Determine the stiffness of the members. k_m

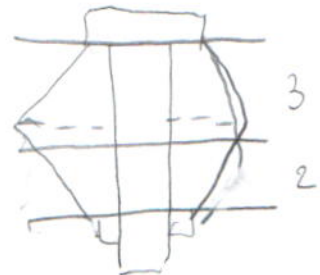
Given

3 in - Aluminum plate

7/8 $\rightarrow$ H $\rightarrow$ 3/4

2 in - Cast iron

7/8 in 9 UNC



1)

$$L = 1 + H_{nut}$$

$$L = 3 + 2 + .75 = 5.75 \quad L = 6 \text{ in}$$

2)

$$L = 5.75$$

$$L_t = L \leq 6 \rightarrow 2d + 1/4 \rightarrow 2(7/8) + 1/4 = 2$$

$$l_d = L - L_t \rightarrow 5.75 - 2 = 3.75$$

$$l_t = 1 - l_d \rightarrow 5 - 3.75 = 1.25$$

$$E = 10.4, 14.5$$

$$A_d = \frac{\pi d^2}{4} = \frac{\pi (7/8)^2}{4} = .601$$

$$A_t = 8-2 \rightarrow .462 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d}$$

$$k_b = \frac{(.601)(.462)(10.4)(10^6)}{(.601)(1.25) + (.462)(3.75)} = 1162631.02$$

3)

$$E_1 = 10.4$$

$$D_1 = 1.5(d) = 1.31$$

$$t_1 = .75 + 3 = 3.75 \quad t_1 = 2.5 \text{ in}$$

$$.5774 \pi (10.4 \times 10^6) (7/8)$$

$$\ln \left(\frac{(1.155)(3.75 + 1.31 - 7/8)(1.31 + 7/8)}{(1.155)(3.75 + 1.21 + 7/8)(1.31 - 7/8)} \right)$$

$$k_m = 13.05 \times 10^6$$

$$E_2 = 14.5$$

$$D_2 = 1.5(d) = 1.31$$

$$t_2 = .75 + 2 = 2.75 \quad t_2 = 2 \text{ in}$$

$$.5774 \pi (14.5 \times 10^6) (7/8)$$

$$\ln \left(\frac{(1.155)(2.75 + 1.31 - 7/8)(1.31 + 7/8)}{(1.155)(2.75 + 1.31 + 7/8)(1.31 - 7/8)} \right)$$

$$14.56 \times 10^6$$

$$\frac{1}{k_m} = \frac{1}{k_1} + \frac{1}{k_2}$$

$$\frac{1}{k_m} = \frac{1}{13.05 \times 10^6} + \frac{1}{14.56 \times 10^6}$$

$$k_m = 7.8 \times 10^6$$

**MACHINE DESIGN I
FALL 2021**

Title of Assignment: Quiz 9

TOPICS COVERED IN QUIZ 9

Chapter 8. Bolt Design

**MACHINE DESIGN I
FALL 2021**

**SAMPLE QUIZ QUESTIONS, SOLUTION,
AND GRADING SCHEME**

MCEG3304: Machine Design I

Fall 2021

Quiz #9

Total points: 30

Assigned Date: 11/22/2021 Due date: 11/24/2021

Name:

A total of 6 bolts are to be used to resist an external load. The stiffness of each bolt is $k_b=3.4$ Mlbf/in and the stiffness of the members is $k_m=8.6$ Mlbf/in per bolt. The bolts are 3/8 in-24 UNF SAE grade 4. The bolts are preloaded to 75 percent of the proof load. Assume the external load is equally distributed to all the bolts.

(a) Determine the maximum external load that can be applied to the entire joint if the overload safety factor is designed to be 1.5.

(b) With the maximum external load obtained from question (a), determine the service load stress in one bolt.

(c) Determine the maximum external load that can be applied to the entire joint without causing the members to come out of compression.

MCEG3043: Machine Design I

Fall 2021

Quiz #9

Total points: 30

Date: 11/22/2021

Name:

A total of 6 bolts are to be used to resist an external load. The stiffness of each bolt is $k_b = 3.4$ Mlbf/in and the stiffness of the members is $k_m = 8.6$ Mlbf/in per bolt. The bolts are 3/8 in-24 UNF SAE grade 4. The bolts are preloaded to 75 percent of the proof load. Assume the external load is equally distributed to all the bolts.

(a) Determine the maximum external load that can be applied to the entire joint if the overload safety factor is designed to be 1.5.

(b) With the maximum external load obtained from question (a), determine the service load stress in one bolt.

(c) Determine the maximum external load that can be applied to the entire joint without causing the members to come out of compression.

$$(a) \quad n_L = \frac{S_p A_t - F_i}{C P} = 1.5$$

$$(2) \quad S_p = 65 \text{ kpsi} \quad (\text{Table 8-9})$$

$$(2) \quad A_t = 0.0878 \text{ in}^2 \quad (\text{Table 8-2})$$

$$(2) \quad F_i = 0.75 \times S_p A_t = 0.75 \times 65 \times 0.0878 = 4.28 \text{ kip}$$

$$(2) \quad C = \frac{k_b}{k_b + k_m} = \frac{3.4}{3.4 + 8.6} = 0.283$$

$$(3) \quad P = P_{\max} / N$$

$$(4) \quad n_L = \frac{65 \times 0.0878 - 4.28}{0.283 \cdot P_{\max} / 6} = 1.5$$

$$P = \frac{5.04}{6} = 0.84 \text{ kip}$$

$$(3) \quad P_{\max} = 20.17 \text{ kip}$$

$$(b) \quad \sigma_b = \frac{F_i + C P}{A_t} = \frac{4.28 + 0.283 \times 20.17 / 6}{0.0878} = 59.58 \text{ kpsi}$$

(c)

$$(c) \quad n_0 = \frac{F_i}{p(1-c)} = 1 \quad (3)$$

$$\frac{4.28}{p(1-c)} = 1$$

$$(2) \quad \frac{P_{max}}{6} (1-0.283)$$

$$(2) \quad P_{max} = 35.82 \text{ kip}$$

ABOVE AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #9

Total points: 30

26/30

Assigned Date: 11/22/2021 Due date: 11/24/2021

Name: Safiatou Zampaligre

A total of 6 bolts are to be used to resist an external load. The stiffness of each bolt is $k_b = 3.4$ Mlbf/in and the stiffness of the members is $k_m = 8.6$ Mlbf/in per bolt. The bolts are 3/8 in-24 UNF SAE grade 4. The bolts are preloaded to 75 percent of the proof load. Assume the external load is equally distributed to all the bolts.

(a) Determine the maximum external load that can be applied to the entire joint if the overload safety factor is designed to be 1.5.

(b) With the maximum external load obtained from question (a), determine the service load stress in one bolt.

(c) Determine the maximum external load that can be applied to the entire joint without causing the members to come out of compression.

$k_b = 3.4 \text{ Mlbf/in}$
 $k_m = 8.6 \text{ Mlbf/in}$
 3/8 in-24 UNF SAE grade 4
 $M = 1.5$
 $N = 6 \text{ bolt}$

a) Maximum External Load.
 $A_t = 0.0878 \text{ in}^2$

$S_p = 65 \text{ Kpsi}$

* Preload

$F_i = 0.75 S_p A_t$
 $= 0.75 \times 65 \times 0.0878$
 $= 4.28 \text{ Kpsi}$

* stiffness constant

$C = \frac{k_b}{k_b + k_m} = \frac{3.4}{3.4 + 8.6} = 0.283$

$C = 0.283$

$N_L = \frac{S_p A_t - F_i}{C \cdot k_{max} / N}$

$\Rightarrow P_{max} = \left(\frac{S_p A_t - F_i}{N L_v} \right) \frac{N}{C}$

$P_{max} = \left(\frac{65 \times 0.0878 - 4.28}{1.5} \right) \times \frac{6}{0.283}$

$P_{max} = 20.17 \text{ Kpsi}$

b) determine the service stress on one bolt

$$\sigma_b = \frac{F_i + Cp}{A_t}$$

$$p = P_{total}/N$$

$$= \frac{4.28 + 0.283 \times (20 \times 1.7)}{0.0878}$$

$$\sigma_b = 1048.24 \text{ Kpsi}$$

c) determine the maximum External Load

$$m_0 = 1.5$$

$$m_0 = \frac{F_i}{(1-c)P_{max}/N} \Rightarrow P_{max} = \frac{F_i}{(1-c)m_0} \times N$$

$$P_{max} = \frac{4.28}{(1-0.283) \times 1.5} \times 6 = 23.88 \text{ Kpsi}$$

$$P_{max} = 23.88 \text{ Kpsi}$$

$$P_{max} = 7.16 \text{ Kpsi}$$

AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #9

Total points: 30

24/30

Assigned Date: 11/22/2021 Due date: 11/24/2021

Name: Mavika Lee

A total of 6 bolts are to be used to resist an external load. The stiffness of each bolt is $k_b = 3.4 \text{ Mlbf/in}$ and the stiffness of the members is $k_m = 8.6 \text{ Mlbf/in}$ per bolt. The bolts are 3/8 in-24 UNF SAE grade 4. The bolts are preloaded to 75 percent of the proof load. Assume the external load is equally distributed to all the bolts.

(a) Determine the maximum external load that can be applied to the entire joint if the overload safety factor is designed to be 1.5.

(b) With the maximum external load obtained from question (a), determine the service load stress in one bolt.

(c) Determine the maximum external load that can be applied to the entire joint without causing the members to come out of compression.

- 3/8 in - 24 UNF SAE grade 4

- From Table 8-2 $\Rightarrow A_t = .0878 \text{ in}^2$, $d = .375 \text{ in}$ } given

- from Table 8-9 $\Rightarrow S_p = 45 \text{ Kpsi}$

a.) $P_{\text{total}} = ?$

$$C = k_b / (k_b + k_m) \Rightarrow 3.4 / (3.4 + 8.6) = .283 \quad \checkmark$$

$$F_i = .75 F_p \Rightarrow F_p = A_t \cdot S_p \Rightarrow .0878 \times 45 \quad \checkmark$$

$$F_i = .75 \times .0878 \times 45 \Rightarrow = 4.2803 \text{ Kips} \quad \checkmark$$

$$n_s = S_p A_t - F_i / C P_{\text{total}} \Rightarrow 1.5 = \frac{45 \times .0878 - 4.2803}{.283 \times P_{\text{total}}} = 3.361 \text{ Kips}$$

$$P_{\text{total}} = P \cdot N$$

-3

b.) $P = 3.361 / 6 = .560 \text{ Kips/bolt}$

$$\sigma_b = F_b / A_t \Rightarrow = C P_{\text{total}} + F_i / A_t$$

$$= .283 \times .5601 + 4.2803 / .0878 = 50.550 \text{ Kpsi}$$

c.) $n_s = F_i / P(1-C)$

$$1 = \frac{4.2803}{P_{\text{total}}(1-.283)} \Rightarrow P_{\text{total}} = 5.97 \text{ Kips}$$

$$P_{\text{total}} = P \cdot N - 2$$

BELOW AVERAGE

MCEG3304: Machine Design I

Fall 2021

Quiz #9

Total points: 30

Assigned Date: 11/22/2021 Due date: 11/24/2021

Name: Joel Lewis

A total of 6 bolts are to be used to resist an external load. The stiffness of each bolt is $k_b = 3.4$ Mlbf/in and the stiffness of the members is $k_m = 8.6$ Mlbf/in per bolt. The bolts are 3/8 in-24 UNF SAE grade 4. The bolts are preloaded to 75 percent of the proof load. Assume the external load is equally distributed to all the bolts.

(a) Determine the maximum external load that can be applied to the entire joint if the overload safety factor is designed to be 1.5.

(b) With the maximum external load obtained from question (a), determine the service load stress in one bolt.

(c) Determine the maximum external load that can be applied to the entire joint without causing the members to come out of compression.

a.) $A_t = 0.0878 \text{ in}^2$, $d = 0.375 \text{ in}$, $S_p = 65 \text{ kpsi}$, $F_i = 0.75 F_p$

$$C = \frac{k_b}{k_b + k_m} = \frac{3.4}{3.4 + 8.6} = 0.283 \quad \checkmark$$

$$F_p = A_t S_p \quad F_i = 0.75 \cdot 0.0878 \cdot 65 = 4.2803 \text{ kips} \quad \checkmark$$

$$n_L = \frac{S_p A_t - F_i}{C P} = 1.5 = \frac{65 \cdot 0.0878 - 4.2803}{0.283 \cdot P_t}$$

$$P_t = 3.36 \text{ kips} \quad \checkmark \quad P_{\text{total}} = P \cdot N = 3$$

b.) $P = 3.36 \div 6 = 0.5601 \text{ kips for 1 bolt}$

$$\sigma_b = \frac{F_b}{A_t} \quad \sigma_b = \frac{F_i + C P}{A_t} = \frac{4.2803 + (0.283)(0.5601)}{0.0878} = 50.55 \text{ kpsi}$$

c.) $n_o = \frac{F_i}{P(1-C)} = \frac{4.2803}{3.36(1-0.283)} = 1.77 \text{ kpsi}$

$n_o = 1$ -5

MACHINE DESIGN I
Fall 2021

HOMEWORK 1

TOPICS COVERED IN HOMEWORK 1

Free-body Diagram

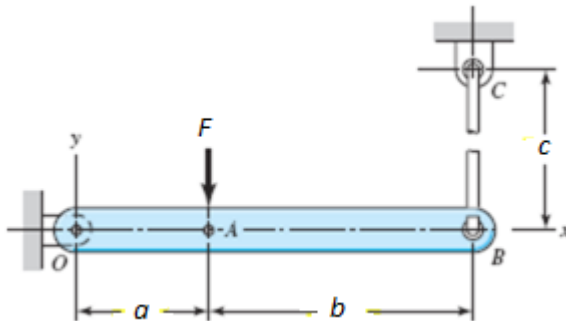
**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS, SOLUTION,
AND GRADING SCHEME**

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the given figure, $F = 200$ lbf, $a = 10$ in, $b = 8$ in, and $c = 11$ in.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break	3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Difficulty: Medium
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1.

Award: 8.00 points

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points O and B in lbf?

The force at point O is lbf.

The force at point B is lbf.

Explanation:

Taking the moment about point O ,

$$\sum M_o = R_B(18 \text{ in}) - Fa$$

$$\sum M_o = R_B(18 \text{ in}) - (200 \text{ lbf} \times 10 \text{ in})$$

Therefore, the reaction force at point B is

$$R_B = \frac{Fa}{18} = \frac{200 \text{ lbf} \times 10 \text{ in}}{18 \text{ in}} = 111.11 \text{ lbf}$$

The sum of equilibrium of forces is

$$\sum F = R_B - F + R_o$$

Therefore, the reaction force at point O is

$$R_o = -R_B + F = -111.11 \text{ lbf} + 200 \text{ lbf} = 88.89 \text{ lb}$$

The force at point O is 88.89 lbf.

The force at point B is 111.11 lbf.

References**Numeric Response**

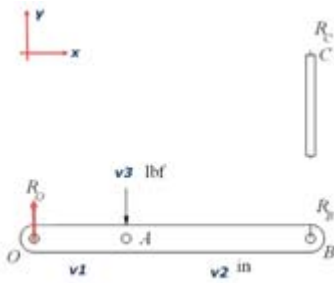
Problem 03.001.a - Difficulty: Medium
Determine the specified reaction force(s)

2.

Award: 9.00 points

Problem 03.001.b - Determine the specified reaction force(s)

What will be the direction of force R_B if the direction of force R_O is as shown in the figure? In the image provided below $v_3 = 200$ lbf, $v_1 = 10$ in, and $v_2 = 8$ in.



- ☐ The direction of the force R_B will be negative.
- ☒ The direction of the force R_B will be positive.

Taking the moment about point O ,

$$\sum M_o = R_B(18 \text{ in}) - (200 \text{ lbf} \times 10 \text{ in})$$

Therefore, the reaction force at point B is

$$R_B = \frac{F_a}{18} = \frac{200 \text{ lbf} \times 10 \text{ in}}{18 \text{ in}} = 111.11 \text{ lbf}$$

The force R_B must be positive otherwise the bar AB would rotate clockwise around point O .

The direction of the force R_B will be positive.

References

Multiple
Choice

Problem 03.001.b - Determine the specified reaction force(s)

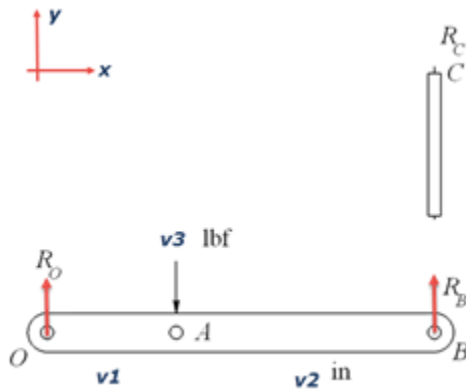
Difficulty: Medium

3.

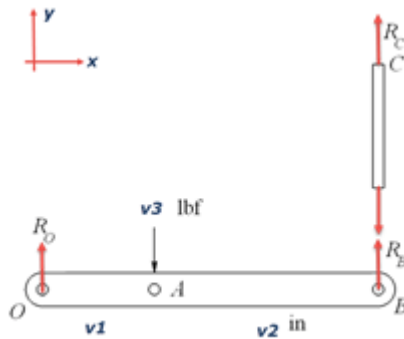
Award: 9.00 points

Problem 03.001.c - Determine the specified reaction force(s)

What is the direction of force R_C ? In the image provided below $v_3 = 200$ lbf, $v_1 = 10$ in, and $v_2 = 8$ in.



- ☐ The direction of force R_C is positive.
☐ The direction of force R_C is negative.



The sum of equilibrium of forces in the vertical direction is

$$\sum R_c - R_b = 0 \rightarrow R_c = R_b$$

Therefore, the direction of force R_C is positive.

The direction of force R_C is positive.

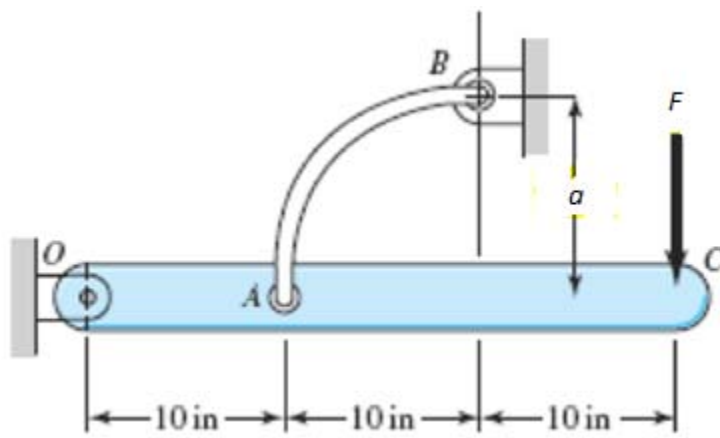
References

Multiple
Choice

Problem 03.001.c - Difficulty: Medium
Determine the
specified reaction
force(s)

**Problem 03.002 - Determine the specified reaction force(s) -
DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS**

Determine the magnitude and direction of each force. In the figure shown, $a = 9$ in and $F = 100$ lbf.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Problem 03.002 - Difficulty: Medium
Determine the
specified reaction
force(s) -
DEPENDENT
MULTI-PART
PROBLEM -
ASSIGN ALL
PARTS

4.

Award: 10.00 points

Problem 03.002.a - Determine the specified reaction force(s)

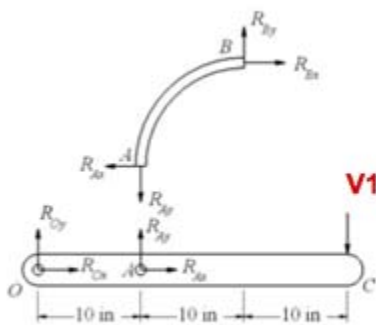
What are the values of forces at points A and B in lbf?

$$R_{Ax} = 333.3333 \pm 1\% \text{ lbf}$$

$$R_{Bx} = 333.3333 \pm 1\% \text{ lbf}$$

$$R_{Ay} = 300 \pm 1\% \text{ lbf}$$

$$R_{By} = 300 \pm 1\% \text{ lbf}$$

Explanation:

Consider body AB.

$$\Sigma F_x = 0, R_{Ax} = R_{Bx}$$

$$\Sigma F_y = 0, R_{Ay} = R_{By}$$

$$\Sigma M_B = 0, R_{Ay}(10 \text{ in}) - R_{Ax}(9 \text{ in}) = 0$$

Therefore,

$$R_{Ax} = \frac{(10 \text{ in})R_{Ay}}{9}$$

Now, consider body OAC.

Taking the moment about point O yields

$$R_{Ay}(10 \text{ in}) - F(30 \text{ in}) = \Sigma M_O = 0$$

$$R_{Ay} = 3F = 3 \times 100 \text{ lbf} = 300 \text{ lbf}$$

$$R_{By} = R_{Ay} = 300 \text{ lbf}$$

$$R_{Ax} = \frac{(10 \text{ in})R_{Ay}}{a} = \frac{(10 \text{ in})(300 \text{ lbf})}{9 \text{ in}} = 333.3333 \text{ lbf}$$

$$R_{Ax} = R_{Bx} = 333.3333 \text{ lbf}$$

$$R_{Ax} = 333.3333 \text{ lbf}$$

$$R_{Bx} = 333.3333 \text{ lbf}$$

$$R_{Ay} = 300 \text{ lbf}$$

$$R_{By} = 300 \text{ lbf}$$

References

Numeric Response

Problem 03.002.a - Difficulty: Medium
Determine the
specified reaction
force(s)

5.

Award: 10.00 points

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = -333.3333 \pm 1\% \text{ lbf}$$

$$R_{Oy} = -200 \pm 1\% \text{ lbf}$$

Explanation:

Sum of forces in the x direction

$$\Sigma F_x = 0 \rightarrow R_{Ox} = -R_{Ax} = -\frac{(10 \text{ in})R_{Ay}}{a}$$

$$R_{Ox} = -\frac{(10 \text{ in})(300 \text{ lbf})}{9 \text{ in}} = -333.3333 \text{ lbf}$$

Sum of forces in the y direction

$$\Sigma F_y = 0$$

$$R_{Oy} + R_{Ay} - F = 0$$

$$R_{Oy} = -R_{Ay} + F = -300 \text{ lbf} + 100 \text{ lbf} = -200 \text{ lbf}$$

$$R_{Ox} = -333.3333 \text{ lbf}$$

$$R_{Oy} = -200 \text{ lbf}$$

References**Numeric Response**

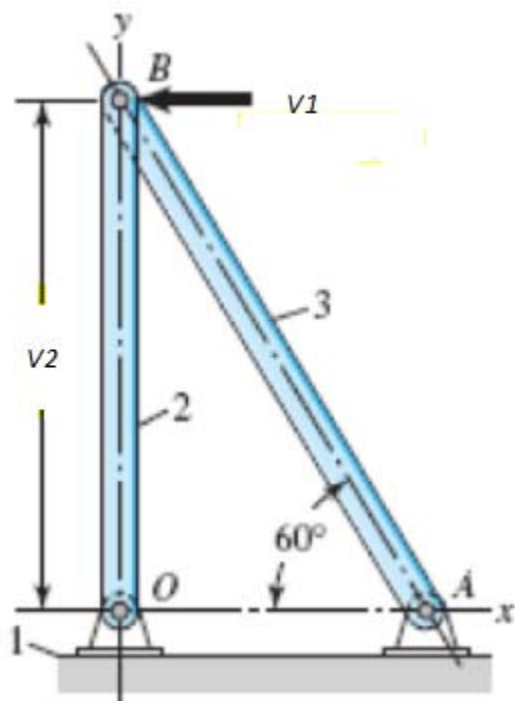
Problem 03.002.b - Difficulty: Medium
Determine the specified reaction force(s)

6.

Award: 20.00 points

Problem 03.003 - Determine the specified reaction force(s)

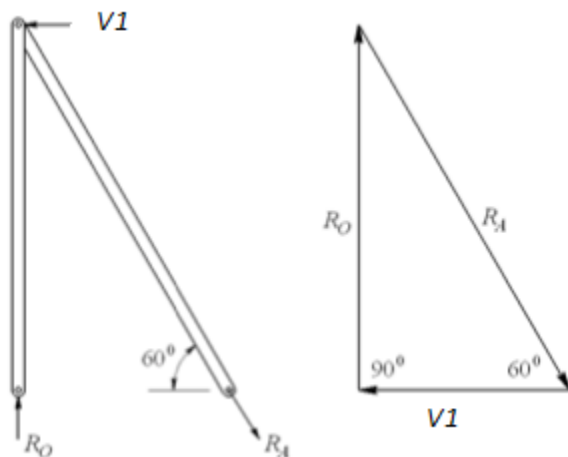
Determine the magnitude and direction of each force at points O and A . In the figure shown, $V_1 = 0.6$ kN and $V_2 = 0.9$ m. What are the resultant values of the forces at points O and A in kN?



$$R_O = 1.0392 \pm 1\% \text{ kN}$$

$$R_A = 1.2 \pm 1\% \text{ kN}$$

Explanation:



$$R_o = \frac{V_1}{\tan 30^\circ}$$

$$R_o = \frac{0.6 \text{ kN}}{0.5773} = 1.0392 \text{ kN}$$

$$R_A = \frac{V_1}{\sin 30^\circ}$$

$$R_A = \frac{0.6 \text{ kN}}{0.5} = 1.2 \text{ kN}$$

$$R_O = 1.0392 \text{ kN}$$

$$R_A = 1.2 \text{ kN}$$

References

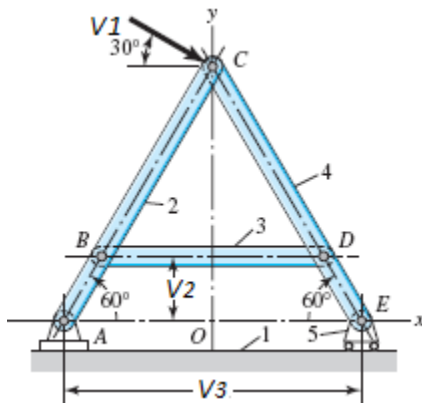
Numeric Response

Problem 03.003 -
Determine the
specified reaction
force(s)

Difficulty: Medium

Problem 03.004 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude of the forces. In the given figure, $V_1 = 800$ N, $V_2 = 2$ m, and $V_3 = 6$ m.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break	Problem 03.004 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Difficulty: Hard
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7.

Award: 8.00 points

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = -692.8 \pm 1\% \text{ N}$$

$$(R_{Ay})_2 = -400 \pm 1\% \text{ N}$$

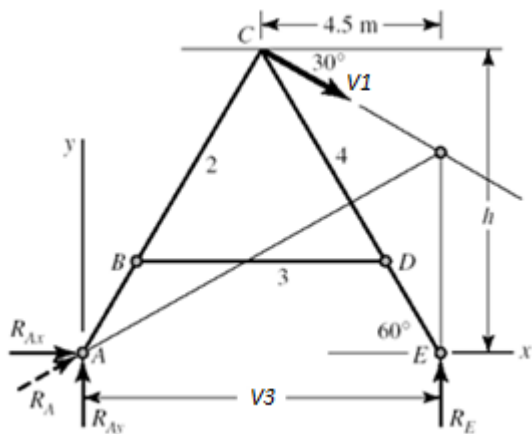
$$(R_{Bx})_2 = 750.9 \pm 1\% \text{ N}$$

$$(R_{By})_2 = 0 \pm 1\% \text{ N}$$

$$(R_{Cx})_2 = -58.1 \pm 1\% \text{ N}$$

$$(R_{Cy})_2 = 400 \pm 1\% \text{ N}$$

Explanation:



The reaction forces at point A are determined as follows:

$$\sum F_x = 0 \quad (R_{Ax})_2 + F \cos 30^\circ = 0$$

$$(R_{Ax})_2 = -F \cos 30^\circ = -V_1 \cos 30^\circ$$

$$(R_{Ax})_2 = -800 \text{ N} \times 0.866 = -692.8 \text{ N}$$

$$\sum F_y = 0 \quad (R_{Ay})_2 + V_1 - V_1 \sin 30^\circ = 0$$

$$(R_{Ay})_2 = V_1 \sin 30^\circ - V_1$$

$$(R_{Ay})_2 = (800 \text{ N} \times 0.5) - 800 \text{ N} = -400 \text{ N}$$

The reaction forces at point B are determined as follows:

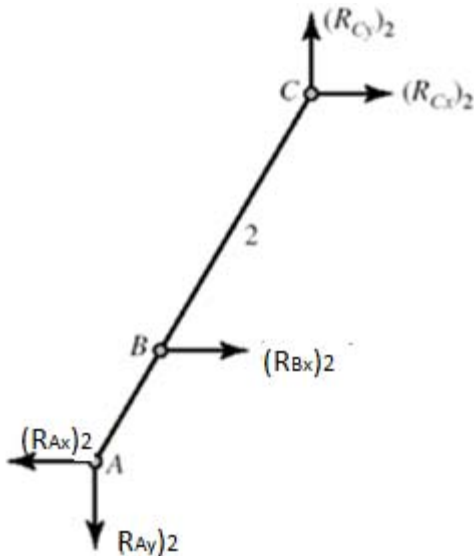
$$h = \frac{\frac{1}{2} \overline{AE}}{\tan 60^\circ} = \frac{\frac{1}{2} 6 \text{ m}}{0.57735} = 5.2 \text{ m}$$

$$\sum M_C = 0 \quad \frac{1}{2} \overline{AE} F - (h - V_2) (R_{Bx})_2 = 0$$

$$(R_{Bx})_2 = \frac{\frac{1}{2} \overline{AE} F}{(h-V_2)} = \frac{\frac{1}{2} V_1 V_3}{\frac{\frac{1}{2} V_3}{\tan 30^\circ} - V_2}$$

$$(R_{Bx})_2 = \frac{\frac{1}{2} (800 \text{ N} \times 6 \text{ m})}{\frac{\frac{1}{2} 6 \text{ m}}{0.57735} - 2 \text{ m}} = 750.9 \text{ N}$$

$$(R_{By})_2 = 0$$



The reaction forces at point C are determined as follows:

$$\sum F_x = 0 \quad (R_{Cx})_2 - F \cos 30^\circ + \frac{\frac{1}{2} \overline{AE} F}{(h-V_2)} = 0$$

$$(R_{Cx})_2 = V_1 \cos 30^\circ - \frac{\frac{1}{2} V_1 V_3}{\frac{\frac{1}{2} V_3}{\tan 30^\circ} - V_2}$$

$$(R_{Cx})_2 = (800 \text{ N} \times \cos 30^\circ) - \frac{\frac{1}{2} (800 \text{ N} \times 6 \text{ m})}{\frac{\frac{1}{2} 6 \text{ m}}{\tan 30^\circ} - 2 \text{ m}} = -58.1 \text{ N}$$

$$\sum F_y = 0 \quad (R_{Cy})_2 = -(V_1 \sin 30^\circ) + V_1 = -(800 \text{ N} \times 0.5) + 800 \text{ N} = 400 \text{ N}$$

$$(R_{Ax})_2 = -692.8 \text{ N}$$

$$(R_{Ay})_2 = -400 \text{ N}$$

$$(R_{Bx})_2 = 750.9 \text{ N}$$

$$(R_{By})_2 = 0 \text{ N}$$

$$(R_{Cx})_2 = -58.1 \text{ N}$$

$$(R_{Cy})_2 = 400 \text{ N}$$

References

**Numeric
Response**

Problem 03.004.a - Difficulty: Hard
Determine the
specified reaction
force(s)

8.

Award: 9.00 points

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = -750.9 \pm 1\% \text{ N}$$

$$(R_{By})_3 = 0 \pm 1\% \text{ N}$$

$$(R_{Dx})_3 = 750.9 \pm 1\% \text{ N}$$

$$(R_{Dy})_3 = 0 \pm 1\% \text{ N}$$

Explanation:

The reaction forces at points *B* and *D* in the x direction are equal and opposite to each other, and equal in magnitude to

$$(R_{Bx})_2 = 750.9 \text{ N}$$

Therefore,

$$(R_{Bx})_3 = -(R_{Bx})_2 = -750.9 \text{ N}$$

$$(R_{By})_3 = 0$$

$$(R_{Dx})_3 = (R_{Bx})_2 = 750.9 \text{ N}$$

$$(R_{Dy})_3 = 0$$

$$(R_{Bx})_3 = -750.9 \text{ N}$$

$$(R_{By})_3 = 0 \text{ N}$$

$$(R_{Dx})_3 = 750.9 \text{ N}$$

$$(R_{Dy})_3 = 0 \text{ N}$$

References**Numeric Response**

Problem 03.004.b - Difficulty: Hard
Determine the specified reaction force(s)

9.

Award: 8.00 points

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$$(R_{Cx})_4 = 750.9 \pm 1\% \text{ N}$$

$$(R_{Cy})_4 = -799.98 \pm 1\% \text{ N}$$

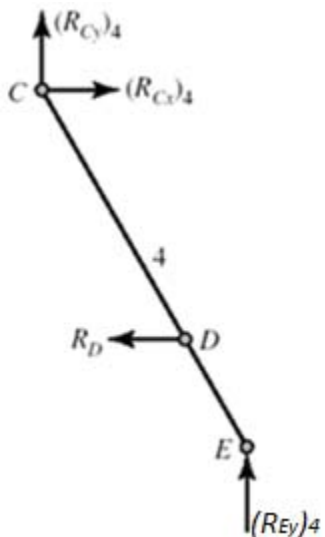
$$(R_{Dx})_4 = -750.9 \pm 1\% \text{ N}$$

$$(R_{Dy})_4 = 0 \pm 1\% \text{ N}$$

$$(R_{Ex})_4 = 0 \pm 1\% \text{ N}$$

$$(R_{Ey})_4 = 799.98 \pm 1\% \text{ N}$$

Explanation:



The reaction force at point D has the same magnitude as that of element 3, but with the opposite sign.

$$(R_{Dx})_4 = -(R_{Dx})_3 = -750.9 \text{ N}$$

$$(R_{Dy})_4 = 0$$

Because point E is on a horizontal roller,

$$(R_{Ex})_4 = 0$$

To find the vertical reaction at point E ,

$$h = \frac{\frac{1}{2} \overline{AE}}{\tan 30^\circ} = \frac{\frac{1}{2} 6 \text{ m}}{\tan 30^\circ} = 5.2 \text{ m}$$

$$\sum M_A = 0 \quad \overline{AE} (R_{Ey})_4 - h(F \cos 30^\circ) - \frac{1}{2} \overline{AE} (F \sin 30^\circ) = 0$$

$$(R_{Ey})_4 = \frac{h(F \cos 30^\circ) + \frac{1}{2} \overline{AE} (F \sin 30^\circ)}{\overline{AE}}$$

$$(R_{Ey})_4 = \frac{5.2 \text{ m}(800 \text{ N} \times \cos 30^\circ) + \frac{1}{2} 6 \text{ m}(800 \text{ N} \times \sin 30^\circ)}{6 \text{ m}} = 799.98 \text{ N}$$

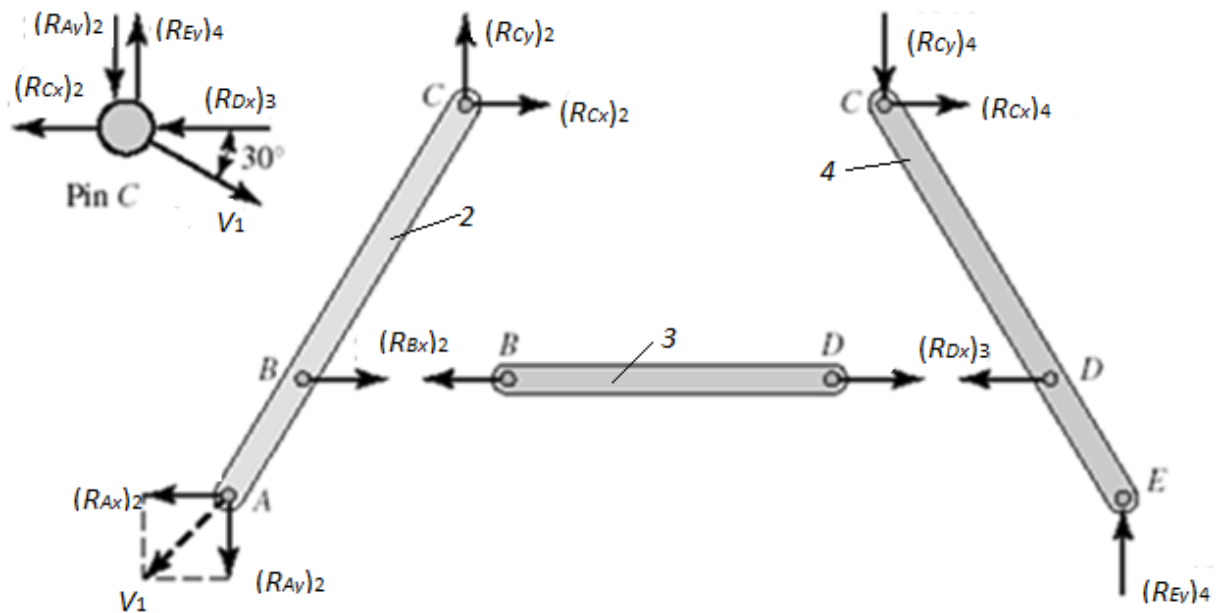
The reaction forces at point C are determined as follows:

$$\sum F_x = 0 \quad (R_{Cx})_4 + (R_{Dx})_4 = 0$$

$$(R_{Cx})_4 = -(R_{Dx})_4 = 750.9 \text{ N}$$

$$\sum F_y = 0 \quad (R_{Cy})_4 + (R_{Ey})_4 = 0$$

$$(R_{Cy})_4 = -(R_{Ey})_4 = -799.98 \text{ N}$$



$$(R_{Cx})_4 = 750.9 \text{ N}$$

$$(R_{Cy})_4 = -799.98 \text{ N}$$

$$(R_{Dx})_4 = -750.9 \text{ N}$$

$$(R_{Dy})_4 = 0 \text{ N}$$

$$(R_{Ex})_4 = 0 \text{ N}$$

$$(R_{Ey})_4 = 799.98 \text{ N}$$

References

Numeric Response

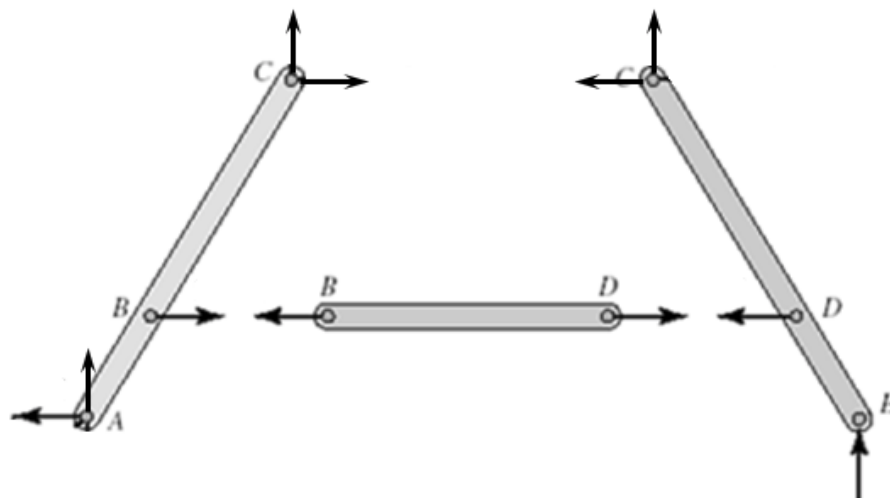
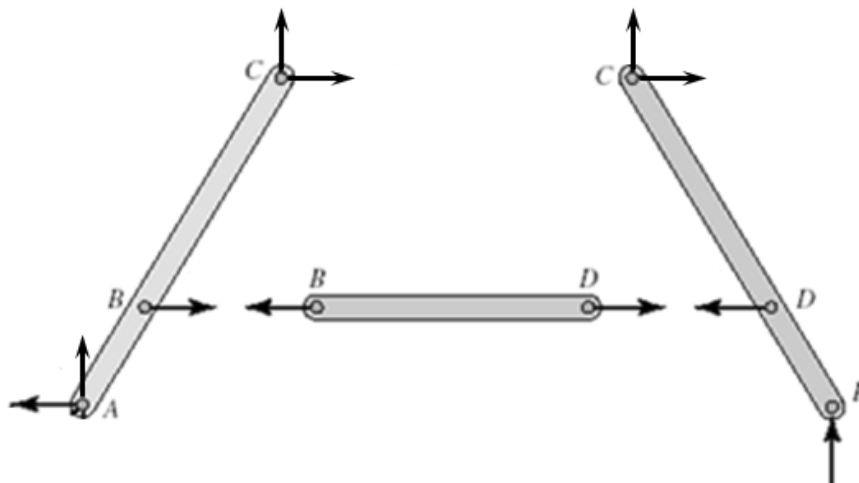
Problem 03.004.c - Difficulty: Hard
Determine the specified reaction force(s)

10.

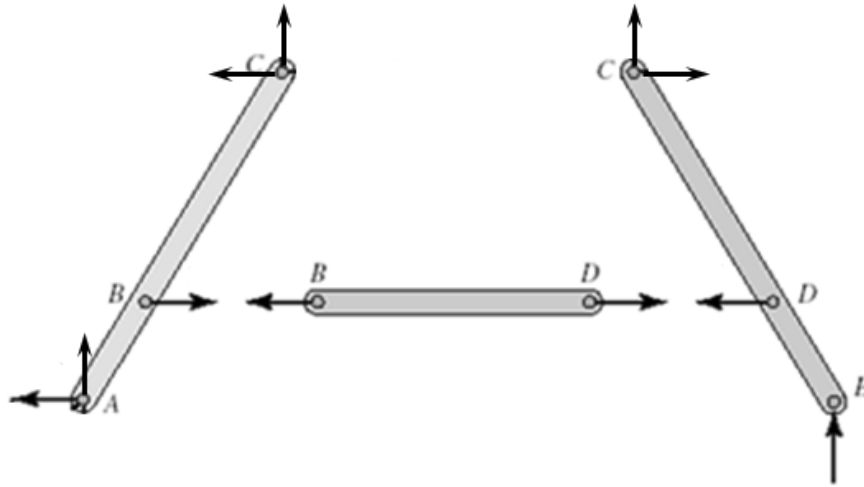
Award: 9.00 points

Problem 03.004.d - Determine the specified reaction force(s).

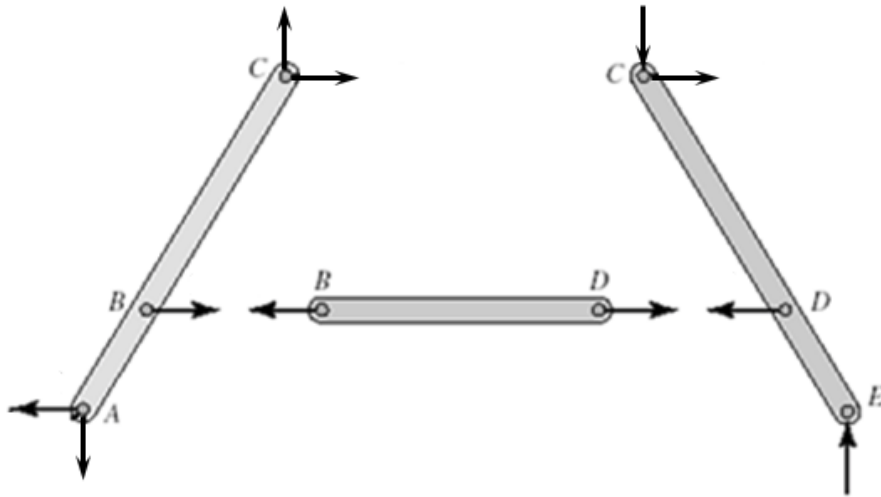
Select the correct set of free-body diagrams from the given choices.



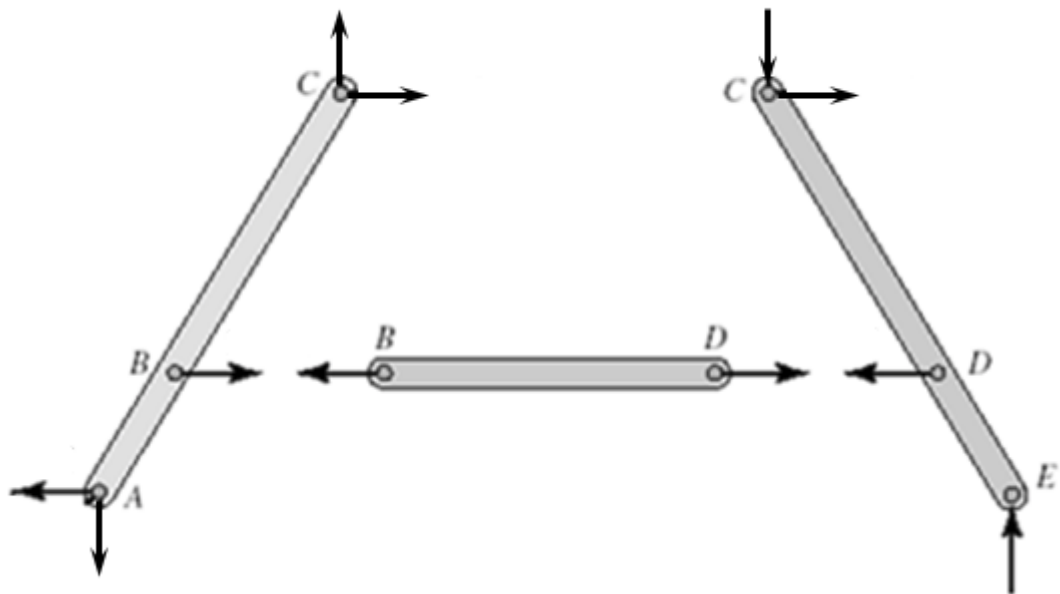
☐



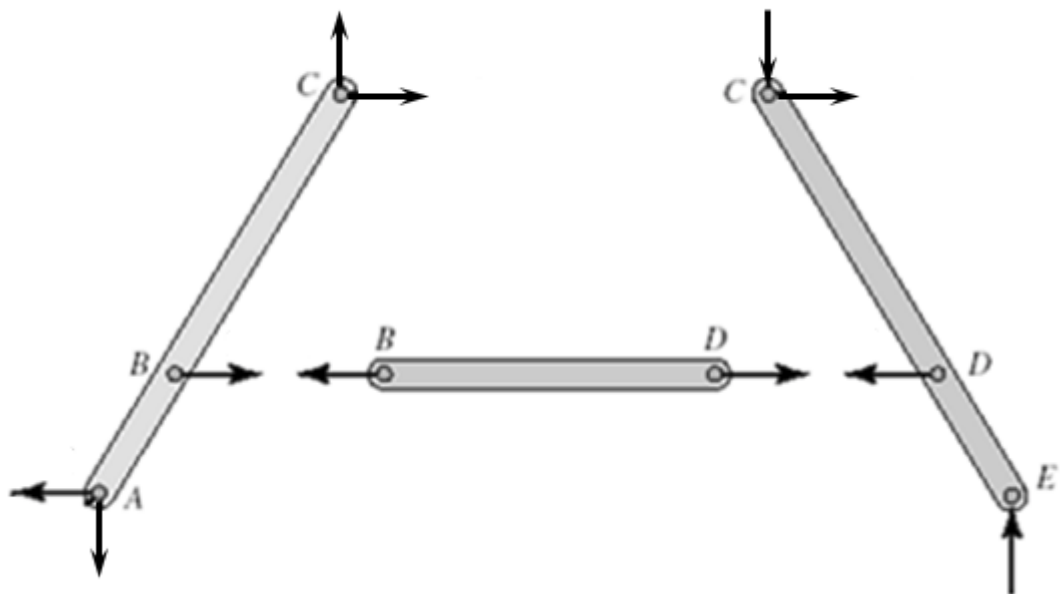
→ ☐



The correct free-body diagram is



The correct free-body diagram is



References

Multiple
Choice

Problem 03.004.d - Difficulty: Hard
Determine the
specified reaction
force(s).

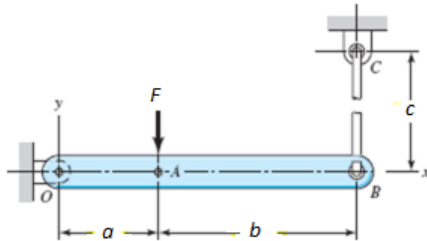
ABOVE AVERAGE

Dupre, Ty

Score: 100/100 Points 100 %

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the given figure, $F = 300$ lbf, $a = 1$ in, $b = 17$ in, and $c = 11$ in.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Section: 03.02 Shear Force and Bending Moments in Beams

1.

Award: 8 out of 8.00 points

 You received credit for this question in a previous attempt

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points *O* and *B* in lbf?

The force at point *O* is **283.33**  lbf.

The force at point *B* is **16.66**  lbf.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.001.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points *O* and *B* in lbf?

The force at point *O* is lbf.

The force at point *B* is lbf.

Explanation:

Taking the moment about point *O*,

$$\Sigma M_O = R_B(18 \text{ in}) - Fa$$

$$\Sigma M_O = R_B(18 \text{ in}) - (300 \text{ lbf} \times 1 \text{ in})$$

Therefore, the reaction force at point *B* is

$$R_B = Fa/18 = 300 \text{ lbf} \times 1 \text{ in} / 18 \text{ in} = 16.67 \text{ lbf}$$

The sum of equilibrium of forces is

$$\Sigma F = R_B - F + R_O$$

Therefore, the reaction force at point *O* is

$$R_O = -R_B + F = -16.67 \text{ lbf} + 300 \text{ lbf} = 283.33 \text{ lbf}$$

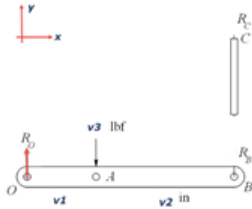
2.

Award: 9 out of 9.00 points

 You received credit for this question in a previous attempt

Problem 03.001.b - Determine the specified reaction force(s)

What will be the direction of force R_B if the direction of force R_O is as shown in the figure? In the image provided below $v_3 = 300$ lbf, $v_1 = 1$ in, and $v_2 = 17$ in.



- ☐ The direction of the force R_B will be negative.
- ✓ ☒ The direction of the force R_B will be positive.

Taking the moment about point O ,

$$\sum M_O = R_B(18 \text{ in}) - (300 \text{ lbf} \times 1 \text{ in})$$

Therefore, the reaction force at point B is

$$R_B = \frac{Fa}{l} = \frac{300 \text{ lbf} \times 1 \text{ in}}{18 \text{ in}} = 16.67 \text{ lbf}$$

The force R_B must be positive otherwise the bar AB would rotate clockwise around point O .

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.001.b -
Determine the
specified reaction
force(s)

Section: 03.02 Shear Force
and Bending Moments in
Beams

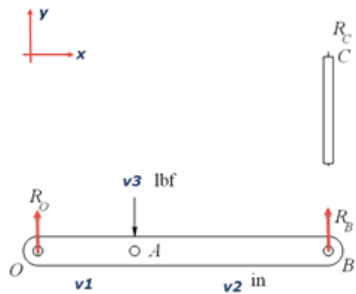
3.

Award: 9 out of 9.00 points

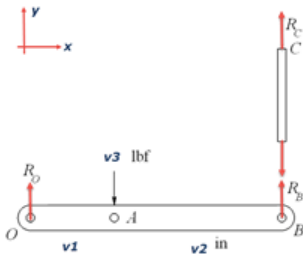
 You received credit for this question in a previous attempt

Problem 03.001.c - Determine the specified reaction force(s)

What is the direction of force R_C ? In the image provided below $v_3 = 300 \text{ lbf}$, $v_1 = 1 \text{ in}$, and $v_2 = 17 \text{ in}$.



- ☐ The direction of force R_C is negative.
- ✓ ☒ The direction of force R_C is positive.



The sum of equilibrium of forces in the vertical direction is

$$\sum R_C - R_B = 0 \rightarrow R_C = R_B$$

Therefore, the direction of force R_C is positive.

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

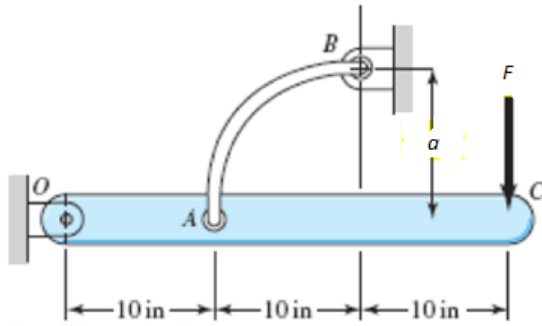
Problem 03.001.c -
Determine the
specified reaction
force(s)

Section: 03.02 Shear Force
and Bending Moments in
Beams

Problem 03.002 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the figure shown, $a = 12$ in and $F = 50$ lbf.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.002 -
Determine the
specified reaction
force(s) - DEPENDENT
MULTI-PART PROBLEM
- ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

4.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.002.a - Determine the specified reaction force(s)

What are the values of forces at points *A* and *B* in lbf?

$$R_{Ax} = \underline{125} \text{ lbf}$$

$$R_{Bx} = \underline{125} \text{ lbf}$$

$$R_{Ay} = \underline{150} \text{ lbf}$$

$$R_{By} = \underline{150} \text{ lbf}$$

References

Numeric Response

Chapter: 03 Load and Stress Analysis

**Problem 03.002.a -
Determine the
specified reaction
force(s)**

Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.002.a - Determine the specified reaction force(s)

What are the values of forces at points *A* and *B* in lbf?

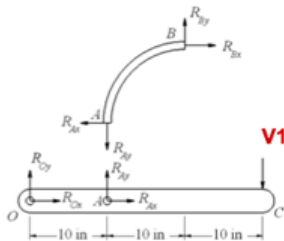
$$R_{Ax} = \underline{125 \pm 1\%} \text{ lbf}$$

$$R_{Bx} = \underline{125 \pm 1\%} \text{ lbf}$$

$$R_{Ay} = \underline{150 \pm 1\%} \text{ lbf}$$

$$R_{By} = \underline{150 \pm 1\%} \text{ lbf}$$

Explanation:

Consider body *AB*.

$$\Sigma F_x = 0, R_{Ax} = R_{Bx}$$

$$\Sigma F_y = 0, R_{Ay} = R_{By}$$

$$\Sigma M_B = 0, R_{Ay}(10 \text{ in}) - R_{Ax}(12 \text{ in}) = 0$$

Therefore,

$$R_{Ax} = (10 \text{ in})R_{Ay}^a$$

Now, consider body *OAC*.

Taking the moment about point *O* yields

$$R_{Ay}(10 \text{ in}) - F(30 \text{ in}) = \Sigma M_O = 0$$

$$R_{Ay} = 3F = 3 \times 50 \text{ lbf} = 150 \text{ lbf}$$

$$R_{By} = R_{Ay} = 150 \text{ lbf}$$

$$R_{Ax} = (10 \text{ in})R_{Ay}^a = (10 \text{ in})(150 \text{ lbf})12 \text{ in} = 125 \text{ lbf}$$

$$R_{Ax} = R_{Bx} = 125 \text{ lbf}$$

5.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = \underline{-125} \text{ lbf}$$

$$R_{Oy} = \underline{-100} \text{ lbf}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.002.b - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = \underline{-125 \pm 1\%} \text{ lbf}$$

$$R_{Oy} = \underline{-100 \pm 1\%} \text{ lbf}$$

Explanation:

Sum of forces in the x direction

$$\Sigma F_x = 0 \rightarrow R_{Ox} = -R_{Ax} = -(10 \text{ in})R_{Ay}^a$$

$$R_{Ox} = -(10 \text{ in})(150 \text{ lbf})12 \text{ in} = -125 \text{ lbf}$$

Sum of forces in the y direction

$$\Sigma F_y = 0$$

$$R_{Oy} + R_{Ay} - F = 0$$

$$R_{Oy} = -R_{Ay} + F = -150 \text{ lbf} + 50 \text{ lbf} = -100 \text{ lbf}$$

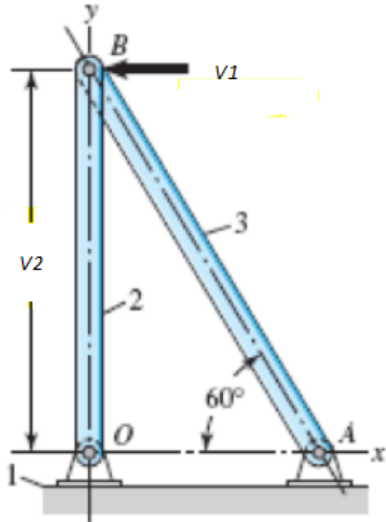
6.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.003 - Determine the specified reaction force(s)

Determine the magnitude and direction of each force at points O and A. In the figure shown, $V_1 = 0.8 \text{ kN}$ and $V_2 = 0.8 \text{ m}$. What are the resultant values of the forces at points O and A in kN?



$$R_O = \underline{1.385} \text{ kN}$$

$$R_A = \underline{1.6} \text{ kN}$$

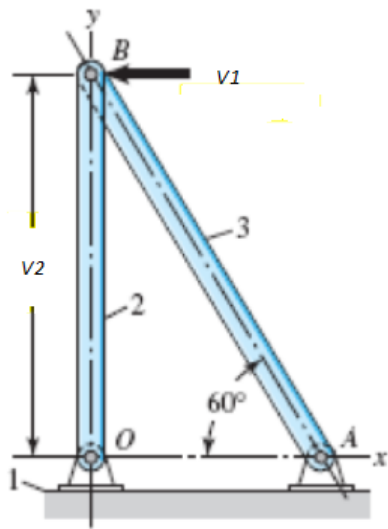
References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.003 - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.003 - Determine the specified reaction force(s)

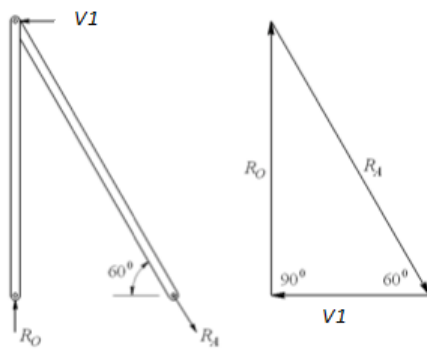
Determine the magnitude and direction of each force at points O and A. In the figure shown, $V_1 = 0.8 \text{ kN}$ and $V_2 = 0.8 \text{ m}$. What are the resultant values of the forces at points O and A in kN?



$$R_O = 1.3856 \pm 1\% \text{ kN}$$

$$R_A = 1.6 \pm 1\% \text{ kN}$$

Explanation:



$$R_O = V_1 \tan 30^\circ$$

$$R_O = 0.8 \text{ kN} \cdot 0.5773 = 1.3856 \text{ kN}$$

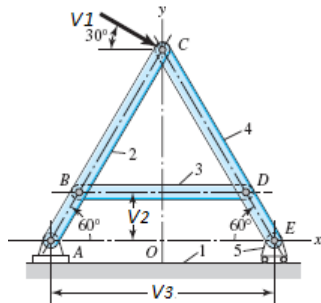
$$R_A = V_1 \sin 30^\circ$$

$$R_A = 0.8 \text{ kN} \cdot 0.5 = 1.6 \text{ kN}$$

Problem 03.004 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude of the forces. In the given figure, $V_1 = 600$ N, $V_2 = 0.9$ m, and $V_3 = 17$ m.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.004 -
Determine the
specified reaction
force(s) - DEPENDENT
MULTI-PART PROBLEM
- ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

7.

Award: 8 out of 8.00 points

 You received credit for this question in a previous attempt

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = \underline{-519.62} \text{ N}$$

$$(R_{Ay})_2 = \underline{-300} \text{ N}$$

$$(R_{Bx})_2 = \underline{368.97} \text{ N}$$

$$(R_{By})_2 = \underline{0} \text{ N}$$

$$(R_{Cx})_2 = \underline{150.65} \text{ N}$$

$$(R_{Cy})_2 = \underline{300} \text{ N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = \underline{-519.6 \pm 1\%} \text{ N}$$

$$(R_{Ay})_2 = \underline{-300 \pm 1\%} \text{ N}$$

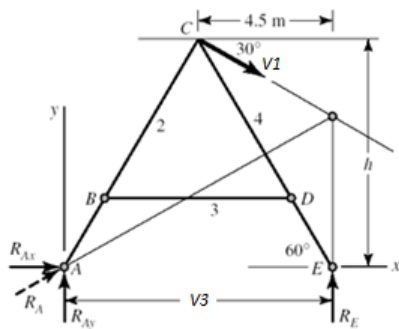
$$(R_{Bx})_2 = \underline{368.97 \pm 1\%} \text{ N}$$

$$(R_{By})_2 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Cx})_2 = \underline{150.63 \pm 1\%} \text{ N}$$

$$(R_{Cy})_2 = \underline{300 \pm 1\%} \text{ N}$$

Explanation:



The reaction forces at point A are determined as follows:

$$\sum F_x = 0 \quad (R_{Ax})_2 + F \cos 30^\circ = 0$$

$$(R_{Ax})_2 = -F \cos 30^\circ = -V_1 \cos 30^\circ$$

$$(R_{Ax})_2 = -600 \text{ N} \times 0.866 = -519.6 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Ay})_2 + V_1 - V_1 \sin 30^\circ = 0$$

$$(R_{Ay})_2 = V_1 \sin 30^\circ - V_1$$

$$(R_{Ay})_2 = (600 \text{ N} \times 0.5) - 600 \text{ N} = -300 \text{ N}$$

The reaction forces at point *B* are determined as follows:

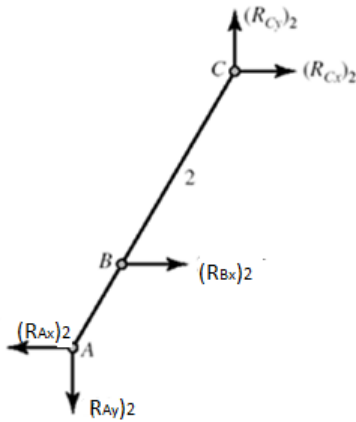
$$h = 12 \text{ m} \tan 30^\circ = 12 \times 0.57735 = 14.72 \text{ m}$$

$$\Sigma M_C = 0 \quad 12 \text{ m} \times 600 \text{ N} - (h - V_2)(R_{Bx})_2 = 0$$

$$(R_{Bx})_2 = 12 \text{ m} \times 600 \text{ N} / (h - V_2) = 12 \times 600 \text{ N} / (14.72 \text{ m} - 0.9 \text{ m}) = 368.97 \text{ N}$$

$$(R_{Bx})_2 = 12(600 \text{ N} \times 17 \text{ m}) / (12 \times 0.57735 - 0.9 \text{ m}) = 368.97 \text{ N}$$

$$(R_{By})_2 = 0$$



The reaction forces at point *C* are determined as follows:

$$\Sigma F_x = 0 \quad (R_{Cx})_2 - F \cos 30^\circ + 12 \text{ m} \times 600 \text{ N} / (h - V_2) = 0$$

$$(R_{Cx})_2 = F \cos 30^\circ - 12 \text{ m} \times 600 \text{ N} / (h - V_2)$$

$$(R_{Cx})_2 = (600 \text{ N} \times \cos 30^\circ) - 12(600 \text{ N} \times 17 \text{ m}) / (12 \times 0.57735 - 0.9 \text{ m}) = 150.63 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Cy})_2 = -(V_1 \sin 30^\circ) + V_1 = -(600 \text{ N} \times 0.5) + 600 \text{ N} = 300 \text{ N}$$

8.

Award: 9 out of 9.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = \underline{-368.97} \text{ N}$$

$$(R_{By})_3 = \underline{0} \text{ N}$$

$$(R_{Dx})_3 = \underline{368.97} \text{ N}$$

$$(R_{Dy})_3 = \underline{0} \text{ N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.b - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = \underline{-368.97 \pm 1\%} \text{ N}$$

$$(R_{By})_3 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Dx})_3 = \underline{368.97 \pm 1\%} \text{ N}$$

$$(R_{Dy})_3 = \underline{0 \pm 1\%} \text{ N}$$

Explanation:

The reaction forces at points *B* and *D* in the x direction are equal and opposite to each other, and equal in magnitude to

$$(R_{Bx})_2 = 368.97 \text{ N}$$

Therefore,

$$(R_{Bx})_3 = -(R_{Bx})_2 = -368.97 \text{ N}$$

$$(R_{By})_3 = 0$$

$$(R_{Dx})_3 = (R_{Bx})_2 = 368.97 \text{ N}$$

$$(R_{Dy})_3 = 0$$

9.

Award: 8 out of 8.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$$(R_{Cx})_4 = \underline{368.97} \text{ N}$$

$$(R_{Cy})_4 = \underline{-599.99} \text{ N}$$

$$(R_{Dx})_4 = \underline{-368.97} \text{ N}$$

$$(R_{Dy})_4 = \underline{0} \text{ N}$$

$$(R_{Ex})_4 = \underline{0} \text{ N}$$

$$(R_{Ey})_4 = \underline{599.99} \text{ N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.c - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$$(R_{Cx})_4 = \underline{368.97 \pm 1\%} \text{ N}$$

$$(R_{Cy})_4 = \underline{-599.99 \pm 1\%} \text{ N}$$

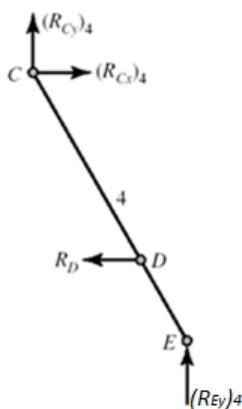
$$(R_{Dx})_4 = \underline{-368.97 \pm 1\%} \text{ N}$$

$$(R_{Dy})_4 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Ex})_4 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Ey})_4 = \underline{599.99 \pm 1\%} \text{ N}$$

Explanation:



The reaction force at point D has the same magnitude as that of element 3, but with the opposite sign.

$$(R_{Dx})_4 = -(R_{Dx})_3 = -368.97 \text{ N}$$

$$(R_{Dy})_4 = 0$$

Because point E is on a horizontal roller,

$$(R_{Ex})_4 = 0$$

To find the vertical reaction at point E ,

$$h = 12 \tan 30^\circ = 12 \times 0.577 = 6.92 \text{ m}$$

$$\sum M_A = 0 \quad -AE(R_{Ey})_4 - h(F \cos 30^\circ) - 12(F \sin 30^\circ) = 0$$

$$(R_{Ey})_4 = \frac{h(F \cos 30^\circ) + 12(F \sin 30^\circ)}{AE}$$

$$(R_{Ey})_4 = \frac{6.92 \text{ m}(600 \text{ N} \times \cos 30^\circ) + 12 \text{ m}(600 \text{ N} \times \sin 30^\circ)}{17 \text{ m}} = 599.99 \text{ N}$$

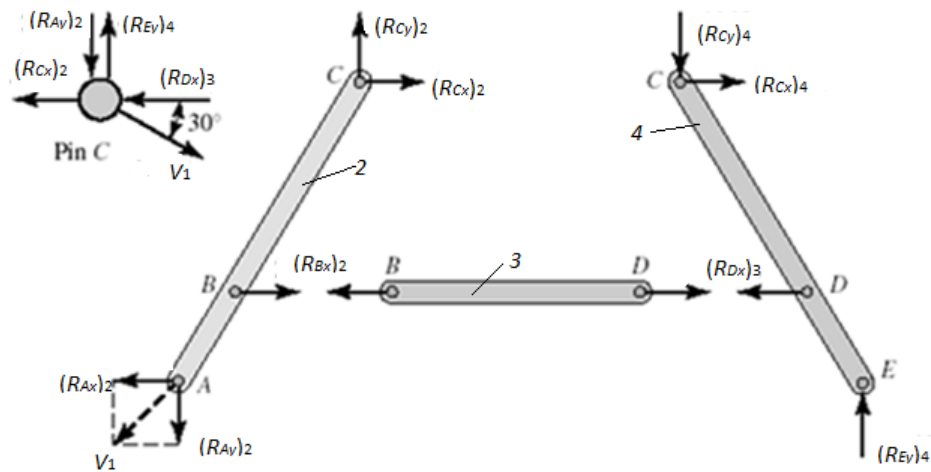
The reaction forces at point C are determined as follows:

$$\sum F_x = 0 \quad (R_{Cx})_4 + (R_{Dx})_4 = 0$$

$$(R_{Cx})_4 = -(R_{Dx})_4 = 368.97 \text{ N}$$

$$\sum F_y = 0 \quad (R_{Cy})_4 + (R_{Ey})_4 = 0$$

$$(R_{Cy})_4 = -(R_{Ey})_4 = -599.99 \text{ N}$$



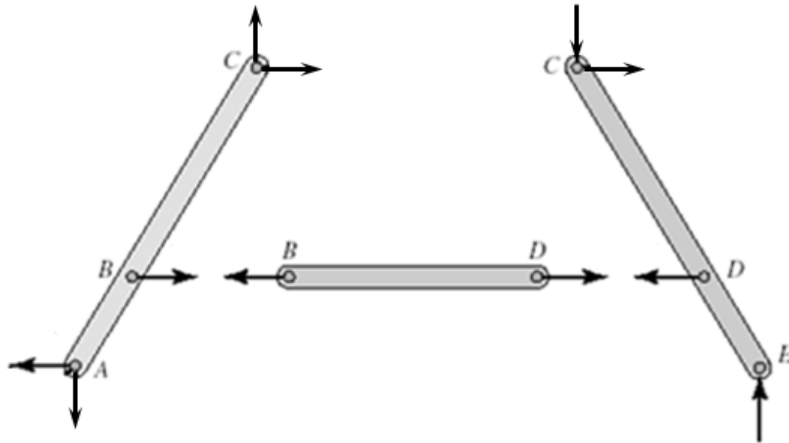
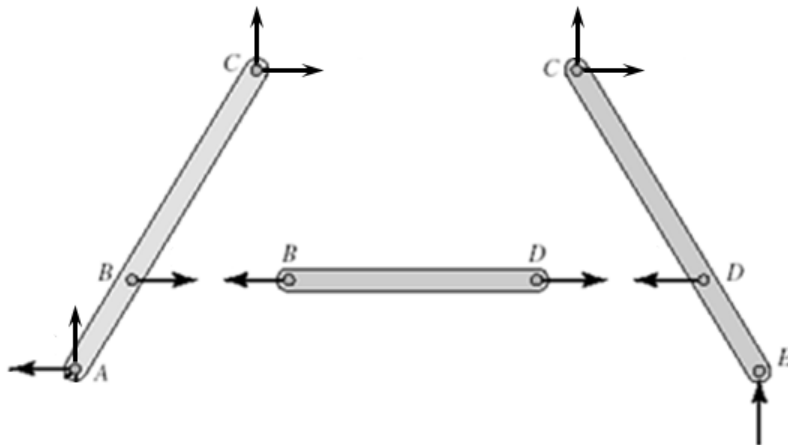
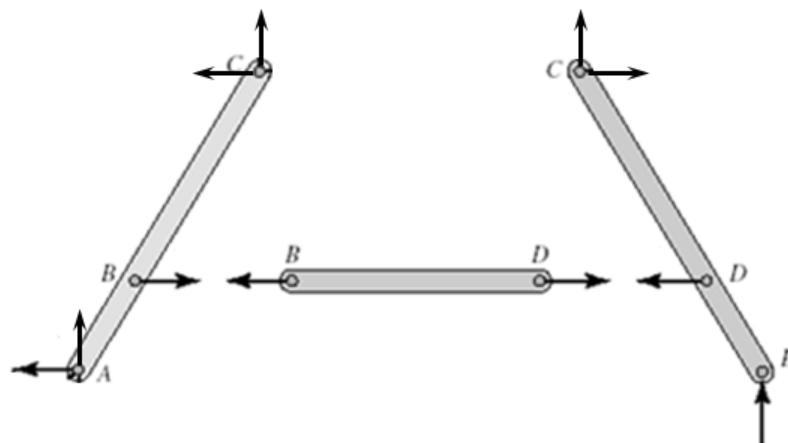
10.

Award: 9 out of 9.00 points

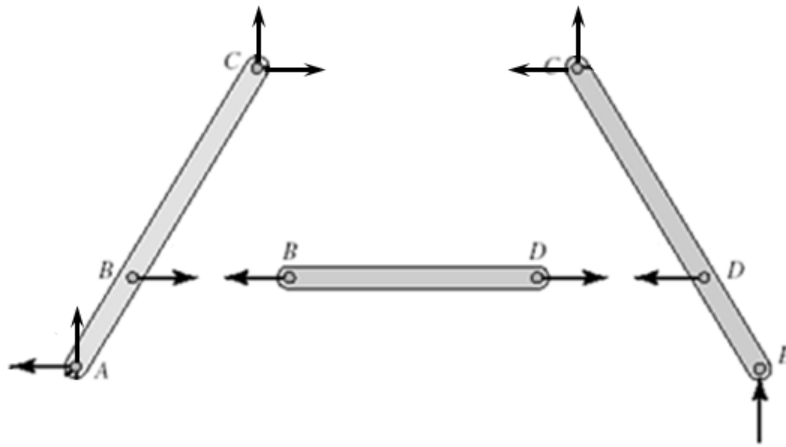
 You received credit for this question in a previous attempt

Problem 03.004.d - Determine the specified reaction force(s).

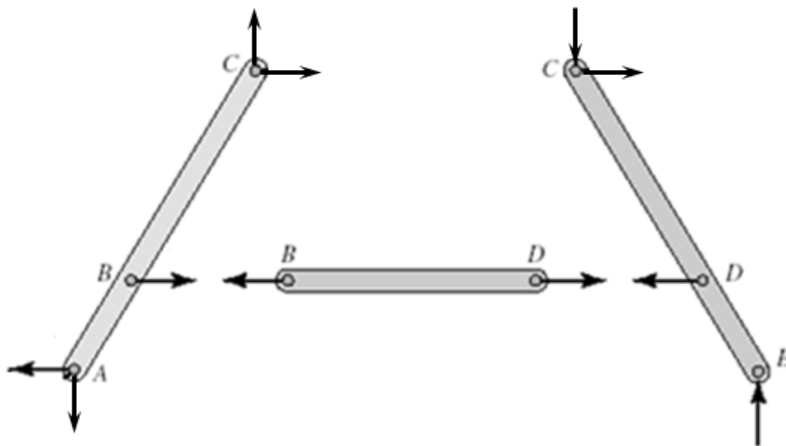
Select the correct set of free-body diagrams from the given choices.

✓ ☒☐☐

○



The correct free-body diagram is



References

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.004.d - Determine the specified reaction force(s).

Section: 03.02 Shear Force and Bending Moments in Beams

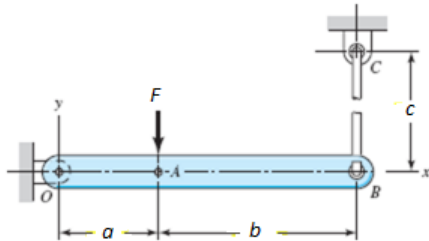
AVERAGE

Newke, christopher

Score: 88.33/100 Points 88.33 %

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the given figure, $F = 200$ lbf, $a = 11$ in, $b = 7$ in, and $c = 11$ in.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Section: 03.02 Shear Force and Bending Moments in Beams

1.

Award: 8 out of 8.00 points

 You received credit for this question in a previous attempt

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points *O* and *B* in lbf?

The force at point *O* is **77.8**  lbf.
 The force at point *B* is **122.2**  lbf.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.001.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points *O* and *B* in lbf?

The force at point *O* is **77.78 ± 1.01** lbf.
 The force at point *B* is **122.22 ± 1.01** lbf.

Explanation:Taking the moment about point *O*,

$$\sum M_O = R_B(18 \text{ in}) - Fa$$

$$\sum M_O = R_B(18 \text{ in}) - (200 \text{ lbf} \times 11 \text{ in})$$

Therefore, the reaction force at point *B* is

$$R_B = \frac{Fa}{18} = \frac{200 \text{ lbf} \times 11 \text{ in}}{18 \text{ in}} = 122.22 \text{ lbf}$$

The sum of equilibrium of forces is

$$\sum F = R_B - F + R_O$$

Therefore, the reaction force at point *O* is

$$R_O = -R_B + F = -122.22 \text{ lbf} + 200 \text{ lbf} = 77.78 \text{ lbf}$$

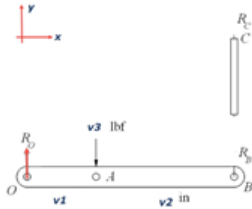
2.

Award: 9 out of 9.00 points

 You received credit for this question in a previous attempt

Problem 03.001.b - Determine the specified reaction force(s)

What will be the direction of force R_B if the direction of force R_O is as shown in the figure? In the image provided below $v_3 = 200$ lbf, $v_1 = 11$ in, and $v_2 = 7$ in.



- ☐ The direction of the force R_B will be negative.
- ✓ ☒ The direction of the force R_B will be positive.

Taking the moment about point O ,

$$\sum M_O = R_B(18 \text{ in}) - (200 \text{ lbf} \times 11 \text{ in})$$

Therefore, the reaction force at point B is

$$R_B = \frac{Fa}{18} = \frac{200 \text{ lbf} \times 11 \text{ in}}{18 \text{ in}} = 122.22 \text{ lbf}$$

The force R_B must be positive otherwise the bar AB would rotate clockwise around point O .

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.001.b -
Determine the
specified reaction
force(s)

Section: 03.02 Shear Force
and Bending Moments in
Beams

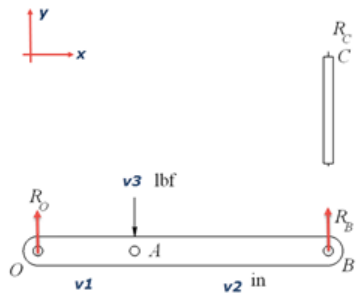
3.

Award: 9 out of 9.00 points

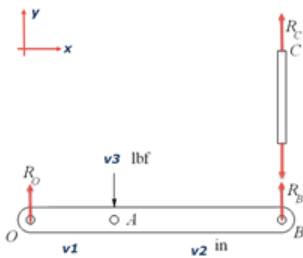
 You received credit for this question in a previous attempt

Problem 03.001.c - Determine the specified reaction force(s)

What is the direction of force R_C ? In the image provided below $v_3 = 200$ lbf, $v_1 = 11$ in, and $v_2 = 7$ in.



- ✓ ☒ The direction of force R_C is positive.
- ☐ The direction of force R_C is negative.



The sum of equilibrium of forces in the vertical direction is

$$\sum R_c - R_b = 0 \rightarrow R_c = R_b$$

Therefore, the direction of force R_C is positive.

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

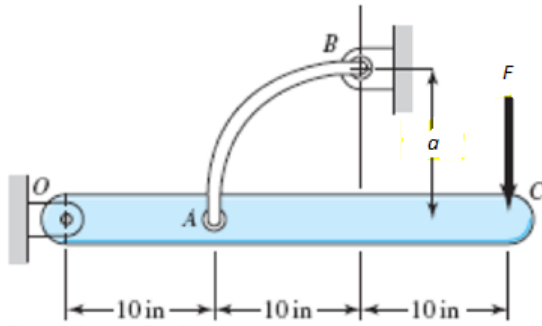
Problem 03.001.c -
Determine the
specified reaction
force(s)

Section: 03.02 Shear Force
and Bending Moments in
Beams

Problem 03.002 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the figure shown, $a = 11$ in and $F = 100$ lbf.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.002 -
Determine the
specified reaction
force(s) - DEPENDENT
MULTI-PART PROBLEM
- ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

4.

Award: 10 out of 10.00 points

 You received credit for this question in a previous attempt

Problem 03.002.a - Determine the specified reaction force(s)

What are the values of forces at points A and B in lbf?

$$R_{Ax} = \underline{272.7} \text{ lbf}$$

$$R_{Bx} = \underline{272.7} \text{ lbf}$$

$$R_{Ay} = \underline{300} \text{ lbf}$$

$$R_{By} = \underline{300} \text{ lbf}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.002.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.002.a - Determine the specified reaction force(s)

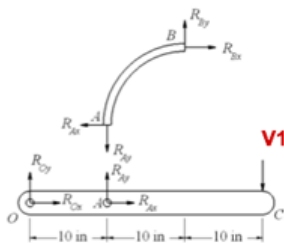
What are the values of forces at points A and B in lbf?

$$R_{Ax} = \underline{272.7273 \pm 1\%} \text{ lbf}$$

$$R_{Bx} = \underline{272.7273 \pm 1\%} \text{ lbf}$$

$$R_{Ay} = \underline{300 \pm 1\%} \text{ lbf}$$

$$R_{By} = \underline{300 \pm 1\%} \text{ lbf}$$

Explanation:Consider body AB .

$$\Sigma F_x = 0, R_{Ax} = R_{Bx}$$

$$\Sigma F_y = 0, R_{Ay} = R_{By}$$

$$\Sigma M_B = 0, R_{Ay}(10 \text{ in}) - R_{Ax}(10 \text{ in}) = 0$$

Therefore,

$$R_{Ax} = \frac{(10 \text{ in})R_{Ay}}{a}$$

Now, consider body OAC.

Taking the moment about point O yields

$$R_{Ay}(10 \text{ in}) - F(30 \text{ in}) = \Sigma M_O = 0$$

$$R_{Ay} = 3F = 3 \times 100 \text{ lbf} = 300 \text{ lbf}$$

$$R_{By} = R_{Ay} = 300 \text{ lbf}$$

$$R_{Ax} = \frac{(10 \text{ in})R_{Ay}}{a} = \frac{(10 \text{ in})(300 \text{ lbf})}{11 \text{ in}} = 272.72727 \text{ lbf}$$

$$R_{Ax} = R_{Bx} = 272.7273 \text{ lbf}$$

5.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = \underline{-272.7273} \text{ lbf}$$

$$R_{Oy} = \underline{-200} \text{ lbf}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.002.b - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = \underline{-272.7273 \pm 1\%} \text{ lbf}$$

$$R_{Oy} = \underline{-200 \pm 1\%} \text{ lbf}$$

Explanation:

Sum of forces in the x direction

$$\Sigma F_x = 0 \rightarrow R_{Ox} = -R_{Ax} = -\frac{(10 \text{ in})R_{Ay}}{a}$$

$$R_{Ox} = -\frac{(10 \text{ in})(300 \text{ lbf})}{11 \text{ in}} = -272.7273 \text{ lbf}$$

Sum of forces in the y direction

$$\Sigma F_y = 0$$

$$R_{Oy} + R_{Ay} - F = 0$$

$$R_{Oy} = -R_{Ay} + F = -300 \text{ lbf} + 100 \text{ lbf} = -200 \text{ lbf}$$

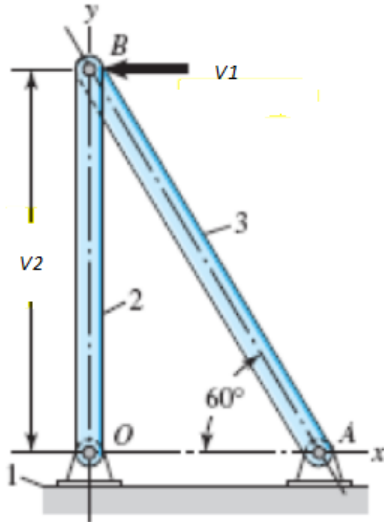
6.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.003 - Determine the specified reaction force(s)

Determine the magnitude and direction of each force at points O and A. In the figure shown, $V_1 = 0.5 \text{ kN}$ and $V_2 = 0.8 \text{ m}$. What are the resultant values of the forces at points O and A in kN?



$$R_O = \underline{0.866} \text{ kN}$$

$$R_A = \underline{1} \text{ kN}$$

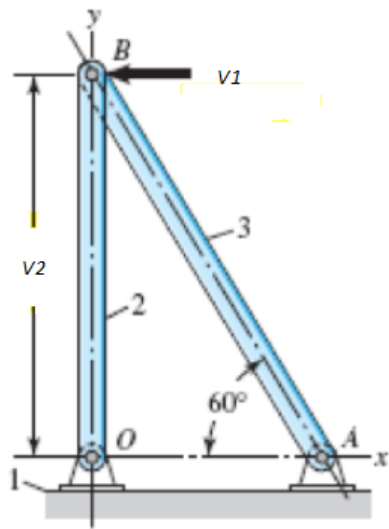
References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.003 - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.003 - Determine the specified reaction force(s)

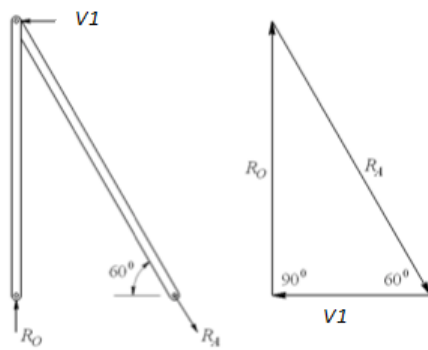
Determine the magnitude and direction of each force at points O and A. In the figure shown, $V_1 = 0.5 \text{ kN}$ and $V_2 = 0.8 \text{ m}$. What are the resultant values of the forces at points O and A in kN?



$$R_O = 0.866 \pm 1\% \text{ kN}$$

$$R_A = 1 \pm 1\% \text{ kN}$$

Explanation:



$$R_O = \frac{V_1}{\tan 30^\circ}$$

$$R_O = \frac{0.5 \text{ kN}}{0.5773} = 0.866 \text{ kN}$$

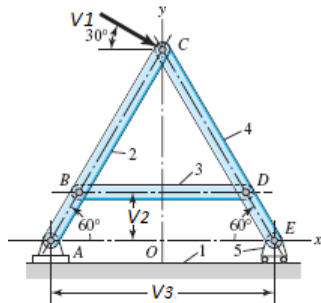
$$R_A = \frac{V_1}{\sin 30^\circ}$$

$$R_A = \frac{0.5 \text{ kN}}{0.5} = 1 \text{ kN}$$

Problem 03.004 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude of the forces. In the given figure, $V_1 = 300$ N, $V_2 = 1.5$ m, and $V_3 = 14$ m.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.004 -
Determine the
specified reaction
force(s) - DEPENDENT
MULTI-PART PROBLEM
- ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

7.

Award: 8 out of 8.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = \underline{-259.81} \text{ N}$$

$$(R_{Ay})_2 = \underline{-150} \text{ N}$$

$$(R_{Bx})_2 = \underline{197.66} \text{ N}$$

$$(R_{By})_2 = \underline{0} \text{ N}$$

$$(R_{Cx})_2 = \underline{62.14} \text{ N}$$

$$(R_{Cy})_2 = \underline{150} \text{ N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = \underline{-259.8 \pm 1\%} \text{ N}$$

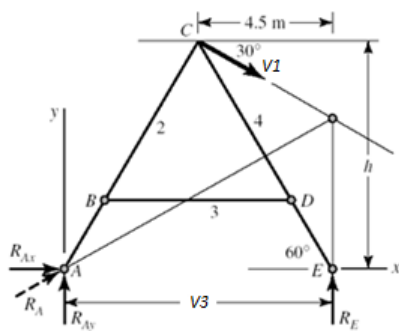
$$(R_{Ay})_2 = \underline{-150 \pm 1\%} \text{ N}$$

$$(R_{Bx})_2 = \underline{197.66 \pm 1\%} \text{ N}$$

$$(R_{By})_2 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Cx})_2 = \underline{62.14 \pm 1\%} \text{ N}$$

$$(R_{Cy})_2 = \underline{150 \pm 1\%} \text{ N}$$

Explanation:

The reaction forces at point A are determined as follows:

$$\sum F_x = 0 \quad (R_{Ax})_2 + F \cos 30^\circ = 0$$

$$(R_{Ax})_2 = -F \cos 30^\circ = -V_1 \cos 30^\circ$$

$$(R_{Ax})_2 = -300 \text{ N} \times 0.866 = -259.8 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Ay})_2 + V_1 - V_1 \sin 30^\circ = 0$$

$$(R_{Ay})_2 = V_1 \sin 30^\circ - V_1$$

$$(R_{Ay})_2 = (300 \text{ N} \times 0.5) - 300 \text{ N} = -150 \text{ N}$$

The reaction forces at point B are determined as follows:

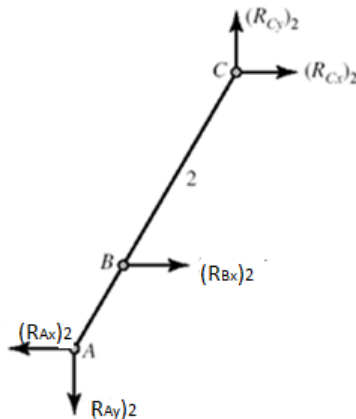
$$h = \frac{\frac{1}{2}AE}{\tan 30^\circ} = \frac{\frac{1}{2}14 \text{ m}}{0.57735} = 12.12 \text{ m}$$

$$\Sigma M_C = 0 \quad \frac{1}{2}AE F - (h - V_2)(R_{Bx})_2 = 0$$

$$(R_{Bx})_2 = \frac{\frac{1}{2}AE F}{(h - V_2)} = \frac{\frac{1}{2}V_1 V_3}{\frac{\frac{1}{2}V_3}{\tan 30^\circ} - V_2}$$

$$(R_{Bx})_2 = \frac{\frac{1}{2}(300 \text{ N} \times 14 \text{ m})}{\frac{\frac{1}{2}14 \text{ m}}{0.57735} - 1.5 \text{ m}} = 197.66 \text{ N}$$

$$(R_{By})_2 = 0$$



The reaction forces at point C are determined as follows:

$$\Sigma F_x = 0 \quad (R_{Cx})_2 - F \cos 30^\circ + \frac{\frac{1}{2}AE F}{(h - V_2)} = 0$$

$$(R_{Cx})_2 = V_1 \cos 30^\circ - \frac{\frac{1}{2}V_1 V_3}{\frac{\frac{1}{2}V_3}{\tan 30^\circ} - V_2}$$

$$(R_{Cx})_2 = (300 \text{ N} \times \cos 30^\circ) - \frac{\frac{1}{2}(300 \text{ N} \times 14 \text{ m})}{\frac{\frac{1}{2}14 \text{ m}}{\tan 30^\circ} - 1.5 \text{ m}} = 62.14 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Cy})_2 = - (V_1 \sin 30^\circ) + V_1 = - (300 \text{ N} \times 0.5) + 300 \text{ N} = 150 \text{ N}$$

8.

Award: 0 out of 9.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = \underline{86.6} \times \text{N}$$

$$(R_{By})_3 = \underline{150} \times \text{N}$$

$$(R_{Dx})_3 = \underline{673.2} \times \text{N}$$

$$(R_{Dy})_3 = \underline{-300} \times \text{N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.b - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = \underline{-197.66 \pm 1\%} \text{ N}$$

$$(R_{By})_3 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Dx})_3 = \underline{197.66 \pm 1\%} \text{ N}$$

$$(R_{Dy})_3 = \underline{0 \pm 1\%} \text{ N}$$

Explanation:

The reaction forces at points *B* and *D* in the x direction are equal and opposite to each other, and equal in magnitude to

$$(R_{Bx})_2 = 197.66 \text{ N}$$

Therefore,

$$(R_{Bx})_3 = - (R_{Bx})_2 = - 197.66 \text{ N}$$

$$(R_{By})_3 = 0$$

$$(R_{Dx})_3 = (R_{Bx})_2 = 197.66 \text{ N}$$

$$(R_{Dy})_3 = 0$$

9.

Award: 5.33 out of 8.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$(R_{Cx})_4 = 0$ ✖ N
 $(R_{Cy})_4 = 299.99$ ✖ N
 $(R_{Dx})_4 = -197.66$ ✔ N
 $(R_{Dy})_4 = 0$ ✔ N
 $(R_{Ex})_4 = 0$ ✔ N
 $(R_{Ey})_4 = 300$ ✔ N

References

Numeric Response Chapter: 03 Load and Stress Analysis

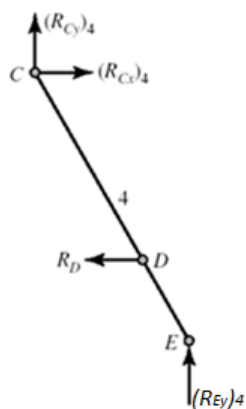
Problem 03.004.c - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$(R_{Cx})_4 = 197.66 \pm 1\%$ N
 $(R_{Cy})_4 = -299.99 \pm 1\%$ N
 $(R_{Dx})_4 = -197.66 \pm 1\%$ N
 $(R_{Dy})_4 = 0 \pm 1\%$ N
 $(R_{Ex})_4 = 0 \pm 1\%$ N
 $(R_{Ey})_4 = 299.99 \pm 1\%$ N

Explanation:



The reaction force at point D has the same magnitude as that of element 3, but with the opposite sign.

$$(R_{Dx})_4 = - (R_{Dx})_3 = -197.66 \text{ N}$$

$$(R_{Dy})_4 = 0$$

Because point E is on a horizontal roller,

$$(R_{Ex})_4 = 0$$

To find the vertical reaction at point E ,

$$h = \frac{\frac{1}{2}AE}{\tan 30^\circ} = \frac{\frac{1}{2}14 \text{ m}}{\tan 30^\circ} = 12.12 \text{ m}$$

$$\Sigma M_A = 0 \quad AE(R_{Ey})_4 - h(F \cos 30^\circ) - \frac{1}{2}AE(F \sin 30^\circ) = 0$$

$$(R_{Ey})_4 = \frac{h(F \cos 30^\circ) + \frac{1}{2}AE(F \sin 30^\circ)}{AE}$$

$$(R_{Ey})_4 = \frac{12.12 \text{ m} (300 \text{ N} \times \cos 30^\circ) + \frac{1}{2}14 \text{ m} (300 \text{ N} \times \sin 30^\circ)}{14 \text{ m}} = 299.99 \text{ N}$$

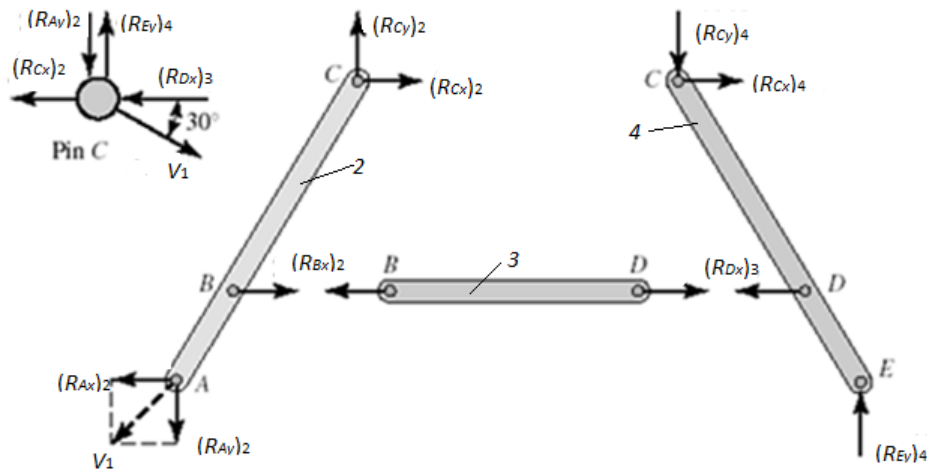
The reaction forces at point C are determined as follows:

$$\Sigma F_x = 0 \quad (R_{Cx})_4 + (R_{Dx})_4 = 0$$

$$(R_{Cx})_4 = -(R_{Dx})_4 = 197.66 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Cy})_4 + (R_{Ey})_4 = 0$$

$$(R_{Cy})_4 = -(R_{Ey})_4 = -299.99 \text{ N}$$



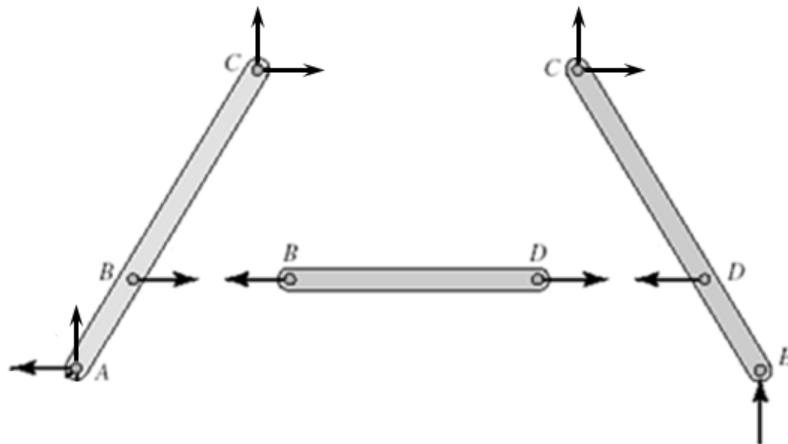
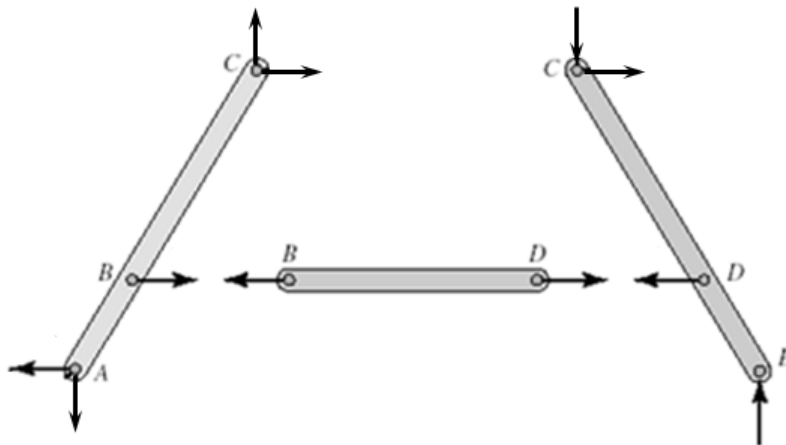
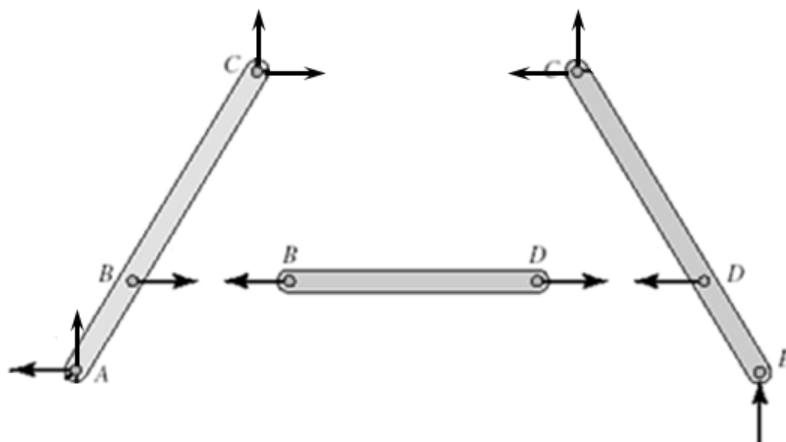
10.

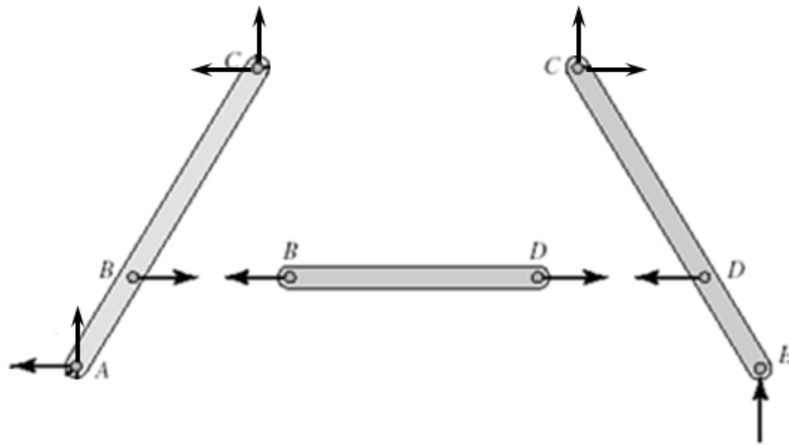
Award: 9 out of 9.00 points

 You received credit for this question in a previous attempt

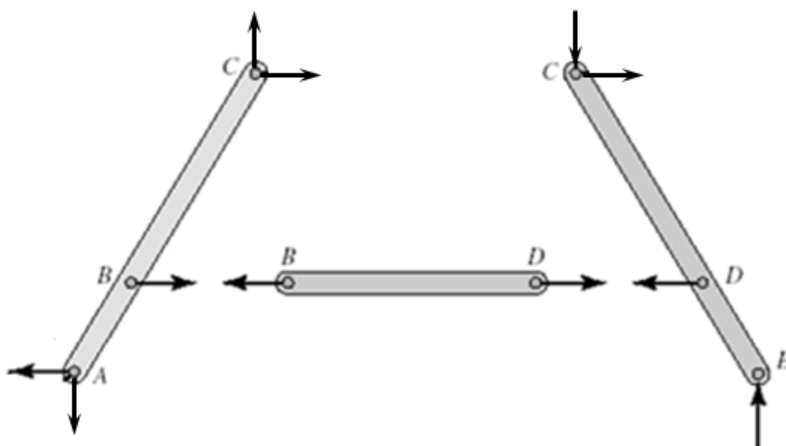
Problem 03.004.d - Determine the specified reaction force(s).

Select the correct set of free-body diagrams from the given choices.

☐☒☐



The correct free-body diagram is



References

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.004.d - Determine the specified reaction force(s).

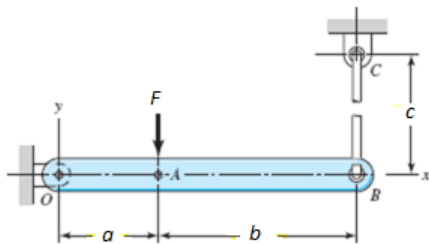
Section: 03.02 Shear Force and Bending Moments in Beams

BELOW AVERAGE

Score: 45/100 Points 45 %

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the given figure, $F = 200$ lbf, $a = 11$ in, $b = 7$ in, and $c = 10$ in.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

3.Problem 03.001 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Section: 03.02 Shear Force and Bending Moments in Beams

1.

Award: 8 out of 8.00 points

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points *O* and *B* in lbf?The force at point *O* is 77.78 ✓ lbf.The force at point *B* is 122.22 ✓ lbf.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.001.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.001.a - Determine the specified reaction force(s)

What is the value of the forces at points *O* and *B* in lbf?The force at point *O* is 77.78 ± 1.01 lbf.The force at point *B* is 122.22 ± 1.01 lbf.**Explanation:**Taking the moment about point *O*,

$$\Sigma M_O = R_B(18 \text{ in}) - Fa$$

$$\Sigma M_O = R_B(18 \text{ in}) - (200 \text{ lbf} \times 11 \text{ in})$$

Therefore, the reaction force at point *B* is

$$R_B = Fa/18 = 200 \text{ lbf} \times 11 \text{ in} / 18 \text{ in} = 122.22 \text{ lbf}$$

The sum of equilibrium of forces is

$$\Sigma F = R_B - F + R_O$$

Therefore, the reaction force at point *O* is

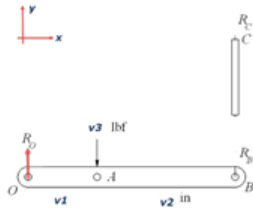
$$R_O = -R_B + F = -122.22 \text{ lbf} + 200 \text{ lbf} = 77.78 \text{ lbf}$$

2.

Award: 9 out of 9.00 points

Problem 03.001.b - Determine the specified reaction force(s)

What will be the direction of force R_B if the direction of force R_O is as shown in the figure? In the image provided below $v_3 = 200$ lbf, $v_1 = 11$ in, and $v_2 = 7$ in.



- ☐ The direction of the force R_B will be negative.
- ✓ ☒ The direction of the force R_B will be positive.

Taking the moment about point O ,

$$\sum M_O = R_B(18 \text{ in}) - (200 \text{ lbf} \times 11 \text{ in})$$

Therefore, the reaction force at point B is

$$R_B = \frac{F \cdot a}{b} = \frac{200 \text{ lbf} \times 11 \text{ in}}{18 \text{ in}} = 122.22 \text{ lbf}$$

The force R_B must be positive otherwise the bar AB would rotate clockwise around point O .

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.001.b -
Determine the
specified reaction
force(s)

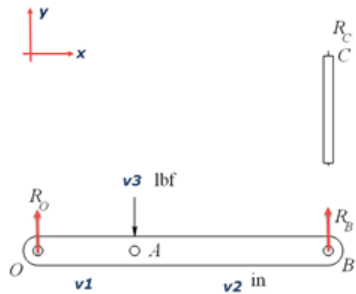
Section: 03.02 Shear Force
and Bending Moments in
Beams

3.

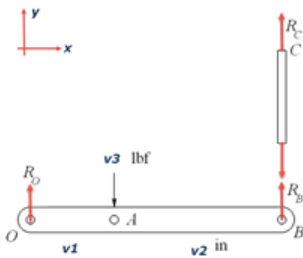
Award: 9 out of 9.00 points

Problem 03.001.c - Determine the specified reaction force(s)

What is the direction of force R_C ? In the image provided below $v_3 = 200$ lbf, $v_1 = 11$ in, and $v_2 = 7$ in.



- ✓ ☒ The direction of force R_C is positive.
- ☐ The direction of force R_C is negative.



The sum of equilibrium of forces in the vertical direction is

$$\sum R_c - R_b = 0 \rightarrow R_c = R_b$$

Therefore, the direction of force R_C is positive.

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

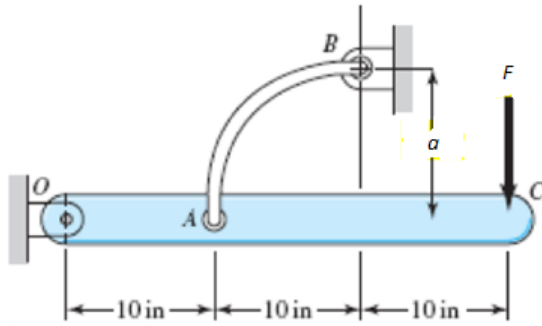
Problem 03.001.c -
Determine the
specified reaction
force(s)

Section: 03.02 Shear Force
and Bending Moments in
Beams

Problem 03.002 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude and direction of each force. In the figure shown, $a = 7$ in and $F = 150$ lbf.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.002 -
Determine the
specified reaction
force(s) - DEPENDENT
MULTI-PART PROBLEM
- ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

4.

Award: 5 out of 10.00 points

Problem 03.002.a - Determine the specified reaction force(s)

What are the values of forces at points A and B in lbf?

$$R_{Ax} = \underline{300} \text{ } \times \text{ lbf}$$

$$R_{Bx} = \underline{300} \text{ } \times \text{ lbf}$$

$$R_{Ay} = \underline{450} \text{ } \checkmark \text{ lbf}$$

$$R_{By} = \underline{450} \text{ } \checkmark \text{ lbf}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.002.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.002.a - Determine the specified reaction force(s)

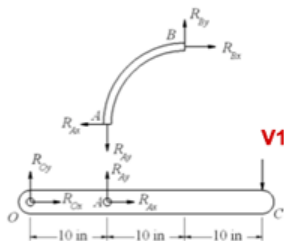
What are the values of forces at points A and B in lbf?

$$R_{Ax} = \underline{642.8571 \pm 1\%} \text{ lbf}$$

$$R_{Bx} = \underline{642.8571 \pm 1\%} \text{ lbf}$$

$$R_{Ay} = \underline{450 \pm 1\%} \text{ lbf}$$

$$R_{By} = \underline{450 \pm 1\%} \text{ lbf}$$

Explanation:

Consider body AB.

$$\Sigma F_x = 0, R_{Ax} = R_{Bx}$$

$$\Sigma F_y = 0, R_{Ay} = R_{By}$$

$$\Sigma M_B = 0, R_{Ay}(10 \text{ in}) - R_{Ax}(7 \text{ in}) = 0$$

Therefore,

$$R_{Ax} = (10 \text{ in})R_{Ay}^a$$

Now, consider body *OAC*.

Taking the moment about point *O* yields

$$R_{Ay}(10 \text{ in}) - F(30 \text{ in}) = \Sigma M_O = 0$$

$$R_{Ay} = 3F = 3 \times 150 \text{ lbf} = 450 \text{ lbf}$$

$$R_{By} = R_{Ay} = 450 \text{ lbf}$$

$$R_{Ax} = (10 \text{ in})R_{Ay}^a = (10 \text{ in})(450 \text{ lbf})7 \text{ in} = 642.85714 \text{ lbf}$$

$$R_{Ax} = R_{Bx} = 642.8571 \text{ lbf}$$

5.

Award: 5 out of 10.00 points

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = \underline{-300} \text{ lbf}$$

$$R_{Oy} = \underline{-300} \text{ lbf}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.002.b - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.002.b - Determine the specified reaction force(s)

What are the values of the x and y forces at point O in lbf?

$$R_{Ox} = \underline{-642.8571 \pm 1\%} \text{ lbf}$$

$$R_{Oy} = \underline{-300 \pm 1\%} \text{ lbf}$$

Explanation:

Sum of forces in the x direction

$$\Sigma F_x = 0 \rightarrow R_{Ox} = -R_{Ax} = -(10 \text{ in})R_{Ay}^a$$

$$R_{Ox} = -(10 \text{ in})(450 \text{ lbf})7 \text{ in} = -642.8571 \text{ lbf}$$

Sum of forces in the y direction

$$\Sigma F_y = 0$$

$$R_{Oy} + R_{Ay} - F = 0$$

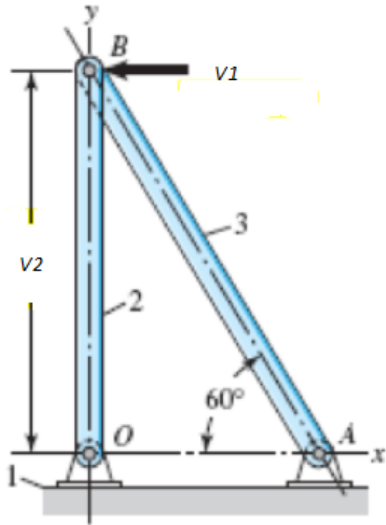
$$R_{Oy} = -R_{Ay} + F = -450 \text{ lbf} + 150 \text{ lbf} = -300 \text{ lbf}$$

6.

Award: 0 out of 20.00 points

Problem 03.003 - Determine the specified reaction force(s)

Determine the magnitude and direction of each force at points O and A. In the figure shown, $V_1 = 0.8 \text{ kN}$ and $V_2 = 0.7 \text{ m}$. What are the resultant values of the forces at points O and A in kN?



$$R_O = \underline{0.2586} \text{ kN}$$

$$R_A = \underline{0.2986} \text{ kN}$$

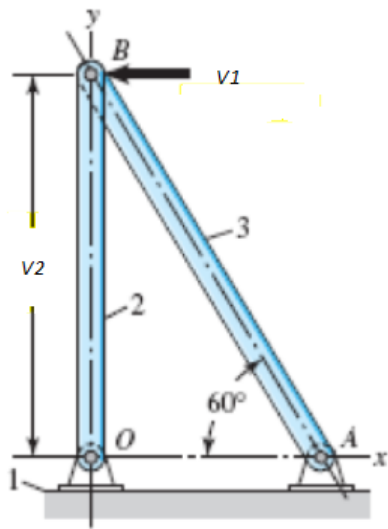
References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.003 - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.003 - Determine the specified reaction force(s)

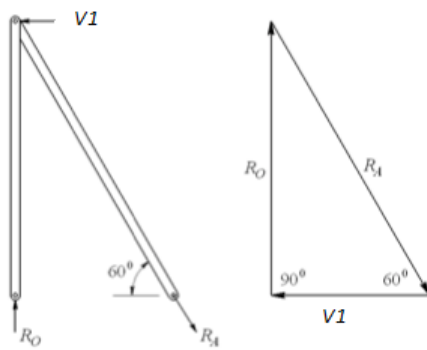
Determine the magnitude and direction of each force at points O and A. In the figure shown, $V_1 = 0.8 \text{ kN}$ and $V_2 = 0.7 \text{ m}$. What are the resultant values of the forces at points O and A in kN?



$$R_O = 1.3856 \pm 1\% \text{ kN}$$

$$R_A = 1.6 \pm 1\% \text{ kN}$$

Explanation:



$$R_O = V_1 \tan 30^\circ$$

$$R_O = 0.8 \text{ kN} \cdot 0.5773 = 1.3856 \text{ kN}$$

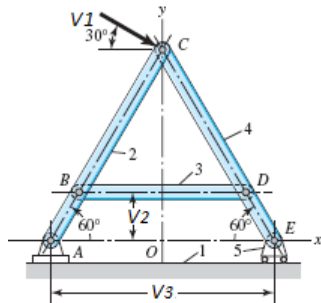
$$R_A = V_1 \sin 30^\circ$$

$$R_A = 0.8 \text{ kN} \cdot 0.5 = 1.6 \text{ kN}$$

Problem 03.004 - Determine the specified reaction force(s) - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Determine the magnitude of the forces. In the given figure, $V_1 = 500$ N, $V_2 = 1.8$ m, and $V_3 = 12$ m.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.004 -
Determine the
specified reaction
force(s) - DEPENDENT
MULTI-PART PROBLEM
- ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = \boxed{433.01} \text{ N}$$

$$(R_{Ay})_2 = \boxed{n/r} \text{ N}$$

$$(R_{Bx})_2 = \boxed{0} \text{ N}$$

$$(R_{By})_2 = \boxed{n/r} \text{ N}$$

$$(R_{Cx})_2 = \boxed{n/r} \text{ N}$$

$$(R_{Cy})_2 = \boxed{n/r} \text{ N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.a - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.a - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 2.

$$(R_{Ax})_2 = \boxed{-433 \pm 1\%} \text{ N}$$

$$(R_{Ay})_2 = \boxed{-250 \pm 1\%} \text{ N}$$

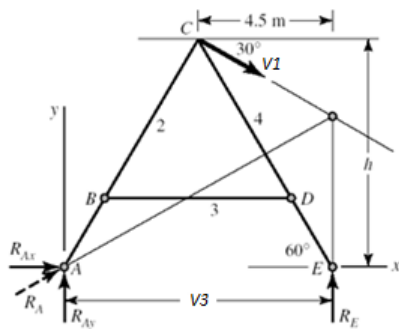
$$(R_{Bx})_2 = \boxed{349.15 \pm 1\%} \text{ N}$$

$$(R_{By})_2 = \boxed{0 \pm 1\%} \text{ N}$$

$$(R_{Cx})_2 = \boxed{83.85 \pm 1\%} \text{ N}$$

$$(R_{Cy})_2 = \boxed{250 \pm 1\%} \text{ N}$$

Explanation:



The reaction forces at point A are determined as follows:

$$\sum F_x = 0 \quad (R_{Ax})_2 + F \cos 30^\circ = 0$$

$$(R_{Ax})_2 = -F \cos 30^\circ = -V_1 \cos 30^\circ$$

$$(R_{Ax})_2 = -500 \text{ N} \times 0.866 = -433 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Ay})_2 + V_1 - V_1 \sin 30^\circ = 0$$

$$(R_{Ay})_2 = V_1 \sin 30^\circ - V_1$$

$$(R_{Ay})_2 = (500 \text{ N} \times 0.5) - 500 \text{ N} = -250 \text{ N}$$

The reaction forces at point *B* are determined as follows:

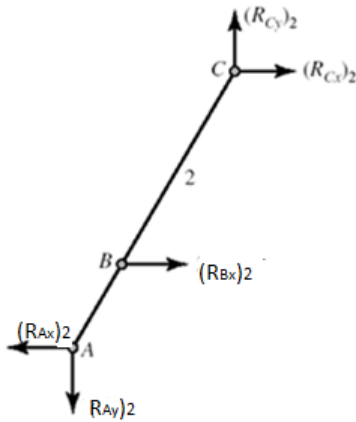
$$h = 12 \tan 30^\circ = 12 \text{ m} \cdot 0.57735 = 10.39 \text{ m}$$

$$\Sigma M_C = 0 \quad 12 \text{ AEF} - (h - V_2)(R_{Bx})_2 = 0$$

$$(R_{Bx})_2 = 12 \text{ AEF} / (h - V_2) = 12 V_1 V_3 / (12 V_3 \tan 30^\circ - V_2)$$

$$(R_{Bx})_2 = 12(500 \text{ N} \times 12 \text{ m}) / (12 \text{ m} \cdot 0.57735 - 1.8 \text{ m}) = 349.15 \text{ N}$$

$$(R_{By})_2 = 0$$



The reaction forces at point *C* are determined as follows:

$$\Sigma F_x = 0 \quad (R_{Cx})_2 - F \cos 30^\circ + 12 \text{ AEF} / (h - V_2) = 0$$

$$(R_{Cx})_2 = V_1 \cos 30^\circ - 12 V_1 V_3 / (12 V_3 \tan 30^\circ - V_2)$$

$$(R_{Cx})_2 = (500 \text{ N} \times \cos 30^\circ) - 12(500 \text{ N} \times 12 \text{ m}) / (12 \text{ m} \cdot \tan 30^\circ - 1.8 \text{ m}) = 83.85 \text{ N}$$

$$\Sigma F_y = 0 \quad (R_{Cy})_2 = - (V_1 \sin 30^\circ) + V_1 = - (500 \text{ N} \times 0.5) + 500 \text{ N} = 250 \text{ N}$$

8.

Award: 0 out of 9.00 points

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = \frac{n}{L} \times \text{N}$$

$$(R_{By})_3 = \frac{n}{L} \times \text{N}$$

$$(R_{Dx})_3 = \frac{n}{L} \times \text{N}$$

$$(R_{Dy})_3 = \frac{n}{L} \times \text{N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.b - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.b - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 3.

$$(R_{Bx})_3 = -349.15 \pm 1\% \text{ N}$$

$$(R_{By})_3 = 0 \pm 1\% \text{ N}$$

$$(R_{Dx})_3 = 349.15 \pm 1\% \text{ N}$$

$$(R_{Dy})_3 = 0 \pm 1\% \text{ N}$$

Explanation:

The reaction forces at points *B* and *D* in the *x* direction are equal and opposite to each other, and equal in magnitude to

$$(R_{Bx})_2 = 349.15 \text{ N}$$

Therefore,

$$(R_{Bx})_3 = -(R_{Bx})_2 = -349.15 \text{ N}$$

$$(R_{By})_3 = 0$$

$$(R_{Dx})_3 = (R_{Bx})_2 = 349.15 \text{ N}$$

$$(R_{Dy})_3 = 0$$

9.

Award: 0 out of 8.00 points

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$$(R_{Cx})_4 = \underline{n/r} \text{ N}$$

$$(R_{Cy})_4 = \underline{n/r} \text{ N}$$

$$(R_{Dx})_4 = \underline{0} \text{ N}$$

$$(R_{Dy})_4 = \underline{n/r} \text{ N}$$

$$(R_{Ex})_4 = \underline{n/r} \text{ N}$$

$$(R_{Ey})_4 = \underline{n/r} \text{ N}$$

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.004.c - Determine the specified reaction force(s) Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.004.c - Determine the specified reaction force(s)

Compute the x and y components of each force acting on element 4.

$$(R_{Cx})_4 = \underline{349.15 \pm 1\%} \text{ N}$$

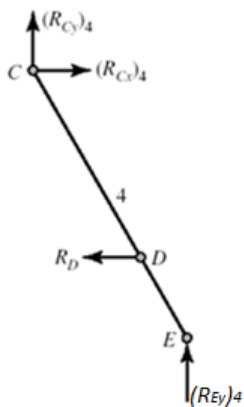
$$(R_{Cy})_4 = \underline{-499.99 \pm 1\%} \text{ N}$$

$$(R_{Dx})_4 = \underline{-349.15 \pm 1\%} \text{ N}$$

$$(R_{Dy})_4 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Ex})_4 = \underline{0 \pm 1\%} \text{ N}$$

$$(R_{Ey})_4 = \underline{499.99 \pm 1\%} \text{ N}$$

Explanation:

The reaction force at point D has the same magnitude as that of element 3, but with the opposite sign.

$$(R_{Dx})_4 = -(R_{Dx})_3 = -349.15 \text{ N}$$

$$(R_{Dy})_4 = 0$$

Because point E is on a horizontal roller,

$$(R_{Ex})_4 = 0$$

To find the vertical reaction at point E ,

$$h = 12 \tan 30^\circ = 12 \tan 30^\circ = 10.39 \text{ m}$$

$$\sum M_A = 0 \quad -AE(R_{Ey})_4 - h(F \cos 30^\circ) - 12(F \sin 30^\circ) = 0$$

$$(R_{Ey})_4 = \frac{h(F \cos 30^\circ) + 12(F \sin 30^\circ)}{AE}$$

$$(R_{Ey})_4 = \frac{10.39 \text{ m}(500 \text{ N} \times \cos 30^\circ) + 12 \text{ m}(500 \text{ N} \times \sin 30^\circ)}{12 \text{ m}} = 499.99 \text{ N}$$

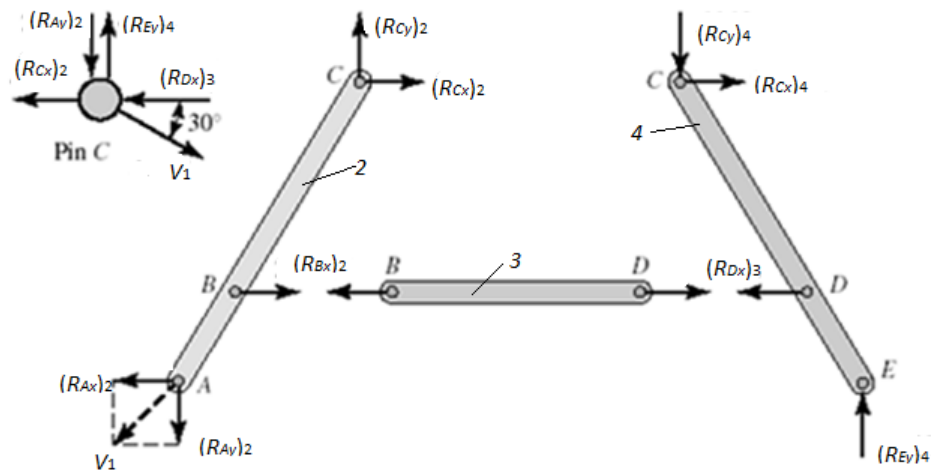
The reaction forces at point C are determined as follows:

$$\sum F_x = 0 \quad (R_{Cx})_4 + (R_{Dx})_4 = 0$$

$$(R_{Cx})_4 = -(R_{Dx})_4 = 349.15 \text{ N}$$

$$\sum F_y = 0 \quad (R_{Cy})_4 + (R_{Ey})_4 = 0$$

$$(R_{Cy})_4 = -(R_{Ey})_4 = -499.99 \text{ N}$$

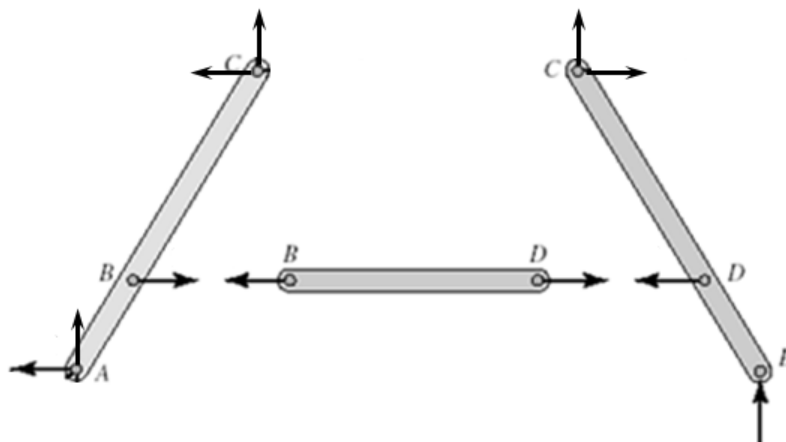
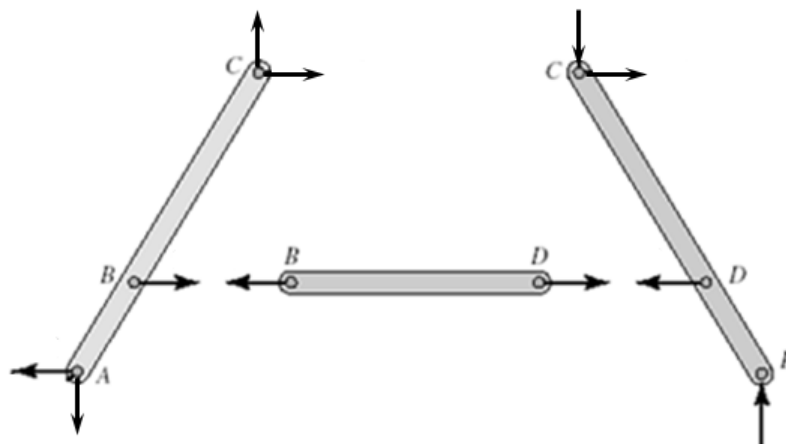
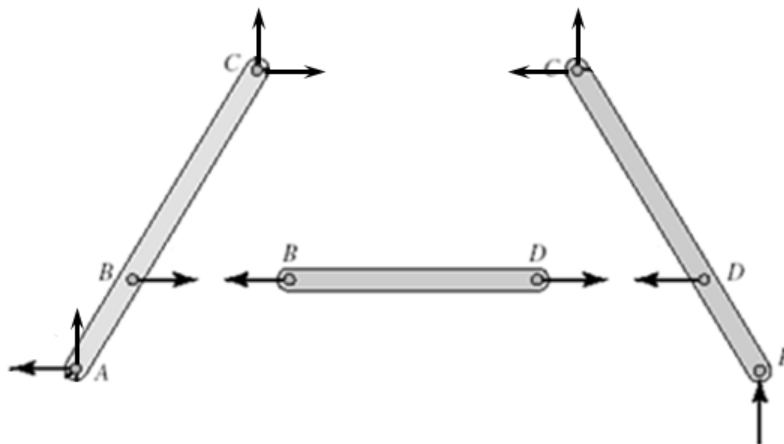


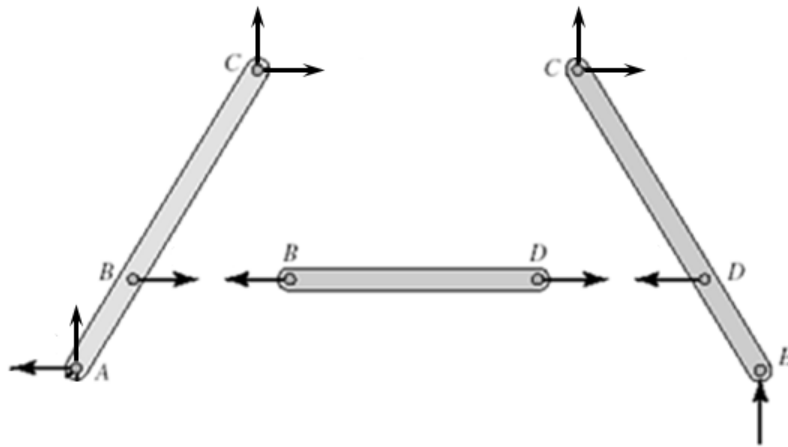
10.

Award: 9 out of 9.00 points

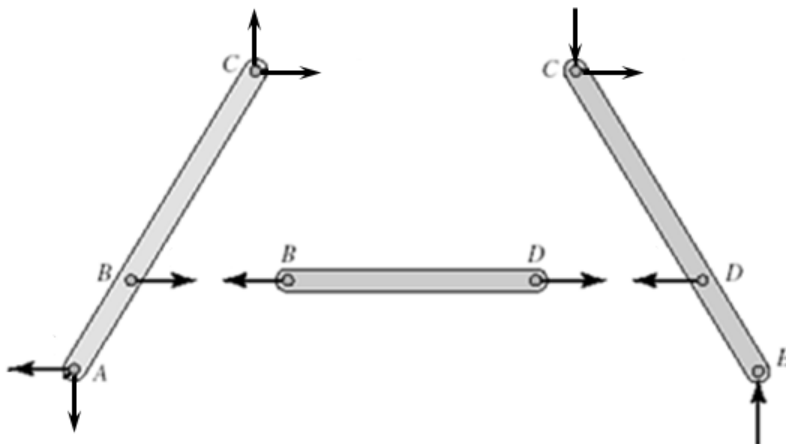
Problem 03.004.d - Determine the specified reaction force(s).

Select the correct set of free-body diagrams from the given choices.





The correct free-body diagram is



References

Multiple Choice

Chapter: 03 Load and Stress
Analysis

Problem 03.004.d -
Determine the
specified reaction
force(s).

Section: 03.02 Shear Force
and Bending Moments in
Beams

MACHINE DESIGN I
Fall 2021

HOMEWORK 2

TOPICS COVERED IN HOMEWORK 2

Shear force and bending moment diagram

**MACHINE DESIGN I
FALL 2021**

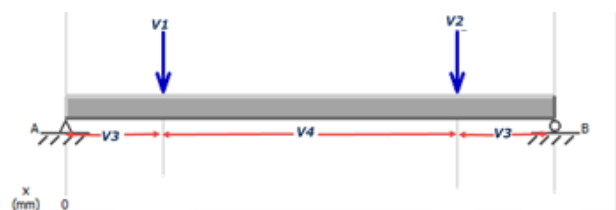
**SAMPLE HOMEWORK QUESTIONS, SOLUTION,
AND GRADING SCHEME**

1.

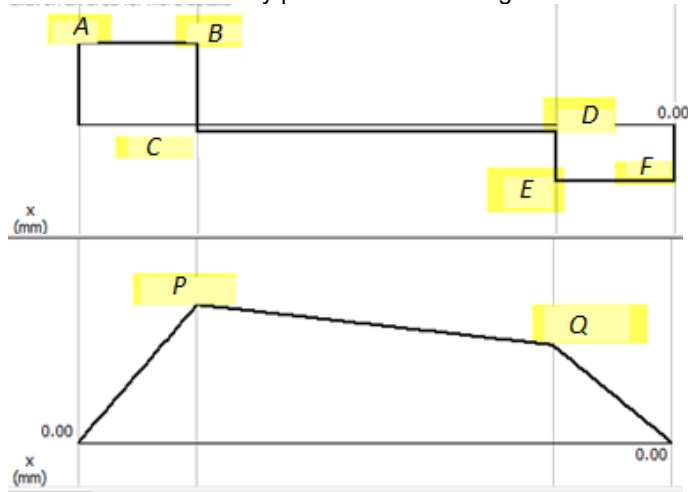
Award: 30.00 points

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 7 \text{ kN}$, $V_2 = 7 \text{ kN}$, $V_3 = 300 \text{ mm}$, and $V_4 = 900 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = 7 ± 0.125 kN and reaction force R_2 (right) = 7 ± 0.125 kN.

In the shear-force and bending-moment diagrams given,

- $A = 7 \pm 0.125 \text{ kN}$
 $B = 7 \pm 0.125 \text{ kN}$
 $C = 0 \pm 0.125 \text{ kN}$
 $D = 0 \pm 0.125 \text{ kN}$
 $E = -7 \pm 0.125 \text{ kN}$
 $F = -7 \pm 0.125 \text{ kN}$
 $P = 2100 \pm 0.25 \text{ kN}\cdot\text{mm}$
 $Q = 2100 \pm 0.25 \text{ kN}\cdot\text{mm}$

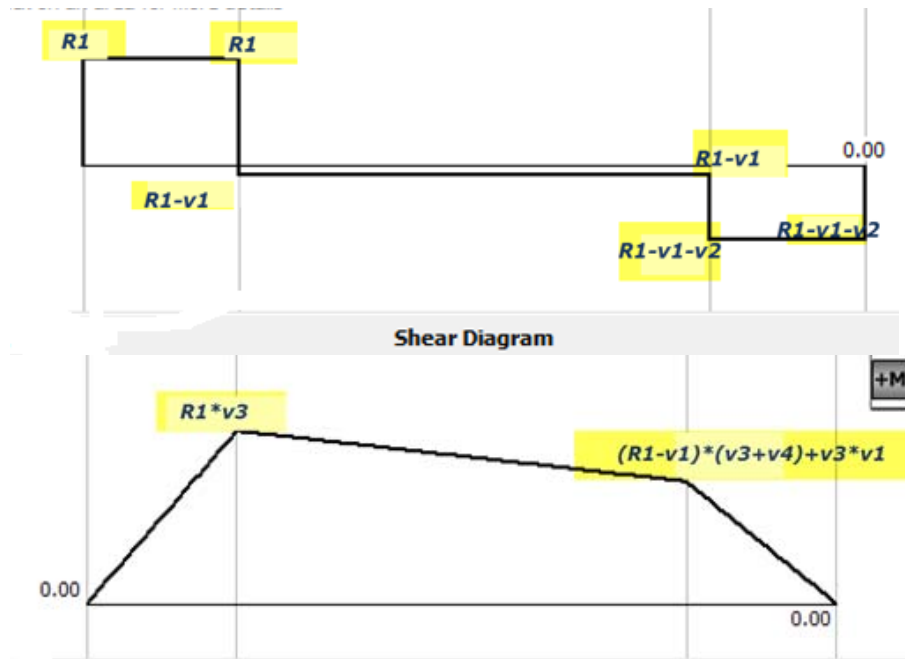
Explanation:

Taking the moment about point A,

$$\sum M_A = 0 = R_2(v_4 + 2v_3) - v_2(v_3 + v_4) - v_1(v_3)$$

$$R_2 = \frac{v_2(v_3 + v_4) + v_1(v_3)}{(v_4 + 2v_3)} = \frac{7 \text{ kN}(300 \text{ mm} + 900 \text{ mm}) + 7 \text{ kN}(300)}{(900 \text{ mm} + 2 \times 300 \text{ mm})} = 7 \text{ kN}$$

$$R_1 = v_1 + v_2 - R_2 = (7 + 7 - 7) \text{ kN} = 7 \text{ kN}$$



In the bending-moment diagram shown above, the bending-moment at each point is determined as follows:

$$P = R_1 v_3 = 7 \text{ kN} \times 300 \text{ mm} = 2100 \text{ kN}\cdot\text{mm}$$

$$Q = (R_1 - v_1)(v_3 + v_4) + (v_3 v_1) = (7 \text{ kN} - 7 \text{ kN})(300 + 900) \text{ mm} + 7 \text{ kN}(300 \text{ mm}) = 2100 \text{ kN}\cdot\text{mm}$$

The reaction forces are R_1 (left) = 7 kN and R_2 (right) = 7 kN.

In the shear-force and bending-moment diagrams given,

$$A = 7 \text{ kN}$$

$$B = 7 \text{ kN}$$

$$C = 0 \text{ kN}$$

$$D = 0 \text{ kN}$$

$$E = -7 \text{ kN}$$

$$F = -7 \text{ kN}$$

$$P = 2100 \text{ kN}\cdot\text{mm}$$

$$Q = 2100 \text{ kN}\cdot\text{mm}$$

References

Numeric Response

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

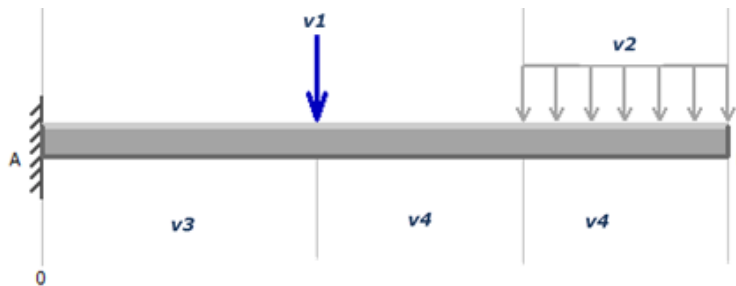
Difficulty: Medium

2.

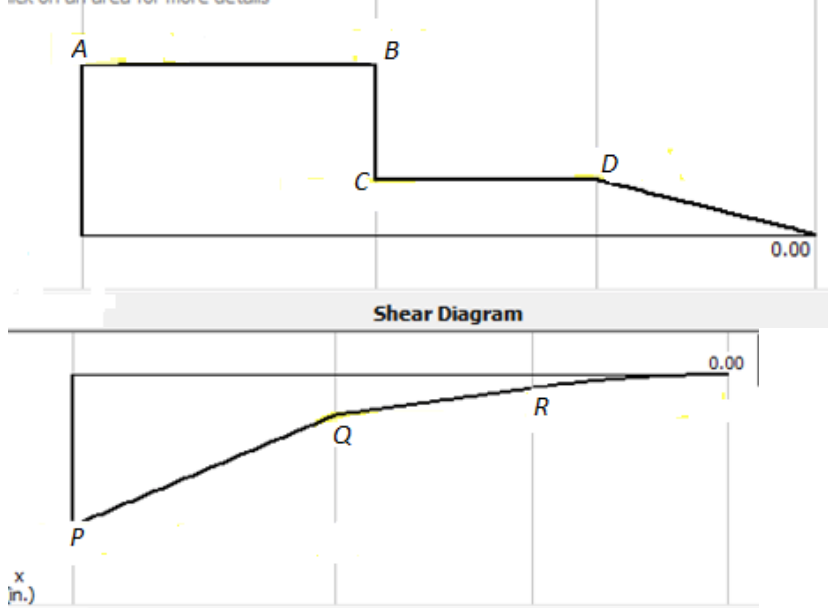
Award: 30.00 points

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 700$ lbf, $V_2 = 48$ lbf/in, $V_3 = 10$ in, and $V_4 = 5$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_0 = 940 \pm 0.125$ lbf.

The bending moment $M_0 = -11200 \pm 2\%$ lbf-in.

In the shear-force and bending-moment diagrams given,

$A = 940 \pm 0.125$ lbf

$B = 940 \pm 0.125$ lbf

$C = 240 \pm 0.125$ lbf

$D = 240 \pm 0.125$ lbf

$P = -11200 \pm 0.25$ lbf-in

$Q = -1800 \pm 0.25$ lbf-in

$R = -600 \pm 0.25$ lbf-in

Explanation:

The bending-moments and the reaction force are

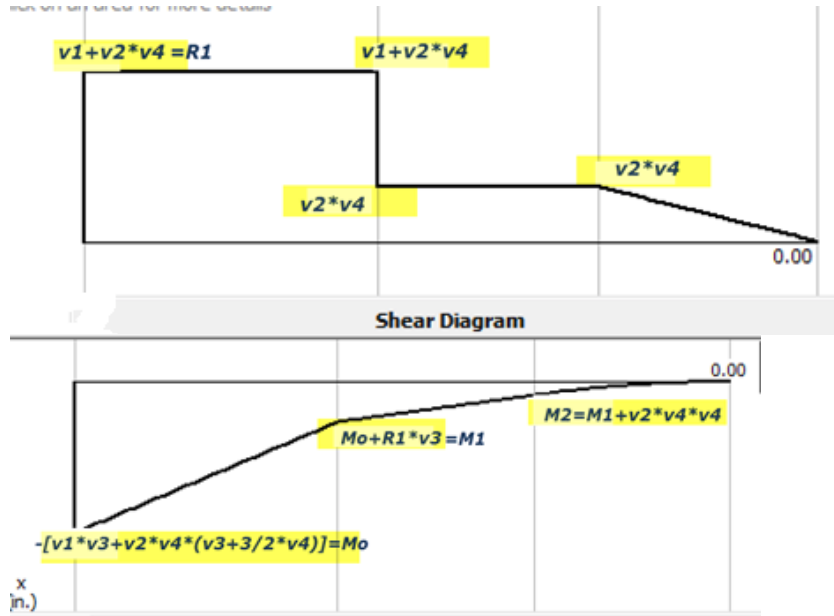
$$M_o = -v_1 v_3 - v_2 v_4 \left(\frac{3}{2} v_4 + v_3 \right)$$

$$M_o = -(700 \text{ lbf} \times 10 \text{ in}) - (48 \text{ lbf/in} \times 5 \text{ in}) \left(\frac{3}{2} 5 \text{ in} + 10 \text{ in} \right) = -11200 \text{ lbf}\cdot\text{in}$$

$$R_o = v_1 + v_2 * v_4 = 700 \text{ lbf} + 48 \text{ lbf/in} \times 5 \text{ in} = 940 \text{ lbf}$$

$$M_1 = M_o + R_o * v_3 = -11200 \text{ lbf}\cdot\text{in} + (940 \text{ lbf} \times 10 \text{ in}) = -1800 \text{ lbf}\cdot\text{in}$$

$$M_2 = M_1 + (R_o - v_1) v_4 = -1800 \text{ lbf}\cdot\text{in} + (940 \text{ lbf} - 700 \text{ lbf}) 5 \text{ in} = -600 \text{ lbf}\cdot\text{in}$$



Reaction force $R_o = 940 \text{ lbf}$

The bending moment $M_o = -11200 \text{ lbf}\cdot\text{in}$.

In the shear-force and bending-moment diagrams given,

- A = 940 lbf
- B = 940 lbf
- C = 240 lbf
- D = 240 lbf
- P = -11200 lbf·in
- Q = -1800 lbf·in
- R = -600 lbf·in

References

Numeric Response

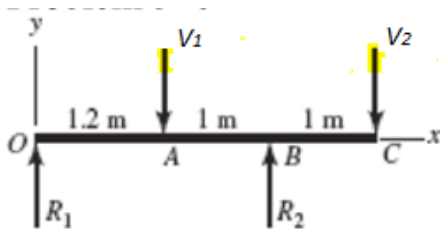
Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

Difficulty: Medium

Problem 03.007 - Find the reactions and plot the shear-force and bending-moment diagrams - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 1 \text{ kN}$ and $V_2 = 2 \text{ kN}$.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

**References****Section Break**

Problem 03.007 - Find the reactions and plot the shear-force and bending-moment diagrams - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

3.

Award: 10.00 points

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = \boxed{-0.4545 \pm 2\%}$ kN and reaction force $R_2 = \boxed{3.4545 \pm 2\%}$ kN

Explanation:

Taking the moment about point B ,

$$\Sigma M_B = 0$$

$$-(2.2 \text{ m})R_1 + 1(V_1) - 1(V_2) = 0$$

$$R_1 = \frac{V_2 - V_1}{-2.2 \text{ m}} = \frac{2 \text{ kN} - 1 \text{ kN}}{-2.2 \text{ m}} = -0.45 \text{ kN}$$

Sum of forces along the vertical direction

$$\Sigma F_y = 0$$

$$R_1 + R_2 - V_1 - V_2 = 0$$

$$R_2 = V_1 + V_2 - R_1 = (1 + 2 - (-0.45)) \text{ kN} = 3.45 \text{ kN}$$

The reaction forces $R_1 = -0.45 \text{ kN}$ and $R_2 = 3.45 \text{ kN}$.

References

Numeric Response

Problem 03.007.a -
Find the reactions and
plot the shear-force and
bending-moment
diagrams.

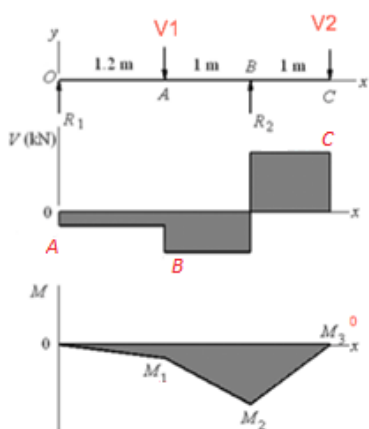
Difficulty: Medium

4.

Award: 30.00 points

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in $\text{kN}\cdot\text{m}$. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

$$A = -0.45 \pm 2\% \text{ kN}$$

$$B = -1.45 \pm 2\% \text{ kN}$$

$$C = 2 \pm 2\% \text{ kN}$$

$$M_1 = -0.55 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_2 = -2 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_3 = 0 \pm 2\% \text{ kN}\cdot\text{m}$$

Explanation:

The bending-moments are

$$M_1 = R_1(1.2 \text{ m}) = -0.45 \text{ kN} \times 1.2 \text{ m} = -0.55 \text{ kN}\cdot\text{m}$$

$$M_2 = -(-(R_1 - V_1)2.2 \text{ m} - 1.2 \text{ m}(V_1))$$

$$M_2 = (R_1)2.2 \text{ m} - (V_1) = (-0.45 \text{ kN} \times 2.2 \text{ m}) - 1 \text{ kN} = -2 \text{ kN}\cdot\text{m}$$

In the shear-force and bending-moment diagrams given,

$$A = -0.45 \text{ kN}$$

$$B = -1.45 \text{ kN}$$

$$C = 2 \text{ kN}$$

$$M_1 = -0.55 \text{ kN}\cdot\text{m}$$

$$M_2 = -2 \text{ kN}\cdot\text{m}$$

$$M_3 = 0 \text{ kN}\cdot\text{m}$$

References

Numeric Response

Problem 03.007.b -
Find the reactions and
plot the shear-force and
bending-moment
diagrams.

Difficulty: Medium

ABOVE AVERAGE

Carter, Nehemiah

Score: 100/100 Points 100 %

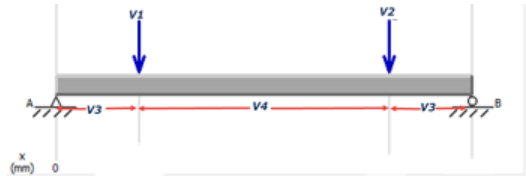
1.

Award: 30 out of 30.00 points

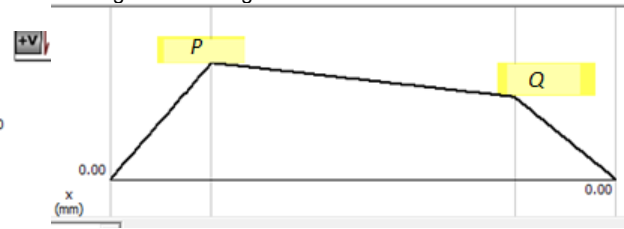
 You received credit for this question in a previous attempt

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 5 \text{ kN}$, $V_2 = 5 \text{ kN}$, $V_3 = 200 \text{ mm}$, and $V_4 = 1100 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = 5  kN and reaction force R_2 (right) = 5  kN.

In the shear-force and bending-moment diagrams given,

$A = \underline{5}$  kN
 $B = \underline{5}$  kN
 $C = \underline{0}$  kN
 $D = \underline{0}$  kN
 $E = \underline{-5}$  kN
 $F = \underline{-5}$  kN
 $P = \underline{1000}$  kN·mm
 $Q = \underline{1000}$  kN·mm

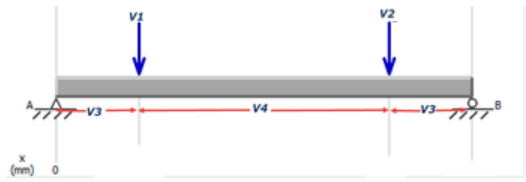
References

Numeric Response Chapter: 03 Load and Stress Analysis

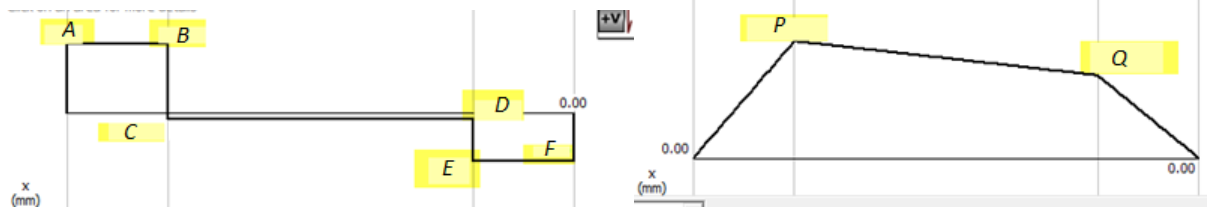
Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 5 \text{ kN}$, $V_2 = 5 \text{ kN}$, $V_3 = 200 \text{ mm}$, and $V_4 = 1100 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = 5 ± 0.125 kN and reaction force R_2 (right) = 5 ± 0.125 kN.

In the shear-force and bending-moment diagrams given,

- A = 5 ± 0.125 kN
 B = 5 ± 0.125 kN
 C = 0 ± 0.125 kN
 D = 0 ± 0.125 kN
 E = -5 ± 0.125 kN
 F = -5 ± 0.125 kN
 P = 1000 ± 0.25 kN·mm
 Q = 1000 ± 0.25 kN·mm

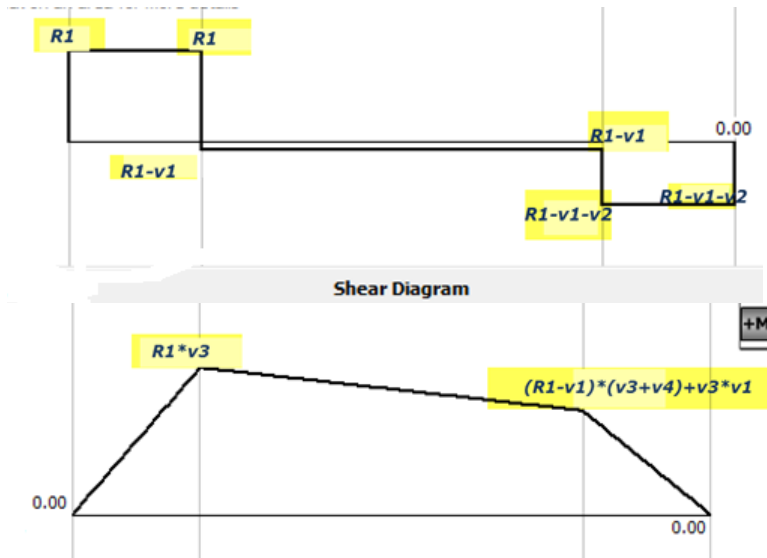
Explanation:

Taking the moment about point A,

$$\sum M_A = 0 = R_2(v_4 + 2v_3) - v_2(v_3 + v_4) - v_1(v_3)$$

$$R_2 = \frac{v_2(v_3 + v_4) + v_1(v_3)}{(v_4 + 2v_3)} = \frac{5 \text{ kN}(200 \text{ mm} + 1100 \text{ mm}) + 5 \text{ kN}(200)}{(1100 \text{ mm} + 2 \times 200 \text{ mm})} = 5 \text{ kN}$$

$$R_1 = v_1 + v_2 - R_2 = (5 + 5 - 5) \text{ kN} = 5 \text{ kN}$$



In the bending-moment diagram shown above, the bending-moment at each point is determined as follows:

$$P = R_1 v_3 = 5 \text{ kN} \times 200 \text{ mm} = 1000 \text{ kN·mm}$$

$$Q = (R_1 - v_1)(v_3 + v_4) + (v_3 v_1) = (5 \text{ kN} - 5 \text{ kN})(200 + 1100) \text{ mm} + 5 \text{ kN}(200 \text{ mm}) = 1000 \text{ kN·mm}$$



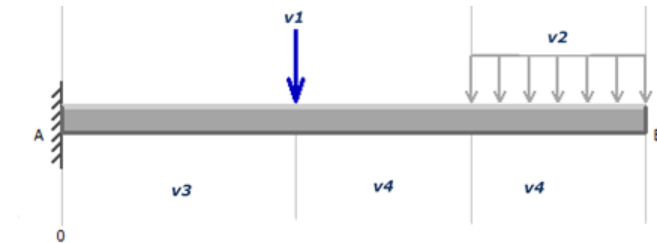
2.

Award: 30 out of 30.00 points

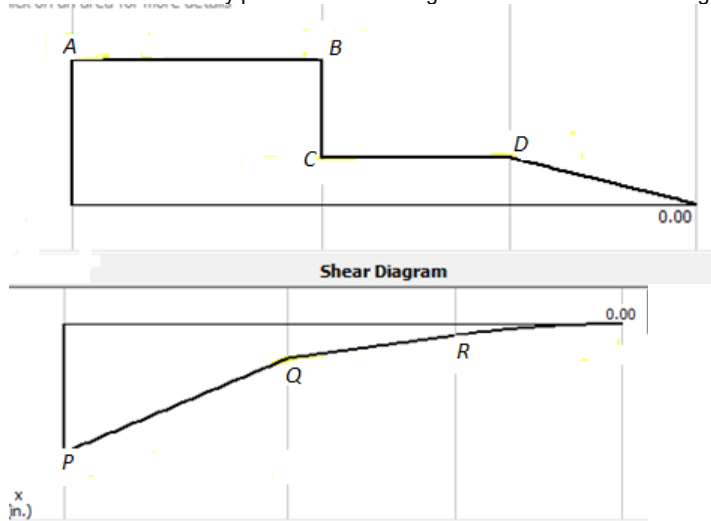
 You did not receive full credit for this question in a previous attempt

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 350$ lbf, $V_2 = 34$ lbf/in, $V_3 = 12$ in, and $V_4 = 4$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_D = 486$ lbf.

The bending moment $M_D = -6648$ lbf-in.

In the shear-force and bending-moment diagrams given,

$A = 486$ lbf

$B = 486$ lbf

$C = 136$ lbf

$D = 136$ lbf

$P = -6648$ lbf-in

$Q = -816$ lbf-in

$R = -272$ lbf-in

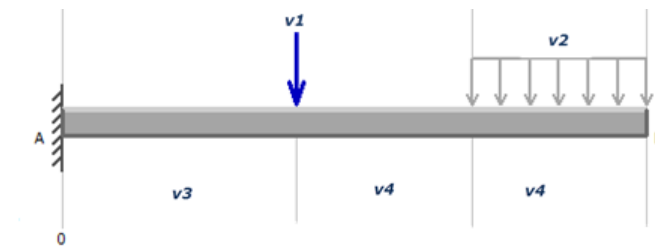
References

Numeric Response Chapter: 03 Load and Stress Analysis

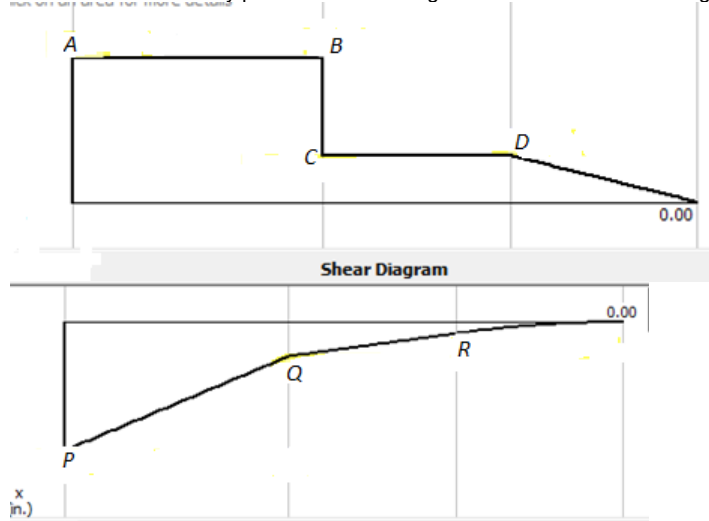
Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 350$ lbf, $V_2 = 34$ lbf/in, $V_3 = 12$ in, and $V_4 = 4$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_0 = 486 \pm 0.125$ lbf.

The bending moment $M_0 = -6648 \pm 2\%$ lbf-in.

In the shear-force and bending-moment diagrams given,

$A = 486 \pm 0.125$ lbf

$B = 486 \pm 0.125$ lbf

$C = 136 \pm 0.125$ lbf

$D = 136 \pm 0.125$ lbf

$P = -6648 \pm 0.25$ lbf-in

$Q = -816 \pm 0.25$ lbf-in

$R = -272 \pm 0.25$ lbf-in

Explanation:

The bending-moments and the reaction force are

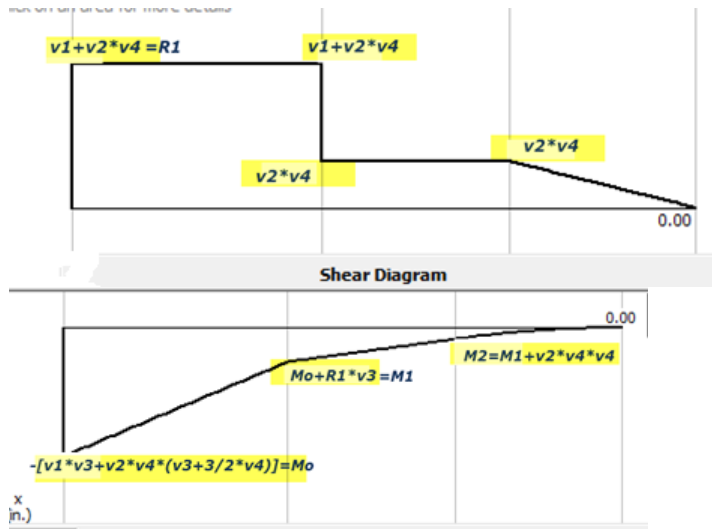
$$M_o = -v_1 v_3 - v_2 v_4 \left(\frac{3}{2} v_4 + v_3 \right)$$

$$M_o = -(350 \text{ lbf} \times 12 \text{ in}) - (34 \text{ lbf/in} \times 4 \text{ in}) \left(\frac{3}{2} 4 \text{ in} + 12 \text{ in} \right) = -6648 \text{ lbf-in}$$

$$R_o = v_1 + v_2 * v_4 = 350 \text{ lbf} + 34 \text{ lbf/in} \times 4 \text{ in} = 486 \text{ lbf}$$

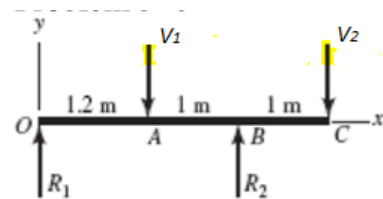
$$M_1 = M_o + R_o * v_3 = -6648 \text{ lbf-in} + (486 \text{ lbf} \times 12 \text{ in}) = -816 \text{ lbf-in}$$

$$M_2 = M_1 + (R_o - v_1) v_4 = -816 \text{ lbf-in} + (486 \text{ lbf} - 350 \text{ lbf}) 4 \text{ in} = -272 \text{ lbf-in}$$



Problem 03.007 - Find the reactions and plot the shear-force and bending-moment diagrams - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 3 \text{ kN}$ and $V_2 = 6 \text{ kN}$.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress
Analysis

Problem 03.007 - Find
the reactions and plot
the shear-force and
bending-moment
diagrams -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

3.

Award: 10 out of 10.00 points

 You received credit for this question in a previous attempt

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = -1.363$  kN and reaction force $R_2 = 10.36$  kN

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = -1.3636 \pm 2\%$ kN and reaction force $R_2 = 10.3636 \pm 2\%$ kN

Explanation:

Taking the moment about point B,

$$\Sigma M_B = 0$$

$$-(2.2 \text{ m})R_1 + 1(V_1) - 1(V_2) = 0$$

$$R_1 = \frac{V_2 - V_1}{-2.2 \text{ m}} = \frac{6 \text{ kN} - 3 \text{ kN}}{-2.2 \text{ m}} = -1.36 \text{ kN}$$

Sum of forces along the vertical direction

$$\Sigma F_y = 0$$

$$R_1 + R_2 - V_1 - V_2 = 0$$

$$R_2 = V_1 + V_2 - R_1 = (3 + 6 - (-1.36)) \text{ kN} = 10.36 \text{ kN}$$

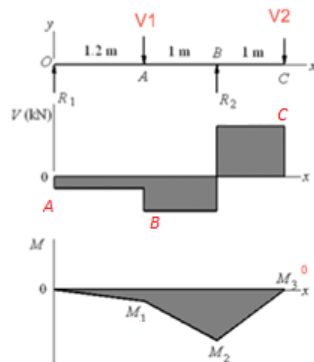
4.

Award: 30 out of 30.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in kN·m. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

A = **-1.36** ✓ kN

B = **-4.36** ✓ kN

C = **6** ✓ kN

M_1 = **-1.64** ✓ kN·m

M_2 = **-6** ✓ kN·m

M_3 = **0** ✓ kN·m

References

Numeric Response

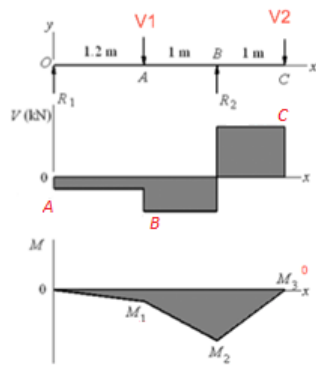
Chapter: 03 Load and Stress Analysis

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in kN·m. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

$$A = -1.36 \pm 2\% \text{ kN}$$

$$B = -4.36 \pm 2\% \text{ kN}$$

$$C = 6 \pm 2\% \text{ kN}$$

$$M_1 = -1.64 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_2 = -6 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_3 = 0 \pm 2\% \text{ kN}\cdot\text{m}$$

Explanation:

The bending-moments are

$$M_1 = R_1(1.2 \text{ m}) = -1.36 \text{ kN} \times 1.2 \text{ m} = -1.64 \text{ kN}\cdot\text{m}$$

$$M_2 = -(-(R_1 - V_1)2.2 \text{ m} - 1.2 \text{ m}(V_1))$$

$$M_2 = (R_1)2.2 \text{ m} - (V_1) = (-1.36 \text{ kN} \times 2.2 \text{ m}) - 3 \text{ kN} = -6 \text{ kN}\cdot\text{m}$$

AVERAGE

Garay, Gabriela

Score: 95/100 Points 95 %

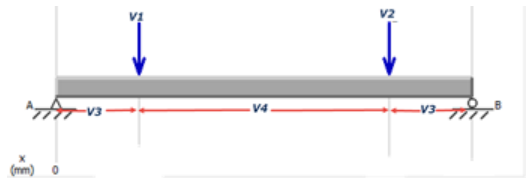
1.

Award: 30 out of 30.00 points

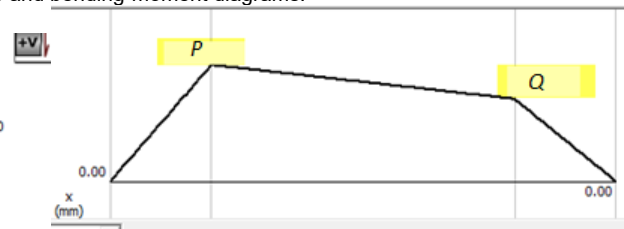
 You did not receive full credit for this question in a previous attempt

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 8 \text{ kN}$, $V_2 = 8 \text{ kN}$, $V_3 = 300 \text{ mm}$, and $V_4 = 900 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = **8**  kN and reaction force R_2 (right) = **8**  kN.

In the shear-force and bending-moment diagrams given,

$A = \mathbf{8}$  kN
 $B = \mathbf{8}$  kN
 $C = \mathbf{0}$  kN
 $D = \mathbf{0}$  kN
 $E = \mathbf{-8}$  kN
 $F = \mathbf{-8}$  kN
 $P = \mathbf{2400}$  kN·mm
 $Q = \mathbf{2400}$  kN·mm

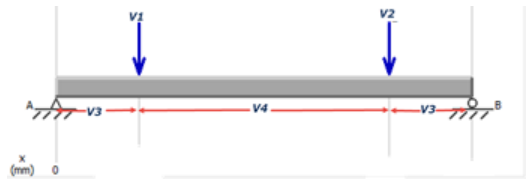
References

Numeric Response Chapter: 03 Load and Stress Analysis

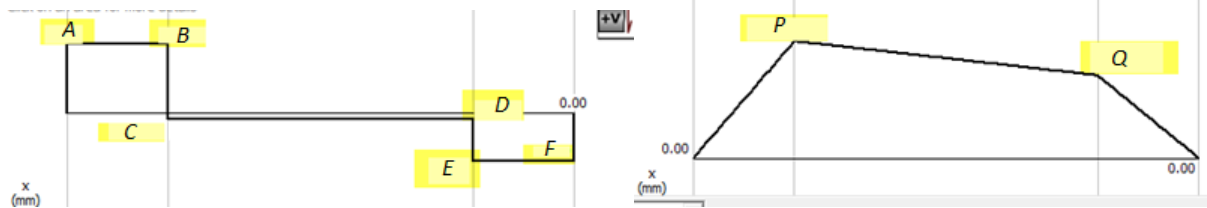
Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 8 \text{ kN}$, $V_2 = 8 \text{ kN}$, $V_3 = 300 \text{ mm}$, and $V_4 = 900 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = 8 ± 0.125 kN and reaction force R_2 (right) = 8 ± 0.125 kN.

In the shear-force and bending-moment diagrams given,

- A = 8 ± 0.125 kN
 B = 8 ± 0.125 kN
 C = 0 ± 0.125 kN
 D = 0 ± 0.125 kN
 E = -8 ± 0.125 kN
 F = -8 ± 0.125 kN
 P = 2400 ± 0.25 kN·mm
 Q = 2400 ± 0.25 kN·mm

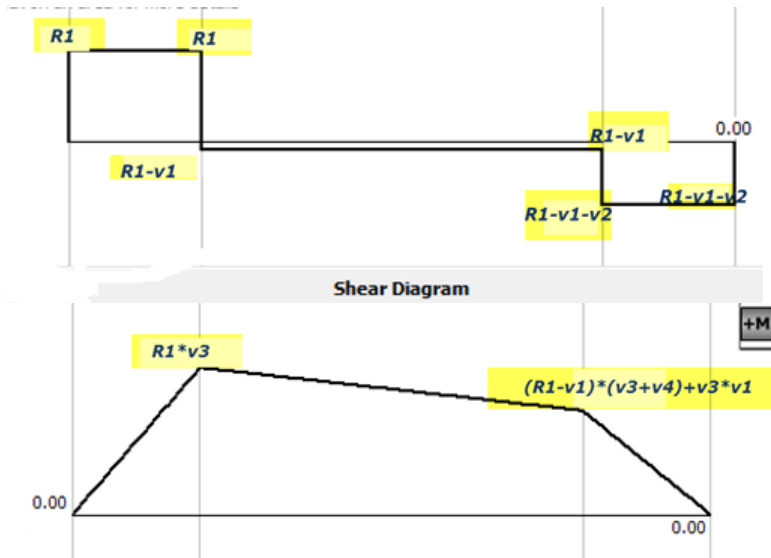
Explanation:

Taking the moment about point A,

$$\sum M_A = 0 = R_2(v_4 + 2v_3) - v_2(v_3 + v_4) - v_1(v_3)$$

$$R_2 = \frac{v_2(v_3 + v_4) + v_1(v_3)}{(v_4 + 2v_3)} = \frac{8 \text{ kN}(300 \text{ mm} + 900 \text{ mm}) + 8 \text{ kN}(300)}{(900 \text{ mm} + 2 \times 300 \text{ mm})} = 8 \text{ kN}$$

$$R_1 = v_1 + v_2 - R_2 = (8 + 8 - 8) \text{ kN} = 8 \text{ kN}$$



In the bending-moment diagram shown above, the bending-moment at each point is determined as follows:

$$P = R_1 v_3 = 8 \text{ kN} \times 300 \text{ mm} = 2400 \text{ kN·mm}$$

$$Q = (R_1 - v_1)(v_3 + v_4) + (v_3 v_1) = (8 \text{ kN} - 8 \text{ kN})(300 + 900) \text{ mm} + 8 \text{ kN}(300 \text{ mm}) = 2400 \text{ kN·mm}$$



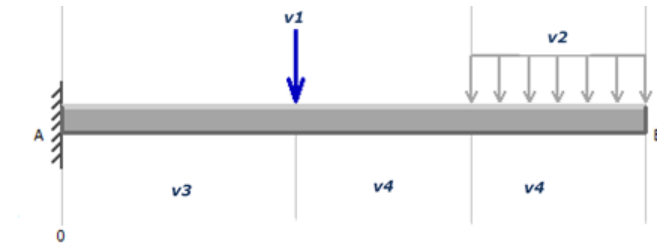
2.

Award: 30 out of 30.00 points

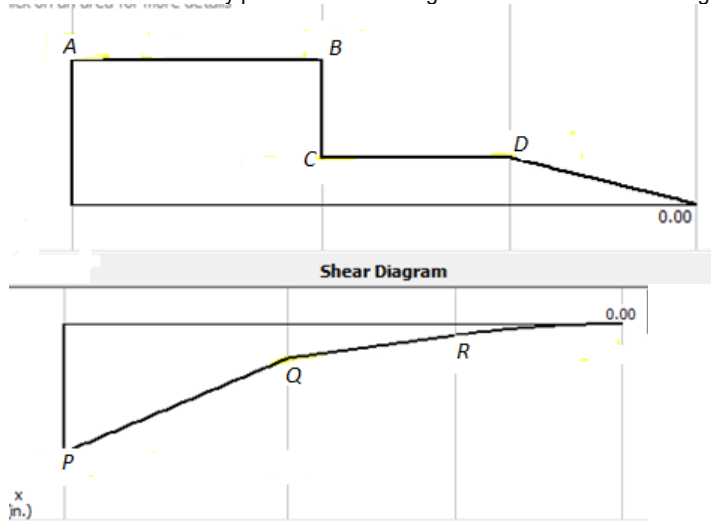
 You did not receive full credit for this question in a previous attempt

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 700$ lbf, $V_2 = 42$ lbf/in, $V_3 = 6$ in, and $V_4 = 7$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_D = 994$ lbf.

The bending moment $M_D = -9051$ lbf-in.

In the shear-force and bending-moment diagrams given,

$A = 994$ lbf

$B = 994$ lbf

$C = 294$ lbf

$D = 294$ lbf

$P = -9051$ lbf-in

$Q = -3087$ lbf-in

$R = -1029$ lbf-in

References

Numeric Response

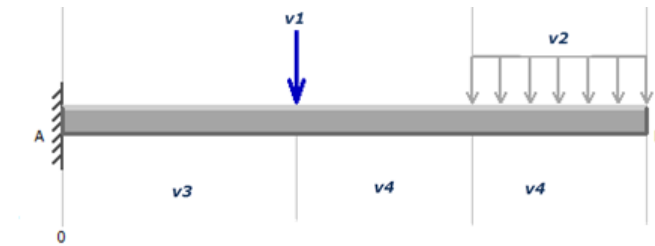
Chapter: 03 Load and Stress Analysis

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

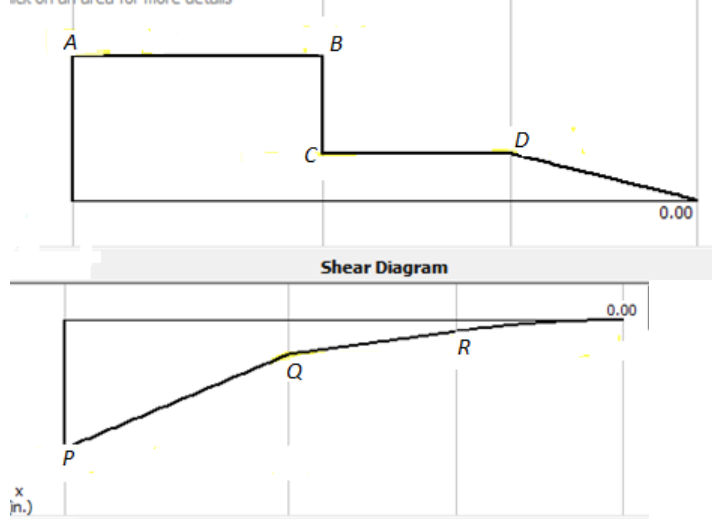
Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 700$ lbf, $V_2 = 42$ lbf/in, $V_3 = 6$ in, and $V_4 = 7$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_0 = 994 \pm 0.125$ lbf.

The bending moment $M_0 = -9051 \pm 2\%$ lbf-in.

In the shear-force and bending-moment diagrams given,

$A = 994 \pm 0.125$ lbf

$B = 994 \pm 0.125$ lbf

$C = 294 \pm 0.125$ lbf

$D = 294 \pm 0.125$ lbf

$P = -9051 \pm 0.25$ lbf-in

$Q = -3087 \pm 0.25$ lbf-in

$R = -1029 \pm 0.25$ lbf-in

Explanation:

The bending-moments and the reaction force are

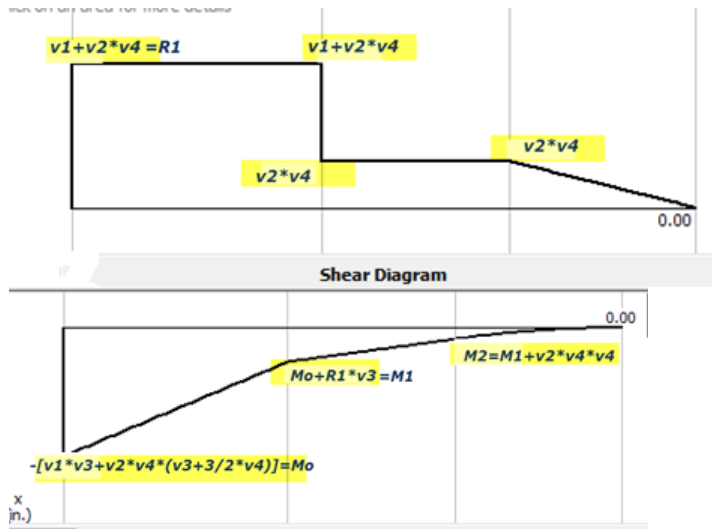
$$M_o = -v_1 v_3 - v_2 v_4 \left(\frac{3}{2} v_4 + v_3 \right)$$

$$M_o = - (700 \text{ lbf} \times 6 \text{ in}) - (42 \text{ lbf/in} \times 7 \text{ in}) \left(\frac{3}{2} 7 \text{ in} + 6 \text{ in} \right) = -9051 \text{ lbf-in}$$

$$R_o = v_1 + v_2 * v_4 = 700 \text{ lbf} + 42 \text{ lbf/in} \times 7 \text{ in} = 994 \text{ lbf}$$

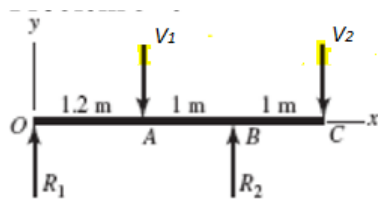
$$M_1 = M_o + R_o * v_3 = -9051 \text{ lbf-in} + (994 \text{ lbf} \times 6 \text{ in}) = -3087 \text{ lbf-in}$$

$$M_2 = M_1 + (R_o - v_1) v_4 = -3087 \text{ lbf-in} + (994 \text{ lbf} - 700 \text{ lbf}) 7 \text{ in} = -1029 \text{ lbf-in}$$



Problem 03.007 - Find the reactions and plot the shear-force and bending-moment diagrams - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 1 \text{ kN}$ and $V_2 = 2 \text{ kN}$. NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.007 - Find the reactions and plot the shear-force and bending-moment diagrams - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Section: 03.02 Shear Force and Bending Moments in Beams

3.

Award: 10 out of 10.00 points

 You received credit for this question in a previous attempt

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = -0.45$  kN and reaction force $R_2 = 3.45$  kN

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = -0.4545 \pm 2\%$ kN and reaction force $R_2 = 3.4545 \pm 2\%$ kN

Explanation:

Taking the moment about point B,

$$\Sigma M_B = 0$$

$$-(2.2 \text{ m})R_1 + 1(V_1) - 1(V_2) = 0$$

$$R_1 = \frac{V_2 - V_1}{-2.2 \text{ m}} = \frac{2 \text{ kN} - 1 \text{ kN}}{-2.2 \text{ m}} = -0.45 \text{ kN}$$

Sum of forces along the vertical direction

$$\Sigma F_y = 0$$

$$R_1 + R_2 - V_1 - V_2 = 0$$

$$R_2 = V_1 + V_2 - R_1 = (1 + 2 - (-0.45)) \text{ kN} = 3.45 \text{ kN}$$

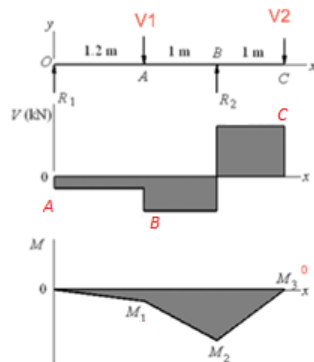
4.

Award: 25 out of 30.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in $\text{kN}\cdot\text{m}$. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

$A = -0.45$ ✓ kN

$B = -0.54$ ✗ kN

$C = 2$ ✓ kN

$M_1 = -0.54$ ✓ $\text{kN}\cdot\text{m}$

$M_2 = -2$ ✓ $\text{kN}\cdot\text{m}$

$M_3 = 0$ ✓ $\text{kN}\cdot\text{m}$

References

Numeric Response

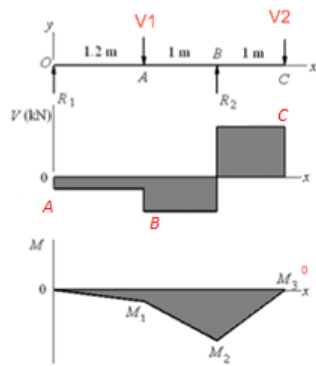
Chapter: 03 Load and Stress Analysis

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in $\text{kN}\cdot\text{m}$. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

$$A = -0.45 \pm 2\% \text{ kN}$$

$$B = -1.45 \pm 2\% \text{ kN}$$

$$C = 2 \pm 2\% \text{ kN}$$

$$M_1 = -0.55 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_2 = -2 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_3 = 0 \pm 2\% \text{ kN}\cdot\text{m}$$

Explanation:

The bending-moments are

$$M_1 = R_1(1.2 \text{ m}) = -0.45 \text{ kN} \times 1.2 \text{ m} = -0.55 \text{ kN}\cdot\text{m}$$

$$M_2 = -(-(R_1 - V_1)2.2 \text{ m} - 1.2 \text{ m}(V_1))$$

$$M_2 = (R_1)2.2 \text{ m} - (V_1) = (-0.45 \text{ kN} \times 2.2 \text{ m}) - 1 \text{ kN} = -2 \text{ kN}\cdot\text{m}$$

BELOW AVERAGE

Woodfork, Myles

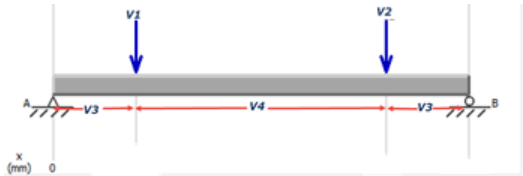
Score: 81.66/100 Points 81.66 %

1.

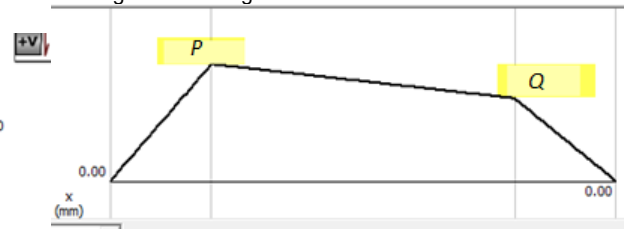
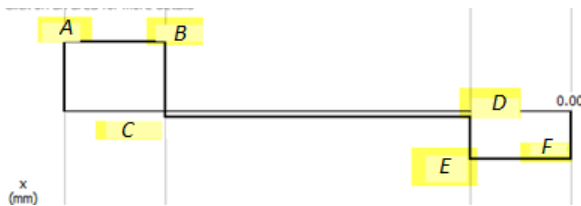
Award: 30 out of 30.00 points

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V1 = 9 \text{ kN}$, $V2 = 9 \text{ kN}$, $V3 = 100 \text{ mm}$, and $V4 = 1300 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = 9 ✓ kN and reaction force R_2 (right) = 9 ✓ kN.

In the shear-force and bending-moment diagrams given,

A = 9 ✓ kN
 B = 9 ✓ kN
 C = 0 ✓ kN
 D = 0 ✓ kN
 E = -9 ✓ kN
 F = -9 ✓ kN
 P = 900 ✓ kN·mm
 Q = 900 ✓ kN·mm

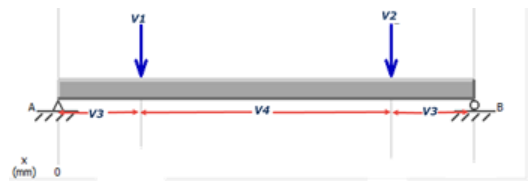
References

Numeric Response Chapter: 03 Load and Stress Analysis

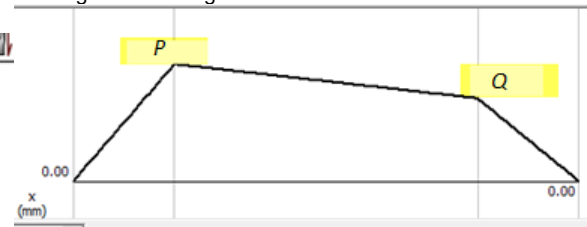
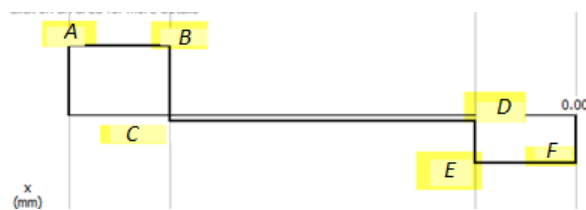
Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.005 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V1 = 9 \text{ kN}$, $V2 = 9 \text{ kN}$, $V3 = 100 \text{ mm}$, and $V4 = 1300 \text{ mm}$.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



Reaction force R_1 (left) = 9 ± 0.125 kN and reaction force R_2 (right) = 9 ± 0.125 kN.

In the shear-force and bending-moment diagrams given,

- A = 9 ± 0.125 kN
- B = 9 ± 0.125 kN
- C = 0 ± 0.125 kN
- D = 0 ± 0.125 kN
- E = -9 ± 0.125 kN
- F = -9 ± 0.125 kN
- P = 900 ± 0.25 kN·mm
- Q = 900 ± 0.25 kN·mm

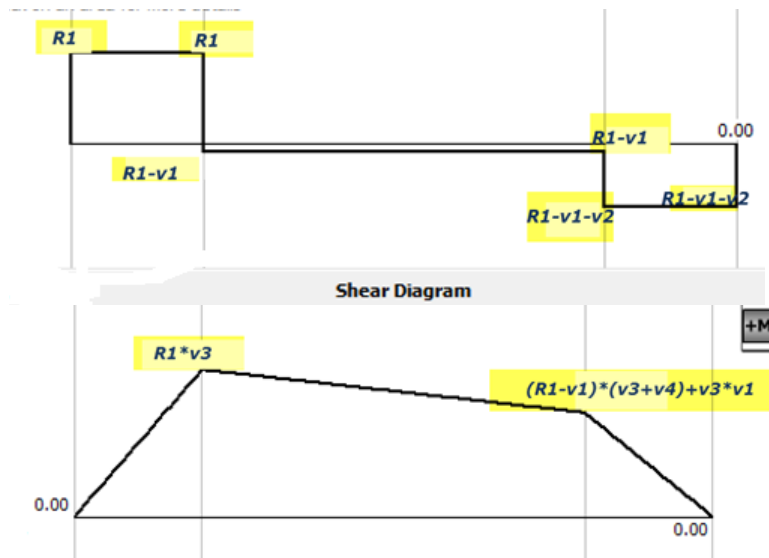
Explanation:

Taking the moment about point A,

$$\sum M_A = 0 = R_2(v_4 + 2v_3) - v_2(v_3 + v_4) - v_1(v_3)$$

$$R_2 = \frac{v_2(v_3 + v_4) + v_1(v_3)}{(v_4 + 2v_3)} = \frac{9 \text{ kN}(100 \text{ mm} + 1300 \text{ mm}) + 9 \text{ kN}(100)}{(1300 \text{ mm} + 2 \times 100 \text{ mm})} = 9 \text{ kN}$$

$$R_1 = v_1 + v_2 - R_2 = (9 + 9 - 9) \text{ kN} = 9 \text{ kN}$$



In the bending-moment diagram shown above, the bending-moment at each point is determined as follows:

$$P = R_1 v_3 = 9 \text{ kN} \times 100 \text{ mm} = 900 \text{ kN·mm}$$

$$Q = (R_1 - v_1)(v_3 + v_4) + (v_3 v_1) = (9 \text{ kN} - 9 \text{ kN})(100 + 1300) \text{ mm} + 9 \text{ kN}(100 \text{ mm}) = 900 \text{ kN·mm}$$

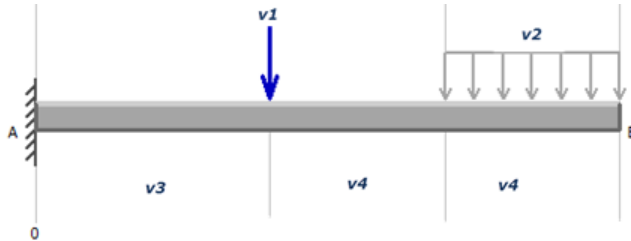


2.

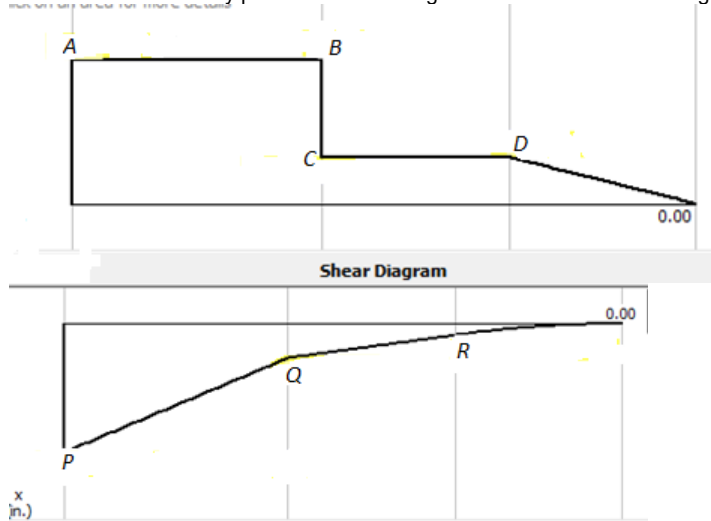
Award: 26.66 out of 30.00 points

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 300$ lbf, $V_2 = 34$ lbf/in, $V_3 = 8$ in, and $V_4 = 6$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_D = 504$ ✓ lbf.

The bending moment $M_D = 5868$ ✗ lbf·in.

In the shear-force and bending-moment diagrams given,

A = 504 ✓ lbf

B = 504 ✓ lbf

C = 204 ✓ lbf

D = 204 ✓ lbf

P = -5868 ✓ lbf·in

Q = -1836 ✓ lbf·in

R = -612 ✓ lbf·in

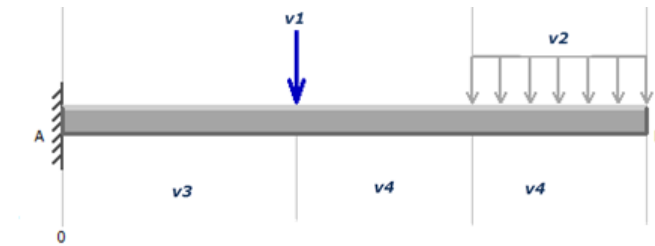
References

Numeric Response Chapter: 03 Load and Stress Analysis

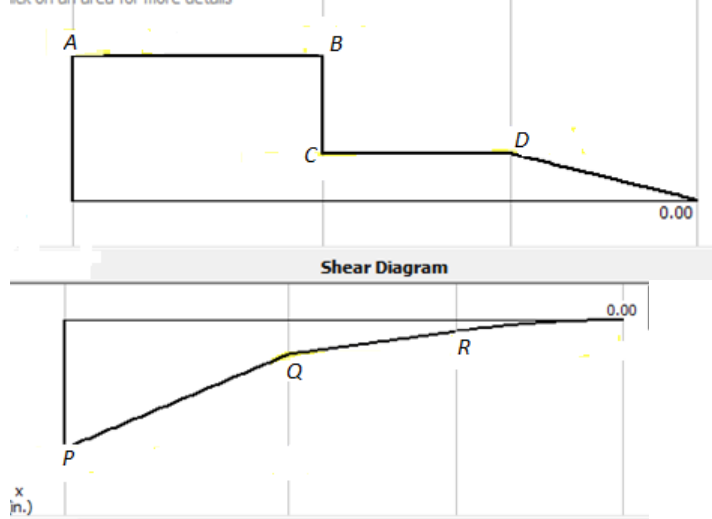
Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.006 - Find the reactions and plot the shear-force and bending-moment diagrams.

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 300$ lbf, $V_2 = 34$ lbf/in, $V_3 = 8$ in, and $V_4 = 6$ in.



Provide values at all key points shown in the given shear-force and bending-moment diagrams.



The reaction force $R_0 = 504 \pm 0.125$ lbf.

The bending moment $M_0 = -5868 \pm 2\%$ lbf-in.

In the shear-force and bending-moment diagrams given,

$A = 504 \pm 0.125$ lbf

$B = 504 \pm 0.125$ lbf

$C = 204 \pm 0.125$ lbf

$D = 204 \pm 0.125$ lbf

$P = -5868 \pm 0.25$ lbf-in

$Q = -1836 \pm 0.25$ lbf-in

$R = -612 \pm 0.25$ lbf-in

Explanation:

The bending-moments and the reaction force are

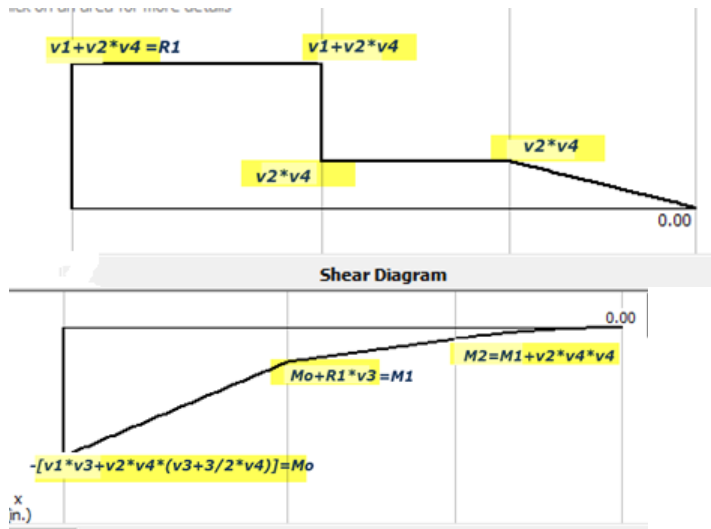
$$M_o = -v_1 v_3 - v_2 v_4 \left(\frac{3}{2} v_4 + v_3 \right)$$

$$M_o = -(300 \text{ lbf} \times 8 \text{ in}) - (34 \text{ lbf/in} \times 6 \text{ in}) \left(\frac{3}{2} 6 \text{ in} + 8 \text{ in} \right) = -5868 \text{ lbf-in}$$

$$R_o = v_1 + v_2 * v_4 = 300 \text{ lbf} + 34 \text{ lbf/in} \times 6 \text{ in} = 504 \text{ lbf}$$

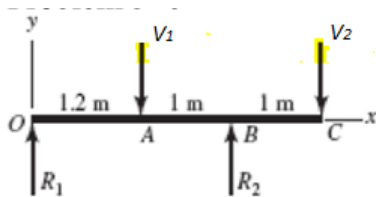
$$M_1 = M_o + R_o * v_3 = -5868 \text{ lbf-in} + (504 \text{ lbf} \times 8 \text{ in}) = -1836 \text{ lbf-in}$$

$$M_2 = M_1 + (R_o - v_1) v_4 = -1836 \text{ lbf-in} + (504 \text{ lbf} - 300 \text{ lbf}) 6 \text{ in} = -612 \text{ lbf-in}$$



Problem 03.007 - Find the reactions and plot the shear-force and bending-moment diagrams -
DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For the beam shown, find the reactions at the supports and plot the shear-force and bending-moment diagrams. $V_1 = 4$ kN and $V_2 = 8$ kN.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress
Analysis

Problem 03.007 - Find
the reactions and plot
the shear-force and
bending-moment
diagrams -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 03.02 Shear Force
and Bending Moments in
Beams

3.

Award: 5 out of 10.00 points

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = -1,8181$ ✖ kN and reaction force $R_2 = 13.8181$ ✔ kN

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams. Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.007.a - Find the reactions and plot the shear-force and bending-moment diagrams.

Find the reaction forces, R_1 and R_2 , in kN.

Reaction forces $R_1 = -1.8182 \pm 2\%$ kN and reaction force $R_2 = 13.8182 \pm 2\%$ kN

Explanation:

Taking the moment about point B,

$$\Sigma M_B = 0$$

$$-(2.2 \text{ m})R_1 + 1(V_1) - 1(V_2) = 0$$

$$R_1 = \frac{V_2 - V_1}{-2.2 \text{ m}} = \frac{8 \text{ kN} - 4 \text{ kN}}{-2.2 \text{ m}} = -1.82 \text{ kN}$$

Sum of forces along the vertical direction

$$\Sigma F_y = 0$$

$$R_1 + R_2 - V_1 - V_2 = 0$$

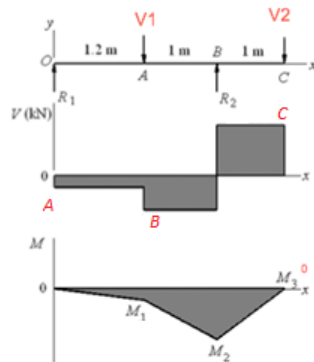
$$R_2 = V_1 + V_2 - R_1 = (4 + 8 - (-1.82)) \text{ kN} = 13.82 \text{ kN}$$

4.

Award: 20 out of 30.00 points

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in $\text{kN}\cdot\text{m}$. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

$A = -5.82$ ✗ kN

$B = -8$ ✗ kN

$C = 8$ ✓ kN

$M_1 = -2.18$ ✓ $\text{kN}\cdot\text{m}$

$M_2 = -8$ ✓ $\text{kN}\cdot\text{m}$

$M_3 = 0$ ✓ $\text{kN}\cdot\text{m}$

References

Numeric Response

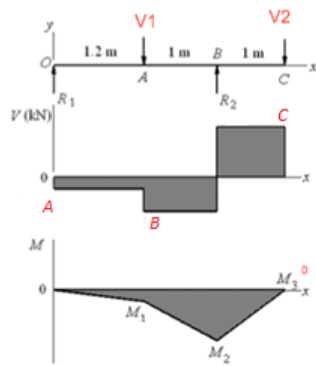
Chapter: 03 Load and Stress Analysis

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Section: 03.02 Shear Force and Bending Moments in Beams

Problem 03.007.b - Find the reactions and plot the shear-force and bending-moment diagrams.

Determine the values of the moments M_1 , M_2 , and M_3 in $\text{kN}\cdot\text{m}$. Provide values at all key points shown in the given shear-force and bending-moment diagrams.



In the shear-force and bending-moment diagrams given,

$$A = -1.82 \pm 2\% \text{ kN}$$

$$B = -5.82 \pm 2\% \text{ kN}$$

$$C = 8 \pm 2\% \text{ kN}$$

$$M_1 = -2.18 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_2 = -8 \pm 2\% \text{ kN}\cdot\text{m}$$

$$M_3 = 0 \pm 2\% \text{ kN}\cdot\text{m}$$

Explanation:

The bending-moments are

$$M_1 = R_1(1.2 \text{ m}) = -1.82 \text{ kN} \times 1.2 \text{ m} = -2.18 \text{ kN}\cdot\text{m}$$

$$M_2 = -(-(R_1 - V_1)2.2 \text{ m} - 1.2 \text{ m}(V_1))$$

$$M_2 = (R_1)2.2 \text{ m} - (V_1) = (-1.82 \text{ kN} \times 2.2 \text{ m}) - 4 \text{ kN} = -8 \text{ kN}\cdot\text{m}$$

MACHINE DESIGN I
Fall 2021

HOMEWORK 3

TOPICS COVERED IN HOMEWORK 3

Mohr's Circle: Problems 3-15, 3-16, 3-18

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 4

TOPICS COVERED IN HOMEWORK 4

Section 3-10: Normal Stresses for Beams in Bending

1.

Award: 20.00 points

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (kpsi, kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ kpsi and $\sigma_2 =$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} = 17 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2} = -7 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} + \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 26.2195 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ ksi}) + (24 \text{ ksi})}{2} - \sqrt{\left(\frac{(10 \text{ ksi}) - (24 \text{ ksi})}{2}\right)^2 + (6 \text{ ksi})^2} = 7.7805 \text{ ksi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(10 \text{ ksi}) - (24 \text{ ksi})}{2}\right)^2 + (6 \text{ ksi})^2} = 9.2195 \text{ ksi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{7 \text{ ksi}}{(6 \text{ ksi})} \right) \right) = 69.6994^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 69.6994^\circ - 45^\circ = 24.6994^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.2195 \text{ ksi}$$

The location of the center of the Mohr's circle C is (17 ksi, 0 ksi).

The principal normal stresses are $\sigma_1 = 26.2195 \text{ ksi}$ and $\sigma_2 = 7.7805 \text{ ksi}$.

The maximum shear stress is 9.2195 ksi.

The angle from the x axis to σ_1 (ϕ_p) is 69.6994° ccw.

The angle from the x axis to τ (ϕ_s) is 24.6994° ccw.

The radius of the Mohr's circle is 9.2195 ksi.

References

Worksheet

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses. Difficulty: Medium

2.

Award: 20.00 points

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (kpsi, kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ kpsi and $\sigma_2 =$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} = 5 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2} = -17 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} + \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = 25.8087 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} - \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = -15.8087 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = 20.8087 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{17 \text{ kpsi}}{(12 \text{ kpsi})} \right) \right) = 72.3912^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 72.3912^\circ - 45^\circ = 27.3912^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 20.8087 \text{ kpsi}$$

The location of the center of the Mohr's circle C is (5 kpsi, 0 kpsi).

The principal normal stresses are $\sigma_1 = 25.8087 \text{ kpsi}$ and $\sigma_2 = -15.8087 \text{ kpsi}$.

The maximum shear stress is 20.8087 kpsi.

The angle from the x axis to σ_1 (ϕ_p) is 72.3912° cw.

The angle from the x axis to τ (ϕ_s) is 27.3912° cw.

The radius of the Mohr's circle is 20.8087 kpsi.

References

Worksheet

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses. Difficulty: Medium

3.

Award: 20.00 points

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (kpsi, kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ kpsi and $\sigma_2 =$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} = 17 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2} = -7 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} + \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 26.2195 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ ksi}) + (24 \text{ ksi})}{2} - \sqrt{\left(\frac{(10 \text{ ksi}) - (24 \text{ ksi})}{2}\right)^2 + (6 \text{ ksi})^2} = 7.7805 \text{ ksi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(10 \text{ ksi}) - (24 \text{ ksi})}{2}\right)^2 + (6 \text{ ksi})^2} = 9.2195 \text{ ksi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{7 \text{ ksi}}{(6 \text{ ksi})} \right) \right) = 69.6994^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 69.6994^\circ - 45^\circ = 24.6994^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.2195 \text{ ksi}$$

The location of the center of the Mohr's circle C is (17 ksi, 0 ksi).

The principal normal stresses are $\sigma_1 = 26.2195 \text{ ksi}$ and $\sigma_2 = 7.7805 \text{ ksi}$.

The maximum shear stress is 9.2195 ksi.

The angle from the x axis to σ_1 (ϕ_p) is 69.6994° ccw.

The angle from the x axis to τ (ϕ_s) is 24.6994° ccw.

The radius of the Mohr's circle is 9.2195 ksi.

References

Worksheet

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses. Difficulty: Medium

4.

Award: 20.00 points

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (MPa, MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ MPa and $\sigma_2 =$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° .

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° .

6. The radius of the Mohr's circle

The radius of the Mohr's circle is MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} = -0.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2} = -7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} + \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.1 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} - \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = -10.1 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.6 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) \rightarrow \phi_p = \frac{1}{2}\tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) = \frac{1}{2}\tan^{-1}\left(\frac{(-6 \text{ MPa})}{0.5((-8 \text{ MPa}) - (7 \text{ MPa}))}\right) = 19.33^\circ = 19.33^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) \rightarrow \phi_s = \frac{1}{2}\tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) = \frac{1}{2}\tan^{-1}\left(-\frac{((-8 \text{ kpsi}) - (7 \text{ kpsi}))}{2(-6 \text{ kpsi})}\right) = -25.67^\circ = 25.67^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.6 \text{ MPa}$$

The location of the center of the Mohr's circle C is $(-0.5 \text{ MPa}, 0 \text{ MPa})$.

The principal normal stresses are $\sigma_1 = 9.1 \text{ MPa}$ and $\sigma_2 = -10.1 \text{ MPa}$.

The maximum shear stress is 9.6 MPa .

The angle from the x axis to σ_1 (ϕ_p) is 19.33° ccw.

The angle from the x axis to τ (ϕ_s) is 25.67° cw.

The radius of the Mohr's circle is 9.6 MPa .

References

Multi-Answer

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses. Difficulty: Medium

5.

Award: 20.00 points

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (MPa, MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ MPa and $\sigma_2 =$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° .

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° .

6. The radius of the Mohr's circle

The radius of the Mohr's circle is MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} = -0.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2} = -7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} + \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.1 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} - \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = -10.1 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.6 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) \rightarrow \phi_p = \frac{1}{2}\tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) = \frac{1}{2}\tan^{-1}\left(\frac{(-6 \text{ MPa})}{0.5((-8 \text{ MPa}) - (7 \text{ MPa}))}\right) = 19.33^\circ = 19.33^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) \rightarrow \phi_s = \frac{1}{2}\tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) = \frac{1}{2}\tan^{-1}\left(-\frac{((-8 \text{ kpsi}) - (7 \text{ kpsi}))}{2(-6 \text{ kpsi})}\right) = -25.67^\circ = 25.67^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.6 \text{ MPa}$$

The location of the center of the Mohr's circle C is $(-0.5 \text{ MPa}, 0 \text{ MPa})$.

The principal normal stresses are $\sigma_1 = 9.1 \text{ MPa}$ and $\sigma_2 = -10.1 \text{ MPa}$.

The maximum shear stress is 9.6 MPa .

The angle from the x axis to σ_1 (ϕ_p) is 19.33° ccw.

The angle from the x axis to τ (ϕ_s) is 25.67° cw.

The radius of the Mohr's circle is 9.6 MPa .

References

Multi-Answer

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses. Difficulty: Medium

ABOVE AVERAGE

Score: **100/100** Points **100** %

1.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (**5**  kpsi, **0**  kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \mathbf{25.8}$  kpsi and $\sigma_2 = \mathbf{-15.81}$  kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is **20.81**  kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is **72.39**  ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is **27.39**  ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is **20.81**  kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ± 0.5 kpsi, 0 ± 0.5 kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 25.81 \pm 0.5$ kpsi and $\sigma_2 = -15.81 \pm 0.5$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.81 ± 0.5 kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $72.39 \pm 1^\circ$ cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $27.39 \pm 1^\circ$ cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.81 ± 0.5 kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} = 5 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2} = -17 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} + \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = 25.8087 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} - \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = -15.8087 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = 20.8087 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{17 \text{ kpsi}}{12 \text{ kpsi}} \right) \right) = 72.3912^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 72.3912^\circ - 45^\circ = 27.3912^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 20.8087 \text{ kpsi}$$

2.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$\sigma_x = 10$ kpsi, $\sigma_y = 24$ kpsi, $\tau_{xy} = 6$ kpsi ccw

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (17  kpsi, 0  kpsi).

2. Principal normal stresses (σ_1 , σ_2)

The principal normal stresses are $\sigma_1 =$ 26.22  kpsi and $\sigma_2 =$ 7.78  kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 9.22  kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 69.7  ° ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 24.7  ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.22  kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (17 ± 0.5 kpsi, 0 ± 0.5 kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 26.22 \pm 0.5$ kpsi and $\sigma_2 = 7.78 \pm 0.5$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 9.22 ± 0.5 kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $69.7 \pm 1^\circ$ ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $24.7 \pm 1^\circ$ ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.22 ± 0.5 kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} = 17 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2} = -7 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} + \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 26.2195 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} - \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 7.7805 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 9.2195 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{7 \text{ kpsi}}{6 \text{ kpsi}} \right) \right) = 69.6994^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 69.6994^\circ - 45^\circ = 24.6994^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.2195 \text{ kpsi}$$

3.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5  kpsi, 0  kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{25.81}$  kpsi and $\sigma_2 = \underline{-15.81}$  kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.81  kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 72.39  ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 27.39  ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.81  kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ± 0.5 kpsi, 0 ± 0.5 kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 25.81 \pm 0.5$ kpsi and $\sigma_2 = -15.81 \pm 0.5$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.81 ± 0.5 kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $72.39 \pm 1^\circ$ cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $27.39 \pm 1^\circ$ cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.81 ± 0.5 kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} = 5 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2} = -17 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} + \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = 25.8087 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ kpsi}) + (22 \text{ kpsi})}{2} - \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = -15.8087 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-12 \text{ kpsi}) - (22 \text{ kpsi})}{2}\right)^2 + (12 \text{ kpsi})^2} = 20.8087 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{17 \text{ kpsi}}{12 \text{ kpsi}} \right) \right) = 72.3912^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 72.3912^\circ - 45^\circ = 27.3912^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 20.8087 \text{ kpsi}$$

4.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is **(-0.5 ✓ MPa, 0 ✓ MPa)**.

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \mathbf{9.1 \text{ ✓ MPa}}$ and $\sigma_2 = \mathbf{-10.1 \text{ ✓ MPa}}$.

3. Maximum shear stress (τ)

The maximum shear stress is **9.6 ✓ MPa**.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is **19.33 ✓ ° ccw ✓**.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is **25.67 ✓ ° cw ✓**.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is **9.6 ✓ MPa**.

References

Multi-Answer

Chapter: 03 Load and Stress Analysis

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (-0.5 ± 0.5 MPa, 0 ± 0.5 MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 9.1 \pm 0.5$ MPa and $\sigma_2 = -10.1 \pm 0.5$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is 9.6 ± 0.5 MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $19.33 \pm 1^\circ$ ccw ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $25.67 \pm 1^\circ$ cw ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.6 ± 0.5 MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} = -0.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2} = -7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} + \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.1 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} - \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = -10.1 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.6 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) \rightarrow \phi_p = \frac{1}{2} \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) = \frac{1}{2} \tan^{-1} \left(\frac{(-6 \text{ MPa})}{0.5((-8 \text{ MPa}) - (7 \text{ MPa}))} \right) = 19.33^\circ = 19.33^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) \rightarrow \phi_s = \frac{1}{2} \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) = \frac{1}{2} \tan^{-1} \left(-\frac{((-8 \text{ kpsi}) - (7 \text{ kpsi}))}{2(-6 \text{ kpsi})} \right) = -25.67^\circ = 25.67^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.6 \text{ MPa}$$

5.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is **(-0.5**  MPa, **0**  MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \mathbf{9.11}$  MPa and $\sigma_2 = \mathbf{-10.1}$  MPa.

3. Maximum shear stress (τ)

The maximum shear stress is **9.6**  MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is **19.33**  ° **ccw** .

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is **25.67**  ° **cw** .

6. The radius of the Mohr's circle

The radius of the Mohr's circle is **9.6**  MPa.

References

Multi-Answer

Chapter: 03 Load and Stress Analysis

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (-0.5 ± 0.5 MPa, 0 ± 0.5 MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 9.1 \pm 0.5$ MPa and $\sigma_2 = -10.1 \pm 0.5$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is 9.6 ± 0.5 MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $19.33 \pm 1^\circ$ ccw ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $25.67 \pm 1^\circ$ cw ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.6 ± 0.5 MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} = -0.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2} = -7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} + \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.1 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} - \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = -10.1 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.6 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) \rightarrow \phi_p = \frac{1}{2} \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) = \frac{1}{2} \tan^{-1} \left(\frac{(-6 \text{ MPa})}{0.5((-8 \text{ MPa}) - (7 \text{ MPa}))} \right) = 19.33^\circ = 19.33^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) \rightarrow \phi_s = \frac{1}{2} \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) = \frac{1}{2} \tan^{-1} \left(-\frac{((-8 \text{ kpsi}) - (7 \text{ kpsi}))}{2(-6 \text{ kpsi})} \right) = -25.67^\circ = 25.67^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.6 \text{ MPa}$$

AVERAGE

Score: **84/100** Points **84** %

1.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (17  kpsi, 0  kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ 26.22  kpsi and $\sigma_2 =$ 7.78  kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 9.22  kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 69.7  ° ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 24.7  ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.22  kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (kpsi, kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ kpsi and $\sigma_2 =$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is kpsi.

4. The angle from the x axis to σ_1 (Φ_p)

The angle from the x axis to σ_1 (Φ_p) is ° ccw.

5. The angle from the x axis to τ (Φ_s)

The angle from the x axis to τ (Φ_s) is ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} = 17 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2} = -7 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} + \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 26.2195 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} - \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 7.7805 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 9.2195 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{7 \text{ kpsi}}{(6 \text{ kpsi})} \right) \right) = 69.6994^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 69.6994^\circ - 45^\circ = 24.6994^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.2195 \text{ kpsi}$$

2.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (17  kpsi, 0  kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ 26.22  kpsi and $\sigma_2 =$ 7.78  kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 9.22  kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 69.7  ° ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 24.7  ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.22  kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (kpsi, kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ kpsi and $\sigma_2 =$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is kpsi.

4. The angle from the x axis to σ_1 (Φ_p)

The angle from the x axis to σ_1 (Φ_p) is ° ccw.

5. The angle from the x axis to τ (Φ_s)

The angle from the x axis to τ (Φ_s) is ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} = 17 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2} = -7 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} + \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 26.2195 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} - \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 7.7805 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 9.2195 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{7 \text{ kpsi}}{(6 \text{ kpsi})} \right) \right) = 69.6994^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 69.6994^\circ - 45^\circ = 24.6994^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.2195 \text{ kpsi}$$

3.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$\sigma_x = 10$ kpsi, $\sigma_y = 24$ kpsi, $\tau_{xy} = 6$ kpsi ccw

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (**17**  kpsi, **0**  kpsi).

2. Principal normal stresses (σ_1 , σ_2)

The principal normal stresses are $\sigma_1 =$ **26.22**  kpsi and $\sigma_2 =$ **7.78**  kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is **9.22**  kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is **69.7**  ° ccw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is **24.7**  ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is **9.22**  kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 10 \text{ kpsi}, \sigma_y = 24 \text{ kpsi}, \tau_{xy} = 6 \text{ kpsi ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (kpsi, kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ kpsi and $\sigma_2 =$ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is kpsi.

4. The angle from the x axis to σ_1 (Φ_p)

The angle from the x axis to σ_1 (Φ_p) is ° ccw.

5. The angle from the x axis to τ (Φ_s)

The angle from the x axis to τ (Φ_s) is ° ccw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is kpsi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} = 17 \text{ kpsi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2} = -7 \text{ kpsi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} + \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 26.2195 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(10 \text{ kpsi}) + (24 \text{ kpsi})}{2} - \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 7.7805 \text{ kpsi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(10 \text{ kpsi}) - (24 \text{ kpsi})}{2}\right)^2 + (6 \text{ kpsi})^2} = 9.2195 \text{ kpsi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{7 \text{ kpsi}}{(6 \text{ kpsi})} \right) \right) = 69.6994^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 69.6994^\circ - 45^\circ = 24.6994^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.2195 \text{ kpsi}$$

4.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (-0.5  MPa, 0  MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{9.1}$  MPa and $\sigma_2 = \underline{-10.1}$  MPa.

3. Maximum shear stress (τ)

The maximum shear stress is 9.6  MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 19.33  ° ccw .

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 25.67  ° cw .

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.6  MPa.

References

Multi-Answer

Chapter: 03 Load and Stress Analysis

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -8 \text{ MPa}, \sigma_y = 7 \text{ MPa}, \tau_{xy} = 6 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (MPa, MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ MPa and $\sigma_2 =$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° ccw ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° cw ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} = -0.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2} = -7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} + \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.1 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-8 \text{ MPa}) + (7 \text{ MPa})}{2} - \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = -10.1 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-8 \text{ MPa}) - (7 \text{ MPa})}{2}\right)^2 + (-6 \text{ MPa})^2} = 9.6 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) \rightarrow \phi_p = \frac{1}{2}\tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) = \frac{1}{2}\tan^{-1}\left(\frac{(-6 \text{ MPa})}{0.5((-8 \text{ MPa}) - (7 \text{ MPa}))}\right) = 19.33^\circ = 19.33^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) \rightarrow \phi_s = \frac{1}{2}\tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) = \frac{1}{2}\tan^{-1}\left(-\frac{((-8 \text{ kpsi}) - (7 \text{ kpsi}))}{2(-6 \text{ kpsi})}\right) = -25.67^\circ = 25.67^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.6 \text{ MPa}$$

5.

Award: 4 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 9 \text{ MPa}, \sigma_y = -6 \text{ MPa}, \tau_{xy} = 3 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is **-0.5** ✖ MPa, **0** ✔ MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{\mathbf{9.1}}$ ✔ MPa and $\sigma_2 = \underline{\mathbf{-10.1}}$ ✖ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is **9.6** ✖ MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is **19.33** ✖ ° **ccw** ✖.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is **25.67** ✖ ° **cw** ✖.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is **9.6** ✖ MPa.

References

Multi-Answer

Chapter: 03 Load and Stress Analysis

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 9 \text{ MPa}, \sigma_y = -6 \text{ MPa}, \tau_{xy} = 3 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (MPa, MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 =$ MPa and $\sigma_2 =$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° cw ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° ccw ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(9 \text{ MPa}) + (-6 \text{ MPa})}{2} = 1.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2} = 7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(9 \text{ MPa}) + (-6 \text{ MPa})}{2} + \sqrt{\left(\frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2}\right)^2 + (-3 \text{ MPa})^2} = 9.58 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(9 \text{ MPa}) + (-6 \text{ MPa})}{2} - \sqrt{\left(\frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2}\right)^2 + (-3 \text{ MPa})^2} = -6.58 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2}\right)^2 + (-3 \text{ MPa})^2} = 8.08 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) \rightarrow \phi_p = \frac{1}{2}\tan^{-1}\left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)}\right) = \frac{1}{2}\tan^{-1}\left(\frac{(-3 \text{ MPa})}{0.5((9 \text{ MPa}) - (-6 \text{ MPa}))}\right) = -10.90^\circ = 10.90^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) \rightarrow \phi_s = \frac{1}{2}\tan^{-1}\left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}}\right) = \frac{1}{2}\tan^{-1}\left(-\frac{((9 \text{ kpsi}) - (-6 \text{ kpsi}))}{2(-3 \text{ kpsi})}\right) = 34.10^\circ = 34.10^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 8.08 \text{ MPa}$$

BELOW AVERAGE

Score: **65/100** Points **65** %

1.

Award: 15 out of 20.00 points

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ✓ kpsi, 0 ✓ kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{25.81}$ ✓ kpsi and $\sigma_2 = \underline{-15.81}$ ✓ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.8 ✓ kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 17.6 ✗ ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 62.61 ✗ ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.8 ✓ kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ ksi}, \sigma_y = 22 \text{ ksi}, \tau_{xy} = 12 \text{ ksi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ± 0.5 ksi, 0 ± 0.5 ksi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 25.81 \pm 0.5$ ksi and $\sigma_2 = -15.81 \pm 0.5$ ksi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.81 ± 0.5 ksi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $72.39 \pm 1^\circ$ cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $27.39 \pm 1^\circ$ cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.81 ± 0.5 ksi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} = 5 \text{ ksi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2} = -17 \text{ ksi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} + \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = 25.8087 \text{ ksi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} - \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = -15.8087 \text{ ksi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = 20.8087 \text{ ksi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{17 \text{ ksi}}{12 \text{ ksi}} \right) \right) = 72.3912^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 72.3912^\circ - 45^\circ = 27.3912^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 20.8087 \text{ ksi}$$

2.

Award: 15 out of 20.00 points

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ✓ kpsi, 0 ✓ kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{25.81}$ ✓ kpsi and $\sigma_2 = \underline{-15.81}$ ✓ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.8 ✓ kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 17.6 ✗ ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 62.61 ✗ ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.8 ✓ kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ ksi}, \sigma_y = 22 \text{ ksi}, \tau_{xy} = 12 \text{ ksi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ± 0.5 ksi, 0 ± 0.5 ksi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 25.81 \pm 0.5$ ksi and $\sigma_2 = -15.81 \pm 0.5$ ksi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.81 ± 0.5 ksi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $72.39 \pm 1^\circ$ cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $27.39 \pm 1^\circ$ cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.81 ± 0.5 ksi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} = 5 \text{ ksi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2} = -17 \text{ ksi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} + \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = 25.8087 \text{ ksi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} - \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = -15.8087 \text{ ksi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = 20.8087 \text{ ksi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{17 \text{ ksi}}{12 \text{ ksi}} \right) \right) = 72.3912^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 72.3912^\circ - 45^\circ = 27.3912^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 20.8087 \text{ ksi}$$

3.

Award: 15 out of 20.00 points

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ kpsi}, \sigma_y = 22 \text{ kpsi}, \tau_{xy} = 12 \text{ kpsi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (5 ✓ kpsi, 0 ✓ kpsi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{25.81}$ ✓ kpsi and $\sigma_2 = \underline{-15.81}$ ✓ kpsi.

3. Maximum shear stress (τ)

The maximum shear stress is 20.8 ✓ kpsi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 17.6 ✗ ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 62.61 ✗ ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 20.8 ✓ kpsi.

References

Worksheet

Chapter: 03 Load and Stress Analysis

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.015 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = -12 \text{ ksi}, \sigma_y = 22 \text{ ksi}, \tau_{xy} = 12 \text{ ksi cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (ksi, ksi).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \text{ ksi}$ and $\sigma_2 = \text{ ksi}$.

3. Maximum shear stress (τ)

The maximum shear stress is ksi.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° cw.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° cw.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is ksi.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} = 5 \text{ ksi}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2} = -17 \text{ ksi}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} + \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = 25.8087 \text{ ksi}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(-12 \text{ ksi}) + (22 \text{ ksi})}{2} - \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = -15.8087 \text{ ksi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(-12 \text{ ksi}) - (22 \text{ ksi})}{2}\right)^2 + (12 \text{ ksi})^2} = 20.8087 \text{ ksi}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$\phi_p = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{CD}{\tau_{xy}} \right) \right) = \frac{1}{2} \left(90 + \tan^{-1} \left(\frac{17 \text{ ksi}}{12 \text{ ksi}} \right) \right) = 72.3912^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$\phi_s = 72.3912^\circ - 45^\circ = 27.3912^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 20.8087 \text{ ksi}$$

4.

Award: 18 out of 20.00 points

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 9 \text{ MPa}, \sigma_y = -6 \text{ MPa}, \tau_{xy} = 3 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (1.5 ✓ MPa, 0 ✓ MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{9.58}$ ✓ MPa and $\sigma_2 = \underline{-6.58}$ ✓ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is 8.07 ✓ MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 10.9 ✓ ° **cw** ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 124.09 ✗ ° **ccw** ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 8.07 ✓ MPa.

References

Multi-Answer

Chapter: 03 Load and Stress Analysis

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 9 \text{ MPa}, \sigma_y = -6 \text{ MPa}, \tau_{xy} = 3 \text{ MPa cw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (MPa, MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \text{ MPa}$ and $\sigma_2 = \text{ MPa}$.

3. Maximum shear stress (τ)

The maximum shear stress is MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is ° cw ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is ° ccw ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(9 \text{ MPa}) + (-6 \text{ MPa})}{2} = 1.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2} = 7.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(9 \text{ MPa}) + (-6 \text{ MPa})}{2} + \sqrt{\left(\frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2}\right)^2 + (-3 \text{ MPa})^2} = 9.58 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(9 \text{ MPa}) + (-6 \text{ MPa})}{2} - \sqrt{\left(\frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2}\right)^2 + (-3 \text{ MPa})^2} = -6.58 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(9 \text{ MPa}) - (-6 \text{ MPa})}{2}\right)^2 + (-3 \text{ MPa})^2} = 8.08 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) \rightarrow \phi_p = \frac{1}{2} \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) = \frac{1}{2} \tan^{-1} \left(\frac{(-3 \text{ MPa})}{0.5((9 \text{ MPa}) - (-6 \text{ MPa}))} \right) = -10.90^\circ = 10.90^\circ \text{ cw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) \rightarrow \phi_s = \frac{1}{2} \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) = \frac{1}{2} \tan^{-1} \left(-\frac{((9 \text{ kpsi}) - (-6 \text{ kpsi}))}{2(-3 \text{ kpsi})} \right) = 34.10^\circ = 34.10^\circ \text{ ccw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 8.08 \text{ MPa}$$

5.

Award: 2 out of 20.00 points

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 6 \text{ MPa}, \sigma_y = -5 \text{ MPa}, \tau_{xy} = 8 \text{ MPa ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (1.5 ✖ MPa, 0 ✔ MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = \underline{9.58}$ ✖ MPa and $\sigma_2 = \underline{-6.58}$ ✖ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is 8.07 ✖ MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is 10.9 ✖ ° ccw ✖.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is 124.09 ✖ ° ccw ✖.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 8.07 ✖ MPa.

References

Multi-Answer

Chapter: 03 Load and Stress Analysis

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Section: 03.06 Mohr's Circle for Plane Stress

Problem 03.016 - Find the center and radius for Mohr's circle and determine the principal normal and shear stresses.

Consider the following plane stress state:

$$\sigma_x = 6 \text{ MPa}, \sigma_y = -5 \text{ MPa}, \tau_{xy} = 8 \text{ MPa ccw}$$

Calculate the following:

1. The coordinates of the center of the Mohr's circle C

The location of the center of the Mohr's circle C is (0.5 ± 0.5 MPa, 0 ± 0.5 MPa).

2. Principal normal stresses (σ_1, σ_2)

The principal normal stresses are $\sigma_1 = 10.21 \pm 0.5$ MPa and $\sigma_2 = -9.21 \pm 0.5$ MPa.

3. Maximum shear stress (τ)

The maximum shear stress is 9.71 ± 0.5 MPa.

4. The angle from the x axis to σ_1 (ϕ_p)

The angle from the x axis to σ_1 (ϕ_p) is $27.75 \pm 1^\circ$ ccw ✓.

5. The angle from the x axis to τ (ϕ_s)

The angle from the x axis to τ (ϕ_s) is $17.25 \pm 1^\circ$ cw ✓.

6. The radius of the Mohr's circle

The radius of the Mohr's circle is 9.71 ± 0.5 MPa.

Explanation:

C is the center of the Mohr's circle.

$$C = \frac{\sigma_x + \sigma_y}{2} = \frac{(6 \text{ MPa}) + (-5 \text{ MPa})}{2} = 0.5 \text{ MPa}$$

$$CD = \frac{\sigma_x - \sigma_y}{2} = \frac{(6 \text{ MPa}) - (-5 \text{ MPa})}{2} = 5.5 \text{ MPa}$$

The principal stresses are

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(6 \text{ MPa}) + (-5 \text{ MPa})}{2} + \sqrt{\left(\frac{(6 \text{ MPa}) - (-5 \text{ MPa})}{2}\right)^2 + (8 \text{ MPa})^2} = 10.21 \text{ MPa}$$

$$\sigma_2 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \frac{(6 \text{ MPa}) + (-5 \text{ MPa})}{2} - \sqrt{\left(\frac{(6 \text{ MPa}) - (-5 \text{ MPa})}{2}\right)^2 + (8 \text{ MPa})^2} = -9.21 \text{ MPa}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + (\tau_{xy})^2} = \sqrt{\left(\frac{(6 \text{ MPa}) - (-5 \text{ MPa})}{2}\right)^2 + (8 \text{ MPa})^2} = 9.71 \text{ MPa}$$

The angle from the x axis to σ_1 (ϕ_p) is

$$2\phi_p = \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) \rightarrow \phi_p = \frac{1}{2} \tan^{-1} \left(\frac{\tau_{xy}}{0.5(\sigma_x - \sigma_y)} \right) = \frac{1}{2} \tan^{-1} \left(\frac{(8 \text{ MPa})}{0.5((6 \text{ MPa}) - (-5 \text{ MPa}))} \right) = 27.75^\circ = 27.75^\circ \text{ ccw}$$

The angle from the x axis to τ (ϕ_s) is

$$2\phi_s = \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) \rightarrow \phi_s = \frac{1}{2} \tan^{-1} \left(-\frac{(\sigma_x - \sigma_y)}{2\tau_{xy}} \right) = \frac{1}{2} \tan^{-1} \left(-\frac{((6 \text{ kpsi}) - (-5 \text{ kpsi}))}{2(8 \text{ kpsi})} \right) = -17.25^\circ = 17.25^\circ \text{ cw}$$

The radius of the Mohr's circle is

$$R = \tau_{\max} = 9.71 \text{ MPa}$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 5

TOPICS COVERED IN HOMEWORK 5

Sections 3-10: Problems 3-35

**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS, SOLUTION,
AND GRADING SCHEME**

Problem 03.035 - Find the cross-sectional properties for a beam

Find the second moment of area, the location of the neutral axis, and the distances from the neutral axis to the top and bottom surfaces. Consider that the section is transmitting a positive bending moment about the z axis, M_z , where $M_z = 10$ kip-in if the dimensions of the section are given in ips units, or $M_z = 1.13$ kN-m if the dimensions are in SI units. Determine the resulting stresses at the top and bottom surfaces and at every abrupt change in the cross section.

References

Section Break

Problem 03.035 - Find the cross-sectional properties for a beam

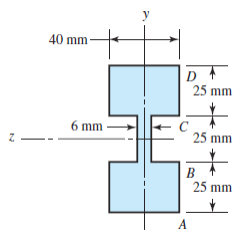
Difficulty: Hard

1.

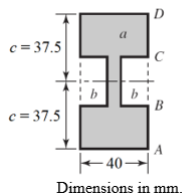
Award: 25.00 points Problems? [Adjust credit](#) for all students.[Required information](#)

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be $2150 \pm 2\%$ mm².The distance from the neutral axis to the top and bottom surfaces is determined to be $37.5 \pm 2\%$ mm.The second moment of area is determined to be $1.36 \pm 2\% \times 10^6$ mm⁴.

Explanation:

Let a be the total area of the entire envelope and b be the area of the side notch.

The total area of the I-section is calculated as follows:

$$A = a - 2b = 40 \text{ mm} \times (2 \times 37.5 \text{ mm}) - (25 \text{ mm} \times 34 \text{ mm}) = 2150 \text{ mm}^2$$

The second-area moment is calculated as follows:

$$I = I_a - 2I_b = \left(\frac{1}{12} \times 40 \text{ mm} \times (75 \text{ mm})^3 \right) - \left(\frac{1}{12} \times 34 \text{ mm} \times (25 \text{ mm})^3 \right) = 1.36 \times 10^6 \text{ mm}^4$$

The area is determined to be 2150 mm².

The distance from the neutral axis to the top and bottom surfaces is determined to be 37.5 mm.

The second moment of inertia is determined to be 1.36×10^6 mm⁴.

Hints

References

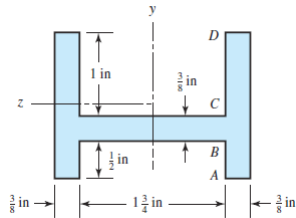
[Hint #1](#)

2.

Award: 25.00 points Problems? [Adjust credit](#) for all students.[Required information](#)

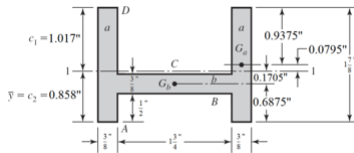
Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure

The area is determined to be $2.0625 \pm 2\%$ in².The distances from the neutral axis to the top and bottom surfaces are determined to be $1.017 \pm 2\%$ in and $0.858 \pm 2\%$ in.The second moment of area is determined to be $0.448 \pm 2\%$ in⁴.**Explanation:**

The total area of the figure is calculated as follows:

$$A = (A_a + A_b) = (0.375 \text{ in} \times 1.875 \text{ in}) + (0.375 \text{ in} \times 1.75 \text{ in}) = 2.0625 \text{ in}^2$$



The neutral axis from the bottom of the figure is calculated as follows:

$$\bar{y} = \frac{2 \times (0.703125 \text{ in}^2 \times (0.9375 \text{ in})) + (0.65625 \text{ in}^2 \times (0.6875 \text{ in}))}{2.0625 \text{ in}^2} = 0.858 \text{ in}$$

The second-area moment of section 'a' is calculated as follows:

$$I_a = \frac{0.375 \text{ in} \times (1.875 \text{ in})^3}{12} = 0.206 \text{ in}^4$$

The second-area moment of section 'b' is calculated as follows:

$$I_b = \frac{1.75 \text{ in} \times (0.375 \text{ in})^3}{12} = 0.00769 \text{ in}^4$$

The total second-area moment of the figure is calculated as follows:

$$I = 2 \times (0.206 \text{ in}^4 + 0.703125 \text{ in}^2 \times (0.0795 \text{ in})^2) + (0.00769 \text{ in}^4 + 0.065625 \text{ in}^2 \times (0.1705 \text{ in})^2) = 0.448 \text{ in}^4$$

The area is determined to be 2.0625 in².

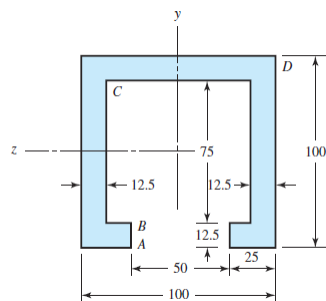
The distances from the neutral axis to the top and bottom surfaces are determined to be 1.017 in and 0.858 in.

The second moment of area is determined to be 0.448 in⁴.[Hints](#)[References](#)[Hint #1](#)

Award: 25.00 points Problems? [Adjust credit](#) for all students.

Required information

From the figure,



The distances from the neutral axis to the top and bottom surfaces are determined to be $42.71 \pm 2\%$ mm and $57.29 \pm 2\%$ mm

The second moment of area is determined to be $4.29 \pm 2\% \times 10^6 \text{ mm}^4$.

Explanation:

The area under section 'a' is calculated as follows:

$$A_g = 12.5 \text{ mm} \times 50 \text{ mm} = 625 \text{ mm}^2$$

The area under section 'b' is calculated as follows:

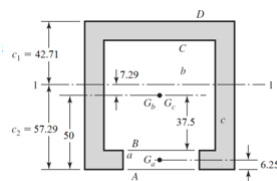
$$A_b = 75 \text{ mm} \times 75 \text{ mm} = 5625 \text{ mm}^2$$

The area under section 'c' is calculated as follows:

$$A_c = 100 \text{ mm} \times 100 \text{ mm} = 10000 \text{ mm}^2$$

The total area of the figure is calculated as follows:

$$A = 10000 \text{ mm}^2 - 5625 \text{ mm}^2 - 625 \text{ mm}^2 = 3750 \text{ mm}^2$$



The distances from the bottom to the neutral axis are calculated for the respective sections are as follows:

$$\bar{y}_a = 6.25 \text{ mm}, \bar{y}_b = 50 \text{ mm}, \bar{y}_c = 50 \text{ mm}$$

$$\bar{y} = \frac{(10000 \text{ mm}^2 \times 50 \text{ mm}) - (5625 \text{ mm}^2 \times 50 \text{ mm}) - (625 \text{ mm}^2 \times 6.25 \text{ mm})}{3750 \text{ mm}^2} = 57.29 \text{ mm}$$

$$c_1 = 100 \text{ mm} - 57.29 \text{ mm} = 42.71 \text{ mm}$$

The second-area moment of section 'a' is calculated as follows:

$$I_a = \frac{50 \text{ mm} \times (12.5 \text{ mm})^3}{12} = 8138 \text{ mm}^4$$

The second-area moment of section 'b' is calculated as follows:

$$I_b = \frac{75 \text{ mm} \times (75 \text{ mm})^3}{12} = 2.637 \times 10^6 \text{ mm}^4$$

The second-area moment of section 'c' is calculated as follows:

$$I_c = \frac{100 \text{ mm} \times (100 \text{ mm})^3}{12} = 8.333 \times 10^6 \text{ mm}^4$$

The total second-area moment of the figure is calculated as follows:

$$I = \left(8.333 \times 10^6 \text{ mm}^4 + \left(10000 \text{ mm}^2 \times (7.29 \text{ mm})^2 \right) \right) - \left(2.637 \times 10^6 \text{ mm}^4 + \left(5625 \text{ mm}^2 \times (7.29 \text{ mm})^2 \right) \right) - \left(8138 \text{ mm}^4 + \left(625 \text{ mm}^2 \times (51.04 \text{ mm})^2 \right) \right)$$

$$I = 4.29 \times 10^6 \text{ mm}^4$$

The area is determined to be 3750 mm^2 .

The distances from the neutral axis to the top and bottom surfaces are determined to be 42.71 mm and 57.29 mm.

The second-area moment is determined to be $4.29 \times 10^6 \text{ mm}^4$.

Hints

References

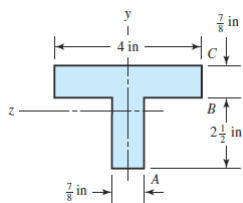
Hint #1

4.

Award: 25.00 points Problems? [Adjust credit](#) for all students.[Required information](#)

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be $5.6875 \pm 2\%$ in².The distances from the neutral axis to the top and bottom surfaces are determined to be $1.087 \pm 2\%$ in and $2.288 \pm 2\%$ in.The second moment of area is determined to be $5.2 \pm 2\%$ in⁴.**Explanation:**

The area under section 'a' is calculated as follows:

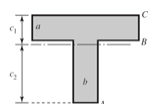
$$A_a = 4 \text{ in} \times 0.875 \text{ in} = 3.5 \text{ in}^2$$

The area under section 'b' is calculated as follows:

$$A_b = 2.5 \text{ in} \times 0.875 \text{ in} = 2.1875 \text{ in}^2$$

The total area of the figure is calculated as follows:

$$A = 3.5 \text{ in}^2 + 2.1875 \text{ in}^2 = 5.6875 \text{ in}^2$$



The distance from the bottom surface is calculated as follows:

$$\bar{y} = \frac{(2.9375 \text{ in}^2 \times 3.5 \text{ in}) + (1.25 \text{ in}^2 \times 2.1875 \text{ in})}{5.6875 \text{ in}^2} = 2.288 \text{ in}$$

$$c_2 = 2.288 \text{ in}$$

$$c_1 = 1.087 \text{ in}$$

The total second-area moment of the figure is calculated as follows:

$$I = \left(\frac{1}{12} \times 4 \text{ in} \times (0.875 \text{ in})^3 \right) + \left(3.5 \text{ in}^2 \times (2.9375 - 2.288 \text{ in})^2 \right) + \left(\frac{1}{12} \times 0.875 \text{ in} \times (2.5 \text{ in})^3 \right) + \left(2.1875 \text{ in}^2 \times (2.288 - 1.25 \text{ in})^2 \right)$$

$$I = 5.2 \text{ in}^4$$

The area is determined to be 5.6875 in².

The distances from the neutral axis to the top and bottom surfaces are determined to be 1.087 in and 2.288 in.

The second-area moment is determined to be 5.20 in⁴.[Hints](#)[References](#)[Hint #1](#)

ABOVE AVERAGE

Score: **100/100** Points **100** %

Problem 03.035 - Find the cross-sectional properties for a beam

Find the second moment of area, the location of the neutral axis, and the distances from the neutral axis to the top and bottom surfaces. Consider that the section is transmitting a positive bending moment about the z axis, M_z , where $M_z = 10$ kip-in if the dimensions of the section are given in ips units, or $M_z = 1.13$ kN-m if the dimensions are in SI units. Determine the resulting stresses at the top and bottom surfaces and at every abrupt change in the cross section.

References

Section Break

Chapter: 03 Load and Stress
AnalysisProblem 03.035 - Find
the cross-sectional
properties for a beamSection: 03.10 Normal Stresses
for Beams in Bending

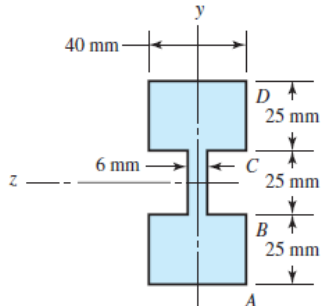
1.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **2150** ✓ mm².

The distance from the neutral axis to the top and bottom surfaces is determined to be **37.5** ✓ mm.

The second moment of area is determined to be **1.36** ✓ × 10⁶ mm⁴.

References

Numeric Response

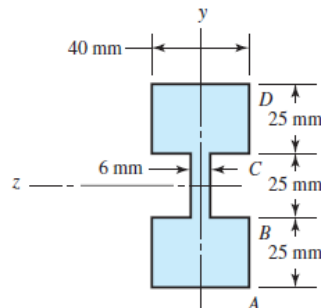
Chapter: 03 Load and Stress Analysis

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,



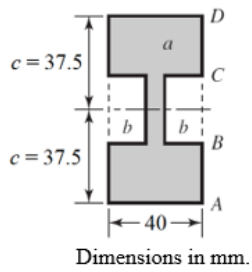
The area is determined to be **2150 ± 2%** mm².

The distance from the neutral axis to the top and bottom surfaces is determined to be **37.5 ± 2%** mm.

The second moment of area is determined to be **1.36 ± 2%** × 10⁶ mm⁴.

Explanation:

Let a be the the total area of the entire envelope and b be the area of the side notch.



The total area of the I-section is calculated as follows:

$$A = a - 2b = 40 \text{ mm} \times (2 \times 37.5 \text{ mm}) - (25 \text{ mm} \times 34 \text{ mm}) = 2150 \text{ mm}^2$$

The second-area moment is calculated as follows:

$$I = I_a - 2I_b = \left(\frac{1}{12} \times 40 \text{ mm} \times (75 \text{ mm})^3 \right) - \left(\frac{1}{12} \times 34 \text{ mm} \times (25 \text{ mm})^3 \right) = 1.36 \times 10^6 \text{ mm}^4$$

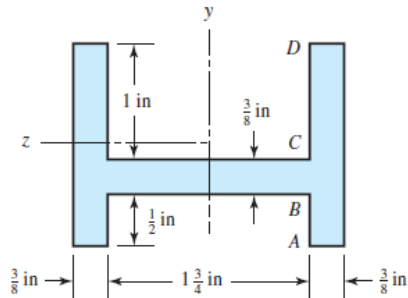
2.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure



The area is determined to be **2.06** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1** in and **0.858** in.

The second moment of area is determined to be **0.448** in⁴.

References

Numeric Response

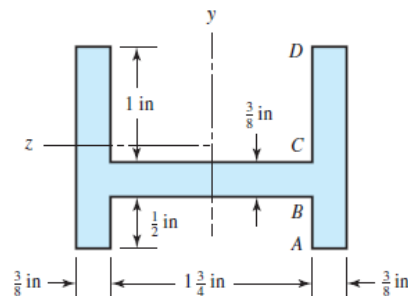
Chapter: 03 Load and Stress Analysis

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure

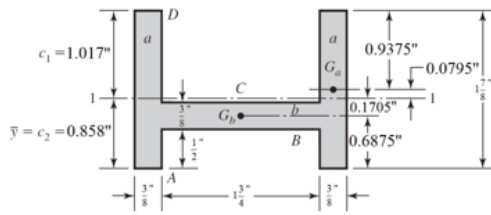


The area is determined to be **2.0625 ± 2%** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.017 ± 2%** in and **0.858 ± 2%** in.

The second moment of area is determined to be **0.448 ± 2%** in⁴.

Explanation:

$$A = (A_a + A_b) = (0.375 \text{ in} \times 1.875 \text{ in}) + (0.375 \text{ in} \times 1.75 \text{ in}) = 2.0625 \text{ in}^2$$

$$\bar{y} = \frac{2 \times (0.703125 \text{ in}^2 \times (0.9375 \text{ in})) + (0.65625 \text{ in}^2 \times (0.6875 \text{ in}))}{2.0625 \text{ in}^2} = 0.858 \text{ in}$$
$$I_a = \frac{0.375 \text{ in} \times (1.875 \text{ in})^3}{12} = 0.206 \text{ in}^4$$
$$I_b = \frac{1.75 \text{ in} \times (0.375 \text{ in})^3}{12} = 0.00769 \text{ in}^4$$
$$I = 2 \times (0.206 \text{ in}^4 + 0.703125 \text{ in}^2 \times (0.0795 \text{ in})^2) + (0.00769 \text{ in}^4 + 0.065625 \text{ in}^2 \times (0.1705 \text{ in})^2) = 0.448 \text{ in}^4$$

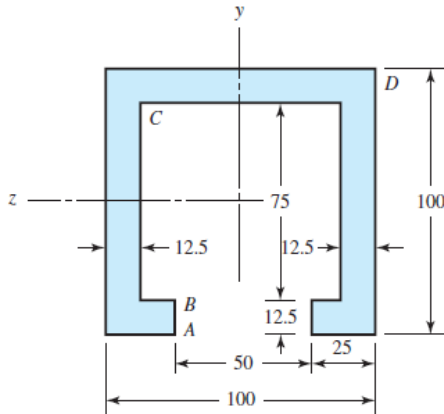
3.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **3750** ✓ mm².

The distances from the neutral axis to the top and bottom surfaces are determined to be **42.71** ✓ mm and **57.29** ✓ mm.

The second moment of area is determined to be **4.29** ✓ × 10⁶ mm⁴.

References

Numeric Response

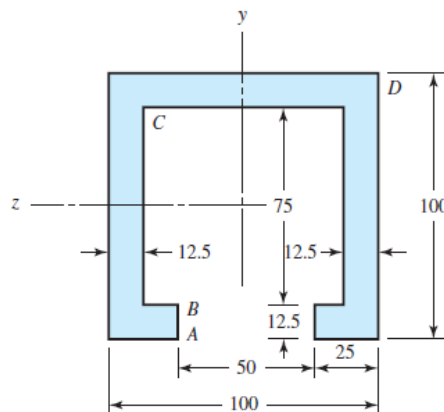
Chapter: 03 Load and Stress Analysis

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **3750 ± 2%** mm².

The distances from the neutral axis to the top and bottom surfaces are determined to be **42.71 ± 2%** mm and **57.29 ± 2%** mm.

The second moment of area is determined to be $4.29 \pm 2\% \times 10^6 \text{ mm}^4$.

Explanation:

The area under section 'a' is calculated as follows:

$$A_a = 12.5 \text{ mm} \times 50 \text{ mm} = 625 \text{ mm}^2$$

The area under section 'b' is calculated as follows:

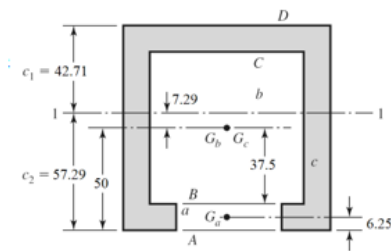
$$A_b = 75 \text{ mm} \times 75 \text{ mm} = 5625 \text{ mm}^2$$

The area under section 'c' is calculated as follows:

$$A_c = 100 \text{ mm} \times 100 \text{ mm} = 10000 \text{ mm}^2$$

The total area of the figure is calculated as follows:

$$A = 10000 \text{ mm}^2 - 5625 \text{ mm}^2 - 625 \text{ mm}^2 = 3750 \text{ mm}^2$$



The distances from the bottom to the neutral axis are calculated for the respective sections as follows:

$$\bar{y}_a = 6.25 \text{ mm}, \bar{y}_b = 50 \text{ mm}, \bar{y}_c = 50 \text{ mm}$$

$$\bar{y} = \frac{(10000 \text{ mm}^2 \times 50 \text{ mm}) - (5625 \text{ mm}^2 \times 50 \text{ mm}) - (625 \text{ mm}^2 \times 6.25 \text{ mm})}{3750 \text{ mm}^2} = 57.29 \text{ mm}$$

$$c_1 = 100 \text{ mm} - 57.29 \text{ mm} = 42.71 \text{ mm}$$

The second-area moment of section 'a' is calculated as follows:

$$I_a = \frac{50 \text{ mm} \times (12.5 \text{ mm})^3}{12} = 8138 \text{ mm}^4$$

The second-area moment of section 'b' is calculated as follows:

$$I_b = \frac{75 \text{ mm} \times (75 \text{ mm})^3}{12} = 2.637 \times 10^6 \text{ mm}^4$$

The second-area moment of section 'c' is calculated as follows:

$$I_c = \frac{100 \text{ mm} \times (100 \text{ mm})^3}{12} = 8.333 \times 10^6 \text{ mm}^4$$

The total second-area moment of the figure is calculated as follows:

$$I = (8.333 \times 10^6 \text{ mm}^4 + (10000 \text{ mm}^2 \times (7.29 \text{ mm})^2)) - (2.637 \times 10^6 \text{ mm}^4 + (5625 \text{ mm}^2 \times (7.29 \text{ mm})^2)) - (8138 \text{ mm}^4 + (625 \text{ mm}^2 \times (51.04 \text{ mm})^2))$$

$$I = 4.29 \times 10^6 \text{ mm}^4$$

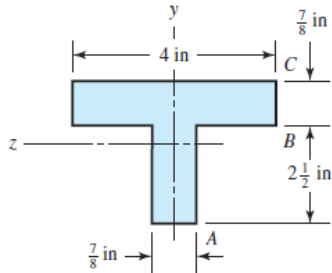
4.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **5.687** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.087** in and **2.288** in.

The second moment of area is determined to be **5.2** in⁴.

References

Numeric Response

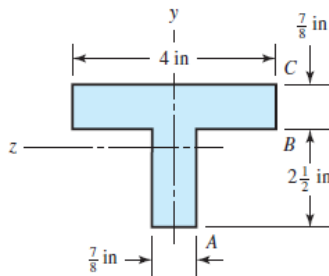
Chapter: 03 Load and Stress Analysis

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **5.6875 ± 2%** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.087 ± 2%** in and **2.288 ± 2%** in.

The second moment of area is determined to be **5.2 ± 2%** in⁴.

Explanation:

The area under section 'a' is calculated as follows:

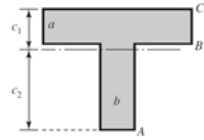
$$A_a = 4 \text{ in} \times 0.875 \text{ in} = 3.5 \text{ in}^2$$

The area under section 'b' is calculated as follows:

$$A_b = 2.5 \text{ in} \times 0.875 \text{ in} = 2.1875 \text{ in}^2$$

The total area of the figure is calculated as follows:

$$A = 3.5 \text{ in}^2 + 2.1875 \text{ in}^2 = 5.6875 \text{ in}^2$$



The distance from the bottom surface is calculated as follows:

$$\bar{y} = \frac{\left(2.9375 \text{ in}^2 \times 3.5 \text{ in} \right) + \left(1.25 \text{ in}^2 \times 2.1875 \text{ in} \right)}{5.6875 \text{ in}^2} = 2.288 \text{ in}$$

$$c_2 = 2.288 \text{ in}$$

$$c_1 = 1.087 \text{ in}$$

The total second-area moment of the figure is calculated as follows:

$$I = \left(\frac{1}{12} \times 4 \text{ in} \times (0.875 \text{ in})^3 \right) + \left(3.5 \text{ in}^2 \times (2.9375 - 2.288 \text{ in})^2 \right) + \left(\frac{1}{12} \times 0.875 \text{ in} \times (2.5 \text{ in})^3 \right) + \left(2.1875 \text{ in}^2 \times (2.288 - 1.25 \text{ in})^2 \right)$$

$$I = 5.2 \text{ in}^4$$

AVERAGE

Score: **93.75/100** Points **93.75** %

Problem 03.035 - Find the cross-sectional properties for a beam

Find the second moment of area, the location of the neutral axis, and the distances from the neutral axis to the top and bottom surfaces. Consider that the section is transmitting a positive bending moment about the z axis, M_z , where $M_z = 10$ kip-in if the dimensions of the section are given in ips units, or $M_z = 1.13$ kN-m if the dimensions are in SI units. Determine the resulting stresses at the top and bottom surfaces and at every abrupt change in the cross section.

References

Section Break

Chapter: 03 Load and Stress
Analysis

Problem 03.035 - Find
the cross-sectional
properties for a beam

Section: 03.10 Normal Stresses
for Beams in Bending

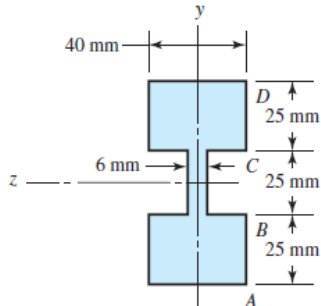
1.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **2150** ✓ mm².

The distance from the neutral axis to the top and bottom surfaces is determined to be **37.5** ✓ mm.

The second moment of area is determined to be **1.36** ✓ × 10⁶ mm⁴.

References

Numeric Response

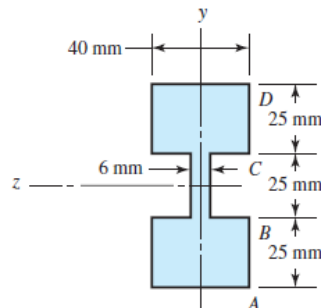
Chapter: 03 Load and Stress Analysis

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,



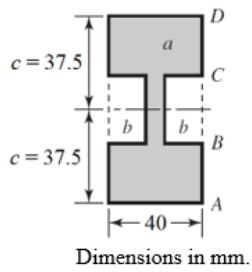
The area is determined to be **2150 ± 2%** mm².

The distance from the neutral axis to the top and bottom surfaces is determined to be **37.5 ± 2%** mm.

The second moment of area is determined to be **1.36 ± 2%** × 10⁶ mm⁴.

Explanation:

Let a be the the total area of the entire envelope and b be the area of the side notch.



The total area of the I-section is calculated as follows:

$$A = a - 2b = 40 \text{ mm} \times (2 \times 37.5 \text{ mm}) - (25 \text{ mm} \times 34 \text{ mm}) = 2150 \text{ mm}^2$$

The second-area moment is calculated as follows:

$$I = I_a - 2I_b = \left(\frac{1}{12} \times 40 \text{ mm} \times (75 \text{ mm})^3 \right) - \left(\frac{1}{12} \times 34 \text{ mm} \times (25 \text{ mm})^3 \right) = 1.36 \times 10^6 \text{ mm}^4$$

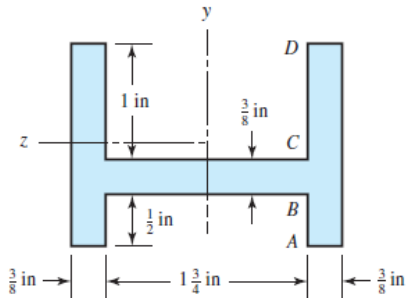
2.

Award: 18.75 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure



The area is determined to be **2.0625** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.017** in and **1** in.

The second moment of area is determined to be **0.448** in⁴.

References

Numeric Response

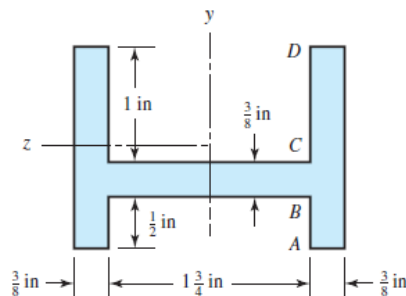
Chapter: 03 Load and Stress Analysis

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure

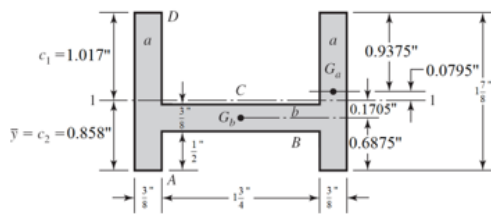


The area is determined to be **2.0625 ± 2%** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.017 ± 2%** in and **0.858 ± 2%** in.

The second moment of area is determined to be **0.448 ± 2%** in⁴.

Explanation:

$$A = (A_a + A_b) = (0.375 \text{ in} \times 1.875 \text{ in}) + (0.375 \text{ in} \times 1.75 \text{ in}) = 2.0625 \text{ in}^2$$

$$\bar{y} = \frac{2 \times (0.703125 \text{ in}^2 \times (0.9375 \text{ in})) + (0.65625 \text{ in}^2 \times (0.6875 \text{ in}))}{2.0625 \text{ in}^2} = 0.858 \text{ in}$$
$$I_a = \frac{0.375 \text{ in} \times (1.875 \text{ in})^3}{12} = 0.206 \text{ in}^4$$
$$I_b = \frac{1.75 \text{ in} \times (0.375 \text{ in})^3}{12} = 0.00769 \text{ in}^4$$
$$I = 2 \times \left(0.206 \text{ in}^4 + 0.703125 \text{ in}^2 \times (0.0795 \text{ in})^2 \right) + \left(0.00769 \text{ in}^4 + 0.065625 \text{ in}^2 \times (0.1705 \text{ in})^2 \right) = 0.448 \text{ in}^4$$

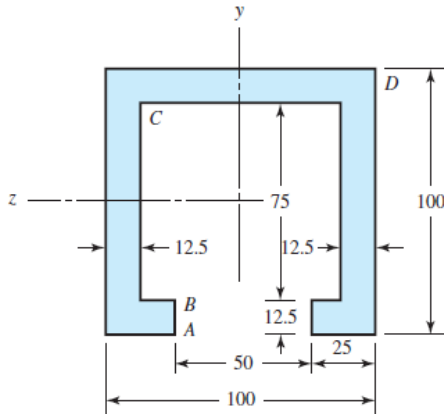
3.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **3750** ✓ mm².

The distances from the neutral axis to the top and bottom surfaces are determined to be **42.71** ✓ mm and **57.29** ✓ mm.

The second moment of area is determined to be **4.29** ✓ × 10⁶ mm⁴.

References

Numeric Response

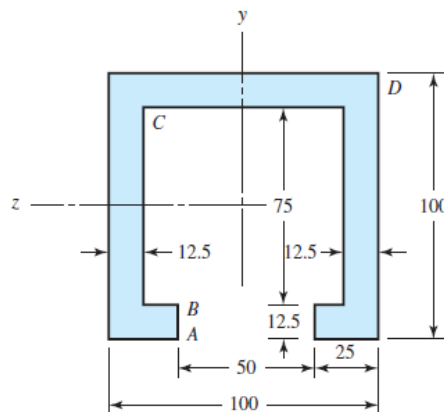
Chapter: 03 Load and Stress Analysis

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **3750 ± 2%** mm².

The distances from the neutral axis to the top and bottom surfaces are determined to be **42.71 ± 2%** mm and **57.29 ± 2%** mm.

The second moment of area is determined to be $4.29 \pm 2\% \times 10^6 \text{ mm}^4$.

Explanation:

The area under section 'a' is calculated as follows:

$$A_a = 12.5 \text{ mm} \times 50 \text{ mm} = 625 \text{ mm}^2$$

The area under section 'b' is calculated as follows:

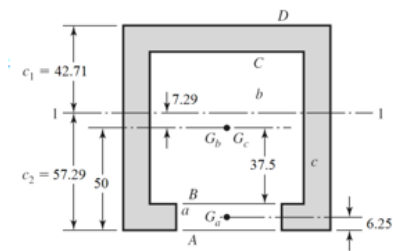
$$A_b = 75 \text{ mm} \times 75 \text{ mm} = 5625 \text{ mm}^2$$

The area under section 'c' is calculated as follows:

$$A_c = 100 \text{ mm} \times 100 \text{ mm} = 10000 \text{ mm}^2$$

The total area of the figure is calculated as follows:

$$A = 10000 \text{ mm}^2 - 5625 \text{ mm}^2 - 625 \text{ mm}^2 = 3750 \text{ mm}^2$$



The distances from the bottom to the neutral axis are calculated for the respective sections as follows:

$$\bar{y}_a = 6.25 \text{ mm}, \bar{y}_b = 50 \text{ mm}, \bar{y}_c = 50 \text{ mm}$$

$$\bar{y} = \frac{(10000 \text{ mm}^2 \times 50 \text{ mm}) - (5625 \text{ mm}^2 \times 50 \text{ mm}) - (625 \text{ mm}^2 \times 6.25 \text{ mm})}{3750 \text{ mm}^2} = 57.29 \text{ mm}$$

$$c_1 = 100 \text{ mm} - 57.29 \text{ mm} = 42.71 \text{ mm}$$

The second-area moment of section 'a' is calculated as follows:

$$I_a = \frac{50 \text{ mm} \times (12.5 \text{ mm})^3}{12} = 8138 \text{ mm}^4$$

The second-area moment of section 'b' is calculated as follows:

$$I_b = \frac{75 \text{ mm} \times (75 \text{ mm})^3}{12} = 2.637 \times 10^6 \text{ mm}^4$$

The second-area moment of section 'c' is calculated as follows:

$$I_c = \frac{100 \text{ mm} \times (100 \text{ mm})^3}{12} = 8.333 \times 10^6 \text{ mm}^4$$

The total second-area moment of the figure is calculated as follows:

$$I = (8.333 \times 10^6 \text{ mm}^4 + (10000 \text{ mm}^2 \times (7.29 \text{ mm})^2)) - (2.637 \times 10^6 \text{ mm}^4 + (5625 \text{ mm}^2 \times (7.29 \text{ mm})^2)) - (8138 \text{ mm}^4 + (625 \text{ mm}^2 \times (51.04 \text{ mm})^2))$$

$$I = 4.29 \times 10^6 \text{ mm}^4$$

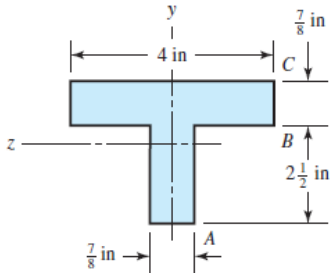
4.

Award: 25 out of 25.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **5.6875** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.087** in and **2.288** in.

The second moment of area is determined to be **5.2** in⁴.

References

Numeric Response

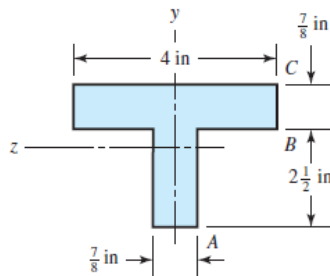
Chapter: 03 Load and Stress Analysis

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,



The area is determined to be **5.6875 ± 2%** in².

The distances from the neutral axis to the top and bottom surfaces are determined to be **1.087 ± 2%** in and **2.288 ± 2%** in.

The second moment of area is determined to be **5.2 ± 2%** in⁴.

Explanation:

The area under section 'a' is calculated as follows:

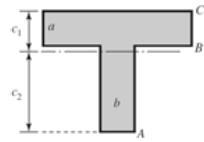
$$A_a = 4 \text{ in} \times 0.875 \text{ in} = 3.5 \text{ in}^2$$

The area under section 'b' is calculated as follows:

$$A_b = 2.5 \text{ in} \times 0.875 \text{ in} = 2.1875 \text{ in}^2$$

The total area of the figure is calculated as follows:

$$A = 3.5 \text{ in}^2 + 2.1875 \text{ in}^2 = 5.6875 \text{ in}^2$$



The distance from the bottom surface is calculated as follows:

$$\bar{y} = \frac{\left(2.9375 \text{ in}^2 \times 3.5 \text{ in} \right) + \left(1.25 \text{ in}^2 \times 2.1875 \text{ in} \right)}{5.6875 \text{ in}^2} = 2.288 \text{ in}$$

$$c_2 = 2.288 \text{ in}$$

$$c_1 = 1.087 \text{ in}$$

The total second-area moment of the figure is calculated as follows:

$$I = \left(\frac{1}{12} \times 4 \text{ in} \times (0.875 \text{ in})^3 \right) + \left(3.5 \text{ in}^2 \times (2.9375 - 2.288 \text{ in})^2 \right) + \left(\frac{1}{12} \times 0.875 \text{ in} \times (2.5 \text{ in})^3 \right) + \left(2.1875 \text{ in}^2 \times (2.288 - 1.25 \text{ in})^2 \right)$$

$$I = 5.2 \text{ in}^4$$

BELOW AVERAGE

Score: **72.92/100** Points **72.92** %

Problem 03.035 - Find the cross-sectional properties for a beam

Find the second moment of area, the location of the neutral axis, and the distances from the neutral axis to the top and bottom surfaces. Consider that the section is transmitting a positive bending moment about the z axis, M_z , where $M_z = 10$ kip-in if the dimensions of the section are given in ips units, or $M_z = 1.13$ kN-m if the dimensions are in SI units. Determine the resulting stresses at the top and bottom surfaces and at every abrupt change in the cross section.

References

Section Break

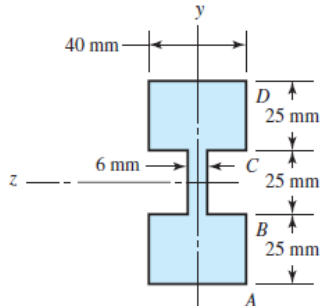
Chapter: 03 Load and Stress
AnalysisProblem 03.035 - Find
the cross-sectional
properties for a beamSection: 03.10 Normal Stresses
for Beams in Bending

1.

Award: 16.67 out of 25.00 points

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be **2150** ✓ mm².The distance from the neutral axis to the top and bottom surfaces is determined to be **37.5** ✓ mm.The second moment of area is determined to be **1.39** ✗ × 10⁶ mm⁴.

References

Numeric Response

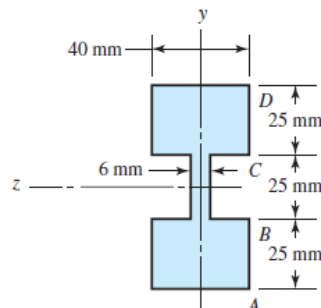
Chapter: 03 Load and Stress Analysis

Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

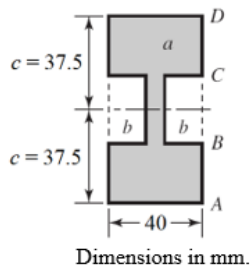
Problem 03.035.a - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be **2150 ± 2%** mm².The distance from the neutral axis to the top and bottom surfaces is determined to be **37.5 ± 2%** mm.The second moment of area is determined to be **1.36 ± 2%** × 10⁶ mm⁴.**Explanation:**

Processing math: 100%

Let a be the the total area of the entire envelope and b be the area of the side notch.



The total area of the I-section is calculated as follows:

$$A = a - 2b = 40 \text{ mm} \times (2 \times 37.5 \text{ mm}) - (25 \text{ mm} \times 34 \text{ mm}) = 2150 \text{ mm}^2$$

The second-area moment is calculated as follows:

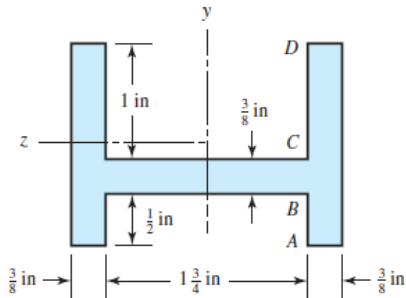
$$I = I_a - 2I_b = \left(\frac{1}{12} \times 40 \text{ mm} \times (75 \text{ mm})^3 \right) - \left(\frac{1}{12} \times 34 \text{ mm} \times (25 \text{ mm})^3 \right) = 1.36 \times 10^6 \text{ mm}^4$$

2.

Award: 25 out of 25.00 points

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure

The area is determined to be **2.0625** in².The distances from the neutral axis to the top and bottom surfaces are determined to be **1.017** in and **0.858** in.The second moment of area is determined to be **0.448** in⁴.

References

Numeric Response

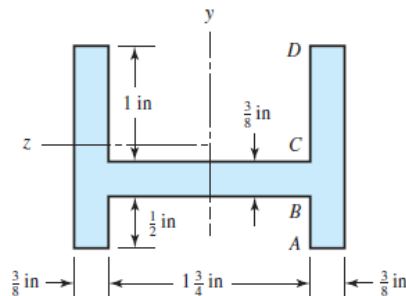
Chapter: 03 Load and Stress Analysis

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.b - Find the second moment of area and the location of the neutral axis.

From the figure

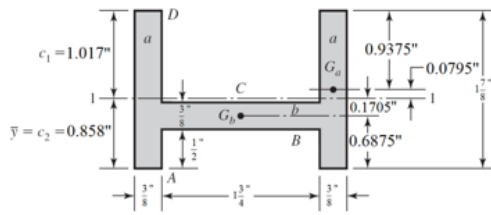
The area is determined to be **2.0625 ± 2%** in².The distances from the neutral axis to the top and bottom surfaces are determined to be **1.017 ± 2%** in and **0.858 ± 2%** in.The second moment of area is determined to be **0.448 ± 2%** in⁴.

Explanation:

Processing math: 100%

The total area of the figure is calculated as follows:

$$A = (A_a + A_b) = (0.375 \text{ in} \times 1.875 \text{ in}) + (0.375 \text{ in} \times 1.75 \text{ in}) = 2.0625 \text{ in}^2$$



The neutral axis from the bottom of the figure is calculated as follows:

$$\bar{y} = \frac{2 \times \left(0.703125 \text{ in}^2 \times (0.9375 \text{ in}) \right) + \left(0.65625 \text{ in}^2 \times (0.6875 \text{ in}) \right)}{2.0625 \text{ in}^2} = 0.858 \text{ in}$$

The second-area moment of section 'a' is calculated as follows:

$$I_a = \frac{0.375 \text{ in} \times (1.875 \text{ in})^3}{12} = 0.206 \text{ in}^4$$

The second-area moment of section 'b' is calculated as follows:

$$I_b = \frac{1.75 \text{ in} \times (0.375 \text{ in})^3}{12} = 0.00769 \text{ in}^4$$

The total second-area moment of the figure is calculated as follows:

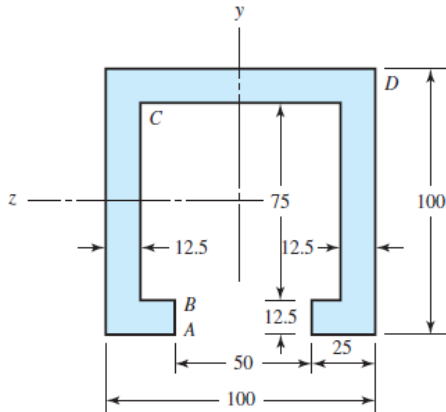
$$I = 2 \times \left(0.206 \text{ in}^4 + 0.703125 \text{ in}^2 \times (0.0795 \text{ in})^2 \right) + \left(0.00769 \text{ in}^4 + 0.65625 \text{ in}^2 \times (0.1705 \text{ in})^2 \right) = 0.448 \text{ in}^4$$

3.

Award: 25 out of 25.00 points

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be **3750** ✓ mm².The distances from the neutral axis to the top and bottom surfaces are determined to be **42.708** ✓ mm and **57.292** ✓ mm.The second moment of area is determined to be **4.293** ✓ × 10⁶ mm⁴.

References

Numeric Response

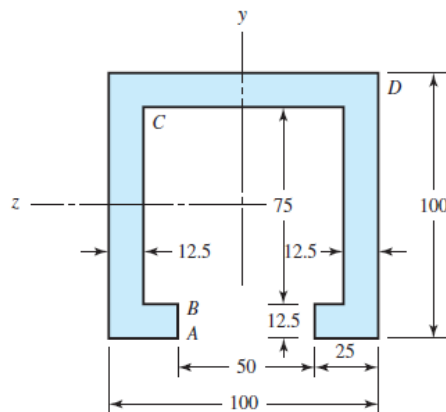
Chapter: 03 Load and Stress Analysis

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.c - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be **3750 ± 2%** mm².

Processing math: 100%

distances from the neutral axis to the top and bottom surfaces are determined to be **42.71 ± 2%** mm and **57.29 ± 2%** mm.

The second moment of area is determined to be $4.29 \pm 2\% \times 10^6 \text{ mm}^4$.

Explanation:

The area under section 'a' is calculated as follows:

$$A_a = 12.5 \text{ mm} \times 50 \text{ mm} = 625 \text{ mm}^2$$

The area under section 'b' is calculated as follows:

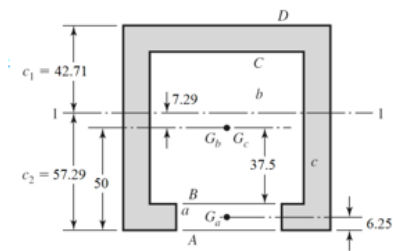
$$A_b = 75 \text{ mm} \times 75 \text{ mm} = 5625 \text{ mm}^2$$

The area under section 'c' is calculated as follows:

$$A_c = 100 \text{ mm} \times 100 \text{ mm} = 10000 \text{ mm}^2$$

The total area of the figure is calculated as follows:

$$A = 10000 \text{ mm}^2 - 5625 \text{ mm}^2 - 625 \text{ mm}^2 = 3750 \text{ mm}^2$$



The distances from the bottom to the neutral axis are calculated for the respective sections as follows:

$$\bar{y}_a = 6.25 \text{ mm}, \bar{y}_b = 50 \text{ mm}, \bar{y}_c = 50 \text{ mm}$$

$$\bar{y} = \frac{(10000 \text{ mm}^2 \times 50 \text{ mm}) - (5625 \text{ mm}^2 \times 50 \text{ mm}) - (625 \text{ mm}^2 \times 6.25 \text{ mm})}{3750 \text{ mm}^2} = 57.29 \text{ mm}$$

$$c_1 = 100 \text{ mm} - 57.29 \text{ mm} = 42.71 \text{ mm}$$

The second-area moment of section 'a' is calculated as follows:

$$I_a = \frac{50 \text{ mm} \times (12.5 \text{ mm})^3}{12} = 8138 \text{ mm}^4$$

The second-area moment of section 'b' is calculated as follows:

$$I_b = \frac{75 \text{ mm} \times (75 \text{ mm})^3}{12} = 2.637 \times 10^6 \text{ mm}^4$$

The second-area moment of section 'c' is calculated as follows:

$$I_c = \frac{100 \text{ mm} \times (100 \text{ mm})^3}{12} = 8.333 \times 10^6 \text{ mm}^4$$

The total second-area moment of the figure is calculated as follows:

$$I = \left(8.333 \times 10^6 \text{ mm}^4 + \left(10000 \text{ mm}^2 \times (7.29 \text{ mm})^2 \right) \right) - \left(2.637 \times 10^6 \text{ mm}^4 + \left(5625 \text{ mm}^2 \times (7.29 \text{ mm})^2 \right) \right) - \left(8138 \text{ mm}^4 + \left(625 \text{ mm}^2 \times (51.04 \text{ mm})^2 \right) \right)$$

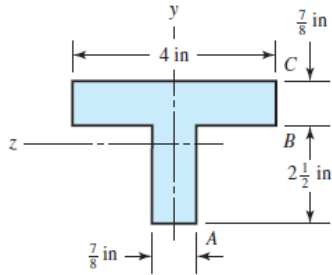
$$I = 4.29 \times 10^6 \text{ mm}^4$$

4.

Award: 6.25 out of 25.00 points

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be **5.6875** ✓ in².The distances from the neutral axis to the top and bottom surfaces are determined to be **0.75** ✗ in and **2.625** ✗ in.The second moment of area is determined to be **0.23** ✗ in⁴.

References

Numeric Response

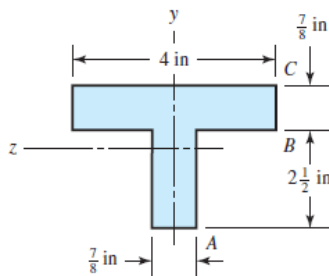
Chapter: 03 Load and Stress Analysis

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.035.d - Find the second moment of area and the location of the neutral axis.

From the figure,

The area is determined to be **5.6875 ± 2%** in².The distances from the neutral axis to the top and bottom surfaces are determined to be **1.087 ± 2%** in and **2.288 ± 2%** in.The second moment of area is determined to be **5.2 ± 2%** in⁴.**Explanation:**

The area under section 'a' is calculated as follows:

Processing math: 100%

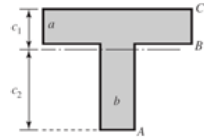
$$A_a = 4 \text{ in} \times 0.875 \text{ in} = 3.5 \text{ in}^2$$

The area under section 'b' is calculated as follows:

$$A_b = 2.5 \text{ in} \times 0.875 \text{ in} = 2.1875 \text{ in}^2$$

The total area of the figure is calculated as follows:

$$A = 3.5 \text{ in}^2 + 2.1875 \text{ in}^2 = 5.6875 \text{ in}^2$$



The distance from the bottom surface is calculated as follows:

$$\bar{y} = \frac{\left(2.9375 \text{ in}^2 \times 3.5 \text{ in} \right) + \left(1.25 \text{ in}^2 \times 2.1875 \text{ in} \right)}{5.6875 \text{ in}^2} = 2.288 \text{ in}$$

$$c_2 = 2.288 \text{ in}$$

$$c_1 = 1.087 \text{ in}$$

The total second-area moment of the figure is calculated as follows:

$$I = \left(\frac{1}{12} \times 4 \text{ in} \times (0.875 \text{ in})^3 \right) + \left(3.5 \text{ in}^2 \times (2.9375 - 2.288 \text{ in})^2 \right) + \left(\frac{1}{12} \times 0.875 \text{ in} \times (2.5 \text{ in})^3 \right) + \left(2.1875 \text{ in}^2 \times (2.288 - 1.25 \text{ in})^2 \right)$$

$$I = 5.2 \text{ in}^4$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 6

TOPICS COVERED IN HOMEWORK 6

Sections 3-10, 3-11, 3-12: Problems 3-36, 3-38, 3-45

**MACHINE DESIGN I
FALL 2021**

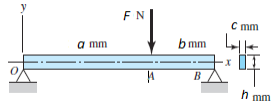
**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

1.

Award: 30.00 points

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 280$ mm, $b = 160$ mm, $c = 21$ mm, $h = 43$ mm, and $F = 4100$ N.



The moment of inertia is $1.391 \pm 2\% \times 10^5 \text{ mm}^4$.

The maximum tensile bending stress due to M is $64.5 \pm 2\% \text{ MPa}$.

The maximum shear stress due to V is $4.3 \pm 2\% \text{ MPa}$.

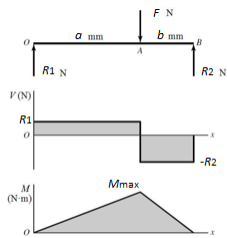
Explanation:

Given: $a = 280$ mm, $b = 160$ mm, $c = 21$ mm, $h = 43$ mm, and $F = 4100$ N

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (21 \text{ mm})(43 \text{ mm})^3 = 1.391 (10^5) \text{ mm}^4$$

$$A = (21 \text{ mm})(43 \text{ mm}) = 903 \text{ mm}^2$$



$M_{\max}$ is at A. At the bottom of the section,

$$R_1 = 1490.9091 \text{ N}$$

$$R_2 = 2609.0909 \text{ N}$$

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(2609.0909 \text{ N} \times 160 \text{ mm}) \left(\frac{43 \text{ mm}}{2} \right)}{1.391(10^5) \text{ mm}^4} = 64.5 \text{ MPa}$$

Due to V , $\tau_{\max}$ is between A and B at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{2609.0909 \text{ N}}{903 \text{ mm}^2} \right) = 4.3 \text{ MPa}$$

The moment of inertia is $1.391 \times 10^5 \text{ mm}^4$.

The maximum tensile bending stress due to M is 64.5 MPa.

The maximum shear stress due to V is 4.3 MPa.

References

Numeric Response

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

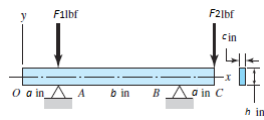
Difficulty: Medium

2.

Award: 30.00 points

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 5$ in, $b = 14.5$ in, $c = 0.825$ in, $h = 2$ in, $F_1 = 3300$ lbf, and $F_2 = 1100$ lbf.



The moment of inertia is $0.55 \pm 2\%$ in⁴.

The maximum tensile bending stress due to M is $10000 \pm 2\%$ psi.

The maximum shear stress due to V is $1000 \pm 2\%$ psi.

Explanation:

Given: $a = 5$ in, $b = 14.5$ in, $c = 0.825$ in, $h = 2$ in, $F_1 = 3300$ lbf, and $F_2 = 1100$ lbf.

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (0.825 \text{ in})(2 \text{ in})^3 = 0.55 \text{ in}^4$$

$$A = (0.825 \text{ in})(2 \text{ in}) = 1.65 \text{ in}^2$$

We know that

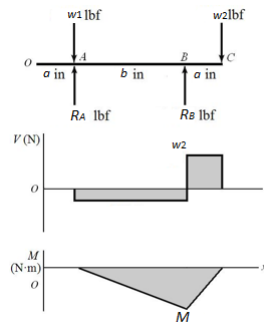
$$\Sigma M_A = 0$$

Therefore,

$$14.5 R_B - (1100 \text{ lbf})(19.5 \text{ in}) = 0$$

$$R_B = 1479 \text{ lbf}$$

$$R_A = 3300 \text{ lbf} - 1479 \text{ lbf} + 1100 \text{ lbf} = 2921 \text{ lbf}$$



$M_{\max}$ is at B . At the bottom of the section,

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(5 \text{ in} \times 1100 \text{ lbf}) \times \left(\frac{2 \text{ in}}{2}\right)}{0.55 \text{ in}^4} = 10000 \text{ psi}$$

Due to V , $\tau_{\max}$ is between B and C at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{1100 \text{ lbf}}{1.65 \text{ in}^2} \right) = 1000 \text{ psi}$$

The moment of inertia is 0.55 in^4 .

The maximum tensile bending stress due to M is 10000 psi .

The maximum shear stress due to V is 1000 psi .

References

Numeric Response

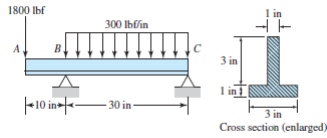
Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

Difficulty: Medium

Problem 03.045 - Determine the maximum tensile, compressive, and shear stresses in a beam - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Consider the beam shown in the given figure.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Problem 03.045 - Determine the maximum tensile, compressive, and shear stresses in a beam - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Hard

3.

Award: 20.00 points

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300 \text{ lbf/in}$, $F = 1800 \text{ lbf}$, $c_1 = 10 \text{ in}$, and $c_2 = 30 \text{ in}$.

The maximum tensile bending stress is $5294 \pm 2\%$ psi.

The maximum compressive bending stress is $-7456 \pm 2\%$ psi.

Explanation:

$$R_1 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} + \frac{40 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 6900 \text{ lbf}$$

$$R_2 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} - \frac{10 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 3900 \text{ lbf}$$

$$a = \frac{3900 \text{ lbf}}{300 \text{ lbf/in}} = 13 \text{ in}$$

$$M_B = -1800 \text{ lbf} \times (10 \text{ in}) = -18000 \text{ lbf}\cdot\text{in}$$

$$M_x = 27 \text{ in} = \frac{1}{2} (3900 \text{ lbf}) \times (13 \text{ in}) = 25350 \text{ lbf}\cdot\text{in}$$

$$\bar{y} = \frac{0.5 \text{ in} \times (3 \text{ in}) + 2.5 \text{ in} \times (3 \text{ in})}{6 \text{ in}} = 1.5 \text{ in}$$

$$I_1 = \frac{1}{12} (3 \text{ in}) \times (1 \text{ in})^3 = 0.25 \text{ in}^4$$

$$I_2 = \frac{1}{12} (1 \text{ in}) \times (3 \text{ in})^3 = 2.25 \text{ in}^4$$

Applying parallel-axis theorem,

$$I_z = (0.25 \text{ in}^4 + 3(1.5 \text{ in} - 0.5 \text{ in})^2) + (2.25 \text{ in}^4 + 3(1.5 \text{ in} - 0.5 \text{ in})^2) = 8.5 \text{ in}^4$$

The maximum tensile and compressive bending stresses are calculated as follows:

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = -\frac{-18000 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5} = -3176 \text{ psi}$$

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = -\frac{-18000 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5} = 5294 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = -1.5 \text{ in, } \sigma_x = -\frac{25350 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5 \text{ in}^4} = 4474 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = 2.5 \text{ in, } \sigma_x = -\frac{25350 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5 \text{ in}^4} = -7456 \text{ psi}$$

Maximum tension = 5294 psi

Maximum compression = -7456 psi

The maximum tensile bending stress is 5294 psi.

The maximum compressive bending stress is -7456 psi.

References

Numeric Response

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Difficulty: Hard

4.

Award: 20.00 points

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is psi.**Explanation:**The maximum shear stress due to V is at B , at the neutral axis.

$$V_{\max} = 5100 \text{ lbf}$$

$$Q = \bar{y}' \times A' = 1.25 \text{ in} \times ((2.5) \times (1) \text{ in}^2) = 3.125 \text{ in}^3$$

$$(\tau_{\max})_V = \frac{VQ}{Ib} = \frac{5100 \text{ lbf} \times (3.125 \text{ in}^3)}{8.5 \text{ in}^4 \times (1 \text{ in})} = 1875 \text{ psi}$$

The maximum shear stress due to V is 1875 psi.**References****Numeric Response**

Problem 03.045.b - Determine the maximum shear stress.

Difficulty: Hard

ABOVE AVERAGE

Score: **100/100** Points **100** %

1.

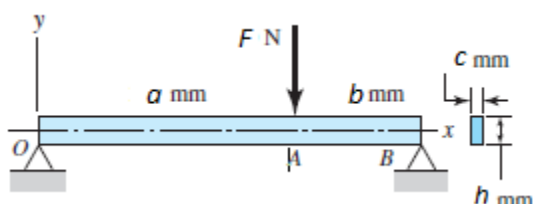
Award: 30 out of 30.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 290$ mm, $b = 155$ mm, $c = 20$ mm, $h = 39$ mm, and $F = 4050$ N.



The moment of inertia is **0.989** ✓ $\times 10^5$ mm⁴.

The maximum tensile bending stress due to M is **80.7** ✓ MPa.

The maximum shear stress due to V is **5.1** ✓ MPa.

References

Numeric Response

Chapter: 03 Load and Stress Analysis

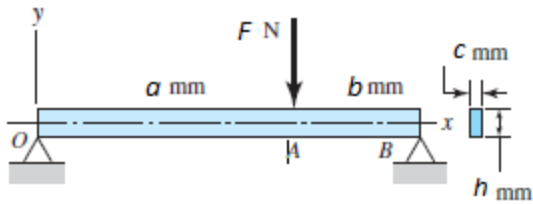
Problem 03.036
- Find the locations/magnitudes of the maximum tensile bending stress.

Section: 03.10
Normal Stresses for Beams in Bending

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum

tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 290$ mm, $b = 155$ mm, $c = 20$ mm, $h = 39$ mm, and $F = 4050$ N.



The moment of inertia is $0.989 \pm 2\% \times 10^5 \text{ mm}^4$.

The maximum tensile bending stress due to M is $80.7 \pm 2\% \text{ MPa}$.

The maximum shear stress due to V is $5.1 \pm 2\% \text{ MPa}$.

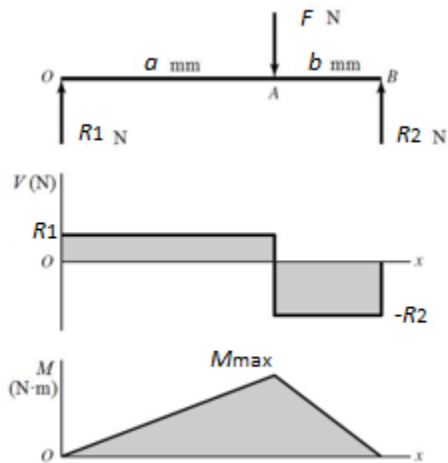
Explanation:

Given: $a = 290$ mm, $b = 155$ mm, $c = 20$ mm, $h = 39$ mm, and $F = 4050$ N

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (20 \text{ mm})(39 \text{ mm})^3 = 0.989 (10^5) \text{ mm}^4$$

$$A = (20 \text{ mm})(39 \text{ mm}) = 780 \text{ mm}^2$$



$M_{\max}$ is at A. At the bottom of the section,

$$R_1 = 1410.6742 \text{ N}$$

$$R_2 = 2639.3258 \text{ N}$$

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(2639.3258 \text{ N} \times 155 \text{ mm}) \left(\frac{39 \text{ mm}}{2} \right)}{0.989 (10^5) \text{ mm}^4} = 80.7 \text{ MPa}$$

Due to V , $\tau_{\max}$ is between A and B at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{2639.3258 \text{ N}}{780 \text{ mm}^2} \right) = 5.1 \text{ MPa}$$

2.

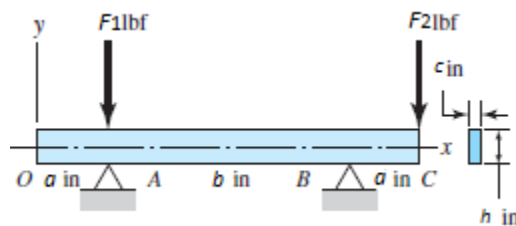
Award: 30 out of 30.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 5$ in, $b = 16.5$ in, $c = 0.725$ in, $h = 2$ in, $F_1 = 2800$ lbf, and $F_2 = 1100$ lbf.



The moment of inertia is **0.4853** ✓ in⁴.

The maximum tensile bending stress due to M is **11379** ✓ psi.

The maximum shear stress due to V is **1138** ✓ psi.

References

Numeric Response

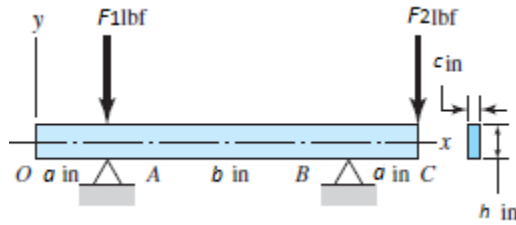
Chapter: 03 Load and Stress Analysis

Problem 03.038 Section: 03.10
- Find the locations/magnitudes of the maximum tensile bending stress. Normal Stresses for Beams in Bending

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum

tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 5$ in, $b = 16.5$ in, $c = 0.725$ in, $h = 2$ in, $F_1 = 2800$ lbf, and $F_2 = 1100$ lbf.



The moment of inertia is $0.4833 \pm 2\%$ in⁴.

The maximum tensile bending stress due to M is $11379 \pm 2\%$ psi.

The maximum shear stress due to V is $1138 \pm 2\%$ psi.

Explanation:

Given: $a = 5$ in, $b = 16.5$ in, $c = 0.725$ in, $h = 2$ in, $F_1 = 2800$ lbf, and $F_2 = 1100$ lbf.

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (0.725 \text{ in})(2 \text{ in})^3 = 0.4833 \text{ in}^4$$

$$A = (0.725 \text{ in})(2 \text{ in}) = 1.45 \text{ in}^2$$

We know that

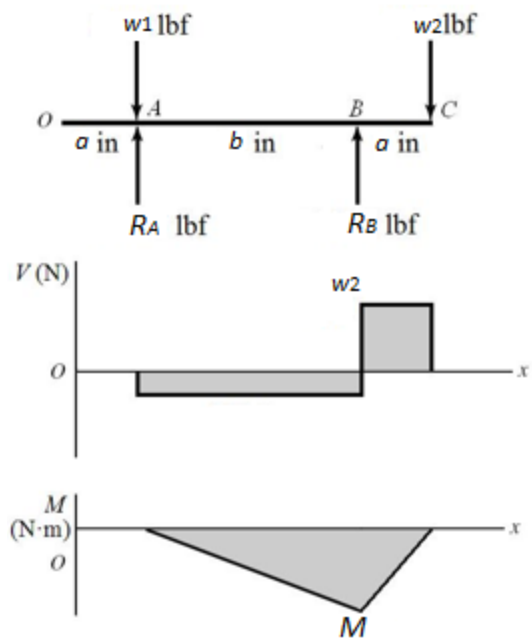
$$\Sigma M_A = 0$$

Therefore,

$$16.5 R_B - (1100 \text{ lbf})(21.5 \text{ in}) = 0$$

$$R_B = 1433 \text{ lbf}$$

$$R_A = 2800 \text{ lbf} - 1433 \text{ lbf} + 1100 \text{ lbf} = 2467 \text{ lbf}$$



$M_{\max}$ is at B . At the bottom of the section,

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(5 \text{ in} \times 1100 \text{ lbf}) \times \left(\frac{2 \text{ in}}{2}\right)}{0.4833 \text{ in}^4} = 11379 \text{ psi}$$

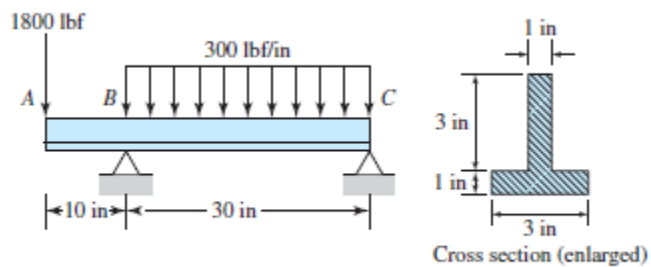
Due to V , $\tau_{\max}$ is between B and C at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{1100 \text{ lbf}}{1.45 \text{ in}^2} \right) = 1138 \text{ psi}$$

Problem 03.045 - Determine the maximum tensile, compressive, and shear stresses in a beam - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Consider the beam shown in the given figure.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break Chapter: 03 Load and Stress Analysis

Problem 03.045 Section: 03.10
- Determine the maximum Normal Stresses for
maximum tensile, compressive, and shear stresses in a
beam - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS
Beams in Bending

3.

Award: 20 out of 20.00 points



You received credit for this question in a previous attempt

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300 \text{ lbf/in}$, $F = 1800 \text{ lbf}$, $c_1 = 10 \text{ in}$, and $c_2 = 30 \text{ in}$.

The maximum tensile bending stress is **5294** psi.

The maximum compressive bending stress is **-7456** psi.

References

Numeric Response

Chapter: 03 Load and Stress Analysis

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300 \text{ lbf/in}$, $F = 1800 \text{ lbf}$, $c_1 = 10 \text{ in}$, and $c_2 = 30 \text{ in}$.

The maximum tensile bending stress is **5294 ± 2%** psi.

The maximum compressive bending stress is **-7456 ± 2%** psi.

Explanation:

$$R_1 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} + \frac{40 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 6900 \text{ lbf}$$

$$R_2 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} - \frac{10 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 3900 \text{ lbf}$$

$$a = \frac{3900 \text{ lbf}}{300 \text{ lbf/in}} = 13 \text{ in}$$

$$M_B = -1800 \text{ lbf} \times (10 \text{ in}) = -18000 \text{ lbf}\cdot\text{in}$$

$$M_{x=27 \text{ in}} = \frac{1}{2} (3900 \text{ lbf}) \times (13 \text{ in}) = 25350 \text{ lbf}\cdot\text{in}$$

$$\bar{y} = \frac{0.5 \text{ in} \times (3 \text{ in}) + 2.5 \text{ in} \times (3 \text{ in})}{6 \text{ in}} = 1.5 \text{ in}$$

$$I_1 = \frac{1}{12} (3 \text{ in}) \times (1 \text{ in})^3 = 0.25 \text{ in}^4$$

$$I_2 = \frac{1}{12} (1 \text{ in}) \times (3 \text{ in})^3 = 2.25 \text{ in}^4$$

Applying parallel-axis theorem,

$$I_z = \left(0.25 \text{ in}^2 + 3(1.5 \text{ in} - 0.5 \text{ in})^2 \right) + \left(2.25 \text{ in}^2 + 3(1.5 \text{ in} - 0.5 \text{ in})^2 \right) = 8.5 \text{ in}^4$$

The maximum tensile and compressive bending stresses are calculated as follows:

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{-18000 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5} = -3176 \text{ psi}$$

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{-18000 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5} = 5294 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = -1.5 \text{ in, } \sigma_x = - \frac{25350 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5 \text{ in}^4} = 4474 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{25350 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5 \text{ in}^4} = -7456 \text{ psi}$$

Maximum tension = 5294 psi

Maximum compression = -7456 psi

4.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is **1875** ✓ psi.**References****Numeric Response**

Chapter: 03 Load and Stress Analysis

Problem 03.045.b - Determine the maximum shear stress.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is **1875 ± 2%** psi.**Explanation:**The maximum shear stress due to V is at B , at the neutral axis.

$$V_{\max} = 5100 \text{ lbf}$$

$$Q = \bar{y}' \times A' = 1.25 \text{ in} \times ((2.5) \times (1) \text{ in}^2) = 3.125 \text{ in}^3$$

$$(\tau_{\max})_V = \frac{VQ}{Ib} = \frac{5100 \text{ lbf} \times (3.125 \text{ in}^3)}{8.5 \text{ in}^4 \times (1 \text{ in})} = 1875 \text{ psi}$$

AVERAGE

Score: **90/100** Points **90** %

1.

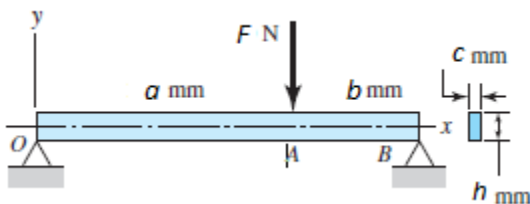
Award: 30 out of 30.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 320$ mm, $b = 160$ mm, $c = 20$ mm, $h = 39$ mm, and $F = 4100$ N.



The moment of inertia is **0.989** ✓ $\times 10^5$ mm⁴.

The maximum tensile bending stress due to M is **86.3** ✓ MPa.

The maximum shear stress due to V is **5.3** ✓ MPa.

References

Numeric Response

Chapter: 03 Load and Stress Analysis

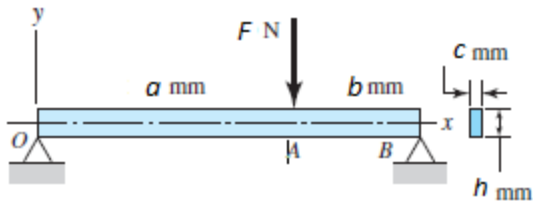
Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum

tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 320$ mm, $b = 160$ mm, $c = 20$ mm, $h = 39$ mm, and $F = 4100$ N.



The moment of inertia is $0.989 \pm 2\% \times 10^5 \text{ mm}^4$.

The maximum tensile bending stress due to M is $86.3 \pm 2\% \text{ MPa}$.

The maximum shear stress due to V is $5.3 \pm 2\% \text{ MPa}$.

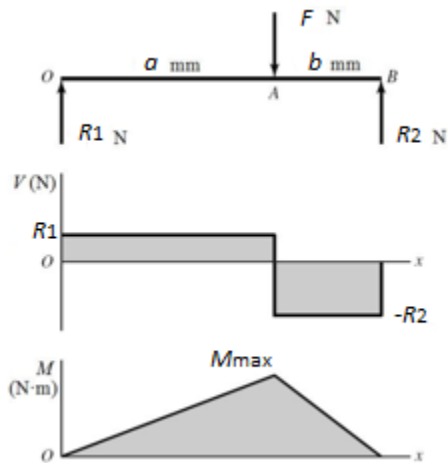
Explanation:

Given: $a = 320$ mm, $b = 160$ mm, $c = 20$ mm, $h = 39$ mm, and $F = 4100$ N

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (20 \text{ mm})(39 \text{ mm})^3 = 0.989 (10^5) \text{ mm}^4$$

$$A = (20 \text{ mm})(39 \text{ mm}) = 780 \text{ mm}^2$$



$M_{\max}$ is at A. At the bottom of the section,

$$R_1 = 1366.6667 \text{ N}$$

$$R_2 = 2733.3333 \text{ N}$$

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(2733.3333 \text{ N} \times 160 \text{ mm}) \left(\frac{39 \text{ mm}}{2} \right)}{0.989(10^5) \text{ mm}^4} = 86.3 \text{ MPa}$$

Due to V , $\tau_{\max}$ is between A and B at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{2733.3333 \text{ N}}{780 \text{ mm}^2} \right) = 5.3 \text{ MPa}$$

2.

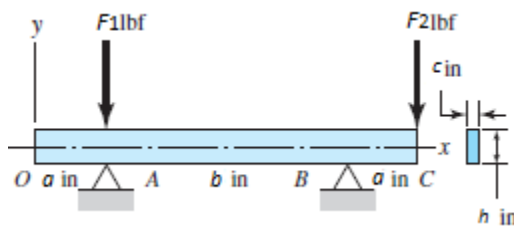
Award: 30 out of 30.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 4.5$ in, $b = 15$ in, $c = 0.7$ in, $h = 1.8$ in, $F_1 = 2900$ lbf, and $F_2 = 1000$ lbf.



The moment of inertia is **0.3402** in⁴.

The maximum tensile bending stress due to M is **11905** psi.

The maximum shear stress due to V is **1190** psi.

References

**Numeric
Response**

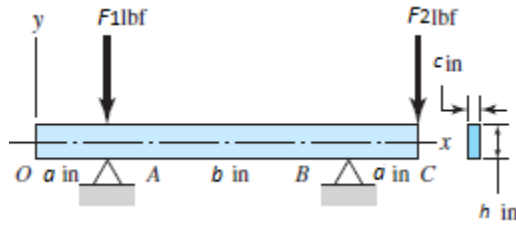
Chapter: 03 Load
and Stress Analysis

Problem 03.038 Section: 03.10
- Find the Normal Stresses for
locations/magn Beams in Bending
itudes of the
maximum
tensile bending
stress.

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum

tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 4.5$ in, $b = 15$ in, $c = 0.7$ in, $h = 1.8$ in, $F_1 = 2900$ lbf, and $F_2 = 1000$ lbf.



The moment of inertia is $0.3402 \pm 2\%$ in⁴.

The maximum tensile bending stress due to M is $11905 \pm 2\%$ psi.

The maximum shear stress due to V is $1190 \pm 2\%$ psi.

Explanation:

Given: $a = 4.5$ in, $b = 15$ in, $c = 0.7$ in, $h = 1.8$ in, $F_1 = 2900$ lbf, and $F_2 = 1000$ lbf.

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (0.7 \text{ in})(1.8 \text{ in})^3 = 0.3402 \text{ in}^4$$

$$A = (0.7 \text{ in})(1.8 \text{ in}) = 1.26 \text{ in}^2$$

We know that

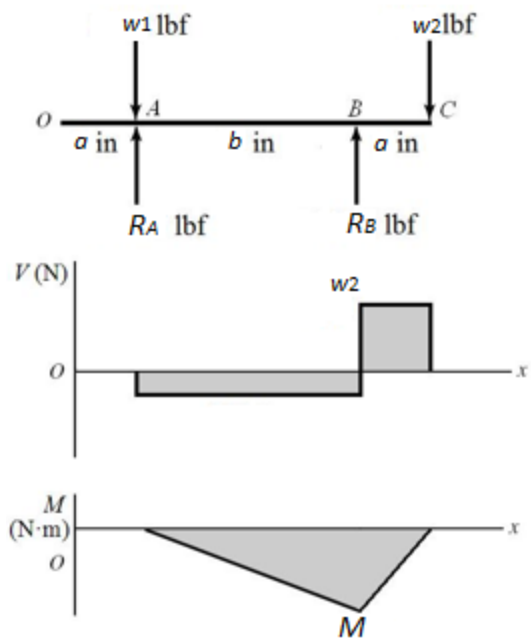
$$\Sigma M_A = 0$$

Therefore,

$$15R_B - (1000 \text{ lbf})(19.5 \text{ in}) = 0$$

$$R_B = 1300 \text{ lbf}$$

$$R_A = 2900 \text{ lbf} - 1300 \text{ lbf} + 1000 \text{ lbf} = 2600 \text{ lbf}$$



$M_{\max}$ is at B . At the bottom of the section,

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(4.5 \text{ in} \times 1000 \text{ lbf}) \times \left(\frac{1.8 \text{ in}}{2}\right)}{0.3402 \text{ in}^4} = 11905 \text{ psi}$$

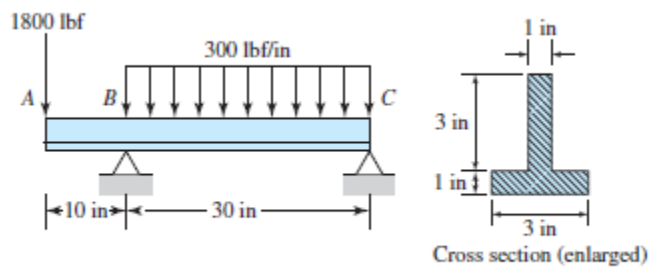
Due to V , $\tau_{\max}$ is between B and C at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{1000 \text{ lbf}}{1.26 \text{ in}^2} \right) = 1190 \text{ psi}$$

Problem 03.045 - Determine the maximum tensile, compressive, and shear stresses in a beam - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Consider the beam shown in the given figure.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break Chapter: 03 Load and Stress Analysis

Problem 03.045 Section: 03.10
- Determine the maximum Normal Stresses for
maximum tensile, compressive, and shear stresses in a
beam - DEPENDENT MULTI-PART PROBLEM -
ASSIGN ALL PARTS Beams in Bending

3.

Award: 10 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300$ lbf/in, $F = 1800$ lbf, $c_1 = 10$ in, and $c_2 = 30$ in.

The maximum tensile bending stress is **5294** ✓ psi.

The maximum compressive bending stress is **1** ✗ psi.

References

Numeric Response

Chapter: 03 Load and Stress Analysis

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300$ lbf/in, $F = 1800$ lbf, $c_1 = 10$ in, and $c_2 = 30$ in.

The maximum tensile bending stress is **5294 ± 2%** psi.

The maximum compressive bending stress is **-7456 ± 2%** psi.

Explanation:

$$R_1 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} + \frac{40 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 6900 \text{ lbf}$$

$$R_2 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} - \frac{10 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 3900 \text{ lbf}$$

$$a = \frac{3900 \text{ lbf}}{300 \text{ lbf/in}} = 13 \text{ in}$$

$$M_B = -1800 \text{ lbf} \times (10 \text{ in}) = -18000 \text{ lbf}\cdot\text{in}$$

$$M_{x=27 \text{ in}} = \frac{1}{2} (3900 \text{ lbf}) \times (13 \text{ in}) = 25350 \text{ lbf}\cdot\text{in}$$

$$\bar{y} = \frac{0.5 \text{ in} \times (3 \text{ in}) + 2.5 \text{ in} \times (3 \text{ in})}{6 \text{ in}} = 1.5 \text{ in}$$

$$I_1 = \frac{1}{12} (3 \text{ in}) \times (1 \text{ in})^3 = 0.25 \text{ in}^4$$

$$I_2 = \frac{1}{12} (1 \text{ in}) \times (3 \text{ in})^3 = 2.25 \text{ in}^4$$

Applying parallel-axis theorem,

$$I_z = \left(0.25 \text{ in}^2 + 3(1.5 \text{ in} - 0.5 \text{ in})^2 \right) + \left(2.25 \text{ in}^2 + 3(1.5 \text{ in} - 0.5 \text{ in})^2 \right) = 8.5 \text{ in}^4$$

The maximum tensile and compressive bending stresses are calculated as follows:

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{-18000 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5} = -3176 \text{ psi}$$

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{-18000 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5} = 5294 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = -1.5 \text{ in, } \sigma_x = - \frac{25350 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5 \text{ in}^4} = 4474 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{25350 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5 \text{ in}^4} = -7456 \text{ psi}$$

Maximum tension = 5294 psi

Maximum compression = -7456 psi

4.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is **1875**  psi.**References****Numeric Response**

Chapter: 03 Load and Stress Analysis

Problem 03.045.b - Determine the maximum shear stress.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is **1875 ± 2%** psi.**Explanation:**The maximum shear stress due to V is at B , at the neutral axis.

$$V_{\max} = 5100 \text{ lbf}$$

$$Q = \bar{y}' \times A' = 1.25 \text{ in} \times ((2.5) \times (1) \text{ in}^2) = 3.125 \text{ in}^3$$

$$(\tau_{\max})_V = \frac{VQ}{Ib} = \frac{5100 \text{ lbf} \times (3.125 \text{ in}^3)}{8.5 \text{ in}^4 \times (1 \text{ in})} = 1875 \text{ psi}$$

BELOW AVERAGE

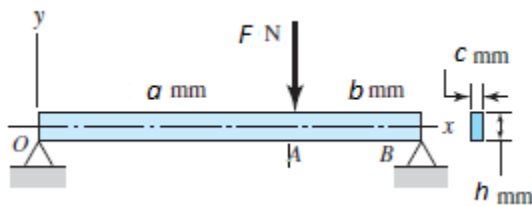
Score: **50/100** Points **50** %

1.

Award: 30 out of 30.00 points

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 320$ mm, $b = 140$ mm, $c = 19$ mm, $h = 41$ mm, and $F = 4800$ N.



The moment of inertia is **1.09** ✓ $\times 10^5$ mm⁴.

The maximum tensile bending stress due to M is **87.8** ✓ MPa.

The maximum shear stress due to V is **6.4** ✓ MPa.

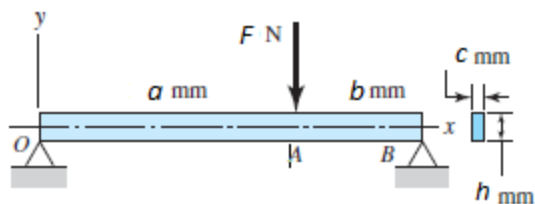
References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.036 Section: 03.10
- Find the locations/magnitudes of the maximum tensile bending stress. Normal Stresses for Beams in Bending

Problem 03.036 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . Parameters are $a = 320$ mm, $b = 140$ mm, $c = 19$ mm, $h = 41$ mm, and $F = 4800$ N.



The moment of inertia is $1.091 \pm 2\% \times 10^5 \text{ mm}^4$.

The maximum tensile bending stress due to M is $87.8 \pm 2\% \text{ MPa}$.

The maximum shear stress due to V is $6.4 \pm 2\% \text{ MPa}$.

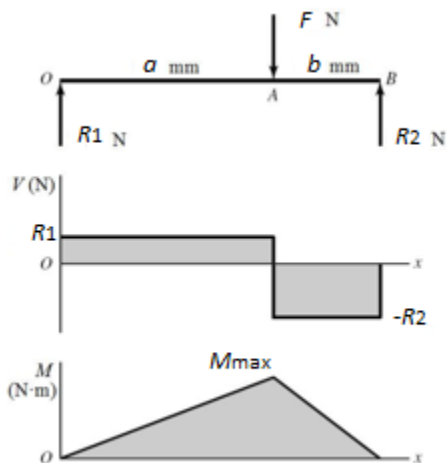
Explanation:

Given: $a = 320 \text{ mm}$, $b = 140 \text{ mm}$, $c = 19 \text{ mm}$, $h = 41 \text{ mm}$, and $F = 4800 \text{ N}$

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (19 \text{ mm})(41 \text{ mm})^3 = 1.091 (10^5) \text{ mm}^4$$

$$A = (19 \text{ mm})(41 \text{ mm}) = 779 \text{ mm}^2$$



$M_{\max}$ is at A. At the bottom of the section,

$$R_1 = 1460.8696 \text{ N}$$

$$R_2 = 3339.1304 \text{ N}$$

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(3339.1304 \text{ N} \times 140 \text{ mm}) \left(\frac{41 \text{ mm}}{2} \right)}{1.091 (10^5) \text{ mm}^4} = 87.8 \text{ MPa}$$

Due to V , $\tau_{\max}$ is between A and B at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{3339.1304 \text{ N}}{779 \text{ mm}^2} \right) = 6.4 \text{ MPa}$$

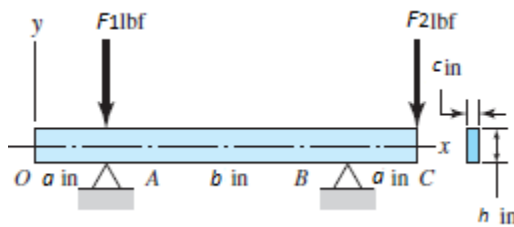


2.

Award: 20 out of 30.00 points

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 5$ in, $b = 15$ in, $c = 0.725$ in, $h = 2$ in, $F_1 = 2800$ lbf, and $F_2 = 1100$ lbf.



The moment of inertia is **0.483** ✓ in⁴.

The maximum tensile bending stress due to M is **7592** ✗ psi.

The maximum shear stress due to V is **1138** ✓ psi.

References

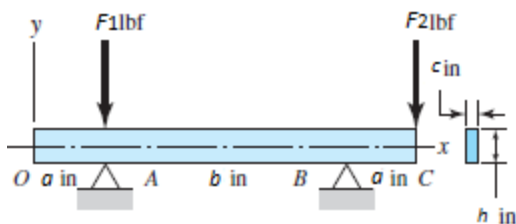
Numeric Response

Chapter: 03 Load and Stress Analysis

Problem 03.038 Section: 03.10
- Find the Normal Stresses for
locations/magn Beams in Bending
itudes of the
maximum
tensile bending
stress.

Problem 03.038 - Find the locations/magnitudes of the maximum tensile bending stress.

For the beam illustrated in the figure, find the locations and magnitudes of the maximum tensile bending stress due to M and the maximum shear stress due to V . The parameters are $a = 5$ in, $b = 15$ in, $c = 0.725$ in, $h = 2$ in, $F_1 = 2800$ lbf, and $F_2 = 1100$ lbf.



The moment of inertia is $0.4833 \pm 2\%$ in⁴.

The maximum tensile bending stress due to M is $11379 \pm 2\%$ psi.

The maximum shear stress due to V is $1138 \pm 2\%$ psi.

Explanation:

Given: $a = 5$ in, $b = 15$ in, $c = 0.725$ in, $h = 2$ in, $F_1 = 2800$ lbf, and $F_2 = 1100$ lbf.

The moment of inertia is calculated as follows:

$$I = \frac{1}{12} (0.725 \text{ in})(2 \text{ in})^3 = 0.4833 \text{ in}^4$$

$$A = (0.725 \text{ in})(2 \text{ in}) = 1.45 \text{ in}^2$$

We know that

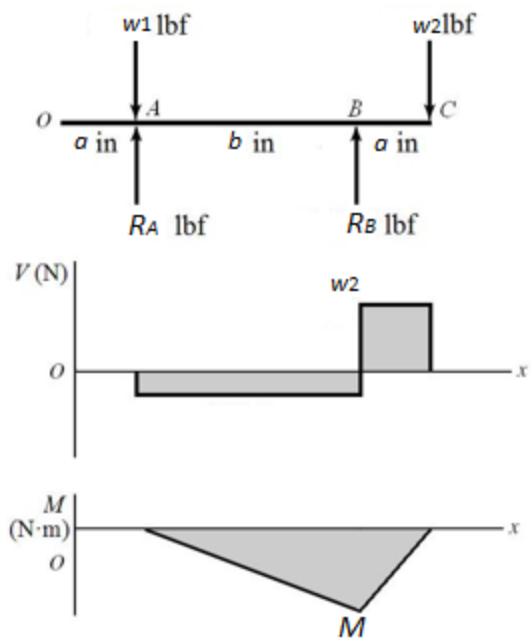
$$\Sigma M_A = 0$$

Therefore,

$$15R_B - (1100 \text{ lbf})(20 \text{ in}) = 0$$

$$R_B = 1467 \text{ lbf}$$

$$R_A = 2800 \text{ lbf} - 1467 \text{ lbf} + 1100 \text{ lbf} = 2433 \text{ lbf}$$



$M_{\max}$ is at B. At the bottom of the section,

$$\sigma_{\max} = \frac{Mc}{I} = \frac{(5 \text{ in} \times 1100 \text{ lbf}) \times \left(\frac{2 \text{ in}}{2}\right)}{0.4833 \text{ in}^4} = 11379 \text{ psi}$$

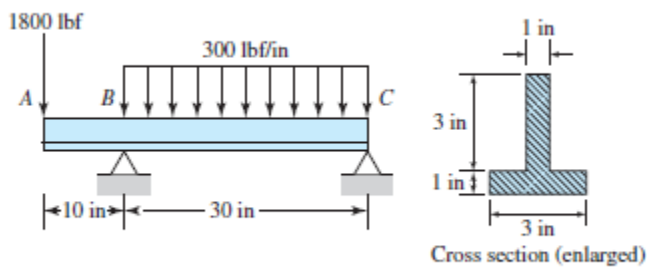
Due to V, $\tau_{\max}$ is between B and C at $y = 0$.

$$\tau_{\max} = \frac{3}{2} \frac{V}{A} = \frac{3}{2} \left(\frac{1100 \text{ lbf}}{1.45 \text{ in}^2} \right) = 1138 \text{ psi}$$

Problem 03.045 - Determine the maximum tensile, compressive, and shear stresses in a beam - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Consider the beam shown in the given figure.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break Chapter: 03 Load and Stress Analysis

Problem 03.045 Section: 03.10
- Determine the maximum Normal Stresses for
maximum tensile, compressive, and shear stresses in a
beam - DEPENDENT MULTI-PART PROBLEM -
ASSIGN ALL PARTS Beams in Bending

3.

Award: 0 out of 20.00 points

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300 \text{ lbf/in}$, $F = 1800 \text{ lbf}$, $c_1 = 10 \text{ in}$, and $c_2 = 30 \text{ in}$.

The maximum tensile bending stress is **4473** ❌ psi.

The maximum compressive bending stress is **7455** ❌ psi.

References

Numeric Response

Chapter: 03 Load and Stress Analysis

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.045.a - Determine the maximum tensile and compressive bending stresses.

Determine the maximum tensile and compressive bending stresses. Take $w = 300 \text{ lbf/in}$, $F = 1800 \text{ lbf}$, $c_1 = 10 \text{ in}$, and $c_2 = 30 \text{ in}$.

The maximum tensile bending stress is **5294 ± 2%** psi.

The maximum compressive bending stress is **-7456 ± 2%** psi.

Explanation:

$$R_1 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} + \frac{40 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 6900 \text{ lbf}$$

$$R_2 = \frac{300 \text{ lbf/in} \times 30 \text{ in}}{2} - \frac{10 \text{ in}}{30 \text{ in}} \times 1800 \text{ lbf} = 3900 \text{ lbf}$$

$$a = \frac{3900 \text{ lbf}}{300 \text{ lbf/in}} = 13 \text{ in}$$

$$M_B = -1800 \text{ lbf} \times (10 \text{ in}) = -18000 \text{ lbf}\cdot\text{in}$$

$$M_{x=27 \text{ in}} = \frac{1}{2} (3900 \text{ lbf}) \times (13 \text{ in}) = 25350 \text{ lbf}\cdot\text{in}$$

$$\bar{y} = \frac{0.5 \text{ in} \times (3 \text{ in}) + 2.5 \text{ in} \times (3 \text{ in})}{6 \text{ in}} = 1.5 \text{ in}$$

$$I_1 = \frac{1}{12} (3 \text{ in}) \times (1 \text{ in})^3 = 0.25 \text{ in}^4$$

$$I_2 = \frac{1}{12} (1 \text{ in}) \times (3 \text{ in})^3 = 2.25 \text{ in}^4$$

Applying parallel-axis theorem,

$$I_z = \left(0.25 \text{ in}^2 + 3(1.5 \text{ in} - 0.5 \text{ in})^2 \right) + \left(2.25 \text{ in}^2 + 3(1.5 \text{ in} - 0.5 \text{ in})^2 \right) = 8.5 \text{ in}^4$$

The maximum tensile and compressive bending stresses are calculated as follows:

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{-18000 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5} = -3176 \text{ psi}$$

$$\text{At } x = 10 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{-18000 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5} = 5294 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = -1.5 \text{ in, } \sigma_x = - \frac{25350 \text{ lbf}\cdot\text{in} \times (-1.5 \text{ in})}{8.5 \text{ in}^4} = 4474 \text{ psi}$$

$$\text{At } x = 27 \text{ in, } y = 2.5 \text{ in, } \sigma_x = - \frac{25350 \text{ lbf}\cdot\text{in} \times (2.5 \text{ in})}{8.5 \text{ in}^4} = -7456 \text{ psi}$$

Maximum tension = 5294 psi

Maximum compression = -7456 psi

4.

Award: 0 out of 20.00 points

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is **3728** ❌ psi.**References****Numeric Response**

Chapter: 03 Load and Stress Analysis

Problem 03.045.b - Determine the maximum shear stress.

Section: 03.10 Normal Stresses for Beams in Bending

Problem 03.045.b - Determine the maximum shear stress.

Determine the maximum shear stress due to V .The maximum shear stress due to V is **1875 ± 2%** psi.**Explanation:**The maximum shear stress due to V is at B , at the neutral axis.

$$V_{\max} = 5100 \text{ lbf}$$

$$Q = \bar{y}' \times A' = 1.25 \text{ in} \times ((2.5) \times (1) \text{ in}^2) = 3.125 \text{ in}^3$$

$$(\tau_{\max})_V = \frac{VQ}{Ib} = \frac{5100 \text{ lbf} \times (3.125 \text{ in}^3)}{8.5 \text{ in}^4 \times (1 \text{ in})} = 1875 \text{ psi}$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 7

TOPICS COVERED IN HOMEWORK 7

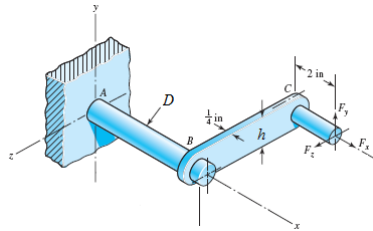
Sections 3-10, 3-11, 3-12: Problems 3-91, 3-95

**MACHINE DESIGN
I FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

Problem 03.091 - Find the principal stresses and maximum shear stress on a cantilevered bar.- DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

The cantilevered bar in the figure is made from a ductile material and is statically loaded with $F_y = 380$ lbf and $F_x = F_z = 0$. The diameter D is 2 in and the height h is 2.5 in. Lengths AB and BC are 9 in. and 5 in., respectively.

**References****Section Break**

Problem 03.091 - Find the principal stresses and maximum shear stress on a cantilevered bar.- DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

1.

Award: 10.00 points

Problem 03.091.a - Identify the location of the critical stress element.

What is the location of the critical stress element?

- ☐ Point A in the center of rod AB
☐ Point B in the center of rod AB
☐ Point B at the top or bottom surface of rod AB
☒ Point A at the top or bottom surface of rod AB

Rod AB experiences constant torsion throughout its length and maximum bending moment at the wall. Both torsional shear stress and bending stress will be maximum on the outer surface. The transverse shear will be very small compared to bending and torsion due to the reasonably high length to diameter ratio. Thus, it will not dominate the determination of the critical location. The critical stress element will be at the wall, at either the top (compression) or the bottom (tension) on the y axis. We will select the bottom element for this analysis.

The critical stress element is located at point A at the top or bottom surface of rod AB.

References**Multiple Choice**

Problem 03.091.a - Identify the location of the critical stress element.

Difficulty: Medium

2.

Award: 20.00 points

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is $0 \pm 2\%$ psi.

The bending stress is $5322.14 \pm 2\%$ psi.

The torsional shear stress is $1209.58 \pm 2\%$ psi.

Explanation:

Transverse shear is zero at the critical stress element. We need to calculate the bending stress and torsional shear stress.

$$\sigma_x = \frac{Mc}{I} = \frac{32M}{\pi d^3} = \frac{32(380 \text{ lb}\cdot\text{ft})(9 + 2) \text{ in.}}{\pi(2 \text{ in.})^3} = 5322.14 \text{ psi}$$

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16(380 \text{ lb}\cdot\text{ft})(5 \text{ in.})}{\pi(2 \text{ in.})^3} = 1209.58 \text{ psi}$$

The transverse shear stress is 0 psi.

The bending stress is 5322.14 psi.

The torsional shear stress is 1209.58 psi.

References

Numeric Response

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Difficulty: Medium

3.

Award: 20.00 points

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = 5584.15 \pm 2\%$ psi and $\sigma_2 = -262.01 \pm 2\%$ psi.

The maximum shear stress is $2923.08 \pm 2\%$ psi.

Explanation:

The principal normal stresses are

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{5322.14 \text{ psi}}{2} + \sqrt{\left(\frac{5322.14 \text{ psi}}{2}\right)^2 + (1209.58 \text{ psi})^2} = 5584.15 \text{ psi}$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{5322.14 \text{ psi}}{2} - \sqrt{\left(\frac{5322.14 \text{ psi}}{2}\right)^2 + (1209.58 \text{ psi})^2} = -262.01 \text{ psi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{5322.14 \text{ psi}}{2}\right)^2 + (1209.58 \text{ psi})^2} = 2923.08 \text{ psi}$$

The principal stresses are $\sigma_1 = 5584.15$ psi and $\sigma_2 = -262.01$ psi.

The maximum shear stress is 2923.08 psi.

References

Numeric Response

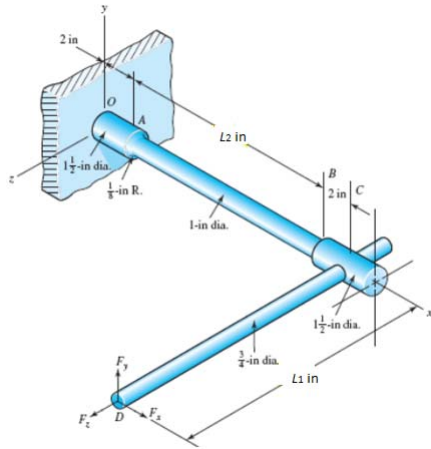
Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

Difficulty: Medium

Problem 03.095 - Find the principal stresses and maximum shear stress on a cantilevered bar - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

The cantilever bar in the figure is made from a ductile material and is statically loaded with $F_y = 100$ lbf and $F_x = F_z = 0$ lbf. The length of the rod DC denoted by L_1 is given to be 11 in. The length of the rod AB denoted by L_2 is given to be 9 in.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.


References
Section Break

Problem 03.095 - Find the principal stresses and maximum shear stress on a cantilevered bar - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

4.

Award: 10.00 points

Problem 03.095.a - Find the critical point at section A.

Where is the critical section at A?

- ☐ On the side along the $-z$ axis
- ☐ On the bottom of the section
- ☐ On the side along the $+z$ axis
- ☐ On the top of the section

The cross section at A will experience bending, torsion, and transverse shear. Both torsional shear stress and bending stress will be a maximum on the outer surface. The transverse shear will be very small compared to bending and torsion, due to the reasonably high length to diameter ratio, so it will not dominate the determination of the critical location. The critical stress element will be at either the top (compression) or the bottom (tension) on the y axis. We'll select the bottom element for this analysis.

The critical section at A is on the bottom of the section.

References
Multiple Choice

Problem 03.095.a - Find the critical point at section A.

Difficulty: Medium

5. Award: 20.00 points

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be $11.2045 \pm 2\%$ kpsi.

The shear stress at A is determined to be $5.6023 \pm 2\%$ kpsi.

Explanation:

The moment at point A is calculated as follows:

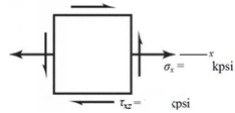
$$M_z = F_y L_2 = 100 \text{ lbf} \times (9 \text{ in} + 2 \text{ in}) = 1100 \text{ lb-in}$$

The torque at A is calculated as follows:

$$T = F_y L_1 = 100 \text{ lbf} \times 11 \text{ in} = 1100 \text{ lb-in}$$

The bending stress at A is in tension at the bottom of the section:

$$\sigma_x = \frac{M_z c}{I} = \frac{32 M_z}{\pi d^3} = \frac{32 \times 1100 \text{ lb-in}}{\pi \times (1 \text{ in})^3} \cdot \frac{1}{1000} = 11.2045 \text{ kpsi}$$



The shear stress at A due to torsion is calculated as follows:

$$\tau_{xz} = \frac{T r}{J} = \frac{16 T}{\pi d^3} = \frac{16 \times 1100 \text{ lb-in}}{\pi \times (1 \text{ in})^3} \cdot \frac{1}{1000} = 5.6023 \text{ kpsi}$$

The bending stress at A is determined to be 11.2045 kpsi.

The shear stress at A is determined to be 5.6023 kpsi.

References

Numeric Response

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Difficulty: Medium

6. Award: 20.00 points

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be $13.525 \pm 2\%$ kpsi.

The principal stress σ_2 is determined to be $-2.3205 \pm 2\%$ kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be $7.9228 \pm 2\%$ kpsi.

Explanation:

The principal normal stresses are as follows:

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{11.2045 \text{ kpsi}}{2} + \sqrt{\left(\frac{11.2045 \text{ kpsi}}{2}\right)^2 + (5.6023 \text{ kpsi})^2} = 13.525 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{11.2045 \text{ kpsi}}{2} - \sqrt{\left(\frac{11.2045 \text{ kpsi}}{2}\right)^2 + (5.6023 \text{ kpsi})^2} = -2.3205 \text{ kpsi}$$

The principal shear stress is as follows:

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{11.2045 \text{ kpsi}}{2}\right)^2 + (5.6023 \text{ kpsi})^2} = 7.9228 \text{ kpsi}$$

The principal stress σ_1 is determined to be 13.525 kpsi.

The principal stress σ_2 is determined to be -2.3205 kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be 7.9228 kpsi.

References

Numeric Response

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

Difficulty: Medium

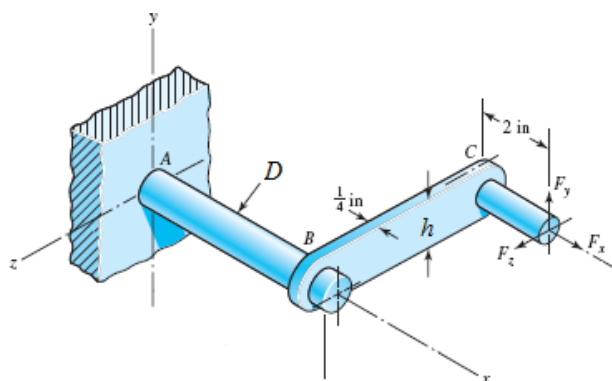
ABOVE AVERAGE

Lee, Malaika

Score: 100/100 Points 100 %

**Problem 03.091 - Find the principal stresses and maximum shear stress on a cantilevered bar.-
DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS**

The cantilevered bar in the figure is made from a ductile material and is statically loaded with $F_y = 140$ lbf and $F_x = F_z = 0$. The diameter D is 2 in and the height h is 2.5 in. Lengths AB and BC are 12 in. and 4 in., respectively.

**References**

Section Break

Chapter: 03 Load and
Stress Analysis

Problem 03.091 -
Find the principal
stresses and
maximum shear
stress on a
cantilevered bar.-
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 03.11 Shear
Stresses for Beams in
Bending

1.

Award: 10 out of 10.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.a - Identify the location of the critical stress element.

What is the location of the critical stress element?

- ☐ Point *B* at the top or bottom surface of rod *AB*
- ☐ Point *A* in the center of rod *AB*
- ✓ ☒ Point *A* at the top or bottom surface of rod *AB*
- ☐ Point *B* in the center of rod *AB*

Rod *AB* experiences constant torsion throughout its length and maximum bending moment at the wall. Both torsional shear stress and bending stress will be maximum on the outer surface. The transverse shear will be very small compared to bending and torsion due to the reasonably high length to diameter ratio. Thus, it will not dominate the determination of the critical location. The critical stress element will be at the wall, at either the top (compression) or the bottom (tension) on the *y* axis. We will select the bottom element for this analysis.

References

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.091.a - Identify the location of the critical stress element.

Section: 03.11 Shear Stresses for Beams in Bending

2.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is 0 ✓ psi.

The bending stress is -2495.55 ✓ psi.

The torsional shear stress is 356.51 ✓ psi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is 0 ± 2% psi.

The bending stress is -2495.55 ± 2% psi.

The torsional shear stress is 356.51 ± 2% psi.

Explanation:

Transverse shear is zero at the critical stress element. We need to calculate the bending stress and torsional shear stress.

$$\sigma_x = -\frac{Mc}{I} = -\frac{32M}{\pi d^3} = -\frac{32(140 \text{ lbf})(12 + 2) \text{ in.}}{\pi(2 \text{ in.})^3} = -2495.55 \text{ psi}$$

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16(140 \text{ lbf})(4 \text{ in.})}{\pi(2 \text{ in.})^3} = 356.51 \text{ psi}$$

3.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = \underline{49.93}$ ✓ psi and $\sigma_2 = \underline{-2545.48}$ ✓ psi.

The maximum shear stress is $\underline{1297.71}$ ✓ psi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = \underline{49.93 \pm 2\%}$ psi and $\sigma_2 = \underline{-2545.48 \pm 2\%}$ psi.

The maximum shear stress is $\underline{1297.71 \pm 2\%}$ psi.

Explanation:

The principal normal stresses are

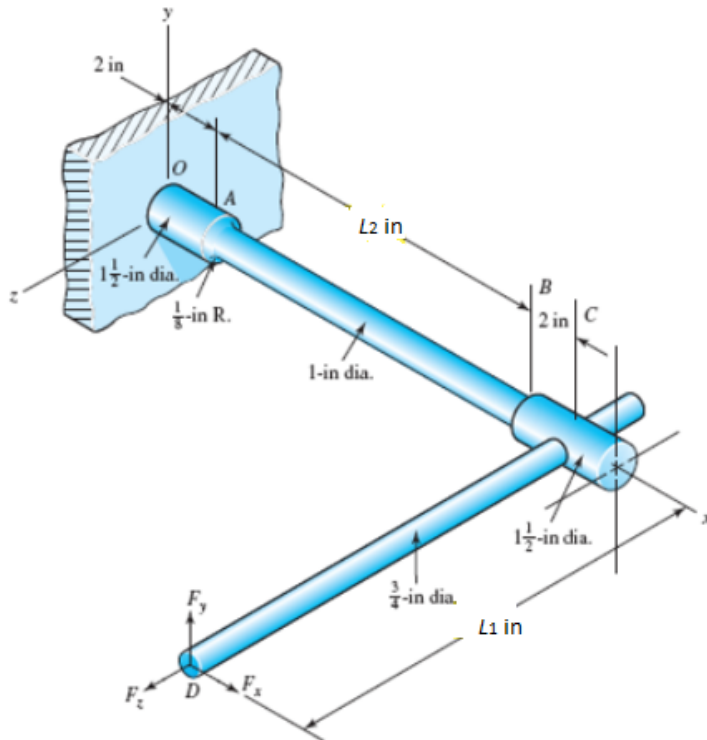
$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{-2495.55 \text{ psi}}{2} + \sqrt{\left(\frac{-2495.55 \text{ psi}}{2}\right)^2 + (356.51 \text{ psi})^2} = 49.93 \text{ psi}$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{-2495.55 \text{ psi}}{2} - \sqrt{\left(\frac{-2495.55 \text{ psi}}{2}\right)^2 + (356.51 \text{ psi})^2} = -2545.48 \text{ psi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{-2495.55 \text{ psi}}{2}\right)^2 + (356.51 \text{ psi})^2} = 1297.71 \text{ psi}$$

The cantilever bar in the figure is made from a ductile material and is statically loaded with $F_y = 200$ lbf and $F_x = F_z = 0$ lbf. The length of the rod DC denoted by L_1 is given to be 12 in. The length of the rod AB denoted by L_2 is given to be 10 in. NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



Section Break	Chapter: 03 Load and Stress Analysis
Problem 03.095 - Find the principal stresses and maximum shear stress on a cantilevered bar - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 03.11 Shear Stresses for Beams in Bending

4.

Award: 10 out of 10.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.a - Find the critical point at section A.

Where is the critical section at A?

- ☐ On the top of the section
- ☐ On the side along the $-z$ axis
- ☐ On the side along the $+z$ axis
- ✓ ☒ On the bottom of the section

The cross section at A will experience bending, torsion, and transverse shear. Both torsional shear stress and bending stress will be a maximum on the outer surface. The transverse shear will be very small compared to bending and torsion, due to the reasonably high length to diameter ratio, so it will not dominate the determination of the critical location. The critical stress element will be at either the top (compression) or the bottom (tension) on the y axis. We'll select the bottom element for this analysis.

[References](#)

Multiple Choice	Chapter: 03 Load and Stress Analysis
Problem 03.095.a - Find the critical point at section A.	Section: 03.11 Shear Stresses for Beams in Bending

5.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be **24.4462** ✓ kpsi.

The shear stress at A is determined to be **12.2231** ✓ kpsi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be **24.4462 ± 2%** kpsi.

The shear stress at A is determined to be **12.2231 ± 2%** kpsi.

Explanation:

The moment at point A is calculated as follows:

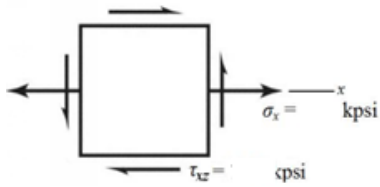
$$M_z = F_y L_2 = 200 \text{ lbf} \times (10 \text{ in} + 2 \text{ in}) = 2400 \text{ lb}\cdot\text{in}$$

The torque at A is calculated as follows:

$$T = F_y L_1 = 200 \text{ lbf} \times 12 \text{ in} = 2400 \text{ lb}\cdot\text{in}$$

The bending stress at A is in tension at the bottom of the section:

$$\sigma_x = \frac{M_z c}{I} = \frac{32 M_z}{\pi d^3} = \frac{32 \times 2400 \text{ lb}\cdot\text{in}}{\pi \times (1 \text{ in})^3} \frac{1}{1000} = 24.4462 \text{ kpsi}$$



The shear stress at A due to torsion is calculated as follows:

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16 \times 2400 \text{ lb}\cdot\text{in}}{\pi \times (1 \text{ in})^3} \frac{1}{1000} = 12.2231 \text{ kpsi}$$

6.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be **29.5092** ✓ kpsi.

The principal stress σ_2 is determined to be **-5.063** ✓ kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be **17.2861** ✓ kpsi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be **29.5092 ± 2%** kpsi.

The principal stress σ_2 is determined to be **-5.063 ± 2%** kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be **17.2861 ± 2%** kpsi.

Explanation:

The principal normal stresses are as follows:

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{24.4462 \text{ kpsi}}{2} + \sqrt{\left(\frac{24.4462 \text{ kpsi}}{2}\right)^2 + (12.2231 \text{ kpsi})^2} = 29.5092 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{24.4462 \text{ kpsi}}{2} - \sqrt{\left(\frac{24.4462 \text{ kpsi}}{2}\right)^2 + (12.2231 \text{ kpsi})^2} = -5.063 \text{ kpsi}$$

The principal shear stress is as follows:

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{24.4462 \text{ kpsi}}{2}\right)^2 + (12.2231 \text{ kpsi})^2} = 17.2861 \text{ kpsi}$$

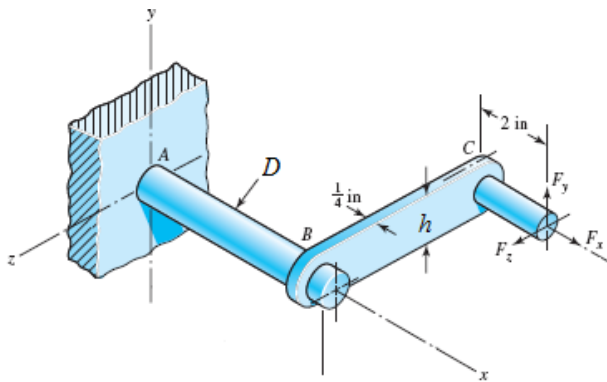
AVERAGE

Garay, Gabriela

Score: 93.33/100 Points 93.33 %

**Problem 03.091 - Find the principal stresses and maximum shear stress on a cantilevered bar.-
DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS**

The cantilevered bar in the figure is made from a ductile material and is statically loaded with $F_y = 260$ lbf and $F_x = F_z = 0$. The diameter D is 3 in and the height h is 3.5 in. Lengths AB and BC are 15 in. and 5 in., respectively.

**References**

Section Break

Chapter: 03 Load and
Stress Analysis

Problem 03.091 -
Find the principal
stresses and
maximum shear
stress on a
cantilevered bar.-
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 03.11 Shear
Stresses for Beams in
Bending

1.

Award: 10 out of 10.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.a - Identify the location of the critical stress element.

What is the location of the critical stress element?

- ✓ ☒ Point *A* at the top or bottom surface of rod *AB*
- ☐ Point *A* in the center of rod *AB*
- ☐ Point *B* at the top or bottom surface of rod *AB*
- ☐ Point *B* in the center of rod *AB*

Rod *AB* experiences constant torsion throughout its length and maximum bending moment at the wall. Both torsional shear stress and bending stress will be maximum on the outer surface. The transverse shear will be very small compared to bending and torsion due to the reasonably high length to diameter ratio. Thus, it will not dominate the determination of the critical location. The critical stress element will be at the wall, at either the top (compression) or the bottom (tension) on the *y* axis. We will select the bottom element for this analysis.

References

Multiple Choice

Chapter: 03 Load and
Stress AnalysisProblem 03.091.a -
Identify the location
of the critical stress
element.Section: 03.11 Shear
Stresses for Beams in
Bending

2.

Award: 13.33 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is 0 ✓ psi.

The bending stress is -166.74 ✗ psi.

The torsional shear stress is 245.22 ✓ psi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is 0 ± 2% psi.

The bending stress is -1667.47 ± 2% psi.

The torsional shear stress is 245.22 ± 2% psi.

Explanation:

Transverse shear is zero at the critical stress element. We need to calculate the bending stress and torsional shear stress.

$$\sigma_x = -\frac{Mc}{I} = -\frac{32M}{\pi d^3} = -\frac{32(260 \text{ lbf})(15 + 2) \text{ in.}}{\pi(3 \text{ in.})^3} = -1667.47 \text{ psi}$$

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16(260 \text{ lbf})(5 \text{ in.})}{\pi(3 \text{ in.})^3} = 245.22 \text{ psi}$$

3.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = \underline{35.31}$ ✓ psi and $\sigma_2 = -\underline{1702.8}$ ✓ psi.

The maximum shear stress is $\underline{869.05}$ ✓ psi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = \underline{35.31 \pm 2\%}$ psi and $\sigma_2 = -\underline{1702.79 \pm 2\%}$ psi.

The maximum shear stress is $\underline{869.05 \pm 2\%}$ psi.

Explanation:

The principal normal stresses are

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{-1667.47 \text{ psi}}{2} + \sqrt{\left(\frac{-1667.47 \text{ psi}}{2}\right)^2 + (245.22 \text{ psi})^2} = 35.31 \text{ psi}$$

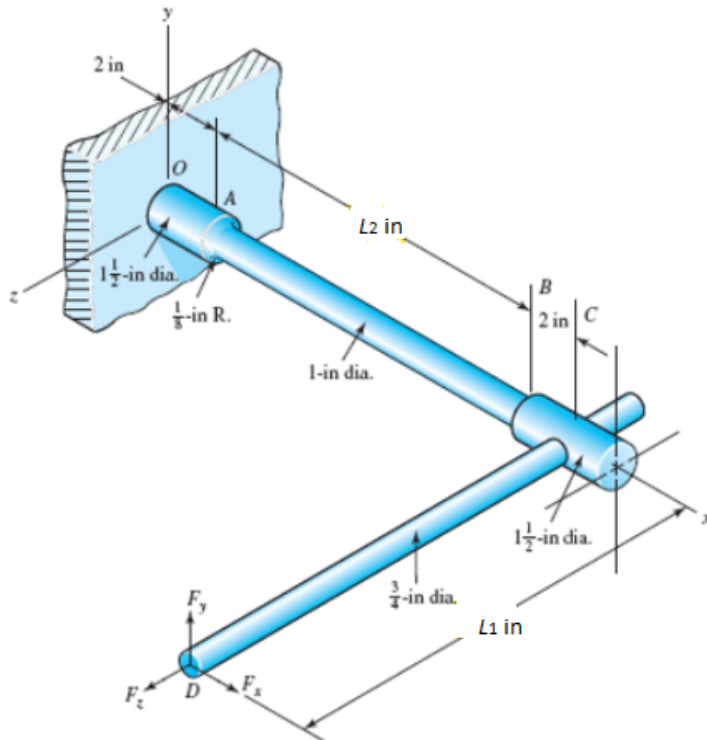
$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{-1667.47 \text{ psi}}{2} - \sqrt{\left(\frac{-1667.47 \text{ psi}}{2}\right)^2 + (245.22 \text{ psi})^2} = -1702.79 \text{ psi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{-1667.47 \text{ psi}}{2}\right)^2 + (245.22 \text{ psi})^2} = 869.05 \text{ psi}$$

The cantilever bar in the figure is made from a ductile material and is statically loaded with $F_y = 150$ lbf and $F_x = F_z = 0$ lbf. The length of the rod DC denoted by L_1 is given to be 10 in. The length of the rod AB denoted by L_2 is given to be 9 in.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



4.

Award: 10 out of 10.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.a - Find the critical point at section A.

Where is the critical section at A?

- ☐ On the side along the + z axis
- ☐ On the side along the - z axis
- ☐ On the top of the section
- ✓ ☒ On the bottom of the section

The cross section at A will experience bending, torsion, and transverse shear. Both torsional shear stress and bending stress will be a maximum on the outer surface. The transverse shear will be very small compared to bending and torsion, due to the reasonably high length to diameter ratio, so it will not dominate the determination of the critical location. The critical stress element will be at either the top (compression) or the bottom (tension) on the y axis. We'll select the bottom element for this analysis.

[References](#)

Multiple Choice

Chapter: 03 Load and Stress Analysis

Problem 03.095.a - Find the critical point at section A.

Section: 03.11 Shear Stresses for Beams in Bending

5.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be **16.8** ✓ kpsi.

The shear stress at A is determined to be **7.6394** ✓ kpsi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be **16.8068 ± 2%** kpsi.

The shear stress at A is determined to be **7.6394 ± 2%** kpsi.

Explanation:

The moment at point A is calculated as follows:

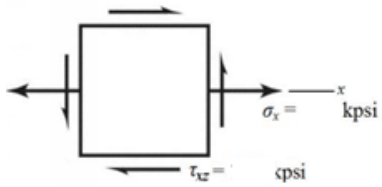
$$M_z = F_y L_2 = 150 \text{ lbf} \times (9 \text{ in} + 2 \text{ in}) = 1650 \text{ lb}\cdot\text{in}$$

The torque at A is calculated as follows:

$$T = F_y L_1 = 150 \text{ lbf} \times 10 \text{ in} = 1500 \text{ lb}\cdot\text{in}$$

The bending stress at A is in tension at the bottom of the section:

$$\sigma_x = \frac{M_z c}{I} = \frac{32 M_z}{\pi d^3} = \frac{32 \times 1650 \text{ lb}\cdot\text{in}}{\pi \times (1 \text{ in})^3} \frac{1}{1000} = 16.8068 \text{ kpsi}$$



The shear stress at A due to torsion is calculated as follows:

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16 \times 1500 \text{ lb}\cdot\text{in}}{\pi \times (1 \text{ in})^3} \frac{1}{1000} = 7.6394 \text{ kpsi}$$

6.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be **19.7602** kpsi.

The principal stress σ_2 is determined to be **-2.9535** kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be **11.3568** kpsi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be **19.7602 ± 2%** kpsi.

The principal stress σ_2 is determined to be **-2.9535 ± 2%** kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be **11.3568 ± 2%** kpsi.

Explanation:

The principal normal stresses are as follows:

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{16.8068 \text{ kpsi}}{2} + \sqrt{\left(\frac{16.8068 \text{ kpsi}}{2}\right)^2 + (7.6394 \text{ kpsi})^2} = 19.7602 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{16.8068 \text{ kpsi}}{2} - \sqrt{\left(\frac{16.8068 \text{ kpsi}}{2}\right)^2 + (7.6394 \text{ kpsi})^2} = -2.9535 \text{ kpsi}$$

The principal shear stress is as follows:

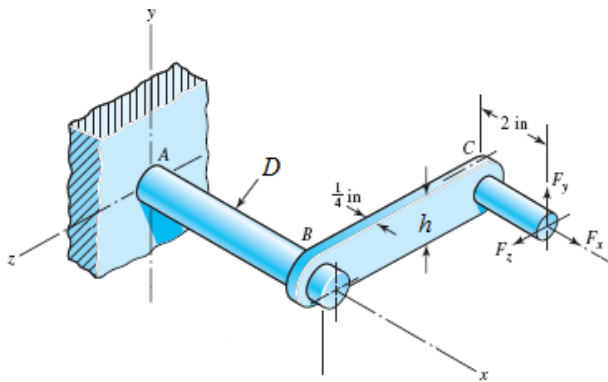
$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{16.8068 \text{ kpsi}}{2}\right)^2 + (7.6394 \text{ kpsi})^2} = 11.3568 \text{ kpsi}$$

BELOW AVERAGE

Zampaligre, Safiatou

Score: **80/100** Points **80** %**Problem 03.091 - Find the principal stresses and maximum shear stress on a cantilevered bar.-
DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS**

The cantilevered bar in the figure is made from a ductile material and is statically loaded with $F_y = 380$ lbf and $F_x = F_z = 0$. The diameter D is 2 in and the height h is 2.5 in. Lengths AB and BC are 10 in. and 4 in., respectively.



References

Section Break

Chapter: 03 Load and
Stress Analysis

Problem 03.091 -
Find the principal
stresses and
maximum shear
stress on a
cantilevered bar.-
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 03.11 Shear
Stresses for Beams in
Bending

1.

Award: 10 out of 10.00 points



You received credit for this question in a previous attempt

Problem 03.091.a - Identify the location of the critical stress element.

What is the location of the critical stress element?

- ☐ Point *A* in the center of rod *AB*
- ☐ Point *B* in the center of rod *AB*
- ✓ ☒ Point *A* at the top or bottom surface of rod *AB*
- ☐ Point *B* at the top or bottom surface of rod *AB*

Rod *AB* experiences constant torsion throughout its length and maximum bending moment at the wall. Both torsional shear stress and bending stress will be maximum on the outer surface. The transverse shear will be very small compared to bending and torsion due to the reasonably high length to diameter ratio. Thus, it will not dominate the determination of the critical location. The critical stress element will be at the wall, at either the top (compression) or the bottom (tension) on the *y* axis. We will select the bottom element for this analysis.

References

Multiple Choice

Chapter: 03 Load and
Stress AnalysisProblem 03.091.a -
Identify the location
of the critical stress
element.Section: 03.11 Shear
Stresses for Beams in
Bending

2.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is 0 ✓ psi.

The bending stress is -5805.97 ✓ psi.

The torsional shear stress is 967.66 ✓ psi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.091.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes and directions for all stresses acting on the critical stress element. (Transverse shear may only be neglected if you can justify this decision.)

The transverse shear stress is 0 ± 2% psi.

The bending stress is -5805.97 ± 2% psi.

The torsional shear stress is 967.66 ± 2% psi.

Explanation:

Transverse shear is zero at the critical stress element. We need to calculate the bending stress and torsional shear stress.

$$\sigma_x = -\frac{Mc}{I} = -\frac{32M}{\pi d^3} = -\frac{32(380 \text{ lbf})(10 + 2) \text{ in.}}{\pi(2 \text{ in.})^3} = -5805.97 \text{ psi}$$

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16(380 \text{ lbf})(4 \text{ in.})}{\pi(2 \text{ in.})^3} = 967.66 \text{ psi}$$

3.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = \underline{157.03}$ ✓ psi and $\sigma_2 = \underline{-5963}$ ✓ psi.

The maximum shear stress is $\underline{3060.02}$ ✓ psi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.091.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stresses are $\sigma_1 = \underline{157.03 \pm 2\%}$ psi and $\sigma_2 = \underline{-5963 \pm 2\%}$ psi.

The maximum shear stress is $\underline{3060.02 \pm 2\%}$ psi.

Explanation:

The principal normal stresses are

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{-5805.97 \text{ psi}}{2} + \sqrt{\left(\frac{-5805.97 \text{ psi}}{2}\right)^2 + (967.66 \text{ psi})^2} = 157.03 \text{ psi}$$

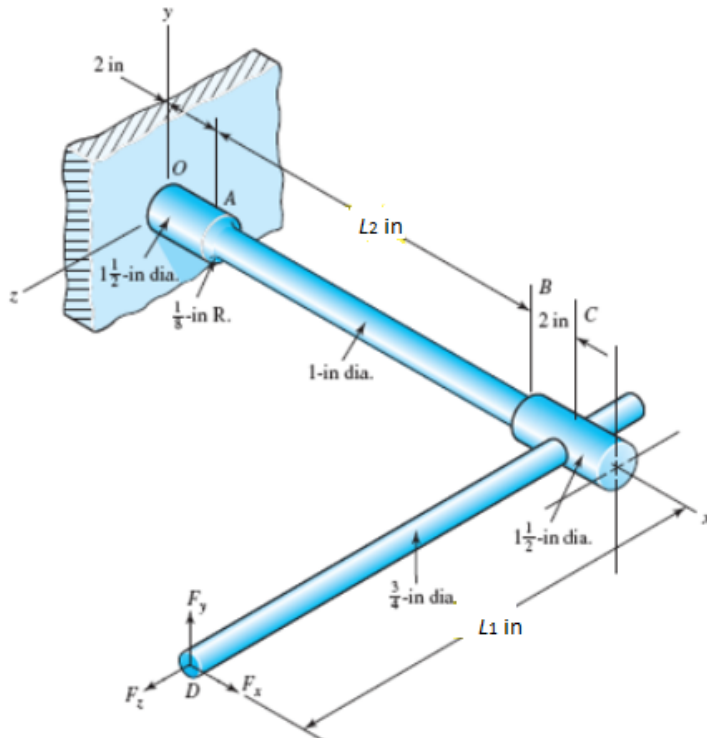
$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{-5805.97 \text{ psi}}{2} - \sqrt{\left(\frac{-5805.97 \text{ psi}}{2}\right)^2 + (967.66 \text{ psi})^2} = -5963 \text{ psi}$$

The maximum shear stress is

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{-5805.97 \text{ psi}}{2}\right)^2 + (967.66 \text{ psi})^2} = 3060.02 \text{ psi}$$

Problem 03.095 - Find the principal stresses and maximum shear stress on a cantilevered bar
- DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

The cantilever bar in the figure is made from a ductile material and is statically loaded with $F_y = 300$ lbf and $F_x = F_z = 0$ lbf. The length of the rod DC denoted by L_1 is given to be 14 in. The length of the rod AB denoted by L_2 is given to be 9 in. NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 03 Load and Stress Analysis

Problem 03.095 - Find the principal stresses and maximum shear stress on a cantilevered bar - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Section: 03.11 Shear Stresses for Beams in Bending

4.

Award: 10 out of 10.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.a - Find the critical point at section A.

Where is the critical section at A?

- ☐ On the top of the section
- ☐ On the side along the + z axis
- ✓ ☒ On the bottom of the section
- ☐ On the side along the - z axis

The cross section at A will experience bending, torsion, and transverse shear. Both torsional shear stress and bending stress will be a maximum on the outer surface. The transverse shear will be very small compared to bending and torsion, due to the reasonably high length to diameter ratio, so it will not dominate the determination of the critical location. The critical stress element will be at either the top (compression) or the bottom (tension) on the y axis. We'll select the bottom element for this analysis.

[References](#)

Multiple Choice Chapter: 03 Load and Stress Analysis

Problem 03.095.a - Find the critical point at section A. Section: 03.11 Shear Stresses for Beams in Bending

5.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be **33.6135** ✓ kpsi.

The shear stress at A is determined to be **21.3904** ✓ kpsi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.095.b - Determine the magnitudes and directions for all stresses acting on the critical stress element.

Determine the magnitudes for all stresses acting on the critical stress element.

The bending stress at A is determined to be **33.6135 ± 2%** kpsi.

The shear stress at A is determined to be **21.3904 ± 2%** kpsi.

Explanation:

The moment at point A is calculated as follows:

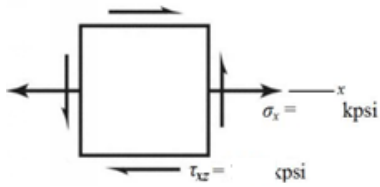
$$M_z = F_y L_2 = 300 \text{ lbf} \times (9 \text{ in} + 2 \text{ in}) = 3300 \text{ lb}\cdot\text{in}$$

The torque at A is calculated as follows:

$$T = F_y L_1 = 300 \text{ lbf} \times 14 \text{ in} = 4200 \text{ lb}\cdot\text{in}$$

The bending stress at A is in tension at the bottom of the section:

$$\sigma_x = \frac{M_z c}{I} = \frac{32 M_z}{\pi d^3} = \frac{32 \times 3300 \text{ lb}\cdot\text{in}}{\pi \times (1 \text{ in})^3} \frac{1}{1000} = 33.6135 \text{ kpsi}$$



The shear stress at A due to torsion is calculated as follows:

$$\tau_{xz} = \frac{Tr}{J} = \frac{16T}{\pi d^3} = \frac{16 \times 4200 \text{ lb}\cdot\text{in}}{\pi \times (1 \text{ in})^3} \frac{1}{1000} = 21.3904 \text{ kpsi}$$

6.

Award: 0 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be kml kpsi.

The principal stress σ_2 is determined to be klm, kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be klm,, kpsi.

References

Numeric Response Chapter: 03 Load and Stress Analysis

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress. Section: 03.11 Shear Stresses for Beams in Bending

Problem 03.095.c - For the critical stress element, determine the principal stresses and the maximum shear stress.

For the critical stress element, determine the principal stresses and the maximum shear stress.

The principal stress σ_1 is determined to be 44.01 ± 2% kpsi.

The principal stress σ_2 is determined to be -10.3965 ± 2% kpsi.

The maximum shear stress $\tau_{\max}$ is determined to be 27.2033 ± 2% kpsi.

Explanation:

The principal normal stresses are as follows:

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{33.6135 \text{ kpsi}}{2} + \sqrt{\left(\frac{33.6135 \text{ kpsi}}{2}\right)^2 + (21.3904 \text{ kpsi})^2} = 44.01 \text{ kpsi}$$

$$\sigma_2 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \frac{33.6135 \text{ kpsi}}{2} - \sqrt{\left(\frac{33.6135 \text{ kpsi}}{2}\right)^2 + (21.3904 \text{ kpsi})^2} = -10.3965 \text{ kpsi}$$

The principal shear stress is as follows:

$$\tau_{\max} = \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + (\tau_{xz})^2} = \sqrt{\left(\frac{33.6135 \text{ kpsi}}{2}\right)^2 + (21.3904 \text{ kpsi})^2} = 27.2033 \text{ kpsi}$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 8

TOPICS COVERED IN HOMEWORK 8

Chapter 4. Deflections: Problems 4-10, 4-11, 4-12, 4-14, 4-16

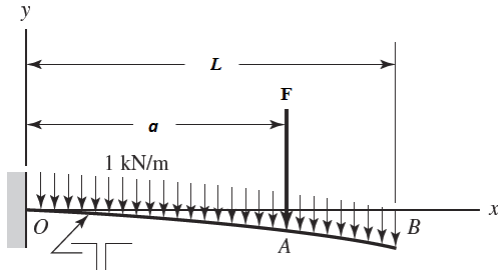
**MACHINE DESIGN
I FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

1. Award: 20.00 points

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size 100 mm × 100 mm × 12 mm mounted back to back. Using superposition, find the deflection at *B* and the maximum stress in the beam. Take the magnitude of force (*F*) to be 2.15 kN, *L* = 2.97 m, and *a* = 1.94 m.

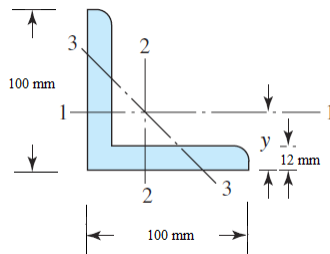


The deflection at *B* is $22.3 \pm 2\%$ mm in the downward direction.

The maximum magnitude of stress in the beam is $147 \pm 2\%$ MPa. The nature of the stress is compressive.

Explanation:

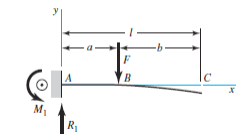
From Table A-6: Properties of Structural-Steel Equal Legs Angles, for each angle, $I_{1-1} = 207 \text{ cm}^4$. Thus, $I = 2(207 \text{ cm}^4) (10^4) = 4.14 \times 10^6 \text{ mm}^4$.



From Table A-9: Shear, Moment, and Deflection of Beams, use beam 2 with $F = 2150 \text{ N}$, $a = 2.97 \text{ m}$, and $l = 1.94 \text{ m}$; and use beam 3 with $w = 1 \text{ N/mm}$ and $l = 3000 \text{ mm}$.

The deflection at point *A* due to point load *F* can be denoted by y_1 .

2 Cantilever—intermediate load

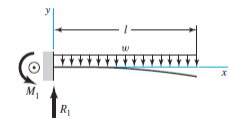


$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_1 = \frac{Fa^2}{6EI} (a - 3l)$$

The deflection at point *A* due to uniformly distributed load *w* can be denoted by y_2 .

3 Cantilever—uniform load



$$y_{\max} = -\frac{wl^4}{8EI}$$

$$y_2 = -\frac{wl^4}{8EI}$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l) - \frac{wl^4}{8EI}$$

$$y_{\max} = \frac{(2150 \text{ N})(1940 \text{ mm})^2}{6(207 \times 10^3 \text{ N/mm}^2)(4.14 \times 10^6 \text{ mm}^4)} [1940 \text{ mm} - 3(2970 \text{ mm})] - \frac{(1 \text{ N/mm})(2970 \text{ mm})^4}{8(207 \times 10^3 \text{ N/mm}^2)(4.14 \times 10^6 \text{ mm}^4)}$$

$$y_{\max} = -22.3 \text{ mm}$$

The deflection at *B* is 22.3 mm in the downward direction.

$$M_O = -Fa - \left(\frac{wl^2}{2} \right)$$

$$M_O = -(2150 \text{ N} \times 1940 \text{ mm}) - \left[\frac{(1 \text{ N/mm})(2970 \text{ mm})^2}{2} \right]$$

$$M_O = -8.5815 \times 10^6 \text{ N-mm}$$

From Table A-6: Properties of Structural-Steel Equal Legs Angles, the distance from centroid to upper surface is $y = 29$ mm. The distance from centroid to bottom surface is $y = 29.0 \text{ mm} - 100 \text{ mm} = -71$ mm. The maximum stress is compressive at the bottom of the beam at the wall. This stress is

$$\sigma_{\max} = -\frac{My}{I} = -\frac{(-8.5815 \times 10^6 \text{ N}\cdot\text{mm})(-71 \text{ mm})}{4.14 \times 10^6 \text{ mm}^4}$$

$$\sigma_{\max} = -147 \text{ MPa}$$

The maximum stress in the beam is 147 MPa and the stress is compressive in nature.

The deflection at B is 22.3 mm in the downward direction.

The maximum stress in the beam is 147 MPa. The nature of the stress is compressive.

References

Multi-Answer

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

Difficulty: Medium

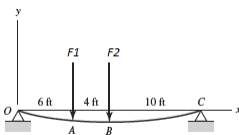
2.

Award: 20.00 points

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

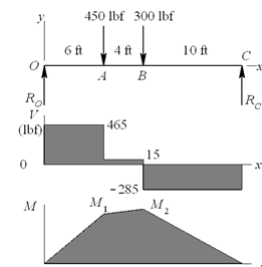
A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed $1/2$ in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is $5 \pm 2\%$ in long and its weight capacity is $6.7 \pm 2\%$ lbf/ft.

Explanation:



$$R_O = \frac{14 \text{ ft}}{20 \text{ ft}} (450 \text{ lbf}) + \frac{10 \text{ ft}}{20 \text{ ft}} (300 \text{ lbf}) = 465 \text{ lbf}$$

$$R_C = \frac{6 \text{ ft}}{20 \text{ ft}} (450 \text{ lbf}) + \frac{10 \text{ ft}}{20 \text{ ft}} (300 \text{ lbf}) = 285 \text{ lbf}$$

$$M_1 = 465 \text{ lbf} \times (6 \times 12) \text{ in} = 33.48 \times 10^3 \text{ lbf}\cdot\text{in}$$

$$M_2 = (33.48 \times 10^3 \text{ lbf}) + 15 \text{ lbf} \times (4 \times 12) \text{ in} = 34.20 \times 10^3 \text{ lbf}\cdot\text{in}$$

$$\sigma_{\max} = \frac{M_{\max}}{Z} \Rightarrow 15 = \frac{34.2 \text{ lbf}\cdot\text{in}}{Z} \quad Z = 2.28 \text{ in}^3$$

For deflections, use beams 5 and 6 of Table A-9:

$$y \Big|_{x=10\text{ft}} = \frac{F_1 a [l - (l/2)]}{6EI} \left[\left(\frac{l}{2} \right)^2 + a^2 - 2l \left(\frac{l}{2} \right) \right] - \frac{F_2 l^3}{48EI}$$

$$-0.5 \text{ in} = \frac{(450 \text{ lbf} \times 72 \text{ in} \times 120 \text{ in})}{6 \times 30 \times 10^6 \text{ psi} \times I \times (240 \text{ in})} (120^2 + 72^2 - 240^2) \text{ in}^2 - \frac{300 \text{ lbf} \times (240 \text{ in}^3)}{48 \times 30 \times 10^6 \text{ psi} \times I}$$

$$I = 12.60 \text{ in}^4 \Rightarrow I/2 = 6.30 \text{ in}^4$$

Select two 5 in-6.7 lbf/ft channels. From Table A-7, $I = 2 \times 7.49 \text{ in}^4 = 14.98 \text{ in}^4$ and $Z = 2 \times 3.00 \text{ in}^3 = 6.00 \text{ in}^3$.

$$y_{\text{midspan}} = \frac{12.60 \text{ in}}{14.98 \text{ in}} \left(-\frac{1}{2} \right) = -0.421 \text{ in}$$

$$\sigma_{\max} = \frac{34.2 \text{ kpsi}}{6.00 \text{ kpsi}} = 5.70 \text{ kpsi}$$

Therefore, this selection suits the required criteria.

The pair of channels that should be selected are 5 in long and its weight capacity is 6.7 lbf/ft.

References

Numeric Response

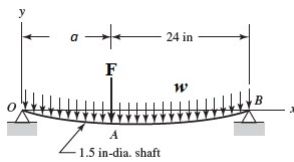
Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

Difficulty: Medium

3. Award: 20.00 points

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 340 lbf, $w = 120$ lbf/ft, and $a = 14$ in.



The magnitude of the deflection of the steel shaft at A is $0.0786 \pm 2\%$ inch in the downward direction.

The magnitude of the deflection at midspan is $0.0836 \pm 2\%$ inch in the downward direction.

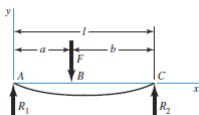
The percentage difference is $6.39 \pm 2\%$ %.

Explanation:

$$I = \frac{\pi}{64} (1.5 \text{ in})^4 = 0.2485 \text{ in}^4 \text{ for the circular cross section of diameter 1.5 in.}$$

From Table A-9: Shear, Moment, and Deflection of Beams, by superposition of beams 6 and 7, at $x = a = 14$ in, with $b = 24$ in and $l = 38$ in. The deflection at point A due to point load F can be denoted by y_1 .

6 Simple supports—intermediate load

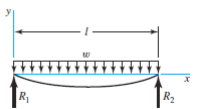


$$y_{AB} = \frac{Fbx}{6EI} [x^2 + b^2 - l^2]$$

$$y_1 = \frac{Fba}{6EI} [a^2 + b^2 - l^2]$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

7 Simple supports—uniform load



$$y = \frac{wx}{24EI} (2lx^2 - x^3 - l^3)$$

$$y_2 = \frac{wa}{24EI} (2la^2 - a^3 - l^3)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y = \frac{Fba}{6EI} [a^2 + b^2 - l^2] + \frac{wa}{24EI} (2la^2 - a^3 - l^3)$$

$$y_A = \frac{(340 \text{ lbf})(24 \text{ in})(14 \text{ in})}{6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(38 \text{ in})} [(14 \text{ in})^2 + (24 \text{ in})^2 - (38 \text{ in})^2] \\ + \frac{\left(\frac{120 \text{ lbf/ft}}{12 \text{ in/ft}}\right)(14 \text{ in})}{24(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)} [2(38 \text{ in})(14 \text{ in})^2 - (14 \text{ in})^3 - (38 \text{ in})^3]$$

$$y_A = -0.0786 \text{ in}$$

The deflection of the steel shaft at A is 0.0786 inch in the downward direction.

$$\text{At } x = \frac{l}{2} = 19 \text{ in,}$$

$$y = \frac{Fa[l - (l/2)]}{6EI} \left[\left(\frac{l}{2}\right)^2 + a^2 - 2l\left(\frac{l}{2}\right) \right] + \frac{w(l/2)}{24EI} \left[2l\left(\frac{l}{2}\right)^2 - \left(\frac{l}{2}\right)^3 - l^3 \right]$$

$$y = \frac{(340 \text{ lbf})(14 \text{ in})(19 \text{ in})}{6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(38 \text{ in})} [(19 \text{ in})^2 + (14 \text{ in})^2 - (38 \text{ in})^2] \\ + \frac{\left(\frac{120 \text{ lbf/ft}}{12 \text{ in/ft}}\right)(19 \text{ in})}{24(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)} [2(38 \text{ in})(19 \text{ in})^2 - (19 \text{ in})^3 - (38 \text{ in})^3] \\ = -0.0836 \text{ in.}$$

The deflection at mid-span is 0.0836 inch in the downward direction.

The percentage difference is obtained by dividing the difference of deflections with deflection at A.

$$\text{Percentage difference} = \frac{(-0.0836 \text{ in}) - (-0.0786 \text{ in})}{-0.0786 \text{ in}} \times 100 = 6.39\%$$

The percentage difference is 6.39%.

The magnitude of the deflection of the steel shaft at A is 0.0786 inch in the downward direction.

The magnitude of the deflection at midspan is 0.0836 inch in the downward direction.

The percentage difference is 6.39%.

References

Multi-Answer

Problem 04.012 - Determine the deflection of a shaft using superposition

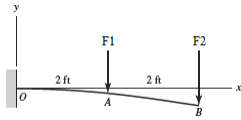
Difficulty: Medium

4.

Award: 20.00 points

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at *B*. Take the magnitude of force (*F*₁) to be 306 lbf and the magnitude of force (*F*₂) to be 209 lbf.



The deflection at *B* is inch in the direction.

Explanation:

For a tubular cross section with inner diameter *d*_i and outer diameter *d*_o, the area moment of inertia is given by

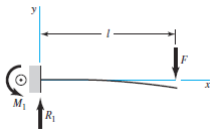
$$I = \frac{\pi}{64} (d_o^4 - d_i^4)$$

$$I = \frac{\pi}{64} [(2 \text{ in})^4 - (1.5 \text{ in})^4] = 0.5369 \text{ in}^4$$

From Table A-5: Physical Constants of Materials, *E* = 10.4 Mpsi.

From Table A-9: Shear, Moment, and Deflection of Beams, beams 1 and 2, by superposition, The deflection at point *A* due to point load *F*₂ can be denoted by *y*₁.

1 Cantilever—end load

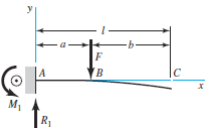


$$y_{\max} = -\frac{Fl^3}{3EI}$$

$$y_l = -\frac{Fl^3}{3EI}$$

The deflection at point *A* due to point load *F*₁ can be denoted by *y*₂.

2 Cantilever—intermediate load



$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_2 = \frac{Fa^2}{6EI} (a - 3l)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_B = -\frac{F_2 l^3}{3EI} + \frac{F_1 a^2}{6EI} (a - 3l) = \frac{-(209 \text{ lbf})(4 \text{ ft})(12 \text{ in/ft})^3}{3(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)} + \frac{(306 \text{ lbf})[(2 \text{ ft})(12 \text{ in/ft})]^2}{6(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)} [(2 \text{ ft})(12 \text{ in/ft}) - 3(4 \text{ ft})(12 \text{ in/ft})]$$

$$y_B = -2 \text{ in}$$

The deflection at *B* is 2 inch in the downward direction.

References

Multi-Answer

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

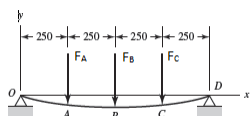
Difficulty: Easy

5.

Award: 20.00 points

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 2.02 mm, given that $F_A = 365$ N, $F_B = 460$ N, and $F_C = 365$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is $31.43 \pm 2\%$ mm.

Explanation:

$$I = \frac{\pi}{64} d^4$$

From Table A-5, $E = 207(10^3)$ MPa.

From Table A-9, beams 5 and 9, with $F_C = F_A = F$, by superposition,

$$y_B = -\frac{F_B l^3}{48EI} + \frac{Fa}{24EI} (4a^2 - 3l^2) \Rightarrow I = \frac{1}{48Ey_B} [-F_B l^3 + 2Fa(4a^2 - 3l^2)]$$

$$I = \frac{1}{48(207 \times 10^3 \text{ MPa})(-2.02 \text{ mm})} \left\{ -460 \text{ N} (1000 \text{ mm})^3 + 2(365) \text{ N} (250 \text{ mm}) [4(250 \text{ mm})^2 - 3(1000 \text{ mm})^2] \right\}$$

$$I = 47924.29 \text{ mm}^4$$

$$d = \sqrt[4]{\frac{64}{\pi} I} = \sqrt[4]{\frac{64}{\pi} (47924.29 \text{ mm}^4)} = 31.43 \text{ mm}$$

The minimum diameter of the steel shaft is 31.43 mm.

References

Numeric Response

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Difficulty: Medium

ABOVE AVERAGE

Dupre, Ty

Score: 100/100 Points 100 %

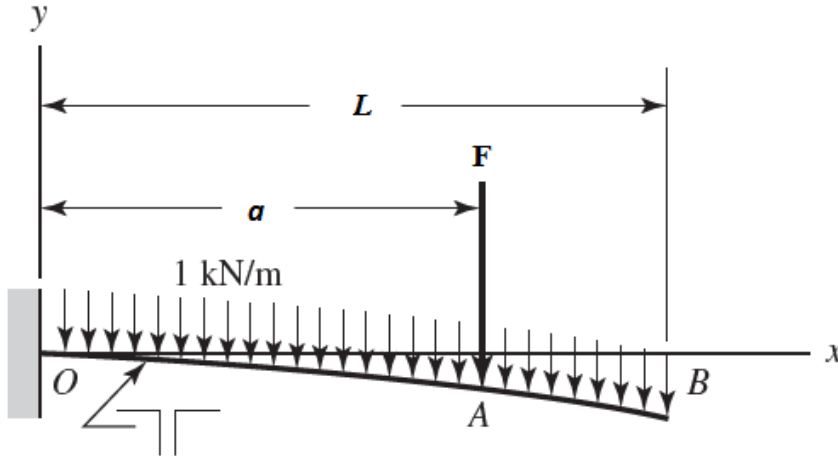
1.

Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size $100 \text{ mm} \times 100 \text{ mm} \times 12 \text{ mm}$ mounted back to back. Using superposition, find the deflection at B and the maximum stress in the beam. Take the magnitude of force (F) to be 2.15 kN , $L = 2.97 \text{ m}$, and $a = 2 \text{ m}$.



The deflection at B is **22.9** ✓ mm in the **downward** ✓ direction.

The maximum magnitude of stress in the beam is **149** ✓ MPa. The nature of the stress is **compressive** ✓.

References

Multi-Answer

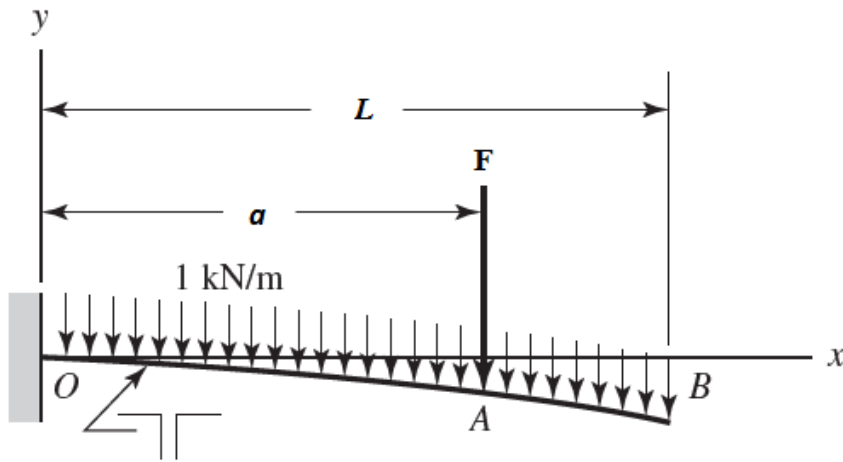
Chapter: 04 Deflection and Stiffness

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

Section: 04.05 Beam Deflections by Superposition

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size $100 \text{ mm} \times 100 \text{ mm} \times 12 \text{ mm}$ mounted back to back. Using superposition, find the deflection at B and the maximum stress in the beam. Take the magnitude of force (F) to be 2.15 kN , $L = 2.97 \text{ m}$, and $a = 2 \text{ m}$.

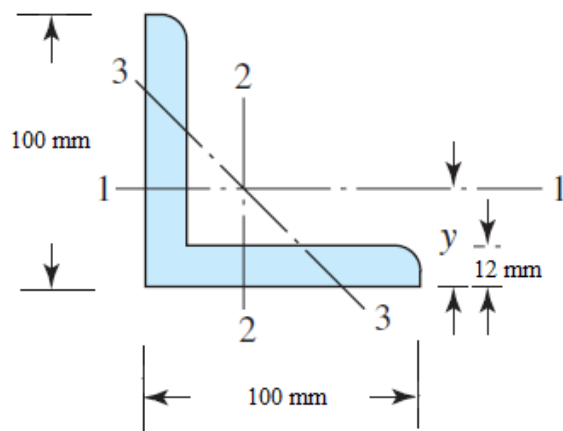


The deflection at B is $22.9 \pm 2\%$ mm in the downward direction.

The maximum magnitude of stress in the beam is $149 \pm 2\%$ MPa. The nature of the stress is compressive.

Explanation:

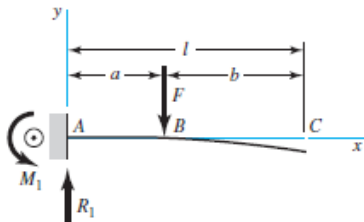
From Table A-6: Properties of Structural-Steel Equal Legs Angles, for each angle, $I_{1-1} = 207 \text{ cm}^4$. Thus, $I = 2(207 \text{ cm}^4)(10^4) = 4.14 \times 10^6 \text{ mm}^4$.



From Table A-9: Shear, Moment, and Deflection of Beams, use beam 2 with $F = 2150 \text{ N}$, $a = 2.97 \text{ mm}$, and $l = 2 \text{ mm}$; and use beam 3 with $w = 1 \text{ N/mm}$ and $l = 3000 \text{ mm}$.

The deflection at point A due to point load F can be denoted by y_1 .

2 Cantilever—intermediate load

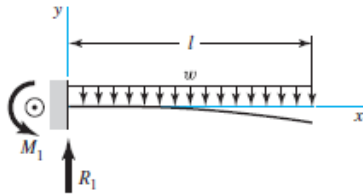


$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_1 = \frac{Fa^2}{6EI} (a - 3l)$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

3 Cantilever—uniform load



$$y_{\max} = -\frac{wl^4}{8EI}$$

$$y_2 = -\frac{wl^4}{8EI}$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_{\max} = \frac{Fa^2}{6EI} \left(a - 3l \right) - \frac{wl^4}{8EI}$$

$$y_{\max} = \frac{(2150 \text{ N})(2000 \text{ mm})^2}{6(207 \times 10^3 \text{ N/mm}^2)(4.14 \times 10^6 \text{ mm}^4)} [2000 \text{ mm} - 3(2970 \text{ mm})] - \frac{(1 \text{ N/mm})(2970 \text{ mm})^4}{8(207 \times 10^3 \text{ N/mm}^2)(4.14 \times 10^6 \text{ mm}^4)}$$

$$y_{\max} = -22.9 \text{ mm}$$

The deflection at B is 22.9 mm in the downward direction.

$$M_O = -Fa - \left(\frac{wl^2}{2} \right)$$

$$M_o = -(2150 \text{ N} \times 2000 \text{ mm}) - \left[\frac{(1 \text{ N/mm})(2970 \text{ mm})^2}{2} \right]$$

$$M_o = -8.7104 \times 10^6 \text{ N}\cdot\text{mm}$$

From Table A-6: Properties of Structural-Steel Equal Legs Angles, the distance from centroid to upper surface is $y = 29 \text{ mm}$. The distance from centroid to bottom surface is $y = 29.0 \text{ mm} - 100 \text{ mm} = -71 \text{ mm}$. The maximum stress is compressive at the bottom of the beam at the wall. This stress is

$$\sigma_{\max} = -\frac{My}{I} = -\frac{(-8.7104 \times 10^6 \text{ N}\cdot\text{mm})(-71 \text{ mm})}{4.14 \times 10^6 \text{ mm}^4}$$

$$\sigma_{\max} = -149 \text{ MPa}$$

The maximum stress in the beam is 149 MPa and the stress is compressive in nature.

2.

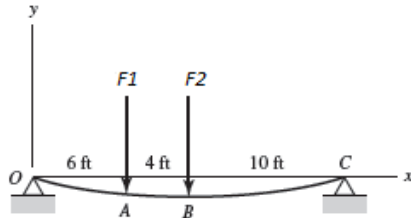
Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed 1/2 in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is **5** in long and its weight capacity is **6.7** lbf/ft.

References

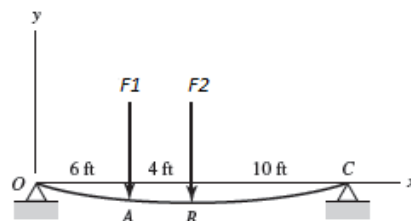
Numeric Response Chapter: 04 Deflection and Stiffness

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints Section: 04.05 Beam Deflections by Superposition

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

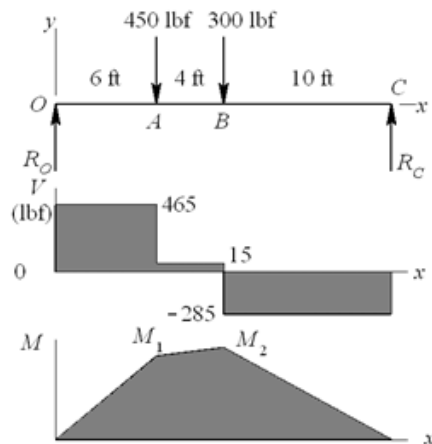
A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed 1/2 in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is **5 ± 2%** in long and its weight capacity is **6.7 ± 2%** lbf/ft.

Explanation:



$$R_O = \frac{14 \text{ ft}}{20 \text{ ft}} (450 \text{ mm}) + \frac{10 \text{ ft}}{20 \text{ ft}} (300 \text{ mm}) = 465 \text{ lbf}$$

$$R_C = \frac{6 \text{ ft}}{20 \text{ ft}} (450 \text{ mm}) + \frac{10 \text{ ft}}{20 \text{ ft}} (300 \text{ mm}) = 285 \text{ lbf}$$

$$M_1 = 465 \text{ lbf} \times (6 \times 12) \text{ in} = 33.48 \times 10^3 \text{ lbf}\cdot\text{in}$$

$$M_2 = (33.48 \times 10^3 \text{ lbf}) + 15 \text{ lbf} \times (4 \times 12) \text{ in} = 34.20 \times 10^3 \text{ lbf}\cdot\text{in}$$

$$\sigma_{\max} = \frac{M_{\max}}{Z} \Rightarrow 15 = \frac{34.2 \text{ lbf}\cdot\text{in}}{Z} \quad Z = 2.28 \text{ in}^3$$

For deflections, use beams 5 and 6 of Table A-9:

$$y \Big|_{x=10\text{ft}} = \frac{F_1 a [l - (l/2)]}{6EI} \left[\left(\frac{l}{2} \right)^2 + a^2 - 2l \frac{l}{2} \right] - \frac{F_2 l^3}{48EI}$$

$$-0.5 \text{ in} = \frac{(450 \text{ lbf} \times 72 \text{ in} \times 120 \text{ in})}{6 \times 30 \times 10^6 \text{ psi} \times I \times (240 \text{ in})} \left(120^2 + 72^2 - 240^2 \right) \text{ in}^2 - \frac{300 \text{ lbf} \times (240 \text{ in}^3)}{48 \times 30 \times 10^6 \text{ psi} \times I}$$

$$I = 12.60 \text{ in}^4 \Rightarrow I/2 = 6.30 \text{ in}^4$$

Select two 5 in-6.7 lbf/ft channels. From Table A-7, $I = 2 \times 7.49 \text{ in}^4 = 14.98 \text{ in}^4$ and $Z = 2 \times 3.00 \text{ in}^3 = 6.00 \text{ in}^3$.

$$y_{\text{midspan}} = \frac{12.60 \text{ in}}{14.98 \text{ in}} \left(-\frac{1}{2} \right) = -0.421 \text{ in}$$

$$\sigma_{\max} = \frac{34.2 \text{ kpsi}}{6.00 \text{ kpsi}} = 5.70 \text{ kpsi}$$

Therefore, this selection suits the required criteria.

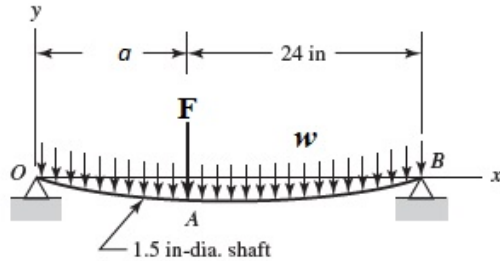
3.

Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 365 lbf, $w = 145$ lbf/ft, and $a = 12$ in.



The magnitude of the deflection of the steel shaft at A is **0.0684** ✓ inch in the **downward** ✓ direction.
 The magnitude of the deflection at midspan is **0.076** ✓ inch in the **downward** ✓ direction.
 The percentage difference is **11.07** ✓ %.

References

Multi-Answer

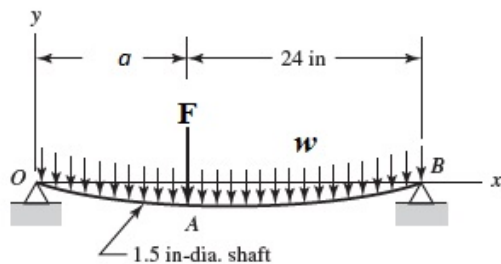
Chapter: 04 Deflection and Stiffness

Problem 04.012 - Determine the deflection of a shaft using superposition

Section: 04.05 Beam Deflections by Superposition

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 365 lbf, $w = 145$ lbf/ft, and $a = 12$ in.



The magnitude of the deflection of the steel shaft at A is **0.0684 ± 2%** ✓ inch in the downward ✓ direction.
 The magnitude of the deflection at midspan is **0.076 ± 2%** ✓ inch in the downward ✓ direction.
 The percentage difference is **11.07 ± 2%** %.

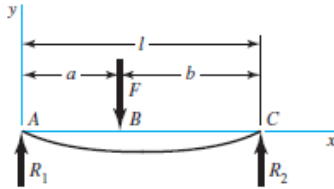
Explanation:

$$I = \frac{\pi}{64}(1.5 \text{ in})^4 = 0.2485 \text{ in}^4 \text{ for the circular cross section of diameter 1.5 in.}$$

From Table A-9: Shear, Moment, and Deflection of Beams, by superposition of beams 6 and 7, at $x = a = 12 \text{ in}$, with $b = 24 \text{ in}$ and $l = 36 \text{ in}$,

The deflection at point A due to point load F can be denoted by y_1 .

6 Simple supports—intermediate load

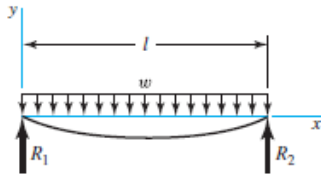


$$y_{AB} = \frac{Fbx}{6EI} [x^2 + b^2 - l^2]$$

$$y_1 = \frac{Fba}{6EI} [a^2 + b^2 - l^2]$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

7 Simple supports—uniform load



$$y = \frac{wx}{24EI} (2lx^2 - x^3 - l^3)$$

$$y_2 = \frac{wa}{24EI} (2la^2 - a^3 - l^3)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y = \frac{Fba}{6EI} [a^2 + b^2 - l^2] + \frac{wa}{24EI} (2la^2 - a^3 - l^3)$$

$$y_A = \frac{(365 \text{ lbf})(24 \text{ in})(12 \text{ in})}{6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(36 \text{ in})} [(12 \text{ in})^2 + (24 \text{ in})^2 - (36 \text{ in})^2] \\ + \frac{\left(\frac{150 \text{ lbf/ft}}{12 \text{ in/ft}}\right)(12 \text{ in})}{(24 \text{ in})(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)} [2(36 \text{ in})(12 \text{ in})^2 - (12 \text{ in})^3 - (36 \text{ in})^3]$$

$$y_A = -0.0684 \text{ in}$$

The deflection of the steel shaft at A is 0.0684 inch in the downward direction.

$$\text{At } x = \frac{l}{2} = 18 \text{ in,}$$

$$y = \frac{Fa[l - (l/2)]}{6EI} \left[\left(\frac{l}{2}\right)^2 + a^2 - 2l\frac{l}{2} \right] + \frac{w(l/2)}{24EI} \left[2l\left(\frac{l}{2}\right)^2 - \left(\frac{l}{2}\right)^3 - l^3 \right]$$

$$y = \frac{(365 \text{ lbf})(12 \text{ in})(18 \text{ in})}{6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(36 \text{ in})} [(18 \text{ in})^2 + (12 \text{ in})^2 - (36 \text{ in})^2]$$

$$+ \frac{\left(\frac{150 \text{ lbf/ft}}{12 \text{ in/ft}}\right)(18 \text{ in})}{24(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)} [2(36 \text{ in})(18 \text{ in})^2 - (18 \text{ in})^3 - (36 \text{ in})^3]$$

$$= -0.076 \text{ in.}$$

The deflection at mid-span is 0.076 inch in the downward direction.

The percentage difference is obtained by dividing the difference of deflections with deflection at A.

$$\text{Percentage difference} = \frac{(-0.076 \text{ in}) - (-0.0684 \text{ in})}{-0.0684 \text{ in}} \times 100 = 11.07\%$$

The percentage difference is 11.07%.

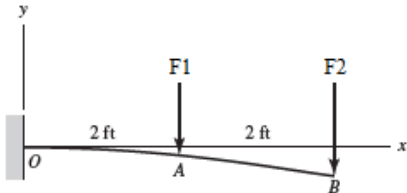
4.

Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at B . Take the magnitude of force (F_1) to be 302 lbf and the magnitude of force (F_2) to be 205 lbf.



The deflection at B is 1.97 ✓ inch in the downward ✓ direction.

References

Multi-Answer

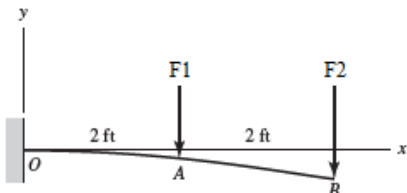
Chapter: 04 Deflection and Stiffness

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

Section: 04.05 Beam Deflections by Superposition

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at B . Take the magnitude of force (F_1) to be 302 lbf and the magnitude of force (F_2) to be 205 lbf.



The deflection at B is 1.97 ± 2% inch in the downward ✓ direction.

Explanation:

For a tubular cross section with inner diameter d_i and outer diameter d_o , the area moment of inertia is given by

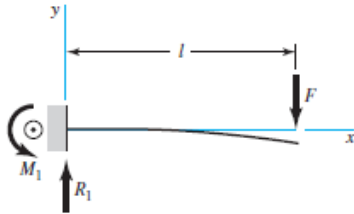
$$I = \frac{\pi}{64} (d_o^4 - d_i^4)$$

$$I = \frac{\pi}{64} [(2 \text{ in})^4 - (1.5 \text{ in})^4] = 0.5369 \text{ in}^4$$

From Table A-5: Physical Constants of Materials, $E = 10.4 \text{ Mpsi}$.

From Table A-9: Shear, Moment, and Deflection of Beams, beams 1 and 2, by superposition, The deflection at point A due to point load F_2 can be denoted by y_1 .

1 Cantilever—end load

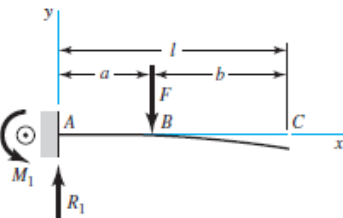


$$y_{\max} = -\frac{Fl^3}{3EI}$$

$$y_1 = -\frac{F_B l^3}{3EI}$$

The deflection at point A due to point load F_1 can be denoted by y_2 .

2 Cantilever—intermediate load



$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_2 = \frac{F_A a^2}{6EI} (a - 3l)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_B = -\frac{F_B l^3}{3EI} + \frac{F_A a^2}{6EI} (a - 3l) = \frac{-(205 \text{ lbf})[(4 \text{ ft})(12 \text{ in/ft})]^3}{3(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)} + \frac{(302 \text{ lbf})[(2 \text{ ft})(12 \text{ in./ft})]^2}{6(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)} [(2 \text{ ft})(12 \text{ in/ft}) - 3(4 \text{ ft})(12 \text{ in/ft})]$$

$$y_B = -1.97 \text{ in}$$

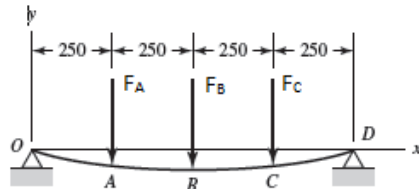
5.

Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 1.97 mm, given that $F_A = 350$ N, $F_B = 450$ N, and $F_C = 350$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is **31.38** mm.

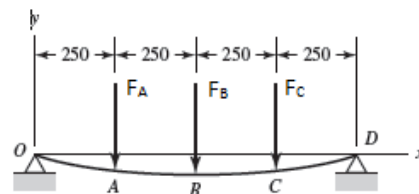
References

Numeric Response Chapter: 04 Deflection and Stiffness

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft Section: 04.05 Beam Deflections by Superposition

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 1.97 mm, given that $F_A = 350$ N, $F_B = 450$ N, and $F_C = 350$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is **31.38 ± 2%** mm.

Explanation:

$$I = \frac{\pi}{64} d^4$$

From Table A-5, $E = 207(10^3)$ MPa.

From Table A-9, beams 5 and 9, with $F_C = F_A = F$, by superposition,

$$y_B = -\frac{F_B l^3}{48EI} + \frac{Fa}{24EI} (4a^2 - 3l^2) \Rightarrow I = \frac{1}{48Ey_B} [-F_B l^3 + 2Fa(4a^2 - 3l^2)]$$

$$I = \frac{1}{48(207 \times 10^3 \text{ MPa})(-1.97 \text{ mm})} \left\{ -450 \text{ N} (1000 \text{ mm})^3 + 2(350) \text{ N} (250 \text{ mm}) [4(250 \text{ mm})^2 - 3(1000 \text{ mm})^2] \right\}$$

$$I = 47576.06 \text{ mm}^4$$

$$d = \sqrt[4]{\frac{64}{\pi} I} = \sqrt[4]{\frac{64}{\pi} (47576.06 \text{ mm}^4)} = 31.38 \text{ mm}$$

AVERAGE

Valentine, Tyler

Score: 91/100 Points 91 %

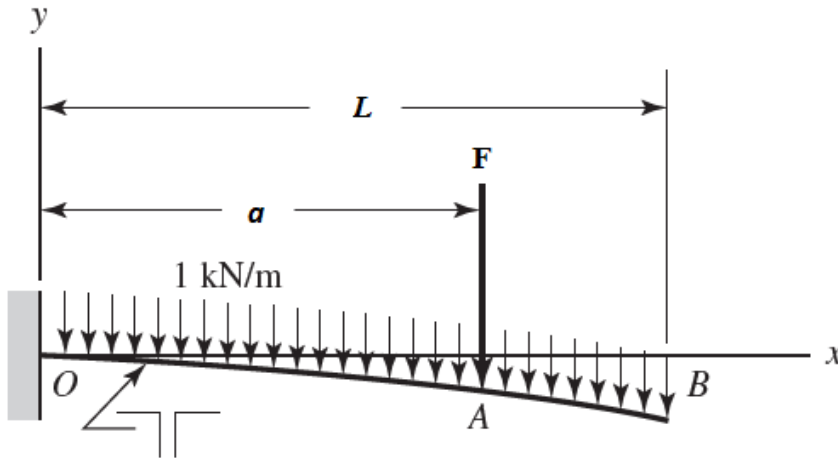
1.

Award: 15 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size $100 \text{ mm} \times 100 \text{ mm} \times 12 \text{ mm}$ mounted back to back. Using superposition, find the deflection at B and the maximum stress in the beam. Take the magnitude of force (F) to be 2.2 kN , $L = 2.94 \text{ m}$, and $a = 1.93 \text{ m}$.



The deflection at B is **21.9** ✓ mm in the **downward** ✓ direction.

The maximum magnitude of stress in the beam is **15** ✖ MPa. The nature of the stress is **compressive** ✓.

References

Multi-Answer

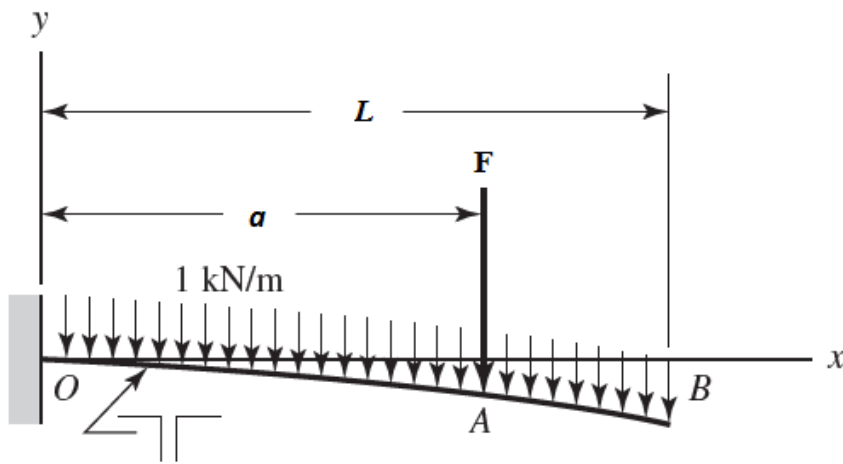
Chapter: 04 Deflection and Stiffness

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

Section: 04.05 Beam Deflections by Superposition

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size $100 \text{ mm} \times 100 \text{ mm} \times 12 \text{ mm}$ mounted back to back. Using superposition, find the deflection at B and the maximum stress in the beam. Take the magnitude of force (F) to be 2.2 kN , $L = 2.94 \text{ m}$, and $a = 1.93 \text{ m}$.

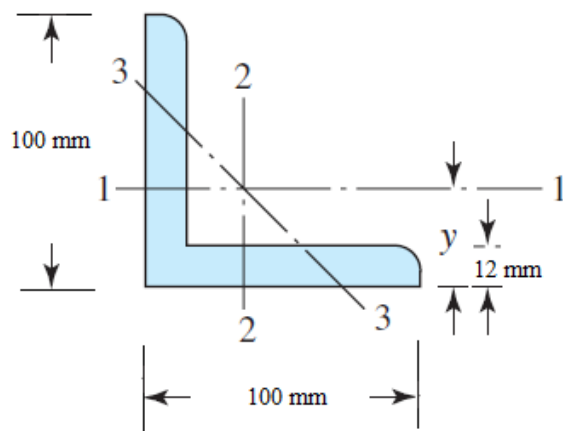


The deflection at B is $21.9 \pm 2\%$ mm in the downward direction. ✓

The maximum magnitude of stress in the beam is $147 \pm 2\%$ MPa. The nature of the stress is compressive. ✓

Explanation:

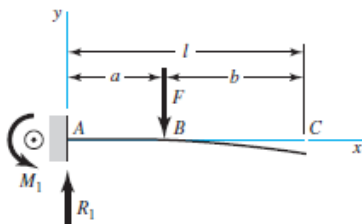
From Table A-6: Properties of Structural-Steel Equal Legs Angles, for each angle, $I_{1-1} = 207 \text{ cm}^4$. Thus, $I = 2(207 \text{ cm}^4)(10^4) = 4.14 \times 10^6 \text{ mm}^4$.



From Table A-9: Shear, Moment, and Deflection of Beams, use beam 2 with $F = 2200 \text{ N}$, $a = 2.94 \text{ mm}$, and $l = 1.93 \text{ mm}$; and use beam 3 with $w = 1 \text{ N/mm}$ and $l = 3000 \text{ mm}$.

The deflection at point A due to point load F can be denoted by y_1 .

2 Cantilever—intermediate load

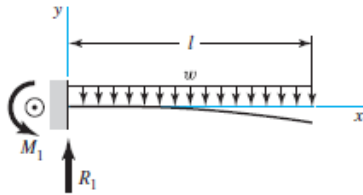


$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_1 = \frac{Fa^2}{6EI} (a - 3l)$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

3 Cantilever—uniform load



$$y_{\max} = -wl^4/8EI$$

$$y_2 = -wl^4/8EI$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_{\max} = Fa^2/6EI \left(a - 3l \right) - wl^4/8EI$$

$$y_{\max} = (2200 \text{ N}) (1930 \text{ mm})^2 / (207 \times 10^3 \text{ N/mm}^2) (4.14 \times 10^6 \text{ mm}^4) [1930 \text{ mm} - 3 (2940 \text{ mm})] - (1 \text{ N/mm}) (2940 \text{ mm})^4 / (8 (207 \times 10^3 \text{ N/mm}^2))$$

$$y_{\max} = -21.9 \text{ mm}$$

The deflection at B is 21.9 mm in the downward direction.

$$M_O = -Fa - wl^2/2$$

$$M_O = - (2200 \text{ N} \times 1930 \text{ mm}) - \left[(1 \text{ N/mm}) (2940 \text{ mm})^2 / 2 \right]$$

$$M_O = -8.5678 \times 10^6 \text{ N}\cdot\text{mm}$$

From Table A-6: Properties of Structural-Steel Equal Legs Angles, the distance from centroid to upper surface is $y = 29 \text{ mm}$. The distance from centroid to bottom surface is $y = 29.0 \text{ mm} - 100 \text{ mm} = -71 \text{ mm}$. The maximum stress is compressive at the bottom of the beam at the wall. This stress is

$$\sigma_{\max} = -My/I = -(-8.5678 \times 10^6 \text{ N}\cdot\text{mm}) (-71 \text{ mm}) / (4.14 \times 10^6 \text{ mm}^4)$$

$$\sigma_{\max} = -147 \text{ MPa}$$

The maximum stress in the beam is 147 MPa and the stress is compressive in nature.

2.

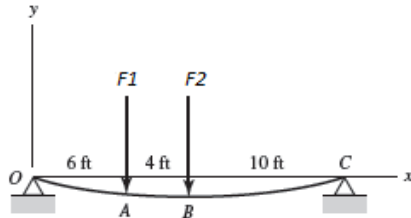
Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed 1/2 in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is **5** in long and its weight capacity is **6.7** lbf/ft.

References

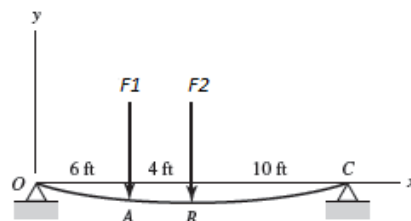
Numeric Response Chapter: 04 Deflection and Stiffness

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints Section: 04.05 Beam Deflections by Superposition

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

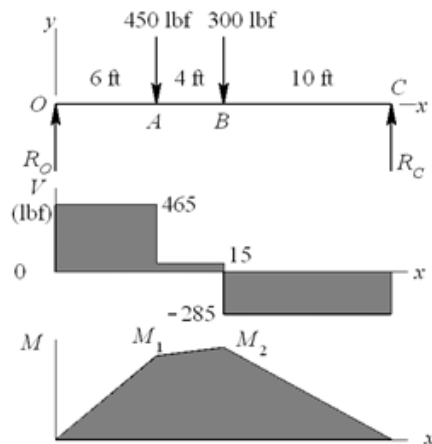
A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed 1/2 in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is **5 ± 2%** in long and its weight capacity is **6.7 ± 2%** lbf/ft.

Explanation:



$$R_O = 14 \text{ ft} \times 20 \text{ ft} \left(450 \text{ mm} \right) + 10 \text{ ft} \times 20 \text{ ft} \left(300 \text{ mm} \right) = 465 \text{ lbf}$$

$$R_C = 6 \text{ ft} \times 20 \text{ ft} \left(450 \text{ mm} \right) + 10 \text{ ft} \times 20 \text{ ft} \left(300 \text{ mm} \right) = 285 \text{ lbf}$$

$$M_1 = 465 \text{ lbf} \times (6 \times 12) \text{ in} = 33.48 \times 10^3 \text{ lbf} \cdot \text{in}$$

$$M_2 = (33.48 \times 10^3 \text{ lbf}) + 15 \text{ lbf} \times (4 \times 12) \text{ in} = 34.20 \times 10^3 \text{ lbf} \cdot \text{in}$$

$$\sigma_{\max} = M_{\max} Z \Rightarrow 15 = 34.2 \text{ lbf} \cdot \text{in} Z \quad Z = 2.28 \text{ in}^3$$

For deflections, use beams 5 and 6 of Table A-9:

$$y|_{x=10\text{ft}} = F_1 a [l - (l/2)]^2 / 6EI + a^2 - 2l^2 - F_2 l^3 / 48EI$$

$$-0.5 \text{ in} = (450 \text{ lbf} \times 72 \text{ in} \times 120 \text{ in}) / 6 \times 30 \times 10^6 \text{ psi} \times l \times \left(240 \text{ in} \left(120^2 + 72^2 - 240^2 \right) \right) \text{ in}^2 - 300 \text{ lbf} \times \left(240 \text{ in}^3 \right) / 48 \times 30 \times 10^6 \text{ psi} \times l$$

$$l = 12.60 \text{ in}^4 \Rightarrow l/2 = 6.30 \text{ in}^4$$

Select two 5 in-6.7 lbf/ft channels. From Table A-7, $I = 2 \times 7.49 \text{ in}^4 = 14.98 \text{ in}^4$ and $Z = 2 \times 3.00 \text{ in}^3 = 6.00 \text{ in}^3$.

$$y_{\text{midspan}} = 12.60 \text{ in} / 14.98 \text{ in} \left(-12 \right) = -0.421 \text{ in}$$

$$\sigma_{\max} = 34.2 \text{ kpsi} / 6.00 \text{ kpsi} = 5.70 \text{ kpsi}$$

Therefore, this selection suits the required criteria.

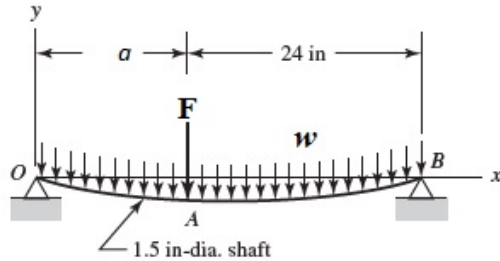
3.

Award: 16 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 340 lbf, $w = 115$ lbf/ft, and $a = 14$ in.



The magnitude of the deflection of the steel shaft at A is **0.07** ✖ inch in the **downward** ✔ direction.
 The magnitude of the deflection at midspan is **0.082** ✔ inch in the **downward** ✔ direction.
 The percentage difference is **6.34** ✔ %.

References

Multi-Answer

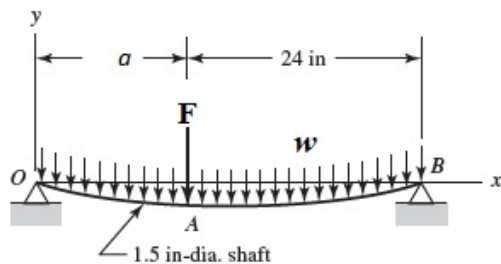
Chapter: 04 Deflection and Stiffness

Problem 04.012 - Determine the deflection of a shaft using superposition

Section: 04.05 Beam Deflections by Superposition

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 340 lbf, $w = 115$ lbf/ft, and $a = 14$ in.



The magnitude of the deflection of the steel shaft at A is **0.0772 ± 2%** inch in the downward ✔ direction.
 The magnitude of the deflection at midspan is **0.0821 ± 2%** inch in the downward ✔ direction.
 The percentage difference is **6.34 ± 2%** %.

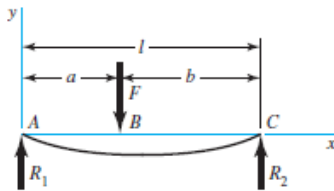
Explanation:

$I = \pi 64 (1.5 \text{ in})^4 = 0.2485 \text{ in}^4$ for the circular cross section of diameter 1.5 in.

From Table A-9: Shear, Moment, and Deflection of Beams, by superposition of beams 6 and 7, at $x = a = 14 \text{ in}$, with $b = 24 \text{ in}$ and $l = 38 \text{ in}$,

The deflection at point A due to point load F can be denoted by y_1 .

6 Simple supports—intermediate load

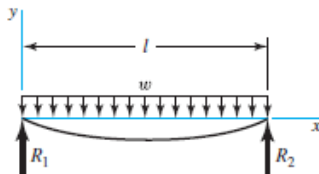


$$y_{AB} = Fbx6EI[lx^2 + b^2 - l^2]$$

$$y_1 = Fba6EI[a^2 + b^2 - l^2]$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

7 Simple supports—uniform load



$$y = wx24E\left(2lx^2 - x^3 - l^3\right)$$

$$y_2 = wa24E\left(2la^2 - a^3 - l^3\right)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y = Fba6EI[a^2 + b^2 - l^2] + wa24E\left(2la^2 - a^3 - l^3\right)$$

$$y_A = (340 \text{ lbf})(24 \text{ in})(14 \text{ in})6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(38 \text{ in})[(14 \text{ in})^2 + (24 \text{ in})^2 - (38 \text{ in})^2]$$

$$+ \left(150 \text{ lbf/ft} \cdot 12 \text{ in/ft}\right)(14 \text{ in})(24 \text{ in})(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)[2(38 \text{ in})(14 \text{ in})^2 - (14 \text{ in})^3 - (38 \text{ in})^3]$$

$$y_A = -0.0772 \text{ in}$$

The deflection of the steel shaft at A is 0.0772 inch in the downward direction.

At $x = l/2 = 19 \text{ in}$,

$$y = Fa[l-(l/2)]6EI\left(\left(\frac{l}{2}\right)^2 + a^2 - 2la\right] + w(l/2)24EI\left[2\left(\frac{l}{2}\right)^2 - \left(\frac{l}{2}\right)^3 - l^3\right]$$

$$y = (340 \text{ lbf})(14 \text{ in})(19 \text{ in})6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(38 \text{ in})[(19 \text{ in})^2 + (14 \text{ in})^2 - (38 \text{ in})^2]$$

$$+ \left(150 \text{ lbf/ft} \times 12 \text{ in/ft}\right)(19 \text{ in})24(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)[2(38 \text{ in})(19 \text{ in})^2 - (19 \text{ in})^3 - (38 \text{ in})^3]$$

$$= -0.0821 \text{ in.}$$

The deflection at mid-span is 0.0821 inch in the downward direction.

The percentage difference is obtained by dividing the difference of deflections with deflection at A.

$$\text{Percentage difference} = (-0.0821 \text{ in}) - (-0.0772 \text{ in}) - 0.0772 \text{ in} \times 100 = 6.34\%$$

The percentage difference is 6.34%.

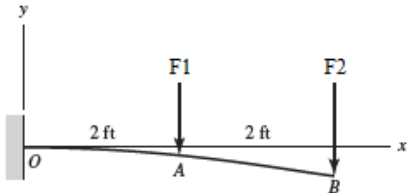
4.

Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at B . Take the magnitude of force (F_1) to be 309 lbf and the magnitude of force (F_2) to be 202 lbf.



The deflection at B is **1.95** inch in the **downward** direction.

References

Multi-Answer

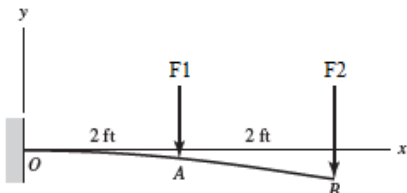
Chapter: 04 Deflection and Stiffness

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

Section: 04.05 Beam Deflections by Superposition

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at B . Take the magnitude of force (F_1) to be 309 lbf and the magnitude of force (F_2) to be 202 lbf.



The deflection at B is **1.95 ± 2%** inch in the downward direction.

Explanation:

For a tubular cross section with inner diameter d_i and outer diameter d_o , the area moment of inertia is given by

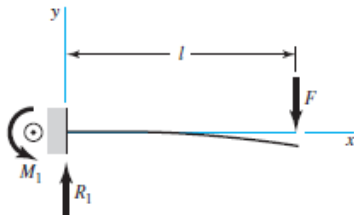
$$I = \pi 64 \left(d_{4o}^4 - d_{4i}^4 \right)$$

$$I = \pi 64 [(2 \text{ in})^4 - (1.5 \text{ in})^4] = 0.5369 \text{ in}^4$$

From Table A-5: Physical Constants of Materials, $E = 10.4 \text{ Mpsi}$.

From Table A-9: Shear, Moment, and Deflection of Beams, beams 1 and 2, by superposition, The deflection at point A due to point load F_2 can be denoted by y_1 .

1 Cantilever—end load

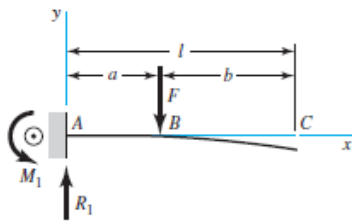


$$y_{\max} = -Fl^3/3EI$$

$$y_1 = -F_B l^3/3EI$$

The deflection at point A due to point load F_1 can be denoted by y_2 .

2 Cantilever—intermediate load



$$y_{\max} = Fa^2/6EI (a - 3l)$$

$$y_2 = F_A a^2/6EI (a - 3l)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_B = -F_B l^3/3EI + F_A a^2/6EI (a - 3l) = -(202 \text{ lbf})[(4 \text{ ft})(12 \text{ in./ft})]^3/(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4) + (309 \text{ lbf})[(2 \text{ ft})(12 \text{ in./ft})]^2/(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)$$

$$y_B = -1.95 \text{ in}$$

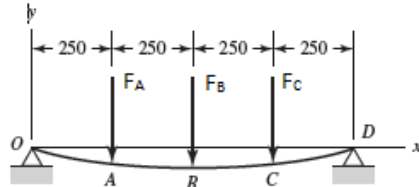
5.

Award: 20 out of 20.00 points

✖ You did not receive full credit for this question in a previous attempt

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 2.04 mm, given that $F_A = 395$ N, $F_B = 450$ N, and $F_C = 395$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is **31.6** mm.

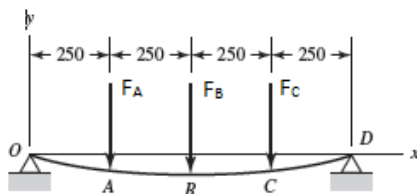
References

Numeric Response Chapter: 04 Deflection and Stiffness

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft Section: 04.05 Beam Deflections by Superposition

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 2.04 mm, given that $F_A = 395$ N, $F_B = 450$ N, and $F_C = 395$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is **31.61 ± 2%** mm.

Explanation:

$$I = \pi 64 d^4$$

From Table A-5, $E = 207(10^3)$ MPa.

From Table A-9, beams 5 and 9, with $F_C = F_A = F$, by superposition,

$$y_B = -\frac{F_B l^3}{48EI} + \frac{Fa^2}{24EI} (4a^2 - 3l^2) \Rightarrow I = 148Ey_B[-F_B l^3 + 2Fa(4a^2 - 3l^2)]$$

$$I = 148(207 \times 10^3 \text{ MPa})(-2.04 \text{ mm}) \left\{ -450 \text{ N} (1000 \text{ mm})^3 + 2(395 \text{ N}) \left[4(250 \text{ mm})^2 - 3(1000 \text{ mm})^2 \right] \right\}$$

$$I = 48996.17 \text{ mm}^4$$

$$d = \sqrt[4]{64\pi I} = \sqrt[4]{64\pi (48996.17 \text{ mm}^4)} = 31.61 \text{ mm}$$

BELOW AVERAGE

Carter, Nehemiah

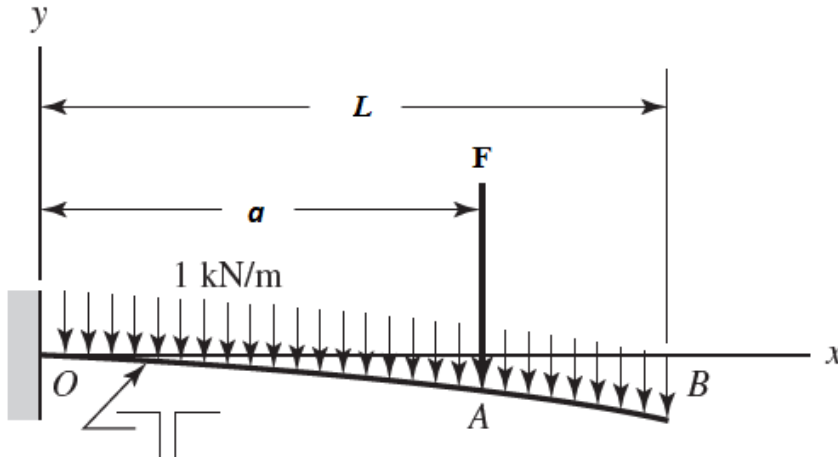
Score: 70/100 Points 70 %

1.

Award: 20 out of 20.00 points

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size $100\text{ mm} \times 100\text{ mm} \times 12\text{ mm}$ mounted back to back. Using superposition, find the deflection at B and the maximum stress in the beam. Take the magnitude of force (F) to be 2.45 kN , $L = 2.92\text{ m}$, and $a = 1.99\text{ m}$.



The deflection at B is **23.4** ✓ mm in the **downward** ✓ direction.

The maximum magnitude of stress in the beam is **157** ✓ MPa. The nature of the stress is **compressive** ✓.

References

Multi-Answer

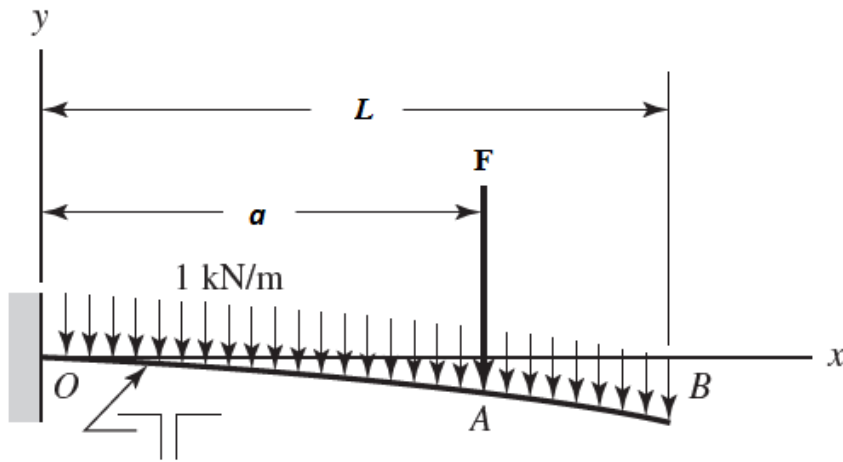
Chapter: 04 Deflection and Stiffness

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

Section: 04.05 Beam Deflections by Superposition

Problem 04.010 - Deflection and stress of a cantilevered beam using superposition

The figure shows a cantilever consisting of steel angles size $100\text{ mm} \times 100\text{ mm} \times 12\text{ mm}$ mounted back to back. Using superposition, find the deflection at B and the maximum stress in the beam. Take the magnitude of force (F) to be 2.45 kN , $L = 2.92\text{ m}$, and $a = 1.99\text{ m}$.

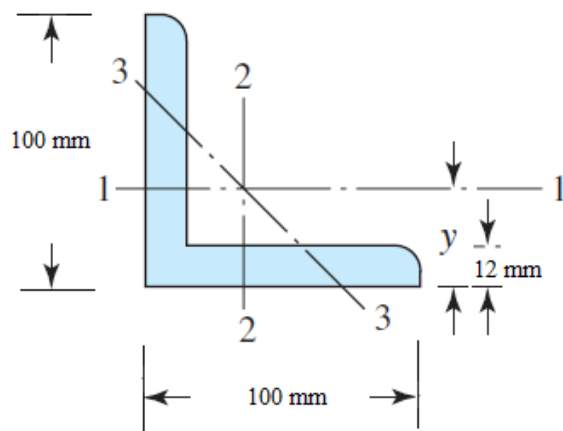


The deflection at B is $23.4 \pm 2\%$ mm in the downward direction.

The maximum magnitude of stress in the beam is $157 \pm 2\%$ MPa. The nature of the stress is compressive.

Explanation:

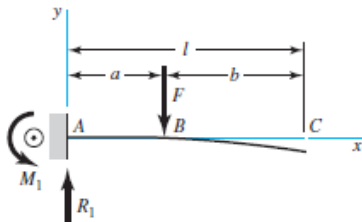
From Table A-6: Properties of Structural-Steel Equal Legs Angles, for each angle, $I_{1-1} = 207 \text{ cm}^4$. Thus, $I = 2(207 \text{ cm}^4)(10^4) = 4.14 \times 10^6 \text{ mm}^4$.



From Table A-9: Shear, Moment, and Deflection of Beams, use beam 2 with $F = 2450 \text{ N}$, $a = 2.92 \text{ mm}$, and $l = 1.99 \text{ mm}$; and use beam 3 with $w = 1 \text{ N/mm}$ and $l = 3000 \text{ mm}$.

The deflection at point A due to point load F can be denoted by y_1 .

2 Cantilever—intermediate load

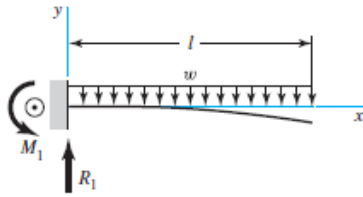


$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_1 = \frac{Fa^2}{6EI} (a - 3l)$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

3 Cantilever—uniform load



$$y_{\max} = -\frac{wl^4}{8EI}$$

$$y_2 = -\frac{wl^4}{8EI}$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_{\max} = \frac{Fa^2}{6EI} \left(a - 3l \right) - \frac{wl^4}{8EI}$$

$$y_{\max} = \frac{(2450 \text{ N}) (1990 \text{ mm})^2}{6(207 \times 10^3 \text{ N/mm}^2) (4.14 \times 10^6 \text{ mm}^4)} [1990 \text{ mm} - 3 (2920 \text{ mm})] - \frac{(1 \text{ N/mm}) (2920 \text{ mm})^4}{8(207 \times 10^3 \text{ N/mm}^2) (4.14 \times 10^6 \text{ mm}^4)}$$

$$y_{\max} = -23.4 \text{ mm}$$

The deflection at B is 23.4 mm in the downward direction.

$$M_O = -Fa - \left(\frac{wl^2}{2} \right)$$

$$M_O = - (2450 \text{ N} \times 1990 \text{ mm}) - \left[\frac{(1 \text{ N/mm}) (2920 \text{ mm})^2}{2} \right]$$

$$M_O = -9.1387 \times 10^6 \text{ N}\cdot\text{mm}$$

From Table A-6: Properties of Structural-Steel Equal Legs Angles, the distance from centroid to upper surface is $y = 29 \text{ mm}$. The distance from centroid to bottom surface is $y = 29.0 \text{ mm} - 100 \text{ mm} = -71 \text{ mm}$. The maximum stress is compressive at the bottom of the beam at the wall. This stress is

$$\sigma_{\max} = -\frac{My}{I} = -\frac{(-9.1387 \times 10^6 \text{ N}\cdot\text{mm}) (-71 \text{ mm})}{4.14 \times 10^6 \text{ mm}^4}$$

$$\sigma_{\max} = -157 \text{ MPa}$$

The maximum stress in the beam is 157 MPa and the stress is compressive in nature.

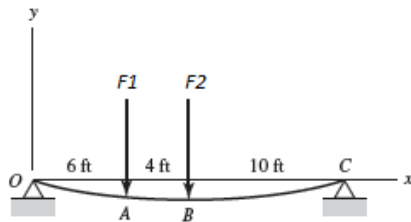
2.

Award: 0 out of 20.00 points

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed 1/2 in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is 2 in long and its weight capacity is 285 lbf/ft.

References**Numeric Response**

Chapter: 04 Deflection and Stiffness

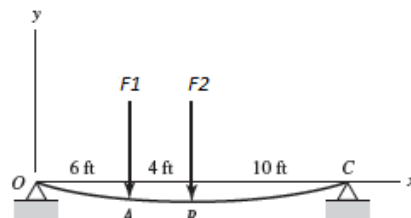
Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

Section: 04.05 Beam Deflections by Superposition

Problem 04.011 - Use superposition to select a pair of channels to meet design constraints

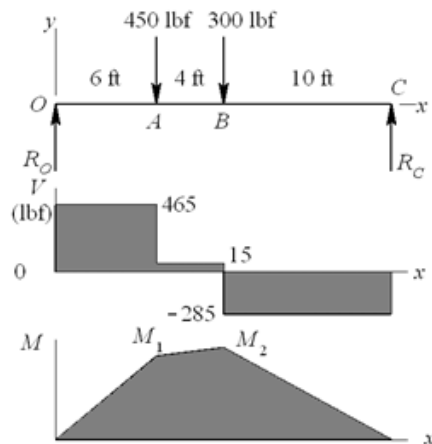
A simply supported beam loaded by two forces is shown in the figure. Select a pair of structural steel channels mounted back to back to support the loads in such a way that the deflection at midspan will not exceed 1/2 in and the maximum stress will not exceed 15 ksi. Use superposition. In the figure, $F_1 = 450$ lbf and $F_2 = 300$ lbf.

Use Table A-7 to select the smallest Imperial units channel that meets the requirements, where dimension a corresponds to the channel length.



The pair of channels that should be selected is 5 ± 2% in long and its weight capacity is 6.7 ± 2% lbf/ft.

Explanation:



$$R_O = \frac{14 \text{ ft}}{20 \text{ ft}} (450 \text{ mm}) + \frac{10 \text{ ft}}{20 \text{ ft}} (300 \text{ mm}) = 465 \text{ lbf}$$

$$R_C = \frac{6 \text{ ft}}{20 \text{ ft}} (450 \text{ mm}) + \frac{10 \text{ ft}}{20 \text{ ft}} (300 \text{ mm}) = 285 \text{ lbf}$$

$$M_1 = 465 \text{ lbf} \times (6 \times 12) \text{ in} = 33.48 \times 10^3 \text{ lbf}\cdot\text{in}$$

$$M_2 = (33.48 \times 10^3 \text{ lbf}) + 15 \text{ lbf} \times (4 \times 12) \text{ in} = 34.20 \times 10^3 \text{ lbf}\cdot\text{in}$$

$$\sigma_{\max} = \frac{M_{\max}}{Z} \Rightarrow 15 = \frac{34.2 \text{ lbf}\cdot\text{in}}{Z} \quad Z = 2.28 \text{ in}^3$$

For deflections, use beams 5 and 6 of Table A-9:

$$y|_{x=10\text{ft}} = \frac{F_1 a [l - (l/2)]}{6EI} \left[\left(\frac{l}{2} \right)^2 + a^2 - 2l \left(\frac{l}{2} \right) \right] - \frac{F_2 l^3}{48EI}$$

$$-0.5 \text{ in} = \frac{(450 \text{ lbf} \times 72 \text{ in} \times 120 \text{ in})}{6 \times 30 \times 10^6 \text{ psi} \times l \times (240 \text{ in})} \left(120^2 + 72^2 - 240^2 \right) \text{ in}^2 - \frac{300 \text{ lbf} \times (240 \text{ in}^3)}{48 \times 30 \times 10^6 \text{ psi} \times l}$$

$$l = 12.60 \text{ in}^4 \Rightarrow l/2 = 6.30 \text{ in}^4$$

Select two 5 in-6.7 lbf/ft channels. From Table A-7, $I = 2 \times 7.49 \text{ in}^4 = 14.98 \text{ in}^4$ and $Z = 2 \times 3.00 \text{ in}^3 = 6.00 \text{ in}^3$.

$$y_{\text{midspan}} = \frac{12.60 \text{ in}}{14.98 \text{ in}} \left(-\frac{1}{2} \right) = -0.421 \text{ in}$$

$$\sigma_{\max} = \frac{34.2 \text{ kpsi}}{6.00 \text{ kpsi}} = 5.70 \text{ kpsi}$$

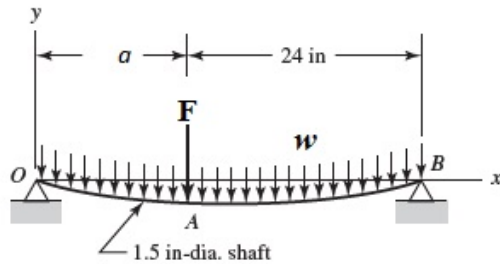
Therefore, this selection suits the required criteria.

3.

Award: 20 out of 20.00 points

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 335 lbf, $w = 125$ lbf/ft, and $a = 11$ in.



The magnitude of the deflection of the steel shaft at A is **0.052** ✓ inch in the **downward** ✓ direction.
 The magnitude of the deflection at midspan is **0.059** ✓ inch in the **downward** ✓ direction.
 The percentage difference is **14** ✓ %.

References**Multi-Answer**

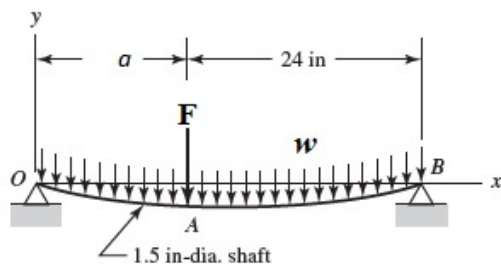
Chapter: 04 Deflection and Stiffness

**Problem 04.012 -
Determine the
deflection of a shaft
using superposition**

Section: 04.05 Beam
Deflections by Superposition

Problem 04.012 - Determine the deflection of a shaft using superposition

Using superposition, find the deflection of the steel shaft at A in the figure. Find the deflection at midspan. By what percentage do these two values differ? Take the magnitude of force (F) to be 335 lbf, $w = 125$ lbf/ft, and $a = 11$ in.



The magnitude of the deflection of the steel shaft at A is **0.0527 ± 2%** ✓ inch in the downward ✓ direction.
 The magnitude of the deflection at midspan is **0.0602 ± 2%** ✓ inch in the downward ✓ direction.
 The percentage difference is **14.13 ± 2%** %.

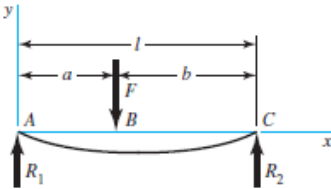
Explanation:

$$I = \frac{\pi}{64}(1.5 \text{ in})^4 = 0.2485 \text{ in}^4 \text{ for the circular cross section of diameter 1.5 in.}$$

From Table A-9: Shear, Moment, and Deflection of Beams, by superposition of beams 6 and 7, at $x = a = 11 \text{ in}$, with $b = 24 \text{ in}$ and $l = 35 \text{ in}$,

The deflection at point A due to point load F can be denoted by y_1 .

6 Simple supports—intermediate load

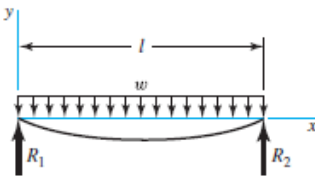


$$y_{AB} = \frac{Fbx}{6EI}[x^2 + b^2 - l^2]$$

$$y_1 = \frac{Fba}{6EI}[a^2 + b^2 - l^2]$$

The deflection at point A due to uniformly distributed load w can be denoted by y_2 .

7 Simple supports—uniform load



$$y = \frac{wx}{24EI}(2lx^2 - x^3 - l^3)$$

$$y_2 = \frac{wa}{24EI}(2la^2 - a^3 - l^3)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y = \frac{Fba}{6EI}[a^2 + b^2 - l^2] + \frac{wa}{24EI}(2la^2 - a^3 - l^3)$$

$$y_A = \frac{(335 \text{ lbf})(24 \text{ in})(11 \text{ in})}{6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(35 \text{ in})}[(11 \text{ in})^2 + (24 \text{ in})^2 - (35 \text{ in})^2]$$

$$+ \frac{\left(\frac{150 \text{ lbf/ft}}{12 \text{ in/ft}}\right)(11 \text{ in})}{(24 \text{ in})(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)}[2(35 \text{ in})(11 \text{ in})^2 - (11 \text{ in})^3 - (35 \text{ in})^3]$$

$$y_A = -0.0527 \text{ in}$$

The deflection of the steel shaft at A is 0.0527 inch in the downward direction.

$$\text{At } x = \frac{l}{2} = 18 \text{ in,}$$

$$y = \frac{Fa[l - (l/2)]}{6EI} \left[\left(\frac{l}{2} \right)^2 + a^2 - 2l\frac{l}{2} \right] + \frac{w(l/2)}{24EI} \left[2l \left(\frac{l}{2} \right)^2 - \left(\frac{l}{2} \right)^3 - l^3 \right]$$

$$y = \frac{(335 \text{ lbf})(11 \text{ in})(18 \text{ in})}{6(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)(35 \text{ in})} [(18 \text{ in})^2 + (11 \text{ in})^2 - (35 \text{ in})^2]$$

$$+ \frac{\left(\frac{150 \text{ lbf/ft}}{12 \text{ in/ft}} \right) (18 \text{ in})}{24(30 \times 10^6 \text{ psi})(0.2485 \text{ in}^4)} [2(35 \text{ in})(18 \text{ in})^2 - (17.5 \text{ in})^3 - (35 \text{ in})^3]$$

$$= -0.0602 \text{ in.}$$

The deflection at mid-span is 0.0602 inch in the downward direction.

The percentage difference is obtained by dividing the difference of deflections with deflection at A.

$$\text{Percentage difference} = \frac{(-0.0602 \text{ in}) - (-0.0527 \text{ in})}{-0.0527 \text{ in}} \times 100 = 14.13\%$$

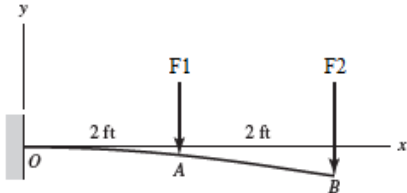
The percentage difference is 14.13%.

4.

Award: 10 out of 20.00 points

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at B . Take the magnitude of force (F_1) to be 306 lbf and the magnitude of force (F_2) to be 207 lbf.



The deflection at B is 2.1 ✖ inch in the downward ✔ direction.

References**Multi-Answer**

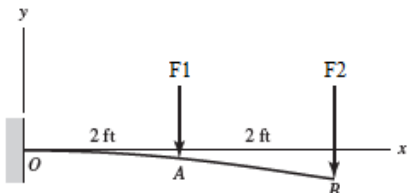
Chapter: 04 Deflection and Stiffness

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

Section: 04.05 Beam Deflections by Superposition

Problem 04.014 - Use superposition to determine the tip deflection of a cantilevered tube

An aluminum tube with outside diameter of 2 in and inside diameter of 1.5 in is cantilevered and loaded as shown. Using the formulas in Table A-9 and superposition, find the deflection at B . Take the magnitude of force (F_1) to be 306 lbf and the magnitude of force (F_2) to be 207 lbf.



The deflection at B is 1.99 ± 2% inch in the downward ✔ direction.

Explanation:

For a tubular cross section with inner diameter d_i and outer diameter d_o , the area moment of inertia is given by

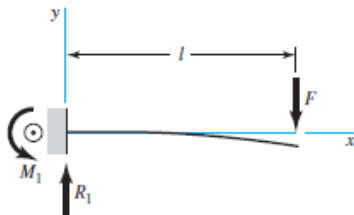
$$I = \frac{\pi}{64} (d_o^4 - d_i^4)$$

$$I = \frac{\pi}{64} [(2 \text{ in})^4 - (1.5 \text{ in})^4] = 0.5369 \text{ in}^4$$

From Table A-5: Physical Constants of Materials, $E = 10.4 \text{ Mpsi}$.

From Table A-9: Shear, Moment, and Deflection of Beams, beams 1 and 2, by superposition, The deflection at point A due to point load F_2 can be denoted by y_1 .

1 Cantilever—end load

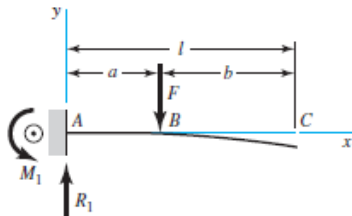


$$y_{\max} = -\frac{Fl^3}{3EI}$$

$$y_1 = -\frac{F_B l^3}{3EI}$$

The deflection at point A due to point load F_1 can be denoted by y_2 .

2 Cantilever—intermediate load



$$y_{\max} = \frac{Fa^2}{6EI} (a - 3l)$$

$$y_2 = \frac{F_A a^2}{6EI} (a - 3l)$$

By the principle of superposition, the total deflection can be obtained by summing up the deflections due to both the loadings.

$$y = y_1 + y_2$$

$$y_B = -\frac{F_B l^3}{3EI} + \frac{F_A a^2}{6EI} (a - 3l) = \frac{-(207 \text{ lbf})[(4 \text{ ft})(12 \text{ in./ft})]^3}{3(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)} + \frac{(306 \text{ lbf})[(2 \text{ ft})(12 \text{ in./ft})]^2}{6(10.4 \times 10^6 \text{ psi})(0.5369 \text{ in}^4)} [(2 \text{ ft})(12 \text{ in./ft}) - 3(4 \text{ ft})(12 \text{ in./ft})]$$

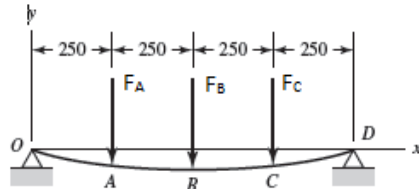
$$y_B = -1.99 \text{ in}$$

5.

Award: 20 out of 20.00 points

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 1.99 mm, given that $F_A = 360$ N, $F_B = 490$ N, and $F_C = 360$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is **32.1** mm.

References**Numeric Response**

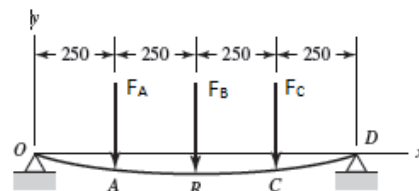
Chapter: 04 Deflection and Stiffness

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Section: 04.05 Beam Deflections by Superposition

Problem 04.016 - Use superposition to determine the minimum diameter of a steel shaft

Using superposition for the bar shown, determine the minimum diameter of a steel shaft for which the maximum deflection is 1.99 mm, given that $F_A = 360$ N, $F_B = 490$ N, and $F_C = 360$ N. The dimensions shown are given in mm.



The minimum diameter of the steel shaft is **31.74 ± 2%** mm.

Explanation:

$$I = \frac{\pi}{64} d^4$$

From Table A-5, $E = 207(10^3)$ MPa.

From Table A-9, beams 5 and 9, with $F_C = F_A = F$, by superposition,

$$y_B = -\frac{F_B l^3}{48EI} + \frac{Fa}{24EI} \left(4a^2 - 3l^2 \right) \Rightarrow I = \frac{1}{48Ey_B} [-F_B l^3 + 2Fa(4a^2 - 3l^2)]$$

$$I = \frac{1}{48(207 \times 10^3 \text{ MPa})(-1.99 \text{ mm})} \left\{ -490 \text{ N} (1000 \text{ mm})^3 + 2(360 \text{ N}) (250 \text{ mm}) [4(250 \text{ mm})^2 - 3(1000 \text{ mm})^2] \right\}$$

$$I = 49816.31 \text{ mm}^4$$

$$d = \sqrt[4]{\frac{64}{\pi} I} = \sqrt[4]{\frac{64}{\pi} (49816.31 \text{ mm}^4)} = 31.74 \text{ mm}$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 9

TOPICS COVERED IN HOMEWORK 9

Chapter 5. Failure Theory: Problems 5-3, 5-4

**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

For a bar of AISI 1030 hot-rolled steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following plane stress states.

[References](#)

Section Break

Difficulty: Medium

1.

Award: 10.00 points

$\sigma_x = 23$ kpsi, and $\sigma_y = 15.5$ kpsi

The factor of safety from the maximum-shear-stress theory is $1.6304 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.8458 \pm 2\%$.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

$\sigma_1 = 23$ kpsi, $\sigma_2 = 15.5$ kpsi, and $\sigma_3 = 0$

$$n = \frac{37.5 \text{ kpsi}}{23 \text{ kpsi} - 0} = 1.6304$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(23 \text{ kpsi})^2 - (23 \text{ kpsi}) \times (15.5 \text{ kpsi}) + (15.5 \text{ kpsi})^2}} = 1.8458$$

The factor of safety from the maximum-shear-stress theory is 1.6304 and the factor of safety from the distortion-energy theory is 1.8458.

[References](#)

Numeric Response

Difficulty: Medium

2.

Award: 10.00 points

$\sigma_x = 15.5$ kpsi, and $\sigma_y = -14.5$ kpsi

The factor of safety from the maximum-shear-stress theory is $1.25 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.44 \pm 2\%$.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

$\sigma_1 = 15.5$ kpsi, $\sigma_2 = 0$, and $\sigma_3 = -14.5$ kpsi

$$n = \frac{37.5 \text{ kpsi}}{15.5 \text{ kpsi} - (-14.5 \text{ kpsi})} = 1.25$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(15.5 \text{ kpsi})^2 - (15.5 \text{ kpsi}) \times (-14.5 \text{ kpsi}) + (-14.5 \text{ kpsi})^2}} = 1.44$$

The factor of safety from the maximum-shear-stress theory is 1.25 and the factor of safety from the distortion-energy theory is 1.44.

[References](#)

Numeric Response

Difficulty: Medium

3.

Award: 10.00 points

 $\sigma_x = 21$ kpsi, and $\tau_{xy} = -9$ kpsiThe factor of safety from the maximum-shear-stress theory is $1.36 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.43 \pm 2\%$.**Explanation:**From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{21 \text{ kpsi}}{2} \pm \sqrt{\left(\frac{21 \text{ kpsi}}{2}\right)^2 + (-9 \text{ kpsi})^2} = 24.33 \text{ kpsi}, -3.33 \text{ kpsi}$$

$$\sigma_1 = 24.33 \text{ kpsi}, \sigma_2 = 0, \sigma_3 = -3.33 \text{ kpsi}$$

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{24.33 \text{ kpsi} - (-3.33 \text{ kpsi})} = 1.36$$

DE

$$\sigma' = \sqrt{(21 \text{ kpsi})^2 + 3(-9 \text{ kpsi})^2} = 26.15 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{26.15 \text{ kpsi}} = 1.43$$

The factor of safety from the maximum-shear-stress theory is 1.36 and the factor of safety from the distortion-energy theory is 1.43.

References**Numeric Response**

Difficulty: Medium

4.

Award: 10.00 points

 $\sigma_x = -12.4$ kpsi, $\sigma_y = 15.5$ kpsi, and $\tau_{xy} = -8.2$ kpsiThe factor of safety from the maximum-shear-stress theory is $1.16 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.34 \pm 2\%$.**Explanation:**From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-12.4 \text{ kpsi}) + (15.5 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-12.4 \text{ kpsi}) - (15.5 \text{ kpsi})}{2}\right)^2 + (-8.2 \text{ kpsi})^2} = 17.73 \text{ kpsi}, -14.63 \text{ kpsi}$$

$$\sigma_1 = 17.73 \text{ kpsi}, \sigma_2 = 0, \sigma_3 = -14.63 \text{ kpsi}$$

According to MSS theory, $\sigma_1 - \sigma_3 = S_y / n$ which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{17.73 \text{ kpsi} - (-14.63 \text{ kpsi})} = 1.16$$

DE

$$\sigma' = \sqrt{(-12.4 \text{ kpsi})^2 - (-12.4 \text{ kpsi}) \times (15.5 \text{ kpsi}) + (15.5 \text{ kpsi})^2 + 3(-8.2 \text{ kpsi})^2} = 28.1 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{28.1 \text{ kpsi}} = 1.34$$

The factor of safety from the maximum-shear-stress theory is 1.16 and the factor of safety from the distortion-energy theory is 1.34.

References**Numeric Response**

Difficulty: Medium

5. Award: 10.00 points

$\sigma_x = -23.2$ kpsi, $\sigma_y = -23.2$ kpsi, and $\tau_{xy} = -14.5$ kpsi

The factor of safety from the maximum-shear-stress theory is $0.99 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.1 \pm 2\%$.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-23.2 \text{ kpsi}) + (-23.2 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-23.2 \text{ kpsi}) - (-23.2 \text{ kpsi})}{2}\right)^2 + (-14.5 \text{ kpsi})^2} = -8.7 \text{ kpsi}, -37.7 \text{ kpsi}$$

$$\sigma_1 = 0, \sigma_2 = -8.7 \text{ kpsi}, \sigma_3 = -37.7 \text{ kpsi}$$

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{0 - (-37.7 \text{ kpsi})} = 0.99$$

DE

$$\sigma' = \sqrt{(-23.2 \text{ kpsi})^2 - (-23.2 \text{ kpsi}) \times (-23.2 \text{ kpsi}) + (-23.2 \text{ kpsi})^2 + 3(-14.5 \text{ kpsi})^2} = 34.19 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{34.19 \text{ kpsi}} = 1.1$$

The factor of safety from the maximum-shear-stress theory is 0.99 and the factor of safety from the distortion-energy theory is 1.1.

References

Numeric Response

Difficulty: Medium

For a bar of AISI 1015 cold-drawn steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following principal stresses.

References

Section Break

Difficulty: Medium

6. Award: 10.00 points

$\sigma_A = 28$ kpsi, and $\sigma_B = 28$ kpsi

The factor of safety from the maximum-shear-stress theory is $1.68 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.68 \pm 2\%$.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = 28$ kpsi, and $\sigma_B = 28$ kpsi

The ordered principle stresses are $\sigma_1 = 28$ kpsi, $\sigma_2 = 28$ kpsi, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(28 \text{ kpsi} - 0)} = 1.68$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(28 \text{ kpsi})^2 - (28 \text{ kpsi}) \times (28 \text{ kpsi}) + (28 \text{ kpsi})^2}} = 1.68$$

The factor of safety from the maximum-shear-stress theory is 1.68 and the factor of safety from the distortion-energy theory is 1.68.

References

Numeric Response

Difficulty: Medium

7.

Award: 10.00 points

 $\sigma_A = 28$ kpsi, and $\sigma_B = -33$ kpsiThe factor of safety from the maximum-shear-stress theory is $0.77 \pm 2\%$, and the factor of safety from the distortion-energy theory is $0.89 \pm 2\%$.**Explanation:**From Table A-20, $S_y = 47$ kpsi. $\sigma_A = 28$ kpsi, and $\sigma_B = -33$ kpsiThe ordered principle stresses are $\sigma_1 = 28$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -33$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(28 \text{ kpsi} - (-33 \text{ kpsi}))} = 0.77$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(28 \text{ kpsi})^2 - (28 \text{ kpsi}) \times (-33 \text{ kpsi}) + (-33 \text{ kpsi})^2}} = 0.89$$

The factor of safety from the maximum-shear-stress theory is 0.77 and the factor of safety from the distortion-energy theory is 0.89.

References**Numeric Response**

Difficulty: Medium

8.

Award: 10.00 points

 $\sigma_A = 28$ kpsi, and $\sigma_B = 13.5$ kpsiThe factor of safety from the maximum-shear-stress theory is $1.68 \pm 2\%$, and the factor of safety from distortion-energy theory is $1.94 \pm 2\%$.**Explanation:**From Table A-20, $S_y = 47$ kpsi. $\sigma_A = 28$ kpsi, and $\sigma_B = 13.5$ kpsiThe ordered principle stresses are $\sigma_1 = 28$ kpsi, $\sigma_2 = 13.5$ kpsi, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(28 \text{ kpsi} - 0)} = 1.68$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(28 \text{ kpsi})^2 - (28 \text{ kpsi}) \times (13.5 \text{ kpsi}) + (13.5 \text{ kpsi})^2}} = 1.94$$

The factor of safety from the maximum-shear-stress theory is 1.68 and the factor of safety from the distortion-energy theory is 1.94.

References**Numeric Response**

Difficulty: Medium

9. Award: 10.00 points

$\sigma_A = -33$ kpsi, and $\sigma_B = -14$ kpsi

The factor of safety from the maximum-shear-stress theory is $1.42 \pm 2\%$, and the factor of safety from the distortion-energy theory is $1.64 \pm 2\%$.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = -33$ kpsi, and $\sigma_B = -14$ kpsi

The ordered principle stresses are $\sigma_1 = 0$, $\sigma_2 = -14$ kpsi, $\sigma_3 = -33$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(0 - (-33 \text{ kpsi}))} = 1.42$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-33 \text{ kpsi})^2 - (-33 \text{ kpsi}) \times (-14 \text{ kpsi}) + (-14 \text{ kpsi})^2}} = 1.64$$

The factor of safety from the maximum-shear-stress theory is 1.42 and the factor of safety from the distortion-energy theory is 1.64.

References

Numeric Response

Difficulty: Medium

10. Award: 10.00 points

$\sigma_A = -46$ kpsi, and $\sigma_B = 10$ kpsi

The factor of safety from the maximum-shear-stress theory is $0.84 \pm 2\%$, and the factor of safety from the distortion-energy theory is $0.91 \pm 2\%$.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = -46$ kpsi, and $\sigma_B = 10$ kpsi

The ordered principle stresses are $\sigma_1 = 10$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -46$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(10 \text{ kpsi} - (-46 \text{ kpsi}))} = 0.84$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-46 \text{ kpsi})^2 - (-46 \text{ kpsi}) \times (10 \text{ kpsi}) + (10 \text{ kpsi})^2}} = 0.91$$

The factor of safety from the maximum-shear-stress theory is 0.84 and the factor of safety from the distortion-energy theory is 0.91.

References

Numeric Response

Difficulty: Medium

ABOVE AVERAGE

Guerrero, Edward

Score: 100/100 Points 100 %

Problem 05.003 - Determination of factor of safety for principal stress states

For a bar of AISI 1030 hot-rolled steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following plane stress states.

References	
Section Break	Chapter: 05 Failures Resulting from Static Loading
Problem 05.003 - Determination of factor of safety for principal stress states	Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

1.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.003.a - case where normal stresses are equal to principal stresses

 $\sigma_x = 25$ kpsi, and $\sigma_y = 15$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.5** , and the factor of safety from the distortion-energy theory is **1.72** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.a - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.a - case where normal stresses are equal to principal stresses

 $\sigma_x = 25$ kpsi, and $\sigma_y = 15$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.5 ± 2%**, and the factor of safety from the distortion-energy theory is **1.7206 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

$\sigma_1 = 25$ kpsi, $\sigma_2 = 15$ kpsi, and $\sigma_3 = 0$

$$n = \frac{37.5 \text{ kpsi}}{25 \text{ kpsi} - 0} = 1.5$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(25 \text{ kpsi})^2 - (25 \text{ kpsi}) \times (15 \text{ kpsi}) + (15 \text{ kpsi})^2}} = 1.7206$$

2.

Award: 10 out of 10.00 points

 You received credit for this question in a previous attempt

Problem 05.003.b - case where normal stresses are equal to principal stresses

 $\sigma_x = 15 \text{ kpsi}$, and $\sigma_y = -14 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.29** , and the factor of safety from the distortion-energy theory is **1.49** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.b - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.b - case where normal stresses are equal to principal stresses

 $\sigma_x = 15 \text{ kpsi}$, and $\sigma_y = -14 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.29 ± 2%**, and the factor of safety from the distortion-energy theory is **1.49 ± 2%**.

Explanation:

 From Table A-20, $S_y = 37.5 \text{ kpsi}$.

 According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

 According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

 $\sigma_1 = 15 \text{ kpsi}$, $\sigma_2 = 0$, and $\sigma_3 = -14 \text{ kpsi}$

$$n = \frac{37.5 \text{ kpsi}}{15 \text{ kpsi} - (-14 \text{ kpsi})} = 1.29$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(15 \text{ kpsi})^2 - (15 \text{ kpsi}) \times (-14 \text{ kpsi}) + (-14 \text{ kpsi})^2}} = 1.49$$

3.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

 $\sigma_x = 19 \text{ kpsi}$, and $\tau_{xy} = -9 \text{ kpsi}$

The factor of safety from the maximum-shear-stress theory is **1.43** , and the factor of safety from the distortion-energy theory is **1.53** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

 $\sigma_x = 19 \text{ kpsi}$, and $\tau_{xy} = -9 \text{ kpsi}$

The factor of safety from the maximum-shear-stress theory is **1.43 ± 2%**, and the factor of safety from the distortion-energy theory is **1.53 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5 \text{ kpsi}$.

$$\sigma_A, \sigma_B = \frac{19 \text{ kpsi}}{2} \pm \sqrt{\left(\frac{19 \text{ kpsi}}{2}\right)^2 + (-9 \text{ kpsi})^2} = 22.59 \text{ kpsi}, -3.59 \text{ kpsi}$$

$\sigma_1 = 22.59 \text{ kpsi}$, $\sigma_2 = 0$, $\sigma_3 = -3.59 \text{ kpsi}$

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{22.59 \text{ kpsi} - (-3.59 \text{ kpsi})} = 1.43$$

DE

$$\sigma' = \sqrt{(19 \text{ kpsi})^2 + 3(-9 \text{ kpsi})^2} = 24.58 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{24.58 \text{ kpsi}} = 1.53$$

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -12.6$ kpsi, $\sigma_y = 15$ kpsi, and $\tau_{xy} = -8.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.16** , and the factor of safety from the distortion-energy theory is **1.33** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -12.6$ kpsi, $\sigma_y = 15$ kpsi, and $\tau_{xy} = -8.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.16 ± 2%**, and the factor of safety from the distortion-energy theory is **1.33 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-12.6 \text{ kpsi}) + (15 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-12.6 \text{ kpsi}) - (15 \text{ kpsi})}{2}\right)^2 + (-8.5 \text{ kpsi})^2} = 17.41 \text{ kpsi}, -15.01 \text{ kpsi}$$

$\sigma_1 = 17.41$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -15.01$ kpsi

According to MSS theory, $\sigma_1 - \sigma_3 = S_y / n$ which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{17.41 \text{ kpsi} - (-15.01 \text{ kpsi})} = 1.16$$

DE

$$\sigma' = \sqrt{(-12.6 \text{ kpsi})^2 - (-12.6 \text{ kpsi}) \times (15 \text{ kpsi}) + (15 \text{ kpsi})^2 + 3(-8.5 \text{ kpsi})^2} = 28.1 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{28.1 \text{ kpsi}} = 1.33$$

5.

Award: 10 out of 10.00 points

 You received credit for this question in a previous attempt

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -26.2$ kpsi, $\sigma_y = -26.2$ kpsi, and $\tau_{xy} = -14$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.93** , and the factor of safety from the distortion-energy theory is **1.06** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -26.2$ kpsi, $\sigma_y = -26.2$ kpsi, and $\tau_{xy} = -14$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.93 ± 2%**, and the factor of safety from the distortion-energy theory is **1.05 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-26.2 \text{ kpsi}) + (-26.2 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-26.2 \text{ kpsi}) - (-26.2 \text{ kpsi})}{2}\right)^2 + (-14 \text{ kpsi})^2} = -12.2 \text{ kpsi}, -40.2 \text{ kpsi}$$

$\sigma_1 = 0$, $\sigma_2 = -12.2$ kpsi, $\sigma_3 = -40.2$ kpsi

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{0 - (-40.2 \text{ kpsi})} = 0.93$$

DE

$$\sigma' = \sqrt{(-26.2 \text{ kpsi})^2 - (-26.2 \text{ kpsi}) \times (-26.2 \text{ kpsi}) + (-26.2 \text{ kpsi})^2 + 3(-14 \text{ kpsi})^2} = 35.7 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{35.7 \text{ kpsi}} = 1.05$$

Problem 05.004 - Determination of factor of safety for principal stress states

For a bar of AISI 1015 cold-drawn steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following principal stresses.

References

Section Break	Chapter: 05 Failures Resulting from Static Loading
Problem 05.004 - Determination of factor of safety for principal stress states	Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

6.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.a - case where normal stresses are equal to principal stresses

 $\sigma_A = 29 \text{ kpsi}$, and $\sigma_B = 29 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.62** , and the factor of safety from the distortion-energy theory is **1.62** .

References

Numeric Response

 Chapter: 05 Failures Resulting
from Static Loading

Problem 05.004.a - case where normal stresses are equal to principal stresses

 Section: 05.04 Maximum-
Shear-Stress Theory for
Ductile Materials

Problem 05.004.a - case where normal stresses are equal to principal stresses

 $\sigma_A = 29 \text{ kpsi}$, and $\sigma_B = 29 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.62 ± 2%**, and the factor of safety from the distortion-energy theory is **1.62 ± 2%**.

Explanation:

 From Table A-20, $S_y = 47 \text{ kpsi}$.

 $\sigma_A = 29 \text{ kpsi}$, and $\sigma_B = 29 \text{ kpsi}$

 The ordered principle stresses are $\sigma_1 = 29 \text{ kpsi}$, $\sigma_2 = 29 \text{ kpsi}$, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(29 \text{ kpsi} - 0)} = 1.62$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(29 \text{ kpsi})^2 - (29 \text{ kpsi}) \times (29 \text{ kpsi}) + (29 \text{ kpsi})^2}} = 1.62$$

Problem 05.004.b - case where normal stresses are equal to principal stresses

$\sigma_A = 29$ kpsi, and $\sigma_B = -32$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.77** , and the factor of safety from the distortion-energy theory is **0.89** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.b - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.b - case where normal stresses are equal to principal stresses

$\sigma_A = 29$ kpsi, and $\sigma_B = -32$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.77 ± 2%**, and the factor of safety from the distortion-energy theory is **0.89 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = 29$ kpsi, and $\sigma_B = -32$ kpsi

The ordered principle stresses are $\sigma_1 = 29$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -32$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(29 \text{ kpsi} - (-32 \text{ kpsi}))} = 0.77$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(29 \text{ kpsi})^2 - (29 \text{ kpsi}) \times (-32 \text{ kpsi}) + (-32 \text{ kpsi})^2}} = 0.89$$

8.

Award: 10 out of 10.00 points

 You received credit for this question in a previous attempt

Problem 05.004.c - case where normal stresses are equal to principal stresses

 $\sigma_A = 29 \text{ kpsi}$, and $\sigma_B = 16.5 \text{ kpsi}$

The factor of safety from the maximum-shear-stress theory is **1.62** , and the factor of safety from distortion-energy theory is **1.87** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.c - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.c - case where normal stresses are equal to principal stresses

 $\sigma_A = 29 \text{ kpsi}$, and $\sigma_B = 16.5 \text{ kpsi}$

The factor of safety from the maximum-shear-stress theory is **1.62 ± 2%**, and the factor of safety from distortion-energy theory is **1.87 ± 2%**.

Explanation:

From Table A-20, $S_y = 47 \text{ kpsi}$.

 $\sigma_A = 29 \text{ kpsi}$, and $\sigma_B = 16.5 \text{ kpsi}$

The ordered principle stresses are $\sigma_1 = 29 \text{ kpsi}$, $\sigma_2 = 16.5 \text{ kpsi}$, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(29 \text{ kpsi} - 0)} = 1.62$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma_1} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(29 \text{ kpsi})^2 - (29 \text{ kpsi}) \times (16.5 \text{ kpsi}) + (16.5 \text{ kpsi})^2}} = 1.87$$

9.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.d - case where normal stresses are equal to principal stresses

 $\sigma_A = -32$ kpsi, and $\sigma_B = -16$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.47** , and the factor of safety from the distortion-energy theory is **1.7** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.d - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.d - case where normal stresses are equal to principal stresses

 $\sigma_A = -32$ kpsi, and $\sigma_B = -16$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.47 ± 2%**, and the factor of safety from the distortion-energy theory is **1.7 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

 $\sigma_A = -32$ kpsi, and $\sigma_B = -16$ kpsi

The ordered principle stresses are $\sigma_1 = 0$, $\sigma_2 = -16$ kpsi, $\sigma_3 = -32$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(0 - (-32 \text{ kpsi}))} = 1.47$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-32 \text{ kpsi})^2 - (-32 \text{ kpsi}) \times (-16 \text{ kpsi}) + (-16 \text{ kpsi})^2}} = 1.7$$

10.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.e - case where normal stresses are equal to principal stresses

 $\sigma_A = -49$ kpsi, and $\sigma_B = 9$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.81** , and the factor of safety from the distortion-energy theory is **0.87** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.e - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.e - case where normal stresses are equal to principal stresses

 $\sigma_A = -49$ kpsi, and $\sigma_B = 9$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.81 ± 2%**, and the factor of safety from the distortion-energy theory is **0.87 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

 $\sigma_A = -49$ kpsi, and $\sigma_B = 9$ kpsi

The ordered principle stresses are $\sigma_1 = 9$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -49$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(9 \text{ kpsi} - (-49 \text{ kpsi}))} = 0.81$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-49 \text{ kpsi})^2 - (-49 \text{ kpsi}) \times (9 \text{ kpsi}) + (9 \text{ kpsi})^2}} = 0.87$$

AVERAGE

Garay, Gabriela

Score: 90/100 Points 90 %

Problem 05.003 - Determination of factor of safety for principal stress states

For a bar of AISI 1030 hot-rolled steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following plane stress states.

References

Section Break	Chapter: 05 Failures Resulting from Static Loading
Problem 05.003 - Determination of factor of safety for principal stress states	Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

1.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.003.a - case where normal stresses are equal to principal stresses

 $\sigma_x = 27.5$ kpsi, and $\sigma_y = 16$ kpsi

 The factor of safety from the maximum-shear-stress theory is **1.36** , and the factor of safety from the distortion-energy theory is **1.56** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.a - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.a - case where normal stresses are equal to principal stresses

 $\sigma_x = 27.5$ kpsi, and $\sigma_y = 16$ kpsi

 The factor of safety from the maximum-shear-stress theory is **1.3636 ± 2%**, and the factor of safety from the distortion-energy theory is **1.5676 ± 2%**.

Explanation:

 From Table A-20, $S_y = 37.5$ kpsi.

 According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

 According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

 $\sigma_1 = 27.5$ kpsi, $\sigma_2 = 16$ kpsi, and $\sigma_3 = 0$

$$n = \frac{37.5 \text{ kpsi}}{27.5 \text{ kpsi} - 0} = 1.3636$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(27.5 \text{ kpsi})^2 - (27.5 \text{ kpsi}) \times (16 \text{ kpsi}) + (16 \text{ kpsi})^2}} = 1.5676$$

2.

Award: 5 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.003.b - case where normal stresses are equal to principal stresses

 $\sigma_x = 16 \text{ kpsi}$, and $\sigma_y = -14.5 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.32** , and the factor of safety from the distortion-energy theory is **1.42** .

References

Numeric Response

 Chapter: 05 Failures Resulting
from Static Loading

Problem 05.003.b - case where normal stresses are equal to principal stresses

 Section: 05.04 Maximum-
Shear-Stress Theory for
Ductile Materials

Problem 05.003.b - case where normal stresses are equal to principal stresses

 $\sigma_x = 16 \text{ kpsi}$, and $\sigma_y = -14.5 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.23 ± 2%**, and the factor of safety from the distortion-energy theory is **1.42 ± 2%**.

Explanation:

 From Table A-20, $S_y = 37.5 \text{ kpsi}$.

 According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

 According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

 $\sigma_1 = 16 \text{ kpsi}$, $\sigma_2 = 0$, and $\sigma_3 = -14.5 \text{ kpsi}$

$$n = \frac{37.5 \text{ kpsi}}{16 \text{ kpsi} - (-14.5 \text{ kpsi})} = 1.23$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(16 \text{ kpsi})^2 - (16 \text{ kpsi}) \times (-14.5 \text{ kpsi}) + (-14.5 \text{ kpsi})^2}} = 1.42$$

3.

Award: 5 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

 $\sigma_x = 21 \text{ kpsi}$, and $\tau_{xy} = -10 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.2** , and the factor of safety from the distortion-energy theory is **1.38** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

 $\sigma_x = 21 \text{ kpsi}$, and $\tau_{xy} = -10 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.29 ± 2%**, and the factor of safety from the distortion-energy theory is **1.38 ± 2%**.

Explanation:

 From Table A-20, $S_y = 37.5 \text{ kpsi}$.

$$\sigma_A, \sigma_B = \frac{21 \text{ kpsi}}{2} \pm \sqrt{\left(\frac{21 \text{ kpsi}}{2}\right)^2 + (-10 \text{ kpsi})^2} = 25 \text{ kpsi}, -4 \text{ kpsi}$$

 $\sigma_1 = 25 \text{ kpsi}$, $\sigma_2 = 0$, $\sigma_3 = -4 \text{ kpsi}$

 According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{25 \text{ kpsi} - (-4 \text{ kpsi})} = 1.29$$

DE

$$\sigma' = \sqrt{(21 \text{ kpsi})^2 + 3(-10 \text{ kpsi})^2} = 27.22 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{27.22 \text{ kpsi}} = 1.38$$

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -10.8$ kpsi, $\sigma_y = 16$ kpsi, and $\tau_{xy} = -9.7$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.13** , and the factor of safety from the distortion-energy theory is **1.3** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -10.8$ kpsi, $\sigma_y = 16$ kpsi, and $\tau_{xy} = -9.7$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.13 ± 2%**, and the factor of safety from the distortion-energy theory is **1.3 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-10.8 \text{ kpsi}) + (16 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-10.8 \text{ kpsi}) - (16 \text{ kpsi})}{2}\right)^2 + (-9.7 \text{ kpsi})^2} = 19.14 \text{ kpsi}, -13.94 \text{ kpsi}$$

$\sigma_1 = 19.14$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -13.94$ kpsi

According to MSS theory, $\sigma_1 - \sigma_3 = S_y / n$ which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{19.14 \text{ kpsi} - (-13.94 \text{ kpsi})} = 1.13$$

DE

$$\sigma' = \sqrt{(-10.8 \text{ kpsi})^2 - (-10.8 \text{ kpsi}) \times (16 \text{ kpsi}) + (16 \text{ kpsi})^2 + 3(-9.7 \text{ kpsi})^2} = 28.8 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{28.8 \text{ kpsi}} = 1.3$$

5.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -25.2$ kpsi, $\sigma_y = -25.2$ kpsi, and $\tau_{xy} = -14.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.94** , and the factor of safety from the distortion-energy theory is **1.05** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -25.2$ kpsi, $\sigma_y = -25.2$ kpsi, and $\tau_{xy} = -14.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.94 ± 2%**, and the factor of safety from the distortion-energy theory is **1.05 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-25.2 \text{ kpsi}) + (-25.2 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-25.2 \text{ kpsi}) - (-25.2 \text{ kpsi})}{2}\right)^2 + (-14.5 \text{ kpsi})^2} = -10.7 \text{ kpsi}, -39.7 \text{ kpsi}$$

$\sigma_1 = 0$, $\sigma_2 = -10.7$ kpsi, $\sigma_3 = -39.7$ kpsi

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{0 - (-39.7 \text{ kpsi})} = 0.94$$

DE

$$\sigma' = \sqrt{(-25.2 \text{ kpsi})^2 - (-25.2 \text{ kpsi}) \times (-25.2 \text{ kpsi}) + 3(-14.5 \text{ kpsi})^2} = 35.58 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{35.58 \text{ kpsi}} = 1.05$$

Problem 05.004 - Determination of factor of safety for principal stress states

For a bar of AISI 1015 cold-drawn steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following principal stresses.

References

Section Break	Chapter: 05 Failures Resulting from Static Loading
Problem 05.004 - Determination of factor of safety for principal stress states	Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

6.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.a - case where normal stresses are equal to principal stresses

 $\sigma_A = 28 \text{ kpsi}$, and $\sigma_B = 28 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.68** , and the factor of safety from the distortion-energy theory is **1.68** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.a - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.a - case where normal stresses are equal to principal stresses

 $\sigma_A = 28 \text{ kpsi}$, and $\sigma_B = 28 \text{ kpsi}$

 The factor of safety from the maximum-shear-stress theory is **1.68 ± 2%**, and the factor of safety from the distortion-energy theory is **1.68 ± 2%**.

Explanation:

 From Table A-20, $S_y = 47 \text{ kpsi}$.

 $\sigma_A = 28 \text{ kpsi}$, and $\sigma_B = 28 \text{ kpsi}$

 The ordered principle stresses are $\sigma_1 = 28 \text{ kpsi}$, $\sigma_2 = 28 \text{ kpsi}$, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(28 \text{ kpsi} - 0)} = 1.68$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(28 \text{ kpsi})^2 - (28 \text{ kpsi}) \times (28 \text{ kpsi}) + (28 \text{ kpsi})^2}} = 1.68$$

Problem 05.004.b - case where normal stresses are equal to principal stresses

$\sigma_A = 28$ kpsi, and $\sigma_B = -31$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.8** , and the factor of safety from the distortion-energy theory is **0.92** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.b - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.b - case where normal stresses are equal to principal stresses

$\sigma_A = 28$ kpsi, and $\sigma_B = -31$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.8 ± 2%**, and the factor of safety from the distortion-energy theory is **0.92 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = 28$ kpsi, and $\sigma_B = -31$ kpsi

The ordered principle stresses are $\sigma_1 = 28$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -31$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(28 \text{ kpsi} - (-31 \text{ kpsi}))} = 0.8$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(28 \text{ kpsi})^2 - (28 \text{ kpsi}) \times (-31 \text{ kpsi}) + (-31 \text{ kpsi})^2}} = 0.92$$

8.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.c - case where normal stresses are equal to principal stresses

$\sigma_A = 28$ kpsi, and $\sigma_B = 15.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.68** , and the factor of safety from distortion-energy theory is **1.93** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.c - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.c - case where normal stresses are equal to principal stresses

$\sigma_A = 28$ kpsi, and $\sigma_B = 15.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.68 ± 2%**, and the factor of safety from distortion-energy theory is **1.93 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = 28$ kpsi, and $\sigma_B = 15.5$ kpsi

The ordered principle stresses are $\sigma_1 = 28$ kpsi, $\sigma_2 = 15.5$ kpsi, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(28 \text{ kpsi} - 0)} = 1.68$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma_1} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(28 \text{ kpsi})^2 - (28 \text{ kpsi}) \times (15.5 \text{ kpsi}) + (15.5 \text{ kpsi})^2}} = 1.93$$

9.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.d - case where normal stresses are equal to principal stresses

 $\sigma_A = -31$ kpsi, and $\sigma_B = -14.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.52** , and the factor of safety from the distortion-energy theory is **1.75** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.d - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.d - case where normal stresses are equal to principal stresses

 $\sigma_A = -31$ kpsi, and $\sigma_B = -14.5$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.52 ± 2%**, and the factor of safety from the distortion-energy theory is **1.75 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

 $\sigma_A = -31$ kpsi, and $\sigma_B = -14.5$ kpsi

The ordered principle stresses are $\sigma_1 = 0$, $\sigma_2 = -14.5$ kpsi, $\sigma_3 = -31$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(0 - (-31 \text{ kpsi}))} = 1.52$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-31 \text{ kpsi})^2 - (-31 \text{ kpsi}) \times (-14.5 \text{ kpsi}) + (-14.5 \text{ kpsi})^2}} = 1.75$$

10.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.004.e - case where normal stresses are equal to principal stresses

$\sigma_A = -51$ kpsi, and $\sigma_B = 10$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.77** , and the factor of safety from the distortion-energy theory is **0.83** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.e - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.e - case where normal stresses are equal to principal stresses

$\sigma_A = -51$ kpsi, and $\sigma_B = 10$ kpsi

The factor of safety from the maximum-shear-stress theory is **0.77 ± 2%**, and the factor of safety from the distortion-energy theory is **0.83 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = -51$ kpsi, and $\sigma_B = 10$ kpsi

The ordered principle stresses are $\sigma_1 = 10$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -51$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(10 \text{ kpsi} - (-51 \text{ kpsi}))} = 0.77$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-51 \text{ kpsi})^2 - (-51 \text{ kpsi}) \times (10 \text{ kpsi}) + (10 \text{ kpsi})^2}} = 0.83$$

BELOW AVERAGE

Jackson-Rubio, Darius

Score: 35/100 Points 35 %

Problem 05.003 - Determination of factor of safety for principal stress states

For a bar of AISI 1030 hot-rolled steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following plane stress states.

References

Section Break	Chapter: 05 Failures Resulting from Static Loading
Problem 05.003 - Determination of factor of safety for principal stress states	Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

1.

Award: 0 out of 10.00 points

Problem 05.003.a - case where normal stresses are equal to principal stresses

 $\sigma_x = 27$ kpsi, and $\sigma_y = 14.5$ kpsiThe factor of safety from the maximum-shear-stress theory is **2.36** ✖, and the factor of safety from the distortion-energy theory is **5.65** ✖.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.a - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.a - case where normal stresses are equal to principal stresses

 $\sigma_x = 27$ kpsi, and $\sigma_y = 14.5$ kpsiThe factor of safety from the maximum-shear-stress theory is **1.3889 ± 2%**, and the factor of safety from the distortion-energy theory is **1.6023 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

 $\sigma_1 = 27$ kpsi, $\sigma_2 = 14.5$ kpsi, and $\sigma_3 = 0$

$$n = \frac{37.5 \text{ kpsi}}{27 \text{ kpsi} - 0} = 1.3889$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(27 \text{ kpsi})^2 - (27 \text{ kpsi}) \times (14.5 \text{ kpsi}) + (14.5 \text{ kpsi})^2}} = 1.6023$$

2.

Award: 0 out of 10.00 points

Problem 05.003.b - case where normal stresses are equal to principal stresses

 $\sigma_x = 14.5$ kpsi, and $\sigma_y = -15$ kpsiThe factor of safety from the maximum-shear-stress theory is **4.4** ✖, and the factor of safety from the distortion-energy theory is **2.5** ✖.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.b - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.b - case where normal stresses are equal to principal stresses

 $\sigma_x = 14.5$ kpsi, and $\sigma_y = -15$ kpsiThe factor of safety from the maximum-shear-stress theory is **1.27 ± 2%**, and the factor of safety from the distortion-energy theory is **1.47 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.According to the DE theory, $\sigma' = (\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2)^{1/2} = (\sigma_x^2 - \sigma_x \sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$, which gives $n = \frac{S_y}{\sigma'}$.

MSST

 $\sigma_1 = 14.5$ kpsi, $\sigma_2 = 0$, and $\sigma_3 = -15$ kpsi

$$n = \frac{37.5 \text{ kpsi}}{14.5 \text{ kpsi} - (-15 \text{ kpsi})} = 1.27$$

DE

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{37.5 \text{ kpsi}}{\sqrt{(14.5 \text{ kpsi})^2 - (14.5 \text{ kpsi}) \times (-15 \text{ kpsi}) + (-15 \text{ kpsi})^2}} = 1.47$$

3.

Award: 0 out of 10.00 points

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

 $\sigma_x = 21$ kpsi, and $\tau_{xy} = -10$ kpsiThe factor of safety from the maximum-shear-stress theory is **3.04** ✖, and the factor of safety from the distortion-energy theory is **9.23** ✖.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.c - case of extreme value shear stress with one zero value plane stress

 $\sigma_x = 21$ kpsi, and $\tau_{xy} = -10$ kpsiThe factor of safety from the maximum-shear-stress theory is **1.29 ± 2%**, and the factor of safety from the distortion-energy theory is **1.38 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{21 \text{ kpsi}}{2} \pm \sqrt{\left(\frac{21 \text{ kpsi}}{2}\right)^2 + (-10 \text{ kpsi})^2} = 25 \text{ kpsi}, -4 \text{ kpsi}$$

 $\sigma_1 = 25$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -4$ kpsiAccording to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{25 \text{ kpsi} - (-4 \text{ kpsi})} = 1.29$$

DE

$$\sigma' = \sqrt{(21 \text{ kpsi})^2 + 3(-10 \text{ kpsi})^2} = 27.22 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{27.22 \text{ kpsi}} = 1.38$$

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -11.6$ kpsi, $\sigma_y = 14.5$ kpsi, and $\tau_{xy} = -8.6$ kpsi

The factor of safety from the maximum-shear-stress theory is **3.27** ✖, and the factor of safety from the distortion-energy theory is **3.64** ✖.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.d - case of extreme value shear stress with no zero value plane stresses

$\sigma_x = -11.6$ kpsi, $\sigma_y = 14.5$ kpsi, and $\tau_{xy} = -8.6$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.2 ± 2%**, and the factor of safety from the distortion-energy theory is **1.38 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5$ kpsi.

$$\sigma_A, \sigma_B = \frac{(-11.6 \text{ kpsi}) + (14.5 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-11.6 \text{ kpsi}) - (14.5 \text{ kpsi})}{2}\right)^2 + (-8.6 \text{ kpsi})^2} = 17.08 \text{ kpsi}, -14.18 \text{ kpsi}$$

$\sigma_1 = 17.08$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -14.18$ kpsi

According to MSS theory, $\sigma_1 - \sigma_3 = S_y / n$ which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{17.08 \text{ kpsi} - (-14.18 \text{ kpsi})} = 1.2$$

DE

$$\sigma' = \sqrt{(-11.6 \text{ kpsi})^2 - (-11.6 \text{ kpsi}) \times (14.5 \text{ kpsi}) + (14.5 \text{ kpsi})^2 + 3(-8.6 \text{ kpsi})^2} = 27.1 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{27.1 \text{ kpsi}} = 1.38$$

5.

Award: 0 out of 10.00 points

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

$$\sigma_x = -24.8 \text{ kpsi}, \sigma_y = -24.8 \text{ kpsi}, \text{ and } \tau_{xy} = -15 \text{ kpsi}$$

The factor of safety from the maximum-shear-stress theory is **1.13** ✖, and the factor of safety from the distortion-energy theory is **1.14** ✖.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.003.e - case of extreme value shear stress with no zero value plane stresses

$$\sigma_x = -24.8 \text{ kpsi}, \sigma_y = -24.8 \text{ kpsi}, \text{ and } \tau_{xy} = -15 \text{ kpsi}$$

The factor of safety from the maximum-shear-stress theory is **0.94 ± 2%**, and the factor of safety from the distortion-energy theory is **1.04 ± 2%**.

Explanation:

From Table A-20, $S_y = 37.5 \text{ kpsi}$.

$$\sigma_A, \sigma_B = \frac{(-24.8 \text{ kpsi}) + (-24.8 \text{ kpsi})}{2} \pm \sqrt{\left(\frac{(-24.8 \text{ kpsi}) - (-24.8 \text{ kpsi})}{2}\right)^2 + (-15 \text{ kpsi})^2} = -9.8 \text{ kpsi}, -39.8 \text{ kpsi}$$

$$\sigma_1 = 0, \sigma_2 = -9.8 \text{ kpsi}, \sigma_3 = -39.8 \text{ kpsi}$$

According to the MSS theory, $\sigma_1 - \sigma_3 = S_y / n$, which gives $n = \frac{S_y}{(\sigma_1 - \sigma_3)}$.

MSST

$$n = \frac{37.5 \text{ kpsi}}{0 - (-39.8 \text{ kpsi})} = 0.94$$

DE

$$\sigma' = \sqrt{(-24.8 \text{ kpsi})^2 - (-24.8 \text{ kpsi}) \times (-24.8 \text{ kpsi}) + (-24.8 \text{ kpsi})^2 + 3(-15 \text{ kpsi})^2} = 35.92 \text{ kpsi}$$

$$n = \frac{37.5 \text{ kpsi}}{35.92 \text{ kpsi}} = 1.04$$

Problem 05.004 - Determination of factor of safety for principal stress states

For a bar of AISI 1015 cold-drawn steel, determine the factor of safety using the distortion-energy and maximum-shear-stress theories for the following principal stresses.

References

Section Break	Chapter: 05 Failures Resulting from Static Loading
Problem 05.004 - Determination of factor of safety for principal stress states	Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

6.

Award: 10 out of 10.00 points

Problem 05.004.a - case where normal stresses are equal to principal stresses

 $\sigma_A = 31$ kpsi, and $\sigma_B = 31$ kpsiThe factor of safety from the maximum-shear-stress theory is **1.52** ✓, and the factor of safety from the distortion-energy theory is **1.52** ✓.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.a - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.a - case where normal stresses are equal to principal stresses

 $\sigma_A = 31$ kpsi, and $\sigma_B = 31$ kpsiThe factor of safety from the maximum-shear-stress theory is **1.52 ± 2%**, and the factor of safety from the distortion-energy theory is **1.52 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi. $\sigma_A = 31$ kpsi, and $\sigma_B = 31$ kpsiThe ordered principle stresses are $\sigma_1 = 31$ kpsi, $\sigma_2 = 31$ kpsi, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(31 \text{ kpsi} - 0)} = 1.52$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(31 \text{ kpsi})^2 - (31 \text{ kpsi}) \times (31 \text{ kpsi}) + (31 \text{ kpsi})^2}} = 1.52$$

Problem 05.004.b - case where normal stresses are equal to principal stresses

$\sigma_A = 31$ kpsi, and $\sigma_B = -31$ kpsi

The factor of safety from the maximum-shear-stress theory is 0.76 ✓, and the factor of safety from the distortion-energy theory is 0.77 ✗.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.b - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.b - case where normal stresses are equal to principal stresses

$\sigma_A = 31$ kpsi, and $\sigma_B = -31$ kpsi

The factor of safety from the maximum-shear-stress theory is 0.76 ± 2%, and the factor of safety from the distortion-energy theory is 0.88 ± 2%.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = 31$ kpsi, and $\sigma_B = -31$ kpsi

The ordered principle stresses are $\sigma_1 = 31$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -31$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(31 \text{ kpsi} - (-31 \text{ kpsi}))} = 0.76$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(31 \text{ kpsi})^2 - (31 \text{ kpsi}) \times (-31 \text{ kpsi}) + (-31 \text{ kpsi})^2}} = 0.88$$

Problem 05.004.c - case where normal stresses are equal to principal stresses

$\sigma_A = 31$ kpsi, and $\sigma_B = 16$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.52** ✓, and the factor of safety from distortion-energy theory is **4.44** ✗.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.c - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.c - case where normal stresses are equal to principal stresses

$\sigma_A = 31$ kpsi, and $\sigma_B = 16$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.52 ± 2%**, and the factor of safety from distortion-energy theory is **1.75 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = 31$ kpsi, and $\sigma_B = 16$ kpsi

The ordered principle stresses are $\sigma_1 = 31$ kpsi, $\sigma_2 = 16$ kpsi, $\sigma_3 = 0$.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(31 \text{ kpsi} - 0)} = 1.52$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(31 \text{ kpsi})^2 - (31 \text{ kpsi}) \times (16 \text{ kpsi}) + (16 \text{ kpsi})^2}} = 1.75$$

Problem 05.004.d - case where normal stresses are equal to principal stresses

$\sigma_A = -31$ kpsi, and $\sigma_B = -16$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.52** ✓, and the factor of safety from the distortion-energy theory is **1.75** ✓.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.d - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.d - case where normal stresses are equal to principal stresses

$\sigma_A = -31$ kpsi, and $\sigma_B = -16$ kpsi

The factor of safety from the maximum-shear-stress theory is **1.52 ± 2%**, and the factor of safety from the distortion-energy theory is **1.75 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi.

$\sigma_A = -31$ kpsi, and $\sigma_B = -16$ kpsi

The ordered principle stresses are $\sigma_1 = 0$, $\sigma_2 = -16$ kpsi, $\sigma_3 = -31$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(0 - (-31 \text{ kpsi}))} = 1.52$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-31 \text{ kpsi})^2 - (-31 \text{ kpsi}) \times (-16 \text{ kpsi}) + (-16 \text{ kpsi})^2}} = 1.75$$

10.

Award: 5 out of 10.00 points

Problem 05.004.e - case where normal stresses are equal to principal stresses

 $\sigma_A = -46$ kpsi, and $\sigma_B = 11$ kpsiThe factor of safety from the maximum-shear-stress theory is **0.83** ✓, and the factor of safety from the distortion-energy theory is **6.05** ✗.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.004.e - case where normal stresses are equal to principal stresses

Section: 05.04 Maximum-Shear-Stress Theory for Ductile Materials

Problem 05.004.e - case where normal stresses are equal to principal stresses

 $\sigma_A = -46$ kpsi, and $\sigma_B = 11$ kpsiThe factor of safety from the maximum-shear-stress theory is **0.82 ± 2%**, and the factor of safety from the distortion-energy theory is **0.9 ± 2%**.

Explanation:

From Table A-20, $S_y = 47$ kpsi. $\sigma_A = -46$ kpsi, and $\sigma_B = 11$ kpsiThe ordered principle stresses are $\sigma_1 = 11$ kpsi, $\sigma_2 = 0$, $\sigma_3 = -46$ kpsi.

For the maximum-shear-stress theory,

$$n = \frac{S_y}{(\sigma_1 - \sigma_3)} = \frac{47 \text{ kpsi}}{(11 \text{ kpsi} - (-46 \text{ kpsi}))} = 0.82$$

For the distortion-energy theory,

$$n = \frac{S_y}{\sigma'} = \frac{S_y}{\sqrt{\sigma_A^2 - \sigma_A \sigma_B + \sigma_B^2}} = \frac{47 \text{ kpsi}}{\sqrt{(-46 \text{ kpsi})^2 - (-46 \text{ kpsi}) \times (11 \text{ kpsi}) + (11 \text{ kpsi})^2}} = 0.9$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 10

TOPICS COVERED IN HOMEWORK 10

Chapter 5. Failure Theory: Problems 5-36, 5-49, 5-60, 5-64

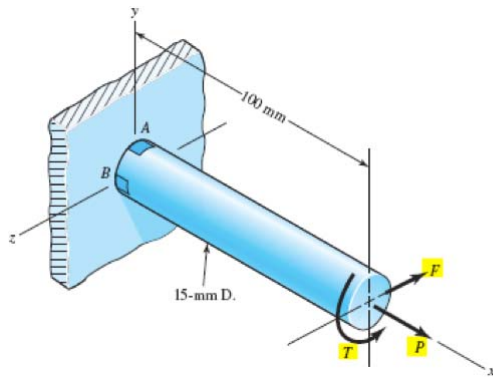
**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

This problem illustrates that the factor of safety for a machine element depends on the particular point selected for analysis. Here you are to compute factors of safety, based upon the distortion-energy theory, for stress elements at A and B of the member shown in the figure. This bar is made of AISI 1006 cold-drawn steel and is loaded by the forces $F = 0.55 \text{ kN}$, $P = 4 \text{ kN}$, and $T = 25 \text{ N-m}$. Given: $S_y = 280 \text{ MPa}$.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

Difficulty: Hard

1.

Award: 5.00 points

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is MPa.

Explanation:

There is no bending stress due to F at A because point A lies on the neutral axis of F .

The axial stress at A is

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

The value of the axial stress at point A is 22.636 MPa.

References

Numeric Response

Problem 05.036.a - calculation of axial stress

Difficulty: Medium

2. Award: 5.00 points

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is $33.576 \pm 2\%$ MPa.**Explanation:**

There are two sources of shear stress at point A.

Transverse shear has a value of

$$\tau_{xz_1} = \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{A} \right)$$

and shear due to torsion has a value of

$$\tau_{xz_2} = \frac{T}{J} = \frac{25 \text{ N} \cdot \text{m} \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right) \times 32}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4}$$

The total shear stress at point A is

$$\tau_{xz} = \tau_{xz_2} + \tau_{xz_1} = \left(\frac{16T}{\pi d^3} + \frac{4F}{3A} \right)$$

$$\tau_{xz} = \tau_{xz_2} + \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N} \cdot \text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} + \frac{4 \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

The value of the shear stress at point A is 33.576 MPa .

References**Numeric Response**

Problem 05.036.b - calculation of shear stress Difficulty: Medium

3. Award: 5.00 points

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is $62.4049 \pm 2\%$ MPa.**Explanation:**

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

The von Mises stress at point A is 62.4049 MPa .

References**Numeric Response**

Problem 05.036.c - calculation of Von Mises stress Difficulty: Medium

4.

Award: 5.00 points

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is .

Explanation:

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16T}{\pi d^3} - \frac{4F}{3A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N}\cdot\text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

$$n_A = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{62.405 \text{ MPa}}$$

$$n_A = 4.487$$

The factor of safety, using DET, at point A is 4.487.

References

Numeric Response

Problem 05.036.d - calculation of factor of safety

Difficulty: Medium

5.

Award: 5.00 points

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point B.

The value of the axial stress at point B is MPa.**Explanation:** P causes point B to be in tension due to the bending moment it causes and P puts the entire cross section in tension.

The axial stress at B is

$$\sigma_x = \frac{P}{A} + \frac{Mc}{I}$$

$$c = \frac{d}{2} = \frac{0.015 \text{ m}}{2}$$

$$c = 7.5 \times 10^{-3} \text{ m}$$

$$I = \frac{\pi}{64} d^4$$

$$I = \frac{\pi}{64} (0.015 \text{ m})^4$$

$$I = 2.485 \times 10^{-9} \text{ m}^4$$

$$M = Pl = 0.55 \text{ kN} \times 0.1 \text{ m}$$

$$M = 0.055 \text{ kN} \cdot \text{m}$$

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{A} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) c}{I} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) \times 7.5 \times 10^{-3} \text{ m}}{2.485 \times 10^{-9} \text{ m}^4} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = 188.629 \text{ MPa}$$

The value of the axial stress at point B is 188.629 MPa.

References**Numeric Response**

Problem 05.036.e - calculation of axial stress

Difficulty: Medium

6.

Award: 5.00 points

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is MPa.

Explanation:

The only source of shear stress at point B is due to torsion:

$$\tau_{xy} = \frac{T r}{J} = \frac{25 \text{ N}\cdot\text{m} \times 32 \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right)}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4} \times \frac{1}{10^6} \text{ MPa}$$

$$\tau_{xy} = 37.726 \text{ MPa}$$

The value of the shear stress at point B is 37.726 MPa .

[References](#)**Numeric Response**

Problem 05.036.f - calculation of shear stress Difficulty: Medium

7.

Award: 5.00 points

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is $198.159 \pm 2\%$ MPa.**Explanation:**The principal stresses are (where S_2 is 0.0)

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_1 = \frac{188.63 \text{ MPa}}{2} + \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (33.58 \text{ MPa})^2}$$

$$\sigma_1 = 188.629 \text{ MPa}$$

$$\sigma_3 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_3 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_3 = \frac{188.63 \text{ MPa}}{2} - \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.7256 \text{ MPa})^2}$$

$$\sigma_3 = -7.265269 \text{ MPa}$$

The von Mises stress at point B is

$$\sigma' = \sqrt{\sigma_1^2 + \sigma_3^2 - \sigma_1\sigma_3} \text{ MPa}$$

$$\sigma' = \sqrt{(188.629 \text{ MPa})^2 + (-7.265269 \text{ MPa})^2 - (188.629 \text{ MPa})(-7.265269 \text{ MPa})}$$

$$\sigma' = 198.159 \text{ MPa}$$

The value of the von Mises stress at point B is 198.159 MPa.

References**Numeric Response**

Problem 05.036.g - calculation of Von Mises stress

Difficulty: Medium

8.

Award: 5.00 points

Problem 05.036.h - calculation of factor of safety

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is .

Explanation:

$$n_B = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{198.159 \text{ MPa}}$$

$$n_B = 1.413$$

The factor of safety, using DET, at point B is 1.413.

References

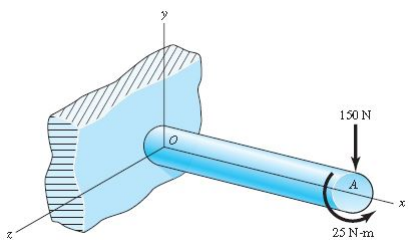
Numeric Response

Problem 05.036.h - calculation of factor of safety

Difficulty: Medium

Problem 05.049

Cantilevered rod OA is 0.5 m long and made from AISI 1015 hot-rolled steel. Constant force and torque are applied as shown.



References

Section Break

Problem 05.049

Difficulty: Medium

9.

Award: 10.00 points

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is $\boxed{20.3894 \pm 2\%}$ mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 190$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau_{\max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$\tau_{\max} = \sqrt{\left(\frac{1200 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2 + \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2}$$

$$\tau_{\max} = \frac{402.63 \text{ N} \cdot \text{m}}{d^3} = \frac{S_y}{2n} = \frac{190(10^6) \text{ Pa}}{2(2)}$$

$$d = \left(\frac{4(402.63 \text{ N} \cdot \text{m})}{190(10^6) \text{ Pa}} \right)^{1/3} = 0.0204 \text{ m} = 20.3894 \text{ mm}$$

The minimum diameter is 20.3894 mm.

References**Numeric Response**

Problem 05.049.a

Difficulty: Medium

10. Award: 10.00 points

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 190$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\sigma' = (\sigma^2 + 3\tau^2)^{1/2}$$

$$\sigma' = \left(\left(\frac{2400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 + 3 \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 \right)^{1/2} = \frac{795.14 \text{ N} \cdot \text{m}}{d^3} = \frac{S_y}{n_y} = \frac{190(10^6) \text{ Pa}}{2}$$

$$d = \left(\frac{2(795.14 \text{ N} \cdot \text{m})}{190(10^6) \text{ Pa}} \right)^{1/3} = 0.0203 \text{ m} = 20.3036 \text{ mm}$$

The minimum diameter is 20.3036 mm.

References**Numeric Response**

Problem 05.049.b

Difficulty: Medium

11.

Award: 20.00 points

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 25$ ksi, $\sigma_2 = -2.0506$ ksi, and $\tau_{\max} = 13.4972$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is $2.0004 \pm 2\%$.

The minimum factor of safety for yielding using distortion-energy theory is $2.0701 \pm 2\%$.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$$\sigma_1 = 25 \text{ ksi}$$

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (25 \text{ ksi}) - 2(13.4972 \text{ ksi})$$

$$\sigma_3 = -1.9943 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(25 \text{ ksi}) - (-1.9943 \text{ ksi})}$$

$$n = 2.0004$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2\right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((25 \text{ ksi})^2 - (25 \text{ ksi})(-2.0506 \text{ ksi}) + (-2.0506 \text{ ksi})^2\right)^{1/2}}$$

$$n = 2.0701$$

The minimum factor of safety for yielding using maximum-shear-stress theory is 2.0004.

The minimum factor of safety for yielding using distortion-energy theory is 2.0701.

References**Numeric Response**

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Difficulty: Medium

12. Award: 20.00 points

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 15$ ksi, $\sigma_2 = -2.8963$ ksi, and $\tau_{\max} = 8.9481$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is $3.0174 \pm 2\%$.

The minimum factor of safety for yielding using distortion-energy theory is $3.2455 \pm 2\%$.

Explanation:**Maximum-Shear-Stress Theory**

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$$\sigma_1 = 15 \text{ ksi}$$

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (15 \text{ ksi}) - 2(8.9481 \text{ ksi})$$

$$\sigma_3 = -2.8962 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(15 \text{ ksi}) - (-2.8962 \text{ ksi})}$$

$$n = 3.0174$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2\right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((15 \text{ ksi})^2 - (15 \text{ ksi})(-2.8963 \text{ ksi}) + (-2.8963 \text{ ksi})^2\right)^{1/2}}$$

$$n = 3.2455$$

The minimum factor of safety for yielding using maximum-shear-stress theory is 3.0174.

The minimum factor of safety for yielding using distortion-energy theory is 3.2455.

References**Numeric Response**

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Difficulty: Medium

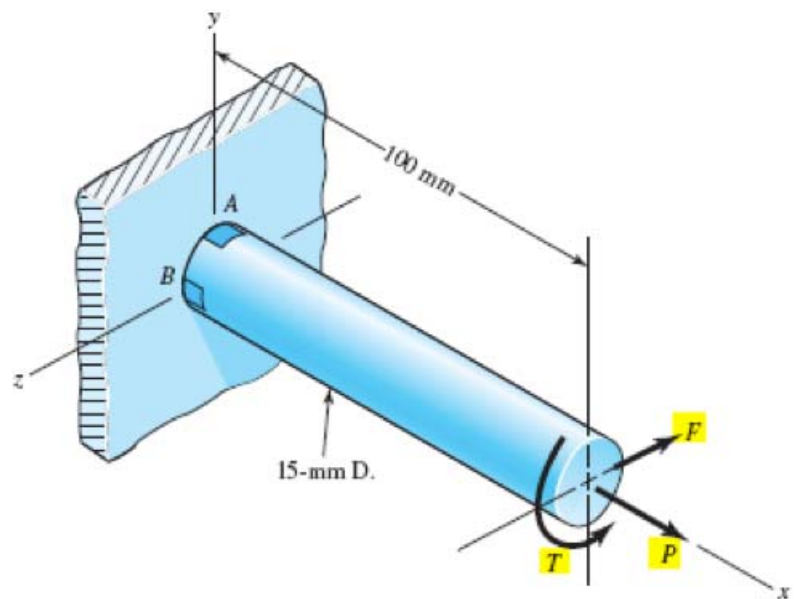
ABOVE AVERAGE

Score: 100/100 Points 100 %

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

This problem illustrates that the factor of safety for a machine element depends on the particular point selected for analysis. Here you are to compute factors of safety, based upon the distortion-energy theory, for stress elements at A and B of the member shown in the figure. This bar is made of AISI 1006 cold-drawn steel and is loaded by the forces $F = 0.55 \text{ kN}$, $P = 4 \text{ kN}$, and $T = 25 \text{ N}\cdot\text{m}$. Given: $S_y = 280 \text{ MPa}$.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

Section: 05.05 Distortion-Energy Theory for Ductile Materials

1.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is **22.63**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.a - calculation of axial stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is **22.636 ± 2%** MPa.

Explanation:

There is no bending stress due to F at A because point A lies on the neutral axis of F .

The axial stress at A is

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

2.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is **33.5**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.b - calculation of shear stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is **33.576 ± 2%** MPa.

Explanation:

There are two sources of shear stress at point A.

Transverse shear has a value of

$$\tau_{xz_1} = \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{A} \right)$$

and shear due to torsion has a value of

$$\tau_{xz_2} = \frac{T r}{J} = \frac{25 \text{ N} \cdot \text{m} \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right) \times 32}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4}$$

The total shear stress at point A is

$$\tau_{xz} = \tau_{xz_2} + \tau_{xz_1} = \left(\frac{16 T}{\pi d^3} + \frac{4 F}{3 A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N}\cdot\text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \frac{1}{(10^6)} \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

3.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is **62.4**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.c - calculation of Von Mises stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is **62.4049 ± 2%** MPa.

Explanation:

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

4.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is **4.487** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.d - calculation of factor of safety

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is **4.487 ± 2%**.

Explanation:

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16T}{\pi d^3} - \frac{4F}{3A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N} \cdot \text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

$$n_A = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{62.405 \text{ MPa}}$$

$$n_A = 4.487$$

5.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point *B*.

The value of the axial stress at point B is **188.629**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.e - calculation of axial stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point *B*.

The value of the axial stress at point B is **188.629 ± 2%** MPa.

Explanation:

F causes point B to be in tension due to the bending moment it causes and *P* puts the entire cross section in tension.

The axial stress at B is

$$\sigma_x = \frac{P}{A} + \frac{Mc}{I}$$

$$c = \frac{d}{2} = \frac{0.015 \text{ m}}{2}$$

$$c = 7.5 \times 10^{-3} \text{ m}$$

$$I = \frac{\pi}{64} d^4$$

$$I = \frac{\pi}{64} (0.015 \text{ m})^4$$

$$I = 2.485 \times 10^{-9} \text{ m}^4$$

$$M = F \ell = 0.55 \text{ kN} \times 0.1 \text{ m}$$

$$M = 0.055 \text{ kN} \cdot \text{m}$$

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{A} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) c}{I} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) \times 7.5 \times 10^{-3} \text{ m}}{2.485 \times 10^{-9} \text{ m}} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = 188.629 \text{ MPa}$$

6.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is **37.726**  MPa.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.f - calculation of shear stress Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is **37.726 ± 2%** MPa.

Explanation:

The only source of shear stress at point B is due to torsion:

$$\tau_{xy} = \frac{T_r}{J} = \frac{25 \text{ N}\cdot\text{m} \times 32 \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right)}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4} \times \frac{1}{10^6} \text{ MPa}$$

$$\tau_{xy} = 37.726 \text{ MPa}$$

7.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is 199.626  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.g - calculation of Von Mises stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is 199.626 ± 2% MPa.

Explanation:

The principal stresses are (where S_2 is 0.0)

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_1 = \frac{188.63 \text{ MPa}}{2} + \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.73 \text{ MPa})^2}$$

$$\sigma_1 = 195.894 \text{ MPa}$$

$$\sigma_3 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_3 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_3 = \frac{188.63 \text{ MPa}}{2} - \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.7256 \text{ MPa})^2}$$

$$\sigma_3 = -7.265269 \text{ MPa}$$

The von Mises stress at point B is

$$\sigma' = \sqrt{\sigma_1^2 + \sigma_3^2 - \sigma_1 \sigma_3} \text{ MPa}$$

$$\sigma' = \sqrt{(195.894 \text{ MPa})^2 + (-7.265269 \text{ MPa})^2 - (195.894 \text{ MPa})(-7.265269 \text{ MPa})}$$

$$\sigma' = 199.626 \text{ MPa}$$

8.

Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt**Problem 05.036.h - calculation of factor of safety**

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is **1.403** .

References**Numeric Response**

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.h - calculation of factor of safety

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.h - calculation of factor of safety

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is **1.403 ± 2%**.

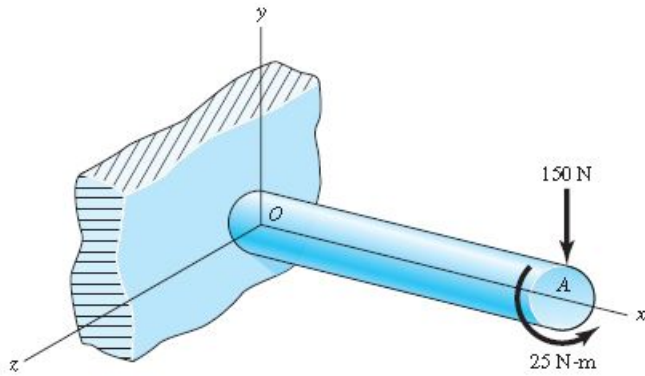
Explanation:

$$n_B = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{199.626 \text{ MPa}}$$

$$n_B = 1.403$$

Problem 05.049

Cantilevered rod OA is 0.5 m long and made from AISI 1020 hot-rolled steel. Constant force and torque are applied as shown.



References

Section Break

Chapter: 05 Failures Resulting
from Static Loading

Problem 05.049

Section: 05.03 Failure Theories

9.

Award: 10 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is **19.72**  mm.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.049.a Section: 05.03 Failure Theories

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is **19.7204 ± 2%** mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 210$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau_{\max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$\tau_{\max} = \sqrt{\left(\frac{1200 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2 + \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2}$$

$$\tau_{\max} = \frac{402.63 \text{ N}\cdot\text{m}}{d^3} = \frac{S_y}{2n} = \frac{210 \left(10^6 \right) \text{ Pa}}{2(2)}$$

$$d = \left(\frac{4(402.63 \text{ N}\cdot\text{m})}{210 \left(10^6 \right) \text{ Pa}} \right)^{1/3} = 0.0197 \text{ m} = 19.7204 \text{ mm}$$

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is **19.63**  mm.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.049.b Section: 05.03 Failure Theories

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is **19.6374 ± 2%** mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 210$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\sigma' = (\sigma^2 + 3\tau^2)^{1/2}$$

$$\sigma' = \left(\left(\frac{2400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 + 3 \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 \right)^{1/2} = \frac{795.14 \text{ N} \cdot \text{m}}{d^3} = \frac{S_y}{n_y} = \frac{210 (10^6) \text{ Pa}}{2}$$

$$d = \left(\frac{2(795.14 \text{ N} \cdot \text{m})}{210 (10^6) \text{ Pa}} \right)^{1/3} = 0.0196 \text{ m} = 19.6374 \text{ mm}$$

11.

Award: 20 out of 20.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 15$ ksi, $\sigma_2 = -1.2303$ ksi, and $\tau_{\max} = 8.0983$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **3.334** .

The minimum factor of safety for yielding using distortion-energy theory is **3.4502** .

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 15$ ksi, $\sigma_2 = -1.2303$ ksi, and $\tau_{\max} = 8.0983$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **3.334 ± 2%**.

The minimum factor of safety for yielding using distortion-energy theory is **3.4502 ± 2%**.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$\sigma_1 = 15$ ksi

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (15 \text{ ksi}) - 2(8.0983 \text{ ksi})$$

$$\sigma_3 = -1.1966 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(15 \text{ ksi}) - (-1.1966 \text{ ksi})}$$

$$n = 3.334$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((15 \text{ ksi})^2 - (15 \text{ ksi})(-1.2303 \text{ ksi}) + (-1.2303 \text{ ksi})^2 \right)^{1/2}}$$

$$n = 3.4502$$

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 25$ ksi, $\sigma_2 = -4.8271$ ksi, and $\tau_{\max} = 14.9135$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **1.8104** .

The minimum factor of safety for yielding using distortion-energy theory is **1.9473** .

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 25$ ksi, $\sigma_2 = -4.8271$ ksi, and $\tau_{\max} = 14.9135$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **1.8104 ± 2%**.

The minimum factor of safety for yielding using distortion-energy theory is **1.9473 ± 2%**.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$\sigma_1 = 25$ ksi

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (25 \text{ ksi}) - 2(14.9135 \text{ ksi})$$

$$\sigma_3 = -4.8271 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(25 \text{ ksi}) - (-4.8271 \text{ ksi})}$$

$$n = 1.8104$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((25 \text{ ksi})^2 - (25 \text{ ksi})(-4.8271 \text{ ksi}) + (-4.8271 \text{ ksi})^2 \right)^{1/2}}$$

$$n = 1.9473$$

AVERAGE

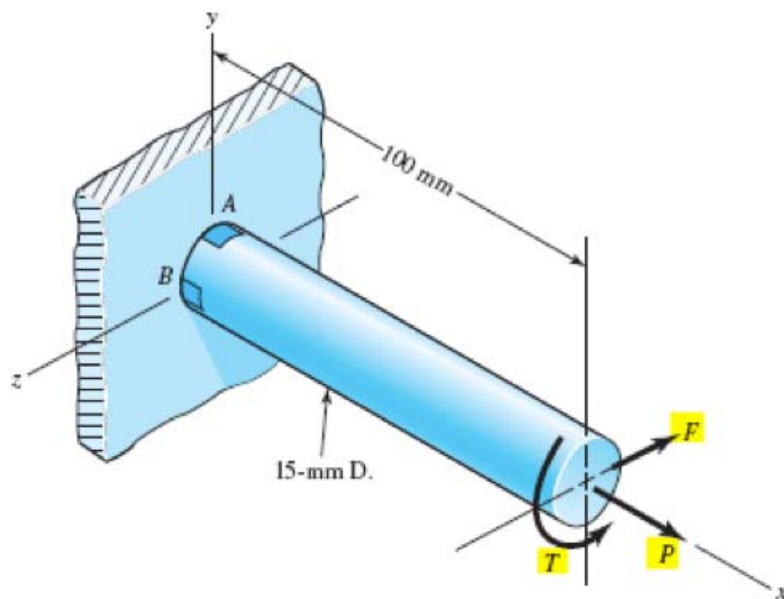
Nylander, Chris

Score: 90/100 Points 90 %

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

This problem illustrates that the factor of safety for a machine element depends on the particular point selected for analysis. Here you are to compute factors of safety, based upon the distortion-energy theory, for stress elements at A and B of the member shown in the figure. This bar is made of AISI 1006 cold-drawn steel and is loaded by the forces $F = 0.55$ kN, $P = 4$ kN, and $T = 25$ N·m. Given: $S_y = 280$ MPa.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

Section: 05.05 Distortion-Energy Theory for Ductile Materials

1.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is **22.63**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.a - calculation of axial stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is **22.636 ± 2%** MPa.

Explanation:

There is no bending stress due to F at A because point A lies on the neutral axis of F .

The axial stress at A is

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

2.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is **33.576**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.b - calculation of shear stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is **33.576 ± 2%** MPa.

Explanation:

There are two sources of shear stress at point A.

Transverse shear has a value of

$$\tau_{xz_1} = \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{A} \right)$$

and shear due to torsion has a value of

$$\tau_{xz_2} = \frac{Tr}{J} = \frac{25 \text{ N} \cdot \text{m} \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right) \times 32}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4}$$

The total shear stress at point A is

$$\tau_{xz} = \tau_{xz_2} + \tau_{xz_1} = \left(\frac{16T}{\pi d^3} + \frac{4F}{3A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N}\cdot\text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \frac{1}{(10^6)} \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

3.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is **62.4049**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.c - calculation of Von Mises stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is **62.4049 ± 2%** MPa.

Explanation:

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

4.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is **4.487** .

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.d - calculation of factor of safety

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is **4.487 ± 2%**.

Explanation:

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16T}{\pi d^3} - \frac{4F}{3A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N} \cdot \text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

$$n_A = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{62.405 \text{ MPa}}$$

$$n_A = 4.487$$

5.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point *B*.

The value of the axial stress at point B is **188.629**  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.e - calculation of axial stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point *B*.

The value of the axial stress at point B is **188.629 ± 2%** MPa.

Explanation:

F causes point B to be in tension due to the bending moment it causes and *P* puts the entire cross section in tension.

The axial stress at B is

$$\sigma_x = \frac{P}{A} + \frac{Mc}{I}$$

$$c = \frac{d}{2} = \frac{0.015 \text{ m}}{2}$$

$$c = 7.5 \times 10^{-3} \text{ m}$$

$$I = \frac{\pi}{64} d^4$$

$$I = \frac{\pi}{64} (0.015 \text{ m})^4$$

$$I = 2.485 \times 10^{-9} \text{ m}^4$$

$$M = Fd = 0.55 \text{ kN} \times 0.1 \text{ m}$$

$$M = 0.055 \text{ kN}\cdot\text{m}$$

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{A} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) c}{I} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) \times 7.5 \times 10^{-3} \text{ m}}{2.485 \times 10^{-9} \text{ m}} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = 188.629 \text{ MPa}$$

6.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is **37.726**  MPa.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.f - calculation of shear stress Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is **37.726 ± 2%** MPa.

Explanation:

The only source of shear stress at point B is due to torsion:

$$\tau_{xy} = \frac{T r}{J} = \frac{25 \text{ N}\cdot\text{m} \times 32 \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right)}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4} \times \frac{1}{10^6} \text{ MPa}$$

$$\tau_{xy} = 37.726 \text{ MPa}$$

7.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is 199.626  MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.g - calculation of Von Mises stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is 199.626 ± 2% MPa.

Explanation:

The principal stresses are (where S_2 is 0.0)

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_1 = \frac{188.63 \text{ MPa}}{2} + \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.73 \text{ MPa})^2}$$

$$\sigma_1 = 195.894 \text{ MPa}$$

$$\sigma_3 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_3 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_3 = \frac{188.63 \text{ MPa}}{2} - \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.7256 \text{ MPa})^2}$$

$$\sigma_3 = -7.265269 \text{ MPa}$$

The von Mises stress at point B is

$$\sigma' = \sqrt{\sigma_1^2 + \sigma_3^2 - \sigma_1 \sigma_3} \text{ MPa}$$

$$\sigma' = \sqrt{(195.894 \text{ MPa})^2 + (-7.265269 \text{ MPa})^2 - (195.894 \text{ MPa})(-7.265269 \text{ MPa})}$$

$$\sigma' = 199.626 \text{ MPa}$$

8.

Award: 4.50 out of 5.00 points

 You did not receive full credit for this question in a previous attempt**Problem 05.036.h - calculation of factor of safety**

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is **1.403** .

References**Numeric Response**

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.h - calculation of factor of safety

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.h - calculation of factor of safety

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is **1.403 ± 2%**.

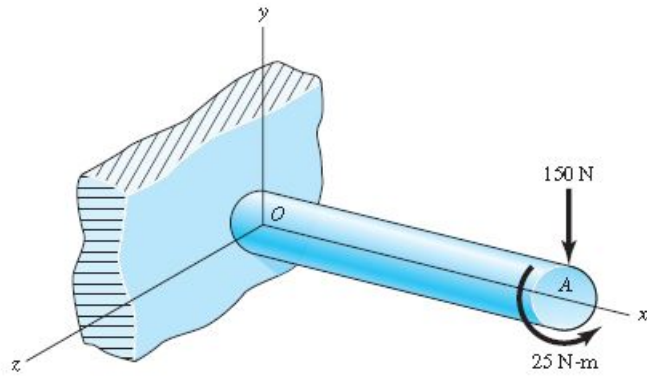
Explanation:

$$n_B = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{199.626 \text{ MPa}}$$

$$n_B = 1.403$$

Problem 05.049

Cantilevered rod OA is 0.5 m long and made from AISI 1050 hot-rolled steel. Constant force and torque are applied as shown.



References

Section Break

Chapter: 05 Failures Resulting
from Static Loading

Problem 05.049

Section: 05.03 Failure Theories

9.

Award: 9 out of 10.00 points

 You did not receive full credit for this question in a previous attempt

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is **16.7943**  mm.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.049.a Section: 05.03 Failure Theories

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is **16.7943 ± 2%** mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 340$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau_{\max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$\tau_{\max} = \sqrt{\left(\frac{1200 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2 + \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2}$$

$$\tau_{\max} = \frac{402.63 \text{ N}\cdot\text{m}}{d^3} = \frac{S_y}{2n} = \frac{340 \left(10^6\right) \text{ Pa}}{2(2)}$$

$$d = \left(\frac{4(402.63 \text{ N}\cdot\text{m})}{340 \left(10^6\right) \text{ Pa}} \right)^{1/3} = 0.0168 \text{ m} = 16.7943 \text{ mm}$$

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is 16.7237  mm.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.049.b Section: 05.03 Failure Theories

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is 16.7237 ± 2% mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 340$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\sigma' = (\sigma^2 + 3\tau^2)^{1/2}$$

$$\sigma' = \left(\left(\frac{2400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 + 3 \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 \right)^{1/2} = \frac{795.14 \text{ N} \cdot \text{m}}{d^3} = \frac{S_y}{n_y} = \frac{340 (10^6) \text{ Pa}}{2}$$

$$d = \left(\frac{2(795.14 \text{ N} \cdot \text{m})}{340 (10^6) \text{ Pa}} \right)^{1/3} = 0.0167 \text{ m} = 16.7237 \text{ mm}$$

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 15$ ksi, $\sigma_2 = -1.2303$ ksi, and $\tau_{\max} = 8.0983$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **3.334** .

The minimum factor of safety for yielding using distortion-energy theory is **3.4502** .

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 15$ ksi, $\sigma_2 = -1.2303$ ksi, and $\tau_{\max} = 8.0983$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **3.334 ± 2%**.

The minimum factor of safety for yielding using distortion-energy theory is **3.4502 ± 2%**.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$\sigma_1 = 15$ ksi

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (15 \text{ ksi}) - 2(8.0983 \text{ ksi})$$

$$\sigma_3 = -1.1966 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(15 \text{ ksi}) - (-1.1966 \text{ ksi})}$$

$$n = 3.334$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1 \sigma_2 + \sigma_2^2 \right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((15 \text{ ksi})^2 - (15 \text{ ksi})(-1.2303 \text{ ksi}) + (-1.2303 \text{ ksi})^2 \right)^{1/2}}$$

$$n = 3.4502$$

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 25$ ksi, $\sigma_2 = -4.8271$ ksi, and $\tau_{\max} = 14.9135$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **1.8104** .

The minimum factor of safety for yielding using distortion-energy theory is **1.9473** .

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 25$ ksi, $\sigma_2 = -4.8271$ ksi, and $\tau_{\max} = 14.9135$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is **1.8104 ± 2%**.

The minimum factor of safety for yielding using distortion-energy theory is **1.9473 ± 2%**.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$\sigma_1 = 25$ ksi

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (25 \text{ ksi}) - 2(14.9135 \text{ ksi})$$

$$\sigma_3 = -4.8271 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(25 \text{ ksi}) - (-4.8271 \text{ ksi})}$$

$$n = 1.8104$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((25 \text{ ksi})^2 - (25 \text{ ksi})(-4.8271 \text{ ksi}) + (-4.8271 \text{ ksi})^2 \right)^{1/2}}$$

$$n = 1.9473$$

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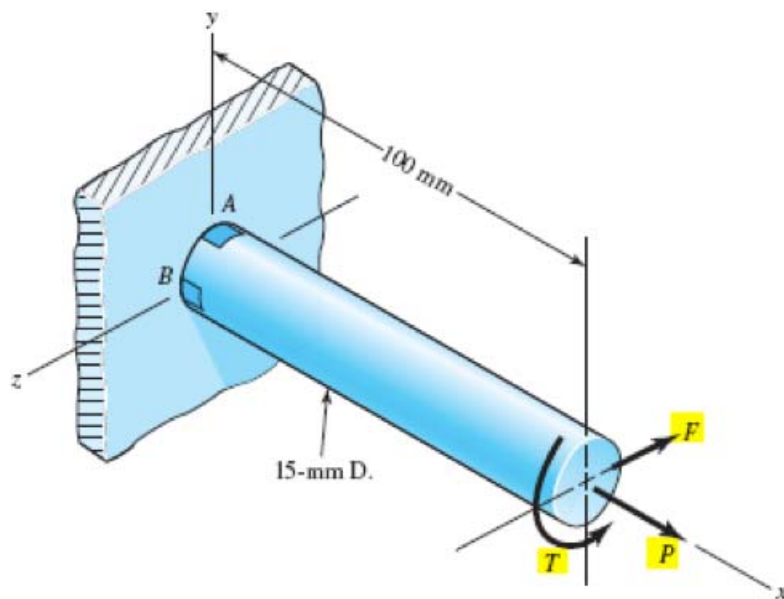
Wright, Winsont

Score: 30/100 Points 30 %

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

This problem illustrates that the factor of safety for a machine element depends on the particular point selected for analysis. Here you are to compute factors of safety, based upon the distortion-energy theory, for stress elements at A and B of the member shown in the figure. This bar is made of AISI 1006 cold-drawn steel and is loaded by the forces $F = 0.55$ kN, $P = 4$ kN, and $T = 25$ N·m. Given: $S_y = 280$ MPa.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.



References

Section Break

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036 - Determination of normal and shear stresses, and resulting factor of safety using Distortion Energy Theory for Ductile Materials - DEPENDENT MULTI-PART PROBLEM – ASSIGN ALL PARTS

Section: 05.05 Distortion-Energy Theory for Ductile Materials

1.

Award: 5 out of 5.00 points

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is **22.63** ✓ MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.a - calculation of axial stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.a - calculation of axial stress

What is the value of the axial stress at point A?

The value of the axial stress at point A is **22.636 ± 2%** MPa.

Explanation:

There is no bending stress due to F at A because point A lies on the neutral axis of F .

The axial stress at A is

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

2.

Award: 0 out of 5.00 points

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is **41.88** ✖ MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.b - calculation of shear stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.b - calculation of shear stress

What is the value of the shear stress at point A?

The value of the shear stress at point A is **33.576 ± 2%** MPa.

Explanation:

There are two sources of shear stress at point A.

Transverse shear has a value of

$$\tau_{xz_1} = \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{A} \right)$$

and shear due to torsion has a value of

$$\tau_{xz_2} = \frac{T r}{J} = \frac{25 \text{ N} \cdot \text{m} \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right) \times 32}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4}$$

The total shear stress at point A is

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 T}{\pi d^3} - \frac{4 F}{3 A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N}\cdot\text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \frac{1}{(10^6)} \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

3.

Award: 0 out of 5.00 points

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is **69.14** ✖ MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.c - calculation of Von Mises stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.c - calculation of Von Mises stress

Determine the value of the von Mises stress at point A.

The von Mises stress at point A is **62.4049 ± 2%** MPa.

Explanation:

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

4.

Award: 0 out of 5.00 points

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is **3.68** ✖.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.d - calculation of factor of safety

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.d - calculation of factor of safety

Determine the factor of safety, using DET, at point A.

The factor of safety, using DET, at point A is **4.487 ± 2%**.

Explanation:

$$\sigma_x = \frac{P}{A}$$

$$\sigma_x = \frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \left(\frac{1}{10^6} \right) \text{ MPa}$$

$$\sigma_x = 22.636 \text{ MPa}$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16T}{\pi d^3} - \frac{4F}{3A} \right)$$

$$\tau_{xz} = \tau_{xz_2} - \tau_{xz_1} = \left(\frac{16 \times 25 \text{ N} \cdot \text{m}}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^3} - \frac{4}{3} \left(\frac{0.55 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} \right) \right) \frac{1}{(10^6)} \text{ MPa}$$

$$\tau_{xz} = 33.576 \text{ MPa}$$

$$\sigma' = \sqrt{\sigma_x^2 + 3\tau_{xy}^2}$$

$$\sigma' = \sqrt{(22.636 \text{ MPa})^2 + 3(33.5757 \text{ MPa})^2}$$

$$\sigma' = 62.4049 \text{ MPa}$$

$$n_A = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{62.405 \text{ MPa}}$$

$$n_A = 4.487$$

5.

Award: 5 out of 5.00 points

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point *B*.The value of the axial stress at point B is **188.62** ✓ MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.e - calculation of axial stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.e - calculation of axial stress

Determine the value of the axial stress at point *B*.The value of the axial stress at point B is **188.629 ± 2%** MPa.

Explanation:

F causes point B to be in tension due to the bending moment it causes and *P* puts the entire cross section in tension.

The axial stress at B is

$$\sigma_x = \frac{P}{A} + \frac{Mc}{I}$$

$$c = \frac{d}{2} = \frac{0.015 \text{ m}}{2}$$

$$c = 7.5 \times 10^{-3} \text{ m}$$

$$I = \frac{\pi}{64} d^4$$

$$I = \frac{\pi}{64} (0.015 \text{ m})^4$$

$$I = 2.485 \times 10^{-9} \text{ m}^4$$

$$M = F \ell = 0.55 \text{ kN} \times 0.1 \text{ m}$$

$$M = 0.055 \text{ kN}\cdot\text{m}$$

$$A = \frac{\pi}{4} \left(\left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right) \frac{\text{m}}{1} \right)^2 = 1.7671 \times 10^{-4} \text{ m}^2$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{A} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) c}{I} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = \left(\frac{4 \text{ kN} \times 1000}{1.7671 \times 10^{-4} \text{ m}^2} + \frac{(0.55 \text{ kN} \times 1000 \times 0.1) \times 7.5 \times 10^{-3} \text{ m}}{2.485 \times 10^{-9} \text{ m}} \right) \frac{1}{(10)^6} \text{ MPa}$$

$$\sigma_x = 188.629 \text{ MPa}$$

6.

Award: 0 out of 5.00 points

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is 37..72 ✖ MPa.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.f - calculation of shear stress Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.f - calculation of shear stress

Determine the value of the shear stress at point B.

The value of the shear stress at point B is 37.726 ± 2% MPa.

Explanation:

The only source of shear stress at point B is due to torsion:

$$\tau_{xy} = \frac{T r}{J} = \frac{25 \text{ N}\cdot\text{m} \times 32 \times \left(\frac{15 \text{ mm}}{2000 \text{ mm}} \right)}{\pi \left(\frac{15 \text{ mm}}{1000 \text{ mm}} \right)^4} \times \frac{1}{10^6} \text{ MPa}$$

$$\tau_{xy} = 37.726 \text{ MPa}$$

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is 200 ✓ MPa.

References

Numeric Response

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.g - calculation of Von Mises stress

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.g - calculation of Von Mises stress

Determine the value of the von Mises stress at point B.

The value of the von Mises stress at point B is 199.626 ± 2% MPa.

Explanation:

The principal stresses are (where S_2 is 0.0)

$$\sigma_1 = \frac{\sigma_x + \sigma_y}{2} + \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_1 = \frac{\sigma_x}{2} + \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_1 = \frac{188.63 \text{ MPa}}{2} + \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.73 \text{ MPa})^2}$$

$$\sigma_1 = 195.894 \text{ MPa}$$

$$\sigma_3 = \frac{\sigma_x + \sigma_y}{2} - \sqrt{\left(\frac{\sigma_x - \sigma_y}{2}\right)^2 + \tau_{xy}^2}$$

$$\sigma_3 = \frac{\sigma_x}{2} - \sqrt{\left(\frac{\sigma_x}{2}\right)^2 + \tau_{xy}^2} \text{ MPa}$$

$$\sigma_3 = \frac{188.63 \text{ MPa}}{2} - \sqrt{\left(\frac{188.63 \text{ MPa}}{2}\right)^2 + (37.7256 \text{ MPa})^2}$$

$$\sigma_3 = -7.265269 \text{ MPa}$$

The von Mises stress at point B is

$$\sigma' = \sqrt{\sigma_1^2 + \sigma_3^2 - \sigma_1 \sigma_3} \text{ MPa}$$

$$\sigma' = \sqrt{(195.894 \text{ MPa})^2 + (-7.265269 \text{ MPa})^2 - (195.894 \text{ MPa})(-7.265269 \text{ MPa})}$$

$$\sigma' = 199.626 \text{ MPa}$$

8.

Award: 5 out of 5.00 points

Problem 05.036.h - calculation of factor of safety

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is **1.403** ✓.

References**Numeric Response**

Chapter: 05 Failures Resulting from Static Loading

Problem 05.036.h - calculation of factor of safety

Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.036.h - calculation of factor of safety

Determine the factor of safety, using DET, at point B.

The factor of safety, using DET, at point B is **1.403 ± 2%**.

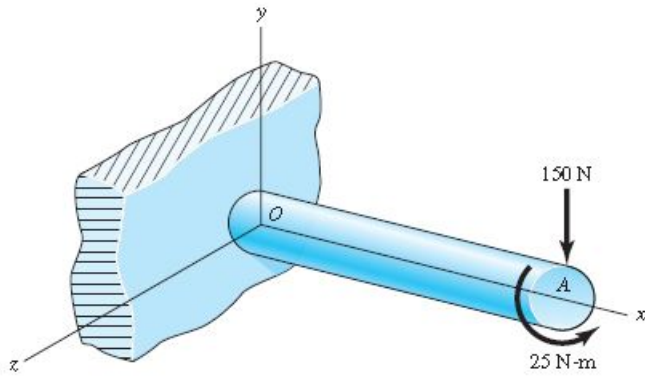
Explanation:

$$n_B = \frac{S_y}{\sigma'} = \frac{280 \text{ MPa}}{199.626 \text{ MPa}}$$

$$n_B = 1.403$$

Problem 05.049

Cantilevered rod OA is 0.5 m long and made from AISI 1018 hot-rolled steel. Constant force and torque are applied as shown.



References

Section Break

Chapter: 05 Failures Resulting
from Static Loading

Problem 05.049

Section: 05.03 Failure Theories

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is **16.3276**  mm.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.049.a Section: 05.03 Failure Theories

Problem 05.049.a

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the maximum-shear-stress failure theory.

The minimum diameter is **19.417 ± 2%** mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 220$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau_{\max} = \sqrt{\left(\frac{\sigma}{2}\right)^2 + \tau^2}$$

$$\tau_{\max} = \sqrt{\left(\frac{1200 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2 + \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3}\right)^2}$$

$$\tau_{\max} = \frac{402.63 \text{ N}\cdot\text{m}}{d^3} = \frac{S_y}{2n} = \frac{220 \left(10^6 \right) \text{ Pa}}{2(2)}$$

$$d = \left(\frac{4(402.63 \text{ N}\cdot\text{m})}{220 \left(10^6 \right) \text{ Pa}} \right)^{1/3} = 0.0194 \text{ m} = 19.4170 \text{ mm}$$

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is **17.34**  mm.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.049.b Section: 05.03 Failure Theories

Problem 05.049.b

Determine the minimum diameter, d , for the rod that will achieve a minimum static factor of safety of 2 using the distortion-energy theory.

The minimum diameter is **19.3353 ± 2%** mm.

Explanation:

Given: $n_y = 2$, $L = 0.5$ m, $F = 150$ N, and $T = 25$ N·m.

From Table A-20, $S_y = 220$ MPa.

$$M_z = FL = 150 \text{ N} \times 0.5 \text{ m} = 75 \text{ N} \cdot \text{m}$$

$$\sigma = \frac{32M_z}{\pi d^3} = \frac{32(75 \text{ N} \cdot \text{m})}{\pi d^3} = \frac{2400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\tau = \frac{16T}{\pi d^3} = \frac{16(25 \text{ N} \cdot \text{m})}{\pi d^3}$$

$$\tau = \frac{400 \text{ N} \cdot \text{m}}{\pi d^3}$$

$$\sigma' = (\sigma^2 + 3\tau^2)^{1/2}$$

$$\sigma' = \left(\left(\frac{2400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 + 3 \left(\frac{400 \text{ N} \cdot \text{m}}{\pi d^3} \right)^2 \right)^{1/2} = \frac{795.14 \text{ N} \cdot \text{m}}{d^3} = \frac{S_y}{n_y} = \frac{220(10^6) \text{ Pa}}{2}$$

$$d = \left(\frac{2(795.14 \text{ N} \cdot \text{m})}{220(10^6) \text{ Pa}} \right)^{1/3} = 0.0193 \text{ m} = 19.3353 \text{ mm}$$

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 20$ ksi, $\sigma_2 = -1.6404$ ksi, and $\tau_{\max} = 10.7977$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is 4.9733 ❌.

The minimum factor of safety for yielding using distortion-energy theory is 2.5732 ✅.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.060- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 20$ ksi, $\sigma_2 = -1.6404$ ksi, and $\tau_{\max} = 10.7977$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is 2.5005 ± 2%.

The minimum factor of safety for yielding using distortion-energy theory is 2.5876 ± 2%.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$\sigma_1 = 20$ ksi

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (20 \text{ ksi}) - 2(10.7977 \text{ ksi})$$

$$\sigma_3 = -1.5955 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(20 \text{ ksi}) - (-1.5955 \text{ ksi})}$$

$$n = 2.5005$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2\right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((20 \text{ ksi})^2 - (20 \text{ ksi})(-1.6404 \text{ ksi}) + (-1.6404 \text{ ksi})^2\right)^{1/2}}$$

$$n = 2.5876$$

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 10$ ksi, $\sigma_2 = -1.9308$ ksi, and $\tau_{\max} = 5.9654$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is 9 ✖.

The minimum factor of safety for yielding using distortion-energy theory is 2.5732 ✖.

References

Numeric Response Chapter: 05 Failures Resulting from Static Loading

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials Section: 05.05 Distortion-Energy Theory for Ductile Materials

Problem 05.064- Determination of factor of safety using Maximum Shear Stress and Distortion Energy Theory for Ductile Materials

Determine the minimum factor of safety for yielding. Use both the maximum-shear-stress theory and the distortion-energy theory and compare the results. The material is 1018 CD steel. $\sigma_1 = 10$ ksi, $\sigma_2 = -1.9308$ ksi, and $\tau_{\max} = 5.9654$ ksi.

The minimum factor of safety for yielding using maximum-shear-stress theory is 4.5261 ± 2%.

The minimum factor of safety for yielding using distortion-energy theory is 4.8683 ± 2%.

Explanation:

Maximum-Shear-Stress Theory

From Table A-20, $S_y = 54$ ksi or 370 MPa.

The state of stress is plane stress. Thus, the three-dimensional principal stresses are

$\sigma_1 = 10$ ksi

$$\sigma_2 = 0$$

From Figure 3-12 and Eq. (5-1),

$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2}$$

$$\sigma_3 = \sigma_1 - 2\tau_{\max}$$

$$\sigma_3 = (10 \text{ ksi}) - 2(5.9654 \text{ ksi})$$

$$\sigma_3 = -1.9308 \text{ ksi}$$

The minimum factor of safety for yielding is found from Eq. (5-3):

$$n = \frac{S_y}{\sigma_1 - \sigma_3}$$

$$n = \frac{54 \text{ ksi}}{(10 \text{ ksi}) - (-1.9308 \text{ ksi})}$$

$$n = 4.5261$$

Distortion-Energy Theory

The minimum factor of safety for yielding is found from Eq. (5-19) with σ' found using Eq. (5-13):

$$n = \frac{S_y}{\sigma'}$$

$$n = \frac{S_y}{\left(\sigma_1^2 - \sigma_1\sigma_2 + \sigma_2^2 \right)^{1/2}}$$

$$n = \frac{54 \text{ ksi}}{\left((10 \text{ ksi})^2 - (10 \text{ ksi})(-1.9308 \text{ ksi}) + (-1.9308 \text{ ksi})^2 \right)^{1/2}}$$

$$n = 4.8683$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 11

TOPICS COVERED IN HOMEWORK 11

Chapter 8 Bolt Design: 8-11, 8-12, 8-14, 8-15, 8-17, 8-18

**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

Problem 08.011 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut is used to clamp together two 20-mm steel plates.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Problem 08.011 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

1.

Award: 2.00 points

Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is $\boxed{55 \pm 2\%}$ mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7.

From Table A–31, the nut height is $H = 12.8$ mm.

$$L > 2t + H$$

$$L = 2 \times 20 \text{ mm} + 12.8 \text{ mm} = 52.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$10 \leq t \leq 11$	35
$12 \leq t \leq 13$	40
$14 \leq t \leq 16$	45
$17 \leq t \leq 18$	50
$19 \leq t \leq 21$	55
$22 \leq t \leq 23$	60
$24 \leq t \leq 25$	65

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t = 2 \times 20 \text{ mm} = 40 \text{ mm}$$

$$l_d = L - L_T = 55 \text{ mm} - 34 \text{ mm} = 21 \text{ mm}$$

$$l_t = l - l_d = 40 \text{ mm} - 21 \text{ mm} = 19 \text{ mm}$$

The length of the bolt L is 55 mm.

References

Numeric Response

Problem 08.011.a - Determining joint and member stiffnesses

Difficulty: Medium

2.

Award: 7.00 points

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is MN/m.

Explanation:

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8-1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 19 \text{ mm}) + (115 \text{ mm}^2 \times 21 \text{ mm})} = 686.1810 \text{ MN/m}$$

The bolt stiffness k_b is 686.1810 MN/m.

References

Numeric Response

Problem 08.011.b - Determining joint and member stiffnesses

Difficulty: Medium

3.

Award: 7.00 points

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8-23) can be used. Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe \frac{Bd}{l}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e^{\left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{40}{1000} \text{ m} \right)} \right)} = 2842.6590 \text{ MN/m}$$

The material stiffness k_m is 2842.6590 MN/m.

References

Numeric Response

Problem 08.011.c - Determining joint and member stiffnesses

Difficulty: Medium

Problem 08.012 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut and a 14R washer under each of bolt head and the nut is used to clamp together two 23-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Problem 08.012 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

4. Award: 2.00 points

Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm and the washer is $w = 3.5$ mm.

$$L > 2t + H + 2w$$

$$L = 2 \times 23 \text{ mm} + 12.8 \text{ mm} + 7 \text{ mm} = 65.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$11 \leq t \leq 12$	45
$13 \leq t \leq 15$	50
$16 \leq t \leq 17$	55
$18 \leq t \leq 20$	60
$21 \leq t \leq 22$	65
$23 \leq t \leq 25$	70

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t + 2w = 2 \times 23 \text{ mm} + 7 \text{ mm} = 53 \text{ mm}$$

$$l_d = L - L_T = 70 \text{ mm} - 34 \text{ mm} = 36 \text{ mm}$$

$$l_t = l - l_d = 53 \text{ mm} - 36 \text{ mm} = 17 \text{ mm}$$

The length of the bolt L is 70 mm.

References

Numeric Response

Problem 08.012.a - Determining joint and member stiffnesses

Difficulty: Medium

5. Award: 7.00 points

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is MN/m.

Explanation:

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8–1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 17 \text{ mm}) + (115 \text{ mm}^2 \times 36 \text{ mm})} = 542.2479 \text{ MN/m}$$

The bolt stiffness k_b is 542.2479 MN/m.

References

Numeric Response

Problem 08.012.b - Determining joint and member stiffnesses

Difficulty: Medium

6. Award: 7.00 points

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is 2693.2914 ± 2% MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used.

Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe \frac{Bd}{l}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e \left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{53}{1000} \text{ m} \right)} \right) = 2693.2914 \text{ MN/m}$$

The material stiffness k_m is 2693.2914 MN/m.

References

Numeric Response

Problem 08.012.c - Determining joint and member stiffnesses

Difficulty: Medium

Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.25-in. steel plate and a 0.5-in. cast-iron plate are compressed with one bolt and nut. The bolt is ½ in-13 UNC.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

7. Award: 2.00 points

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest ¼ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B = 1.25 \text{ in} + 0.5 \text{ in} = 1.75 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437 \text{ in}$.

$$L \geq l + H$$

$$L = 1.75 \text{ in} + 0.437 \text{ in} = 2.187 \text{ in}$$

Rounding up the value L to the nearest ¼ in.

$$L = 2.25 \text{ in}$$

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 2.25 \text{ in} - 1.25 \text{ in} = 1 \text{ in}$$

$$l_t = l - l_d = 1.75 \text{ in} - 1 \text{ in} = 0.75 \text{ in}$$

The length of the bolt is 2.25 in.

References

Numeric Response

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

8. Award: 7.00 points

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is Mlb/in.

Explanation:

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.75 \text{ in}) + (0.1419 \text{ in}^2 \times 1 \text{ in})} = 2.8903 \text{ Mlb/in}$$

The bolt stiffness is 2.8903 Mlb/in.

References

Numeric Response

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

9.

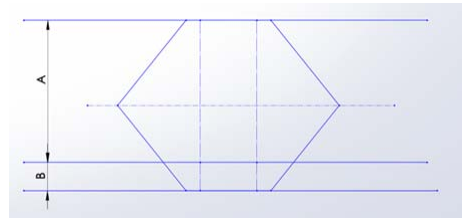
Award: 8.00 points

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$, we must calculate three k 's for the frusta.

For the steel plate, the frustum is in two parts.

Steel plate:

 $t = 0.875$ in, $d = 0.5$ in, $D = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.875 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.875 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 26.5352 \text{ Mlb/in}$$

$$t_2 = A - \frac{l}{2} \text{ in} = 1.25 \text{ in} - \left(\frac{1.75}{2}\right) \text{ in} = 0.375 \text{ in}$$

$$D_2 = 1.5d + 2(l - h) \tan(30^\circ) = 1.5 \times 0.5 \text{ in} + 2(1.75 \text{ in} - 1.25 \text{ in}) \times \tan(30^\circ) = 1.3274 \text{ in}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{(1.155 \times 0.375 \text{ in} + 1.3274 \text{ in} - 0.5 \text{ in})(1.3274 \text{ in} + 0.5 \text{ in})}{(1.155 \times 0.375 \text{ in} + 1.3274 \text{ in} + 0.5 \text{ in})(1.3274 \text{ in} - 0.5 \text{ in})}\right)} = 130.6305 \text{ Mlb/in}$$

For the cast-iron plate, the frustum is in one part. This will generate k_3 .

Cast iron:

 $t_3 = 0.5$ in, $d = 0.5$ in, $D = 0.75$ in, and $E = 14500$ kpsi (from Table 8-8).

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{(1.155 \times 0.5 \text{ in} + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{(1.155 \times 0.5 \text{ in} + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 16.0941 \text{ Mlb/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}\right)^{-1} = \left(\frac{1}{26.5352 \text{ Mlb/in}} + \frac{1}{130.6305 \text{ Mlb/in}} + \frac{1}{16.0941 \text{ Mlb/in}}\right)^{-1} = 9.3044 \text{ Mlb/in}$$

The material stiffness is 9.3044 Mlb/in.

References

Numeric Response

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

Problem 08.015 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.5-in. steel plate and a 0.75-in. gray cast iron plate are compressed with one bolt and nut. The bolt is $\frac{1}{2}$ in-13 UNC. A $\frac{1}{2}$ N American Standard plain washer is used under the head of the bolt and under the nut.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Problem 08.015 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

10. Award: 2.00 points

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest ¼ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B + 2w = 1.5 \text{ in} + 0.75 \text{ in} + 2(0.095 \text{ in}) = 2.44 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437 \text{ in}$ and the washer thickness is $w = 0.095 \text{ in}$.

$$L \geq l + H$$

$$L = 2.44 \text{ in} + 0.437 \text{ in} = 2.877 \text{ in}$$

Rounding up the value of L to the nearest ¼ in.

$$L = 3 \text{ in}$$

The length of the bolt is 3 in.

References

Numeric Response

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

11. Award: 7.00 points

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 3 \text{ in} - 1.25 \text{ in} = 1.75 \text{ in}$$

$$l_t = l - l_d = 2.44 \text{ in} - 1.75 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 1.75 \text{ in})} = 2.1775 \text{ Mlb/in}$$

The bolt stiffness is 2.1775 Mlb/in.

References

Numeric Response

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

12. Award: 8.00 points

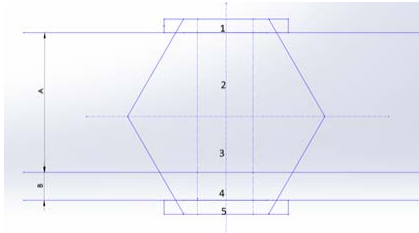
Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **8.7535 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$ and the washer material is different than the bottom piece being joined, we must calculate five k 's for the frusta. Note that the top washer and the top piece are both steel. However, the ID of the washer is different than that of the plates.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the steel plate, the frustum is in two parts.

Steel:

$$t_2 = \frac{A+B}{2} \text{ in} = \left(\frac{1.5 + 0.75}{2}\right) \text{ in} = 1.125 \text{ in}$$

$d = 0.5$ in

$$D_2 = 0.75 \text{ in} + 2(0.095 \text{ in})\tan(30^\circ) = 0.860 \text{ in}$$

$E = 30000$ kpsi

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.125 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.125 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 31.7316 \text{ Mlb/in}$$

$$t_3 = A - t_2 = 1.5 \text{ in} - 1.125 \text{ in} = 0.375 \text{ in}$$

$$D_3 = 0.860 \text{ in} + 2(0.75 \text{ in})\tan(30^\circ) = 1.7261 \text{ in}$$

$E = 30000$ kpsi

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 1.7261 \text{ in} - 0.5 \text{ in})(1.7261 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 1.7261 \text{ in} + 0.5 \text{ in})(1.7261 \text{ in} - 0.5 \text{ in})}\right)} = 218.1482 \text{ Mlb/in}$$

For the cast-iron cylinder, the frustum is in one part. This will generate k_4 .

Cast iron:

$t_4 = 0.75$ in

$d = 0.5$ in

$D = 0.860$ in

$E = 14500$ kpsi (from Table 8-8)

$$k_4 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_4) + D_4 - d)(D_4 + d)}{(1.155(t_4) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.75 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.75 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 17.9471 \text{ Mlb/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{2}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_4}\right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{1}{31.7316 \text{ Mlb/in}} + \frac{1}{218.1482 \text{ Mlb/in}} + \frac{1}{17.9471 \text{ Mlb/in}}\right)^{-1} = 8.7535 \text{ Mlb/in}$$

The material stiffness is 8.7535 Mlb/in.

References

Numeric Response

Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

Problem 08.017 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 175 in thick and are compressed with one bolt and nut. Washers are used under the head of the bolt and under the nut. NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

Washer properties: steel; ID = 0.531 in; OD = 1.062 in; thickness = 0.095 in
 Nut properties: steel; height = 7/16 in
 Bolt properties: 1/2-13 UNC grade 8
 Plate properties: aluminum; $E = 10.4$ Mpsi; $S_U = 47$ kpsi; $S_Y = 25$ kpsi

References

Section Break

Problem 08.017 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

13.

Award: 2.00 points

Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 1/4 in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7.

From Table A–31,

$$l = 2t + 2w = 2(1.75 \text{ in}) + 2(0.095 \text{ in}) = 3.69 \text{ in}$$

where

t = steel plate thickness

The nut height is $H = 0.437$ in and the washer thickness is $w = 0.095$ in.

$$L \geq l + H$$

$$L = 3.69 \text{ in} + 0.437 \text{ in} = 4.127 \text{ in}$$

Rounding up to the nearest 1/4 in,

$$L = 4.25 \text{ in}$$

The length of the bolt is 4.25 in.

References

Numeric Response

Problem 08.017.a - Determining joint and member stiffnesses

Difficulty: Medium

14. Award: 7.00 points

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 4.25 \text{ in} - 1.25 \text{ in} = 3 \text{ in}$$

$$l_t = l - l_d = 3.69 \text{ in} - 3 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 3 \text{ in})} = 1.4892 \text{ Mlb/in}$$

The bolt stiffness is 1.4892 Mlb/in.

References

Numeric Response

Problem 08.017.b - Determining joint and member stiffnesses

Difficulty: Medium

15. Award: 8.00 points

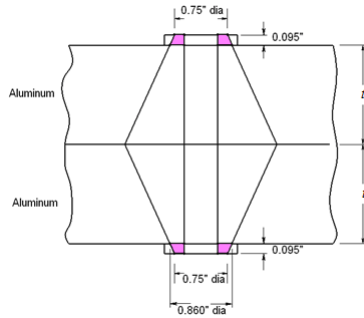
Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is Mlb/in.

Explanation:

Now calculate the member stiffness. As the thickness and material of the plates are the same and the material of the washers are different, we must calculate two k 's for the frusta.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the aluminum plates,

$t_2 = 1.75$ in, $d = 0.5$ in, $D = 0.860$ in, and $E = 10.4$ Mpsi

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (10.4 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.75 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.75 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 9.6397 \text{ Mlb/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{2}{k_1} + \frac{2}{k_2}\right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{2}{9.6397 \text{ Mlb/in}}\right)^{-1} = 4.3498 \text{ Mlb/in}$$

The material stiffness is 4.3498 Mlb/in.

References

Numeric Response

Problem 08.017.c - Determining joint and member stiffnesses

Difficulty: Medium

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 1.75 in (t) thick, and are compressed with one bolt and nut. No washers are used under the head of the bolt but two washers are used under the nut.

Washer properties: steel ID = 0.531 in, OD = 1.062 in, and thickness = 0.095 in.

Nut properties: steel height = 7/16 in

Bolt properties: 1/2-13 UNC grade 8

Plate properties: aluminum $E = 10.3$ Mpsi, $S_U = 47$ kpsi, and $S_Y = 25$ kpsi.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

16. Award: 2.00 points

Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. Following the procedure of Table 8–7, we have

$l = 2t + 2w = 2 \times 1.75 \text{ in} + 2 \times 0.095 \text{ in} = 3.69 \text{ in}$. The nut height is $H = 0.437 \text{ in}$ (from Table A–31), and the washer thickness is $w = 0.095 \text{ in}$.

$L \geq l + H = 3.69 \text{ in} + 0.437 \text{ in} = 4.127 \text{ in}$.

Rounding up the value to the nearest $\frac{1}{4}$ in gives $L = 4.25 \text{ in}$.

This is the required bolt length.

The bolt length is 4.25 in.

References

Numeric Response

Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

17. Award: 7.00 points

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is lb/in.

Explanation:

From Eq. 8–14,

$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$

From Table 8–7, we have

$l_d = L - L_T = 4.25 \text{ in} - 1.25 \text{ in} = 3 \text{ in}$

$l_t = l - l_d = 3.69 \text{ in} - 3 \text{ in} = 0.69 \text{ in}$

$A_d = \frac{\pi (0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$

From Table 8–1, $A_t = 0.1419 \text{ in}^2$.

Therefore, from Eq. (8–17), the bolt stiffness is

$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 (0.1419 \text{ in}^2) (30 \times 10^6 \text{ psi})}{0.1963 \text{ in}^2 (0.69 \text{ in}) + 0.1419 \text{ in}^2 (3 \text{ in})} = 1489180.375 \text{ lb/in}$

The bolt stiffness is 1489180.375 lb/in.

References

Numeric Response

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

18. Award: 8.00 points

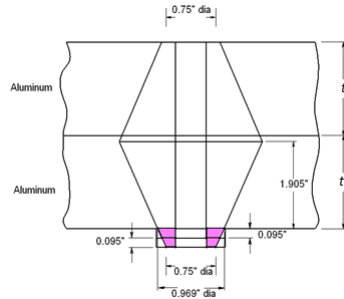
Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is $4202167.937 \pm 2\%$ lb/in.

Explanation:

The member stiffness requires the calculation for each frustum in the stack. As per the figure below, we will need to calculate three k values, the washers, the upper frustum, and the lower frustum.



For the washers,
Steel - $t = 2w = 2(0.095 \text{ in}) = 0.190 \text{ in}$, $d_1 = 0.531 \text{ in}$, $D_1 = 0.75 \text{ in}$, and $E = 30,000 \text{ ksi}$.

$$k_1 = \frac{0.5774 \pi (E) d_1}{\ln \frac{[1.155 (t) + D_1 - d_1] (D_1 + d_1)}{[1.155 (t) + D_1 + d_1] (D_1 - d_1)}} = \frac{0.5774 \pi (30 \times 10^6 \text{ psi}) (0.5 \text{ in})^2}{\ln \frac{[1.155 (0.190 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}] (0.75 \text{ in} + 0.5 \text{ in})}{[1.155 (0.190 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}] (0.75 \text{ in} - 0.5 \text{ in})}} = 53906002.57 \text{ lb/in}$$

For the aluminum plates, the frustum is in two parts.

Aluminum - $t_2 = \frac{t}{2} \text{ in} - 2w = 1.655 \text{ in}$, $d = 0.5 \text{ in}$, $D_2 = 0.75 \text{ in} + 2 \left(0.095 \text{ in} \right) \tan \left(30^\circ \right) = 0.969 \text{ in}$, and $E = 10,300 \text{ ksi}$.

From Eq. (8-20),

$$k_2 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_2) + D_2 - d] (D_2 + d)}{[1.155 (t_2) + D_2 + d] (D_2 - d)}} = \frac{0.5774 \pi (10.3 \times 10^6 \text{ psi}) (0.5 \text{ in})}{\ln \frac{[1.155 (1.655 \text{ in}) + 0.969 \text{ in} - 0.5 \text{ in}] (0.969 \text{ in} + 0.5 \text{ in})}{[1.155 (1.655 \text{ in}) + 0.969 \text{ in} + 0.5 \text{ in}] (0.969 \text{ in} - 0.5 \text{ in})}} = 11809839.12 \text{ lb/in}$$

$t_3 = t/2 \text{ in} = 1.845 \text{ in}$, $D_3 = 0.75 \text{ in}$, $d = 0.5 \text{ in}$, and $E = 10,300 \text{ ksi}$.

$$k_3 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_3) + D_3 - d] (D_3 + d)}{[1.155 (t_3) + D_3 + d] (D_3 - d)}} = \frac{0.5774 \pi (10.3 \times 10^6 \text{ psi}) (0.5 \text{ in})}{\ln \frac{[1.155 (1.845 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}] (0.75 \text{ in} + 0.5 \text{ in})}{[1.155 (1.845 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}] (0.75 \text{ in} - 0.5 \text{ in})}} = 7421346.664 \text{ lb/in}$$

Therefore, the material stiffness is

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{53906002.57} + \frac{1}{11809839.12} + \frac{1}{7421346.664} \right)^{-1} \text{ lb/in} = 4202167.937 \text{ lb/in}$$

The material stiffness is 4202167.937 lb/in.

References

Numeric Response

Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials

Difficulty: Medium

ABOVE AVERAGE

Deshawn Houston

Score: **100/100** Points **100** %

Problem 08.011 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut is used to clamp together two 23-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.011 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints -
Member Stiffness



Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **60** mm.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.011.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints -
Member Stiffness

Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **60 ± 2%** mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm.

$$L > 2t + H$$

$$L = 2 \times 23 \text{ mm} + 12.8 \text{ mm} = 58.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$10 \leq t \leq 11$	35
$12 \leq t \leq 13$	40
$14 \leq t \leq 16$	45
$17 \leq t \leq 18$	50
$19 \leq t \leq 21$	55
$22 \leq t \leq 23$	60
$24 \leq t \leq 25$	65

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t = 2 \times 23 \text{ mm} = 46 \text{ mm}$$

$$l_d = L - L_T = 60 \text{ mm} - 34 \text{ mm} = 26 \text{ mm}$$

$$l_t = l - l_d = 46 \text{ mm} - 26 \text{ mm} = 20 \text{ mm}$$

2.

Award: 7 out of 7.00 points

 You did not receive full credit for this question in a previous attempt

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **603.76**  MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.011.b - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **603.7556 ± 2%** MN/m.

Explanation:

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8–1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 20 \text{ mm}) + (115 \text{ mm}^2 \times 26 \text{ mm})} = 603.7556 \text{ MN/m}$$

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2762.2**  MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.011.c - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2762.2263 ± 2%** MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used. Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe^{\frac{Bd}{t}}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e^{\left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{46}{1000} \text{ m} \right)} \right)} = 2762.2263 \text{ MN/m}$$

Problem 08.012 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut and a 14R washer under each of bolt head and the nut is used to clamp together two 13-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.012 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints -
Member Stiffness



Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **50** mm.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.012.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints -
Member Stiffness

Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **50 ± 2%** mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm and the washer is $w = 3.5$ mm.

$$L > 2t + H + 2w$$

$$L = 2 \times 13 \text{ mm} + 12.8 \text{ mm} + 7 \text{ mm} = 45.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$11 \leq t \leq 12$	45
$13 \leq t \leq 15$	50
$16 \leq t \leq 17$	55
$18 \leq t \leq 20$	60
$21 \leq t \leq 22$	65
$23 \leq t \leq 25$	70

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t + 2w = 2 \times 13 \text{ mm} + 7 \text{ mm} = 33 \text{ mm}$$

$$l_d = L - L_T = 50 \text{ mm} - 34 \text{ mm} = 16 \text{ mm}$$

$$l_t = l - l_d = 33 \text{ mm} - 16 \text{ mm} = 17 \text{ mm}$$

5.

Award: 7 out of 7.00 points



You did not receive full credit for this question in a previous attempt

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **822.1** ✓ MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.012.b - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **822.1146 ± 2%** MN/m.**Explanation:**

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8-1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 17 \text{ mm}) + (115 \text{ mm}^2 \times 16 \text{ mm})} = 822.1146 \text{ MN/m}$$

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2978.5**  MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.012.c - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2978.4955 ± 2%** MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used.

Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe^{\frac{Bd}{t}}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e^{\left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{33}{1000} \text{ m} \right)} \right)} = 2978.4955 \text{ MN/m}$$

Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.25-in. steel plate and a 0.5-in. cast-iron plate are compressed with one bolt and nut. The bolt is $\frac{1}{2}$ in-13 UNC.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
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Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 08.05 Joints - Member Stiffness
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Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **2.25**  in.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Section: 08.05 Joints - Member Stiffness

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B = 1.25 \text{ in} + 0.5 \text{ in} = 1.75 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437$ in.

$$L \geq l + H$$

$$L = 1.75 \text{ in} + 0.437 \text{ in} = 2.187 \text{ in}$$

Rounding up the value L to the nearest $\frac{1}{4}$ in.

$$L = 2.25 \text{ in}$$

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 2.25 \text{ in} - 1.25 \text{ in} = 1 \text{ in}$$

$$l_t = l - l_d = 1.75 \text{ in} - 1 \text{ in} = 0.75 \text{ in}$$

8.

Award: 7 out of 7.00 points

 You did not receive full credit for this question in a previous attempt

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.89**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.8903 ± 2%** Mlb/in.**Explanation:**

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.75 \text{ in}) + (0.1419 \text{ in}^2 \times 1 \text{ in})} = 2.8903 \text{ Mlb/in}$$

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **9.3**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

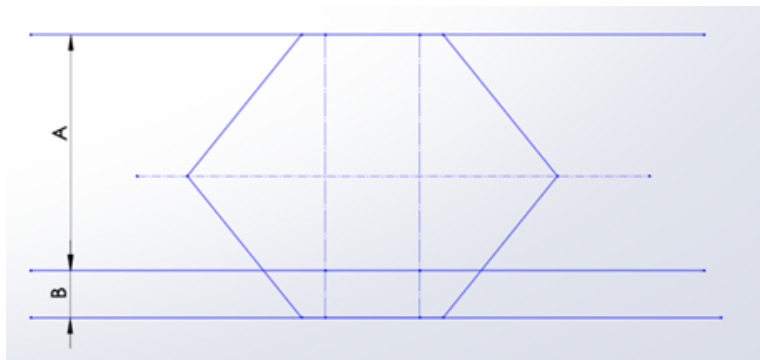
Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **9.3044 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$, we must calculate three k 's for the frusta.



For the steel plate, the frustum is in two parts.

Steel plate:

$t = 0.875$ in, $d = 0.5$ in, $D = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.875 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.875 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 26.5352 \text{ Mlb/in}$$

$$t_2 = A - \frac{l}{2} \text{ in} = 1.25 \text{ in} - \left(\frac{1.75}{2}\right) \text{ in} = 0.375 \text{ in}$$

$$D_2 = 1.5d + 2(l - h) \tan(30^\circ) = 1.5 \times 0.5 \text{ in} + 2(1.75 \text{ in} - 1.25 \text{ in}) \times \tan(30^\circ) = 1.3274 \text{ in}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 1.3274 \text{ in} - 0.5 \text{ in})(1.3274 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 1.3274 \text{ in} + 0.5 \text{ in})(1.3274 \text{ in} - 0.5 \text{ in})}\right)} = 130.6305 \text{ Mlb/in}$$

For the cast-iron plate, the frustum is in one part. This will generate k_3 .

Cast iron:

$t_3 = 0.5 \text{ in}$, $d = 0.5 \text{ in}$, $D = 0.75 \text{ in}$, and $E = 14500 \text{ kpsi}$ (from Table 8–8).

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{(1.155 \times 0.5 \text{ in} + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{(1.155 \times 0.5 \text{ in} + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 16.0941 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{26.5352 \text{ Mlb/in}} + \frac{1}{130.6305 \text{ Mlb/in}} + \frac{1}{16.0941 \text{ Mlb/in}} \right)^{-1} = 9.3044 \text{ Mlb/in}$$

Problem 08.015 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.25-in. steel plate and a 0.5-in. gray cast iron plate are compressed with one bolt and nut. The bolt is ½ in-13 UNC. A ½ N American Standard plain washer is used under the head of the bolt and under the nut.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break
Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.015 -
Determining joint and
member stiffnesses
with dissimilar joint
materials -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS
Section: 08.05 Joints -
Member Stiffness

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **2.5**  in.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Section: 08.05 Joints - Member Stiffness

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B + 2w = 1.25 \text{ in} + 0.5 \text{ in} + 2(0.095 \text{ in}) = 1.94 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437 \text{ in}$ and the washer thickness is $w = 0.095 \text{ in}$.

$$L \geq l + H$$

$$L = 1.94 \text{ in} + 0.437 \text{ in} = 2.377 \text{ in}$$

Rounding up the value of L to the nearest $\frac{1}{4}$ in.

$$L = 2.5 \text{ in}$$

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.6714**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.6714 ± 2%** Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 2.5 \text{ in} - 1.25 \text{ in} = 1.25 \text{ in}$$

$$l_t = l - l_d = 1.94 \text{ in} - 1.25 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 1.25 \text{ in})} = 2.6714 \text{ Mlb/in}$$

Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **9.6576**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

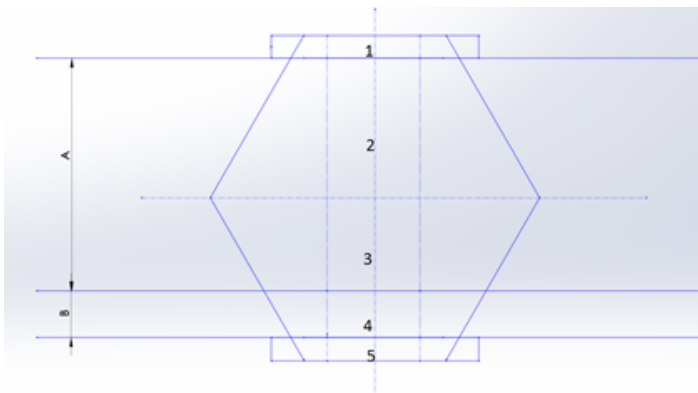
Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **9.6576 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$ and the washer material is different than the bottom piece being joined, we must calculate five k 's for the frusta. Note that the top washer and the top piece are both steel. However, the ID of the washer is different than that of the plates.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the steel plate, the frustum is in two parts.

Steel:

$$t_2 = \frac{A+B}{2} \text{ in} = \left(\frac{1.25+0.5}{2} \right) \text{ in} = 0.875 \text{ in}$$

$$d = 0.5 \text{ in}$$

$$D_2 = 0.75 \text{ in} + 2(0.095 \text{ in})\tan(30^\circ) = 0.860 \text{ in}$$

$$E = 30000 \text{ kpsi}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.875 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.875 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 34.828 \text{ Mlb/in}$$

$$t_3 = A - t_2 = 1.25 \text{ in} - 0.875 \text{ in} = 0.375 \text{ in}$$

$$D_3 = 0.860 \text{ in} + 2(0.5 \text{ in})\tan(30^\circ) = 1.4374 \text{ in}$$

$$E = 30000 \text{ kpsi}$$

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 1.4374 \text{ in} - 0.5 \text{ in})(1.4374 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 1.4374 \text{ in} + 0.5 \text{ in})(1.4374 \text{ in} - 0.5 \text{ in})}\right)} = 152.7975 \text{ Mlb/in}$$

For the cast-iron cylinder, the frustum is in one part. This will generate k_4 .

Cast iron:

$$t_4 = 0.5 \text{ in}$$

$$d = 0.5 \text{ in}$$

$$D = 0.860 \text{ in}$$

$$E = 14500 \text{ kpsi (from Table 8-8)}$$

$$k_4 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_4) + D_4 - d)(D_4 + d)}{(1.155(t_4) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.5 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.5 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 21.8024 \text{ Mlb/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{2}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_4} \right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{1}{34.828 \text{ Mlb/in}} + \frac{1}{152.7975 \text{ Mlb/in}} + \frac{1}{21.8024 \text{ Mlb/in}} \right)^{-1} = 9.6576 \text{ Mlb/in}$$

Problem 08.017 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 1.5 in thick and are compressed with one bolt and nut. Washers are used under the head of the bolt and under the nut.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

Washer properties: steel; ID = 0.531 in; OD = 1.062 in; thickness = 0.095 in

Nut properties: steel; height = 7/16 in

Bolt properties: 1/2-13 UNC grade 8

Plate properties: aluminum; $E = 10.4$ Mpsi; $S_U = 47$ kpsi; $S_Y = 25$ kpsi

References

Section Break

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.017 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints -
Member Stiffness



Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **3.75**  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.017.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints -
Member Stiffness

Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7.

From Table A–31,

$$l = 2t + 2w = 2(1.5 \text{ in}) + 2(0.095 \text{ in}) = 3.19 \text{ in}$$

where

t = steel plate thickness

The nut height is $H = 0.437$ in and the washer thickness is $w = 0.095$ in.

$$L \geq l + H$$

$$L = 3.19 \text{ in} + 0.437 \text{ in} = 3.627 \text{ in}$$

Rounding up to the nearest $\frac{1}{4}$ in,

$$L = 3.75 \text{ in}$$

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is **1.705**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.017.b - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is **1.7048 ± 2%** Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 3.75 \text{ in} - 1.25 \text{ in} = 2.5 \text{ in}$$

$$l_t = l - l_d = 3.19 \text{ in} - 2.5 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 2.5 \text{ in})} = 1.7048 \text{ Mlb/in}$$

Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is **4.516**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.017.c - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

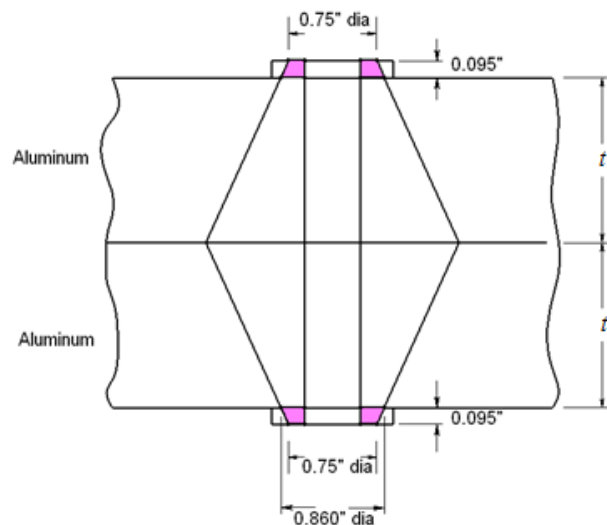
Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is **4.5164 ± 2%** Mlb/in.

Explanation:

Now calculate the member stiffness. As the thickness and material of the plates are the same and the material or the washers are different, we must calculate two k 's for the frusta.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the aluminum plates,

$t_2 = 1.5$ in, $d = 0.5$ in, $D = 0.860$ in, and $E = 10.4$ Mpsi

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_4 - d)(D_4 + d)}{(1.155(t_2) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (10.4 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.5 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.5 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 10.0506 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{2}{k_1} + \frac{2}{k_2}\right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{2}{10.0506 \text{ Mlb/in}}\right)^{-1} = 4.5164 \text{ Mlb/in}$$

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 1.5 in (t) thick, and are compressed with one bolt and nut. No washers are used under the head of the bolt but two washers are used under the nut.

Washer properties: steel ID = 0.531 in, OD = 1.062 in, and thickness = 0.095 in.

Nut properties: steel height = 7/16 in

Bolt properties: ½-13 UNC grade 8

Plate properties: aluminum $E = 10.3$ Mpsi, $S_U = 47$ kpsi, and $S_Y = 25$ kpsi.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS
Section: 08.05 Joints - Member Stiffness



Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is 3.75 ✓ in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.018.a -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints -
Member Stiffness

Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. Following the procedure of Table 8–7, we have

$l = 2t + 2w = 2 \times 1.5 \text{ in} + 2 \times 0.095 \text{ in} = 3.19 \text{ in}$. The nut height is $H = 0.437 \text{ in}$ (from Table A–31), and the washer thickness is $w = 0.095 \text{ in}$.

$L \geq l + H = 3.19 \text{ in} + 0.437 \text{ in} = 3.627 \text{ in}$.

Rounding up the value to the nearest $\frac{1}{4}$ in gives $L = 3.75 \text{ in}$.

This is the required bolt length.

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1704720.9**  lb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1704720.959 ± 2%** lb/in.

Explanation:

From Eq. 8–14,

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7, we have

$$l_d = L - L_T = 3.75 \text{ in} - 1.25 \text{ in} = 2.5 \text{ in}$$

$$l_t = l - l_d = 3.19 \text{ in} - 2.5 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi(0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8–1, $A_t = 0.1419 \text{ in}^2$.

Therefore, from Eq. (8–17), the bolt stiffness is

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 (0.1419 \text{ in}^2) (30 \times 10^6 \text{ psi})}{0.1963 \text{ in}^2 (0.69 \text{ in}) + 0.1419 \text{ in}^2 (2.5 \text{ in})} = 1704720.959 \text{ lb/in}$$

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln \frac{[1.155(t)+D_1-d_1](D_1+d_1)}{[1.155(t)+D_1+d_1](D_1-d_1)}} = \frac{0.5774\pi(30 \times 10^6 \text{ psi})0.5 \text{ in}^2}{\ln \frac{[1.155(0.190 \text{ in})+0.75 \text{ in}-0.5 \text{ in}](0.75 \text{ in}+0.5 \text{ in})}{[1.155(0.190 \text{ in})+0.75 \text{ in}+0.5 \text{ in}](0.75 \text{ in}-0.5 \text{ in})}} = 53906002.57 \text{ lb/in}$$

For the aluminum plates, the frustum is in two parts.

Aluminum – $t_2 = \frac{l}{2}$ in – $2w = 1.405$ in, $d = 0.5$ in, $D_2 = 0.75$ in + $2 \times 2(0.095 \text{ in}) \tan(30^\circ) = 0.969$ in, and $E = 10,300$ ksi.

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi Ed}{\ln \frac{[1.155(t_2)+D_2-d](D_2+d)}{[1.155(t_2)+D_2+d](D_2-d)}} = \frac{0.5774\pi(10.3 \times 10^6 \text{ psi})0.5 \text{ in}}{\ln \frac{[1.155(1.405 \text{ in})+0.969 \text{ in}-0.5 \text{ in}](0.969 \text{ in}+0.5 \text{ in})}{[1.155(1.405 \text{ in})+0.969 \text{ in}+0.5 \text{ in}](0.969 \text{ in}-0.5 \text{ in})}} = 12439209.06 \text{ lb/in}$$

$t_3 = l/2$ in = 1.595 in, $D_3 = 0.75$ in, $d = 0.5$ in, and $E = 10,300$ ksi.

$$k_3 = \frac{0.5774\pi Ed}{\ln \frac{[1.155(t_3)+D_3-d](D_3+d)}{[1.155(t_3)+D_3+d](D_3-d)}} = \frac{0.5774\pi(10.3 \times 10^6)0.5 \text{ in}}{\ln \frac{[1.155(1.595 \text{ in})+0.75 \text{ in}-0.5 \text{ in}](0.75 \text{ in}+0.5 \text{ in})}{[1.155(1.595 \text{ in})+0.75 \text{ in}+0.5 \text{ in}](0.75 \text{ in}-0.5 \text{ in})}} = 7664967.28 \text{ lb/in}$$

Therefore, the material stiffness is

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{53906002.57} + \frac{1}{12439209.06} + \frac{1}{7664967.28} \right)^{-1} \text{ lb/in} = 4359093.885 \text{ lb/in}$$

AVERAGE

Score: 92/100 Points 92 %

Problem 08.011 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut is used to clamp together two 17-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.011 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints -
Member Stiffness



Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **50** mm.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.011.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints -
Member Stiffness

Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **50 ± 2%** mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm.

$$L > 2t + H$$

$$L = 2 \times 17 \text{ mm} + 12.8 \text{ mm} = 46.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$10 \leq t \leq 11$	35
$12 \leq t \leq 13$	40
$14 \leq t \leq 16$	45
$17 \leq t \leq 18$	50
$19 \leq t \leq 21$	55
$22 \leq t \leq 23$	60
$24 \leq t \leq 25$	65

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t = 2 \times 17 \text{ mm} = 34 \text{ mm}$$

$$l_d = L - L_T = 50 \text{ mm} - 34 \text{ mm} = 16 \text{ mm}$$

$$l_t = l - l_d = 34 \text{ mm} - 16 \text{ mm} = 18 \text{ mm}$$

2.

Award: 7 out of 7.00 points



You did not receive full credit for this question in a previous attempt

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **794.67** ✓ MN/m.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.011.b - Determining joint and member stiffnesses

Section: 08.05 Joints - Member Stiffness

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **794.6704 ± 2%** MN/m.

Explanation:

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8–1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 18 \text{ mm}) + (115 \text{ mm}^2 \times 16 \text{ mm})} = 794.6704 \text{ MN/m}$$

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2955.22**  MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.011.c - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2955.2203 ± 2%** MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used. Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe^{\frac{Bd}{t}}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e^{\left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{34}{1000} \text{ m} \right)} \right)} = 2955.2203 \text{ MN/m}$$

Problem 08.012 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut and a 14R washer under each of bolt head and the nut is used to clamp together two 25-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
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Problem 08.012 - Determining joint and member stiffnesses - DEPENDENT MULTI- PART PROBLEM - ASSIGN ALL PARTS	Section: 08.05 Joints - Member Stiffness
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Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **70**  mm.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.012.a - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **70 ± 2%** mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm and the washer is $w = 3.5$ mm.

$$L > 2t + H + 2w$$

$$L = 2 \times 25 \text{ mm} + 12.8 \text{ mm} + 7 \text{ mm} = 69.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$11 \leq t \leq 12$	45
$13 \leq t \leq 15$	50
$16 \leq t \leq 17$	55
$18 \leq t \leq 20$	60
$21 \leq t \leq 22$	65
$23 \leq t \leq 25$	70

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t + 2w = 2 \times 25 \text{ mm} + 7 \text{ mm} = 57 \text{ mm}$$

$$l_d = L - L_T = 70 \text{ mm} - 34 \text{ mm} = 36 \text{ mm}$$

$$l_t = l - l_d = 57 \text{ mm} - 36 \text{ mm} = 21 \text{ mm}$$

5.

Award: 7 out of 7.00 points



You did not receive full credit for this question in a previous attempt

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is 496.9667  MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.012.b - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is 496.9667 ± 2% MN/m.**Explanation:**

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8-1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 21 \text{ mm}) + (115 \text{ mm}^2 \times 36 \text{ mm})} = 496.9667 \text{ MN/m}$$

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2662.084**  MN/m.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.012.c - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **2662.084 ± 2%** MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used.

Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe^{\frac{Bd}{t}}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e^{\left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{57}{1000} \text{ m} \right)} \right)} = 2662.0840 \text{ MN/m}$$

Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.5-in. steel plate and a 0.75-in. cast-iron plate are compressed with one bolt and nut. The bolt is $\frac{1}{2}$ in-13 UNC.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
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Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI- PART PROBLEM - ASSIGN ALL PARTS	Section: 08.05 Joints - Member Stiffness
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Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **2.75**  in.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Section: 08.05 Joints - Member Stiffness

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B = 1.5 \text{ in} + 0.75 \text{ in} = 2.25 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437$ in.

$$L \geq l + H$$

$$L = 2.25 \text{ in} + 0.437 \text{ in} = 2.687 \text{ in}$$

Rounding up the value L to the nearest $\frac{1}{4}$ in.

$$L = 2.75 \text{ in}$$

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 2.75 \text{ in} - 1.25 \text{ in} = 1.5 \text{ in}$$

$$l_t = l - l_d = 2.25 \text{ in} - 1.5 \text{ in} = 0.75 \text{ in}$$

8.

Award: 7 out of 7.00 points



You did not receive full credit for this question in a previous attempt

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.3208** ✓ Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints**Problem 08.014.b -
Determining joint and
member stiffnesses
with dissimilar joint
materials**Section: 08.05 Joints -
Member Stiffness

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.3208 ± 2%** Mlb/in.**Explanation:**

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.75 \text{ in}) + (0.1419 \text{ in}^2 \times 1.5 \text{ in})} = 2.3208 \text{ Mlb/in}$$

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **8.3476**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

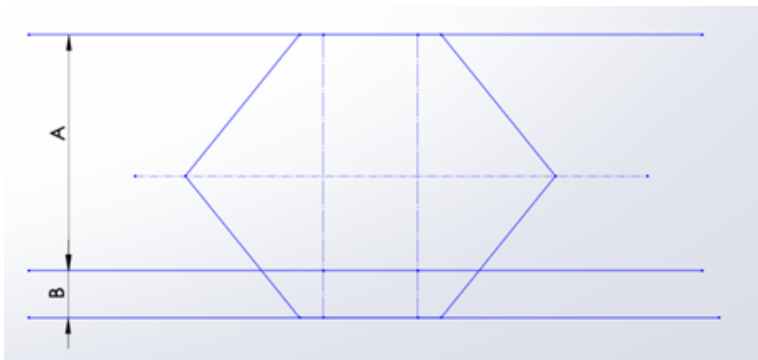
Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **8.3476 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$, we must calculate three k 's for the frusta.



For the steel plate, the frustum is in two parts.

Steel plate:

$t = 1.125$ in, $d = 0.5$ in, $D = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.125 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.125 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 24.4811 \text{ Mlb/in}$$

$$t_2 = A - \frac{l}{2} \text{ in} = 1.5 \text{ in} - \left(\frac{2.25}{2}\right) \text{ in} = 0.375 \text{ in}$$

$$D_2 = 1.5d + 2(l - h) \tan(30^\circ) = 1.5 \times 0.5 \text{ in} + 2(2.25 \text{ in} - 1.5 \text{ in}) \times \tan(30^\circ) = 1.6161 \text{ in}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 1.6161 \text{ in} - 0.5 \text{ in})(1.6161 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 1.6161 \text{ in} + 0.5 \text{ in})(1.6161 \text{ in} - 0.5 \text{ in})}\right)} = 192.0211 \text{ Mlb/in}$$

For the cast-iron plate, the frustum is in one part. This will generate k_3 .

Cast iron:

$t_3 = 0.75 \text{ in}$, $d = 0.5 \text{ in}$, $D = 0.75 \text{ in}$, and $E = 14500 \text{ kpsi}$ (from Table 8–8).

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{(1.155 \times 0.75 \text{ in} + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{(1.155 \times 0.75 \text{ in} + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 13.5612 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{24.4811 \text{ Mlb/in}} + \frac{1}{192.0211 \text{ Mlb/in}} + \frac{1}{13.5612 \text{ Mlb/in}} \right)^{-1} = 8.3476 \text{ Mlb/in}$$

Problem 08.015 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.75-in. steel plate and a 1-in. gray cast iron plate are compressed with one bolt and nut. The bolt is ½ in-13 UNC. A ½ N American Standard plain washer is used under the head of the bolt and under the nut.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.015 -
Determining joint and
member stiffnesses
with dissimilar joint
materials -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints -
Member Stiffness

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **3.5**  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Section: 08.05 Joints -
Member Stiffness

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B + 2w = 1.75 \text{ in} + 1 \text{ in} + 2(0.095 \text{ in}) = 2.94 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437 \text{ in}$ and the washer thickness is $w = 0.095 \text{ in}$.

$$L \geq l + H$$

$$L = 2.94 \text{ in} + 0.437 \text{ in} = 3.377 \text{ in}$$

Rounding up the value of L to the nearest $\frac{1}{4}$ in.

$$L = 3.5 \text{ in}$$

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1.8378**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1.8378 ± 2%** Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 3.5 \text{ in} - 1.25 \text{ in} = 2.25 \text{ in}$$

$$l_t = l - l_d = 2.94 \text{ in} - 2.25 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 2.25 \text{ in})} = 1.8378 \text{ Mlb/in}$$

Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **8.1987**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

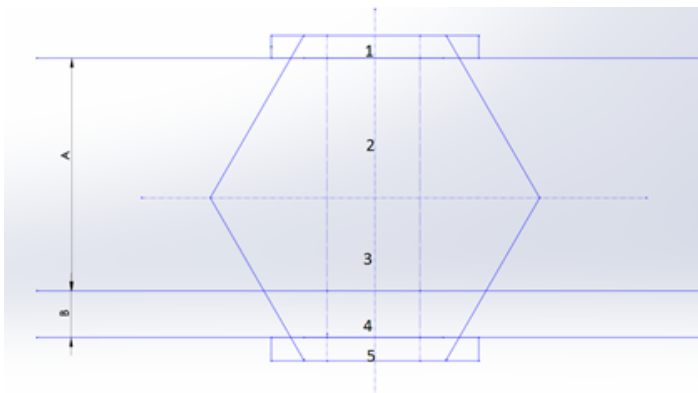
Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **8.1987 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$ and the washer material is different than the bottom piece being joined, we must calculate five k 's for the frusta. Note that the top washer and the top piece are both steel. However, the ID of the washer is different than that of the plates.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the steel plate, the frustum is in two parts.

Steel:

$$t_2 = \frac{A+B}{2} \text{ in} = \left(\frac{1.75+1}{2} \right) \text{ in} = 1.375 \text{ in}$$

$$d = 0.5 \text{ in}$$

$$D_2 = 0.75 \text{ in} + 2(0.095 \text{ in})\tan(30^\circ) = 0.860 \text{ in}$$

$$E = 30000 \text{ kpsi}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.375 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.375 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 29.7424 \text{ Mlb/in}$$

$$t_3 = A - t_2 = 1.75 \text{ in} - 1.375 \text{ in} = 0.375 \text{ in}$$

$$D_3 = 0.860 \text{ in} + 2(1 \text{ in})\tan(30^\circ) = 2.0148 \text{ in}$$

$$E = 30000 \text{ kpsi}$$

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 2.0148 \text{ in} - 0.5 \text{ in})(2.0148 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 2.0148 \text{ in} + 0.5 \text{ in})(2.0148 \text{ in} - 0.5 \text{ in})}\right)} = 293.923 \text{ Mlb/in}$$

For the cast-iron cylinder, the frustum is in one part. This will generate k_4 .

Cast iron:

$$t_4 = 1 \text{ in}$$

$$d = 0.5 \text{ in}$$

$$D = 0.860 \text{ in}$$

$$E = 14500 \text{ kpsi (from Table 8-8)}$$

$$k_4 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_4) + D_4 - d)(D_4 + d)}{(1.155(t_4) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 15.9936 \text{ Mlb/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{2}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_4} \right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{1}{29.7424 \text{ Mlb/in}} + \frac{1}{293.923 \text{ Mlb/in}} + \frac{1}{15.9936 \text{ Mlb/in}} \right)^{-1} = 8.1987 \text{ Mlb/in}$$

Problem 08.017 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 1.25 in thick and are compressed with one bolt and nut. Washers are used under the head of the bolt and under the nut.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

Washer properties: steel; ID = 0.531 in; OD = 1.062 in; thickness = 0.095 in

Nut properties: steel; height = 7/16 in

Bolt properties: 1/2-13 UNC grade 8

Plate properties: aluminum; $E = 10.4$ Mpsi; $S_U = 47$ kpsi; $S_Y = 25$ kpsi

References

Section Break

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

Problem 08.017 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints -
Member Stiffness



Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **3.25** in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.017.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints -
Member Stiffness

Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7.

From Table A–31,

$$l = 2t + 2w = 2(1.25 \text{ in}) + 2(0.095 \text{ in}) = 2.69 \text{ in}$$

where

t = steel plate thickness

The nut height is $H = 0.437$ in and the washer thickness is $w = 0.095$ in.

$$L \geq l + H$$

$$L = 2.69 \text{ in} + 0.437 \text{ in} = 3.127 \text{ in}$$

Rounding up to the nearest $\frac{1}{4}$ in,

$$L = 3.25 \text{ in}$$

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is **1.9933**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.017.b - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is **1.9933 ± 2%** Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 3.25 \text{ in} - 1.25 \text{ in} = 2 \text{ in}$$

$$l_t = l - l_d = 2.69 \text{ in} - 2 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 2 \text{ in})} = 1.9933 \text{ Mlb/in}$$

Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is **4.7458**  Mlb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.017.c - Determining joint and member stiffnesses Section: 08.05 Joints - Member Stiffness

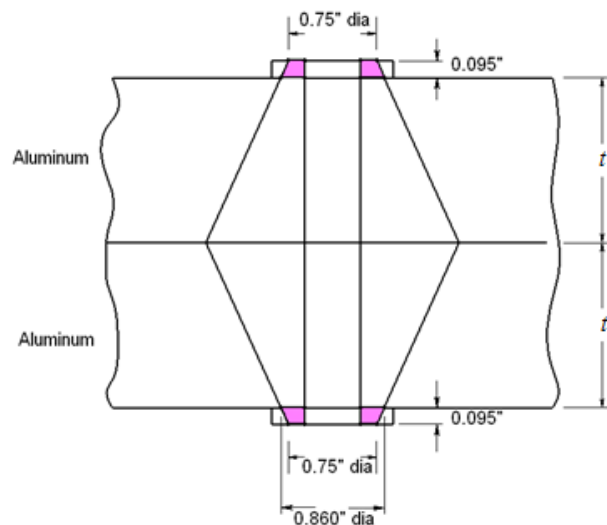
Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is **4.7458 ± 2%** Mlb/in.

Explanation:

Now calculate the member stiffness. As the thickness and material of the plates are the same and the material or the washers are different, we must calculate two k 's for the frusta.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the aluminum plates,

$t_2 = 1.25$ in, $d = 0.5$ in, $D = 0.860$ in, and $E = 10.4$ Mpsi

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_4 - d)(D_4 + d)}{(1.155(t_2) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (10.4 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.25 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.25 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 10.6218 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{2}{k_1} + \frac{2}{k_2}\right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{2}{10.6218 \text{ Mlb/in}}\right)^{-1} = 4.7458 \text{ Mlb/in}$$

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 2 in (t) thick, and are compressed with one bolt and nut. No washers are used under the head of the bolt but two washers are used under the nut.

Washer properties: steel ID = 0.531 in, OD = 1.062 in, and thickness = 0.095 in.

Nut properties: steel height = 7/16 in

Bolt properties: ½-13 UNC grade 8

Plate properties: aluminum $E = 10.3$ Mpsi, $S_U = 47$ kpsi, and $S_Y = 25$ kpsi.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS
Section: 08.05 Joints - Member Stiffness



Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is 4.75  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.018.a -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints -
Member Stiffness

Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. Following the procedure of Table 8–7, we have

$l = 2t + 2w = 2 \times 2 \text{ in} + 2 \times 0.095 \text{ in} = 4.19 \text{ in}$. The nut height is $H = 0.437 \text{ in}$ (from Table A–31), and the washer thickness is $w = 0.095 \text{ in}$.

$L \geq l + H = 4.19 \text{ in} + 0.437 \text{ in} = 4.627 \text{ in}$.

Rounding up the value to the nearest $\frac{1}{4}$ in gives $L = 4.75 \text{ in}$.

This is the required bolt length.

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1322026.683** ✓ lb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1322026.683 ± 2%** lb/in.

Explanation:

From Eq. 8–14,

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7, we have

$$l_d = L - L_T = 4.75 \text{ in} - 1.25 \text{ in} = 3.5 \text{ in}$$

$$l_t = l - l_d = 4.19 \text{ in} - 3.5 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi(0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8–1, $A_t = 0.1419 \text{ in}^2$.

Therefore, from Eq. (8–17), the bolt stiffness is

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 (0.1419 \text{ in}^2) (30 \times 10^6 \text{ psi})}{0.1963 \text{ in}^2 (0.69 \text{ in}) + 0.1419 \text{ in}^2 (3.5 \text{ in})} = 1322026.683 \text{ lb/in}$$

Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **2368**  lb/in.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials Section: 08.05 Joints - Member Stiffness

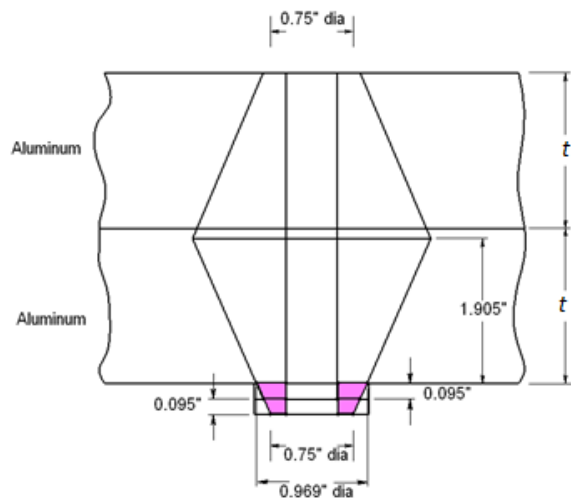
Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **4082730.289 ± 2%** lb/in.

Explanation:

The member stiffness requires the calculation for each frustum in the stack. As per the figure below, we will need to calculate three k values, the washers, the upper frustum, and the lower frustum.



For the washers,
Steel – $t = 2w = 2(0.095 \text{ in}) = 0.190 \text{ in}$, $d_1 = 0.531 \text{ in}$, $D_1 = 0.75 \text{ in}$, and $E = 30,000 \text{ ksi}$.

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln \frac{[1.155(t)+D_1-d_1](D_1+d_1)}{[1.155(t)+D_1+d_1](D_1-d_1)}} = \frac{0.5774\pi(30 \times 10^6 \text{ psi})0.5 \text{ in}^2}{\ln \frac{[1.155(0.190 \text{ in})+0.75 \text{ in}-0.531 \text{ in}](0.75 \text{ in}+0.531 \text{ in})}{[1.155(0.190 \text{ in})+0.75 \text{ in}+0.531 \text{ in}](0.75 \text{ in}-0.531 \text{ in})}} = 53906002.57 \text{ lb/in}$$

For the aluminum plates, the frustum is in two parts.

Aluminum – $t_2 = \frac{l}{2}$ in – $2w = 1.905$ in, $d = 0.5$ in, $D_2 = 0.75$ in + $2 \times 2(0.095 \text{ in}) \tan(30^\circ) = 0.969$ in, and $E = 10,300$ ksi.

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi Ed}{\ln \frac{[1.155(t_2)+D_2-d](D_2+d)}{[1.155(t_2)+D_2+d](D_2-d)}} = \frac{0.5774\pi(10.3 \times 10^6 \text{ psi})0.5 \text{ in}}{\ln \frac{[1.155(1.905 \text{ in})+0.969 \text{ in}-0.5 \text{ in}](0.969 \text{ in}+0.5 \text{ in})}{[1.155(1.905 \text{ in})+0.969 \text{ in}+0.5 \text{ in}](0.969 \text{ in}-0.5 \text{ in})}} = 11343455.71 \text{ lb/in}$$

$t_3 = l/2$ in = 2.095 in, $D_3 = 0.75$ in, $d = 0.5$ in, and $E = 10,300$ ksi.

$$k_3 = \frac{0.5774\pi Ed}{\ln \frac{[1.155(t_3)+D_3-d](D_3+d)}{[1.155(t_3)+D_3+d](D_3-d)}} = \frac{0.5774\pi(10.3 \times 10^6)0.5 \text{ in}}{\ln \frac{[1.155(2.095 \text{ in})+0.75 \text{ in}-0.5 \text{ in}](0.75 \text{ in}+0.5 \text{ in})}{[1.155(2.095 \text{ in})+0.75 \text{ in}+0.5 \text{ in}](0.75 \text{ in}-0.5 \text{ in})}} = 7234488.906 \text{ lb/in}$$

Therefore, the material stiffness is

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{53906002.57} + \frac{1}{11343455.71} + \frac{1}{7234488.906} \right)^{-1} \text{ lb/in} = 4082730.289 \text{ lb/in}$$

BELOW AVERAGE

Darius Rubio

Score: **33/100** Points **33** %

Problem 08.011 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut is used to clamp together two 11-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.011 - Determining joint and member stiffnesses - DEPENDENT MULTI- PART PROBLEM - ASSIGN ALL PARTS	Section: 08.05 Joints - Member Stiffness

Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **35**  mm.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.011.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

Problem 08.011.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **35 ± 2%** mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm.

$$L > 2t + H$$

$$L = 2 \times 11 \text{ mm} + 12.8 \text{ mm} = 34.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$10 \leq t \leq 11$	35
$12 \leq t \leq 13$	40
$14 \leq t \leq 16$	45
$17 \leq t \leq 18$	50
$19 \leq t \leq 21$	55
$22 \leq t \leq 23$	60
$24 \leq t \leq 25$	65

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t = 2 \times 11 \text{ mm} = 22 \text{ mm}$$

$$l_d = L - L_T = 35 \text{ mm} - 34 \text{ mm} = 1 \text{ mm}$$

$$l_t = l - l_d = 22 \text{ mm} - 1 \text{ mm} = 21 \text{ mm}$$

2.

Award: 7 out of 7.00 points

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **1094** ✓ MN/m.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.011.b - Determining joint and member stiffnesses	Section: 08.05 Joints - Member Stiffness

Problem 08.011.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is **1094.6217 ± 2%** MN/m.

Explanation:

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8–1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 21 \text{ mm}) + (115 \text{ mm}^2 \times 1 \text{ mm})} = 1094.6217 \text{ MN/m}$$

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **1153** ✖ MN/m.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.011.c -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

Problem 08.011.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is **3403.4352 ± 2%** MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used. Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe \frac{Bd}{l}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e \left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{22}{1000} \text{ m} \right)} \right) = 3403.4352 \text{ MN/m}$$

Problem 08.012 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

An M 14 × 2 hex-head bolt with a nut and a 14R washer under each of bolt head and the nut is used to clamp together two 13-mm steel plates.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws, Fasteners,
and the Design of
Nonpermanent Joints

Problem 08.012 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints - Member
Stiffness

Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **40**  mm.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.012.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

Problem 08.012.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest 5 mm.

The length of the bolt L is **50 ± 2%** mm.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure to be followed is given in Table 8–7,

From Table A–31, the nut height is $H = 12.8$ mm and the washer is $w = 3.5$ mm.

$$L > 2t + H + 2w$$

$$L = 2 \times 13 \text{ mm} + 12.8 \text{ mm} + 7 \text{ mm} = 45.8 \text{ mm}$$

Material thickness (t)	Bolt length (L)
$11 \leq t \leq 12$	45
$13 \leq t \leq 15$	50
$16 \leq t \leq 17$	55
$18 \leq t \leq 20$	60
$21 \leq t \leq 22$	65
$23 \leq t \leq 25$	70

From Eq. (8–14),

$$L_T = 2(14 \text{ mm}) + 6 \text{ mm} = 34 \text{ mm}$$

From Table 8–7,

$$l = 2t + 2w = 2 \times 13 \text{ mm} + 7 \text{ mm} = 33 \text{ mm}$$

$$l_d = L - L_T = 50 \text{ mm} - 34 \text{ mm} = 16 \text{ mm}$$

$$l_t = l - l_d = 33 \text{ mm} - 16 \text{ mm} = 17 \text{ mm}$$

5.

Award: 0 out of 7.00 points

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is 972.3 ✖ MN/m.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent JointsProblem 08.012.b -
Determining joint and
member stiffnessesSection: 08.05 Joints - Member
Stiffness

Problem 08.012.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness k_b is 822.1146 ± 2% MN/m.

Explanation:

$$A_d = \frac{\pi}{4} \times (14 \text{ mm})^2 = 153.9 \text{ mm}^2$$

From Table 8–1,

$$A_t = 115 \text{ mm}^2$$

The bolt stiffness is calculated as follows:

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{153.9 \text{ mm}^2 \times 115 \text{ mm}^2 \times 207 \text{ GPa}}{(153.9 \text{ mm}^2 \times 17 \text{ mm}) + (115 \text{ mm}^2 \times 16 \text{ mm})} = 822.1146 \text{ MN/m}$$

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is 3332.3 ✖ MN/m.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.012.c -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

Problem 08.012.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness k_m is 2978.4955 ± 2% MN/m.

Explanation:

Now we calculate the member stiffness. Since all members in the stack are steel, Eq. (8–23) can be used.

Note that this equation can only be used when all materials in the joint are the same.

$$k_m = EdAe \frac{Bd}{l}$$

$$E = 207 \text{ GPa} = 207 \times 10^9 \text{ N/m}^2$$

$$d = 14 \text{ mm} = 0.014 \text{ m}$$

$$A = 0.78715$$

$$B = 0.62873$$

$$k_m = 207 \times 10^9 \text{ N/m}^2 \times 0.014 \text{ m} \times 0.78715 \times e \left(\frac{0.62873 \times 0.014 \text{ m}}{\left(\frac{33}{1000} \text{ m} \right)} \right) = 2978.4955 \text{ MN/m}$$

Problem 08.014 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 2-in. steel plate and a 1.25-in. cast-iron plate are compressed with one bolt and nut. The bolt is $\frac{1}{2}$ in-13 UNC.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws, Fasteners,
and the Design of
Nonpermanent Joints

Problem 08.014 -
Determining joint and
member stiffnesses with
dissimilar joint materials
- DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints - Member
Stiffness

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **4**  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.014.a -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

Problem 08.014.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B = 2 \text{ in} + 1.25 \text{ in} = 3.25 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437 \text{ in}$.

$$L \geq l + H$$

$$L = 3.25 \text{ in} + 0.437 \text{ in} = 3.687 \text{ in}$$

Rounding up the value L to the nearest $\frac{1}{4}$ in.

$$L = 3.75 \text{ in}$$

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 3.75 \text{ in} - 1.25 \text{ in} = 2.5 \text{ in}$$

$$l_t = l - l_d = 3.25 \text{ in} - 2.5 \text{ in} = 0.75 \text{ in}$$

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1.5** ✖ Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.014.b -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

Problem 08.014.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1.6648 ± 2%** Mlb/in.

Explanation:

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.75 \text{ in}) + (0.1419 \text{ in}^2 \times 2.5 \text{ in})} = 1.6648 \text{ Mlb/in}$$

Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **7.21**  Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.014.c -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

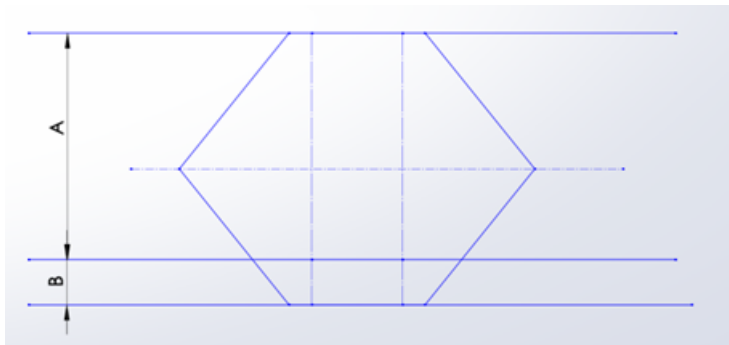
Problem 08.014.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **7.4088 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$, we must calculate three K 's for the frusta.



For the steel plate, the frustum is in two parts.

Steel plate:

$t = 1.625$ in, $d = 0.5$ in, $D = 0.75$ in, and $E = 30,000$ kpsi

$$K_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.625 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.625 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 22.2288 \text{ Mlb/in}$$

$$t_2 = A - \frac{l}{2} \text{ in} = 2 \text{ in} - \left(\frac{3.25}{2}\right) \text{ in} = 0.375 \text{ in}$$

$$D_2 = 1.5d + 2(l - h) \tan(30^\circ) = 1.5 \times 0.5 \text{ in} + 2(3.25 \text{ in} - 2 \text{ in}) \times \tan(30^\circ) = 2.1934 \text{ in}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 2.1934 \text{ in} - 0.5 \text{ in})(2.1934 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 2.1934 \text{ in} + 0.5 \text{ in})(2.1934 \text{ in} - 0.5 \text{ in})}\right)} = 346.0294 \text{ Mlb/in}$$

For the cast-iron plate, the frustum is in one part. This will generate k_3 .

Cast iron:

$t_3 = 1.25 \text{ in}$, $d = 0.5 \text{ in}$, $D = 0.75 \text{ in}$, and $E = 14500 \text{ kpsi}$ (from Table 8–8).

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{(1.155 \times 1.25 \text{ in} + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{(1.155 \times 1.25 \text{ in} + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}\right)} = 11.4813 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}\right)^{-1} = \left(\frac{1}{22.2288 \text{ Mlb/in}} + \frac{1}{346.0294 \text{ Mlb/in}} + \frac{1}{11.4813 \text{ Mlb/in}}\right)^{-1} = 7.4088 \text{ Mlb/in}$$

Problem 08.015 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 1.25-in. steel plate and a 0.5-in. gray cast iron plate are compressed with one bolt and nut. The bolt is ½ in-13 UNC. A ½ N American Standard plain washer is used under the head of the bolt and under the nut.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Chapter: 08 Screws, Fasteners,
and the Design of
Nonpermanent Joints

Problem 08.015 -
Determining joint and
member stiffnesses with
dissimilar joint materials
- DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints - Member
Stiffness

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is **2.5**  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.015.a -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

Problem 08.015.a - Determining joint and member stiffnesses with dissimilar joint materials

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7,

From Table A–31,

$$l = A + B + 2w = 1.25 \text{ in} + 0.5 \text{ in} + 2(0.095 \text{ in}) = 1.94 \text{ in}$$

where

A = steel plate thickness

B = cast-iron plate thickness

The nut height is $H = 0.437 \text{ in}$ and the washer thickness is $w = 0.095 \text{ in}$.

$$L \geq l + H$$

$$L = 1.94 \text{ in} + 0.437 \text{ in} = 2.377 \text{ in}$$

Rounding up the value of L to the nearest $\frac{1}{4}$ in.

$$L = 2.5 \text{ in}$$

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.67**  Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.015.b -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

Problem 08.015.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **2.6714 ± 2%** Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 2.5 \text{ in} - 1.25 \text{ in} = 1.25 \text{ in}$$

$$l_t = l - l_d = 1.94 \text{ in} - 1.25 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 1.25 \text{ in})} = 2.6714 \text{ Mlb/in}$$

Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **9.65** ✓ Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.015.c -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

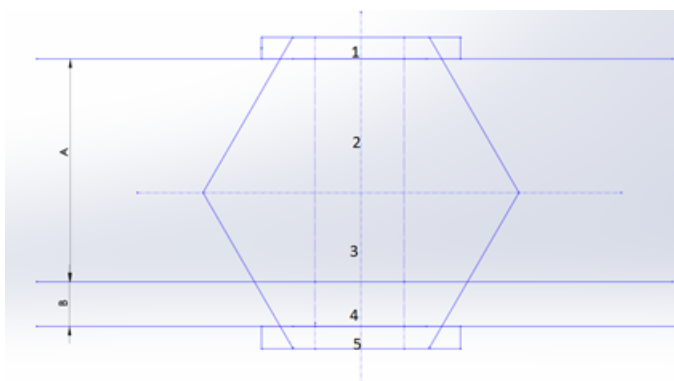
Problem 08.015.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **9.6576 ± 2%** Mlb/in.

Explanation:

Now we calculate the member stiffness. This requires the calculation for each frustum in the stack. As $A \neq B$ and the washer material is different than the bottom piece being joined, we must calculate five k 's for the frusta. Note that the top washer and the top piece are both steel. However, the ID of the washer is different than that of the plates.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the steel plate, the frustum is in two parts.

Steel:

$$t_2 = \frac{A+B}{2} \text{ in} = \left(\frac{1.25 + 0.5}{2} \right) \text{ in} = 0.875 \text{ in}$$

$$d = 0.5 \text{ in}$$

$$D_2 = 0.75 \text{ in} + 2(0.095 \text{ in})\tan(30^\circ) = 0.860 \text{ in}$$

$$E = 30000 \text{ kpsi}$$

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.875 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.875 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 34.828 \text{ Mlb/in}$$

$$t_3 = A - t_2 = 1.25 \text{ in} - 0.875 \text{ in} = 0.375 \text{ in}$$

$$D_3 = 0.860 \text{ in} + 2(0.5 \text{ in})\tan(30^\circ) = 1.4374 \text{ in}$$

$$E = 30000 \text{ kpsi}$$

$$k_3 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_3) + D_3 - d)(D_3 + d)}{(1.155(t_3) + D_3 + d)(D_3 - d)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.375 \text{ in}) + 1.4374 \text{ in} - 0.5 \text{ in})(1.4374 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.375 \text{ in}) + 1.4374 \text{ in} + 0.5 \text{ in})(1.4374 \text{ in} - 0.5 \text{ in})}\right)} = 152.7975 \text{ Mlb/in}$$

For the cast-iron cylinder, the frustum is in one part. This will generate k_4 .

Cast iron:

$$t_4 = 0.5 \text{ in}$$

$$d = 0.5 \text{ in}$$

$$D = 0.860 \text{ in}$$

$$E = 14500 \text{ kpsi (from Table 8–8)}$$

$$k_4 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_4) + D_4 - d)(D_4 + d)}{(1.155(t_4) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (14.5 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 0.5 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 0.5 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 21.8024 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{2}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} + \frac{1}{k_4} \right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{1}{34.828 \text{ Mlb/in}} + \frac{1}{152.7975 \text{ Mlb/in}} + \frac{1}{21.8024 \text{ Mlb/in}} \right)^{-1} = 9.6576 \text{ Mlb/in}$$

Problem 08.017 - Determining joint and member stiffnesses - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 1.25 in thick and are compressed with one bolt and nut. Washers are used under the head of the bolt and under the nut.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

Washer properties: steel; ID = 0.531 in; OD = 1.062 in; thickness = 0.095 in

Nut properties: steel; height = 7/16 in

Bolt properties: 1/2-13 UNC grade 8

Plate properties: aluminum; $E = 10.4$ Mpsi; $S_U = 47$ kpsi; $S_y = 25$ kpsi

References

Section Break

Chapter: 08 Screws, Fasteners,
and the Design of
Nonpermanent Joints

Problem 08.017 -
Determining joint and
member stiffnesses -
DEPENDENT MULTI-
PART PROBLEM -
ASSIGN ALL PARTS

Section: 08.05 Joints - Member
Stiffness

Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is 4.75  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.017.a -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

Problem 08.017.a - Determining joint and member stiffnesses

Determine a suitable length for the bolt, rounded up to the nearest $\frac{1}{4}$ in.

The length of the bolt is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. The procedure is followed as given in Table 8–7.

From Table A–31,

$$l = 2t + 2w = 2(1.25 \text{ in}) + 2(0.095 \text{ in}) = 2.69 \text{ in}$$

where

t = steel plate thickness

The nut height is $H = 0.437$ in and the washer thickness is $w = 0.095$ in.

$$L \geq l + H$$

$$L = 2.69 \text{ in} + 0.437 \text{ in} = 3.127 \text{ in}$$

Rounding up to the nearest $\frac{1}{4}$ in,

$$L = 3.25 \text{ in}$$

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is **1.32** ✖ Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.017.b -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

Problem 08.017.b - Determining joint and member stiffnesses

Determine the bolt stiffness.

The bolt stiffness is **1.9933 ± 2%** Mlb/in.

Explanation:

From Eq. (8–14),

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7,

$$l_d = L - L_T = 3.25 \text{ in} - 1.25 \text{ in} = 2 \text{ in}$$

$$l_t = l - l_d = 2.69 \text{ in} - 2 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi}{4} \times (0.5 \text{ in})^2 = 0.1963 \text{ in}^2$$

From Table 8–1,

$$A_t = 0.1419 \text{ in}^2$$

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 \times 0.1419 \text{ in}^2 \times 30 \times 10^6 \text{ psi}}{(0.1963 \text{ in}^2 \times 0.69 \text{ in}) + (0.1419 \text{ in}^2 \times 2 \text{ in})} = 1.9933 \text{ Mlb/in}$$

Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is **4.19**  Mlb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.017.c -
Determining joint and
member stiffnesses**

Section: 08.05 Joints - Member
Stiffness

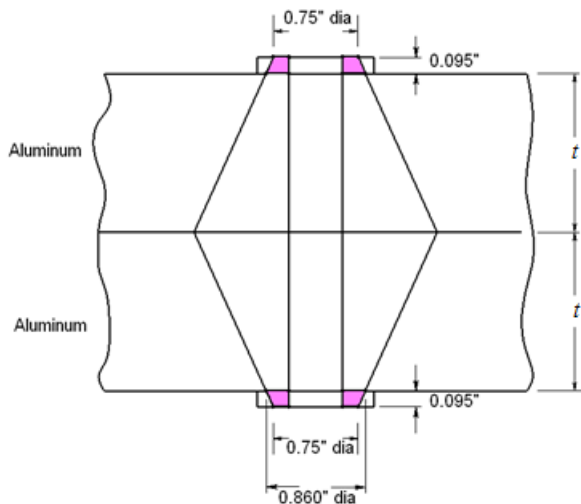
Problem 08.017.c - Determining joint and member stiffnesses

Determine the material stiffness.

The material stiffness is **4.7458 ± 2%** Mlb/in.

Explanation:

Now calculate the member stiffness. As the thickness and material of the plates are the same and the material or the washers are different, we must calculate two k 's for the frusta.



For the washers,

Steel:

$w = 0.095$ in, $d_1 = 0.531$ in (Table A-32), $D_1 = 0.75$ in, and $E = 30,000$ kpsi

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln\left(\frac{(1.155(w) + D_1 - d_1)(D_1 + d_1)}{(1.155(w) + D_1 + d_1)(D_1 - d_1)}\right)} = \frac{0.5774 \times \pi \times (30 \times 10^6 \text{ psi}) \times 0.531 \text{ in}}{\ln\left(\frac{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} - 0.531 \text{ in})(0.75 \text{ in} + 0.531 \text{ in})}{((1.155 \times 0.095 \text{ in}) + 0.75 \text{ in} + 0.531 \text{ in})(0.75 \text{ in} - 0.531 \text{ in})}\right)} = 89.1957 \text{ Mlb/in}$$

For the aluminum plates,

$t_2 = 1.25$ in, $d = 0.5$ in, $D = 0.860$ in, and $E = 10.4$ Mpsi

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_4 - d)(D_4 + d)}{(1.155(t_2) + D_4 + d)(D_4 - d)}\right)} = \frac{0.5774 \times \pi \times (10.4 \times 10^6 \text{ psi}) \times 0.5 \text{ in}}{\ln\left(\frac{((1.155 \times 1.25 \text{ in}) + 0.86 \text{ in} - 0.5 \text{ in})(0.86 \text{ in} + 0.5 \text{ in})}{((1.155 \times 1.25 \text{ in}) + 0.86 \text{ in} + 0.5 \text{ in})(0.86 \text{ in} - 0.5 \text{ in})}\right)} = 10.6218 \text{ Mlb/in}$$

From Eq. (8–18),

$$k_m = \left(\frac{2}{k_1} + \frac{2}{k_2}\right)^{-1} = \left(\frac{2}{89.1957 \text{ Mlb/in}} + \frac{2}{10.6218 \text{ Mlb/in}}\right)^{-1} = 4.7458 \text{ Mlb/in}$$

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Two identical aluminum plates are each 1.5 in (t) thick, and are compressed with one bolt and nut. No washers are used under the head of the bolt but two washers are used under the nut.

Washer properties: steel ID = 0.531 in, OD = 1.062 in, and thickness = 0.095 in.

Nut properties: steel height = 7/16 in

Bolt properties: 1/2-13 UNC grade 8

Plate properties: aluminum $E = 10.3$ Mpsi, $S_U = 47$ kpsi, and $S_Y = 25$ kpsi.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.018 - Determining joint and member stiffnesses with dissimilar joint materials - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS Section: 08.05 Joints - Member Stiffness

Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is **3.62**  in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.018.a -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

Problem 08.018.a - Determining joint and member stiffnesses with dissimilar joint materials

Find a suitable bolt length.

The bolt length is in.

Explanation:

In order to calculate the stiffness constants, you must first determine the length of each member in the frusta. Following the procedure of Table 8–7, we have

$l = 2t + 2w = 2 \times 1.5 \text{ in} + 2 \times 0.095 \text{ in} = 3.19 \text{ in}$. The nut height is $H = 0.437 \text{ in}$ (from Table A–31), and the washer thickness is $w = 0.095 \text{ in}$.

$L \geq l + H = 3.19 \text{ in} + 0.437 \text{ in} = 3.627 \text{ in}$.

Rounding up the value to the nearest $\frac{1}{4}$ in gives $L = 3.75 \text{ in}$.

This is the required bolt length.

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1704972** ✓ lb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.018.b -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

Problem 08.018.b - Determining joint and member stiffnesses with dissimilar joint materials

Determine the bolt stiffness.

The bolt stiffness is **1704720.959 ± 2%** lb/in.

Explanation:

From Eq. 8–14,

$$L_T = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8–7, we have

$$l_d = L - L_T = 3.75 \text{ in} - 1.25 \text{ in} = 2.5 \text{ in}$$

$$l_t = l - l_d = 3.19 \text{ in} - 2.5 \text{ in} = 0.69 \text{ in}$$

$$A_d = \frac{\pi (0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8–1, $A_t = 0.1419 \text{ in}^2$.

Therefore, from Eq. (8–17), the bolt stiffness is

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1963 \text{ in}^2 (0.1419 \text{ in}^2) (30 \times 10^6 \text{ psi})}{0.1963 \text{ in}^2 (0.69 \text{ in}) + 0.1419 \text{ in}^2 (2.5 \text{ in})} = 1704720.959 \text{ lb/in}$$

Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **4207699** ✖ lb/in.

References

Numeric Response

Chapter: 08 Screws,
Fasteners, and the Design of
Nonpermanent Joints

**Problem 08.018.c -
Determining joint and
member stiffnesses
with dissimilar joint
materials**

Section: 08.05 Joints - Member
Stiffness

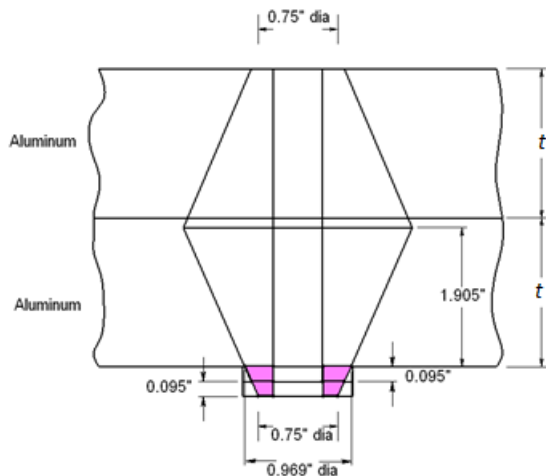
Problem 08.018.c - Determining joint and member stiffnesses with dissimilar joint materials

Determine the material stiffness.

The material stiffness is **4359093.885 ± 2%** lb/in.

Explanation:

The member stiffness requires the calculation for each frustum in the stack. As per the figure below, we will need to calculate three k values, the washers, the upper frustum, and the lower frustum.



For the washers,
Steel – $t = 2w = 2(0.095 \text{ in}) = 0.190 \text{ in}$, $d_1 = 0.531 \text{ in}$, $D_1 = 0.75 \text{ in}$, and $E = 30,000 \text{ ksi}$.

$$k_1 = \frac{0.5774 \pi (E) d_1}{\ln \frac{[1.155(t) + D_1 - d_1](D_1 + d_1)}{[1.155(t) + D_1 + d_1](D_1 - d_1)}} = \frac{0.5774 \pi (30 \times 10^6 \text{ psi}) 0.5 \text{ in}^2}{\ln \frac{[1.155(0.190 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}](0.75 \text{ in} + 0.5 \text{ in})}{[1.155(0.190 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}](0.75 \text{ in} - 0.5 \text{ in})}} = 53906002.57 \text{ lb/in}$$

For the aluminum plates, the frustum is in two parts.

Aluminum – $t_2 = \frac{l}{2}$ in – $2w = 1.405$ in, $d = 0.5$ in, $D_2 = 0.75$ in + $2 \times 2 \left(0.095 \text{ in} \right) \tan \left(30^\circ \right) = 0.969$ in, and $E = 10,300$ ksi.

From Eq. (8-20),

$$k_2 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_2) + D_2 - d] (D_2 + d)}{[1.155 (t_2) + D_2 + d] (D_2 - d)}} = \frac{0.5774 \pi \left(10.3 \times 10^6 \text{ psi} \right) 0.5 \text{ in}}{\ln \frac{[1.155 (1.405 \text{ in}) + 0.969 \text{ in} - 0.5 \text{ in}] (0.969 \text{ in} + 0.5 \text{ in})}{[1.155 (1.405 \text{ in}) + 0.969 \text{ in} + 0.5 \text{ in}] (0.969 \text{ in} - 0.5 \text{ in})}} = 12439209.06 \text{ lb/in}$$

$t_3 = l/2$ in = 1.595 in, $D_3 = 0.75$ in, $d = 0.5$ in, and $E = 10,300$ ksi.

$$k_3 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_3) + D_3 - d] (D_3 + d)}{[1.155 (t_3) + D_3 + d] (D_3 - d)}} = \frac{0.5774 \pi \left(10.3 \times 10^6 \right) 0.5 \text{ in}}{\ln \frac{[1.155 (1.595 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}] (0.75 \text{ in} + 0.5 \text{ in})}{[1.155 (1.595 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}] (0.75 \text{ in} - 0.5 \text{ in})}} = 7664967.28 \text{ lb/in}$$

Therefore, the material stiffness is

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{53906002.57} + \frac{1}{12439209.06} + \frac{1}{7664967.28} \right)^{-1} \text{ lb/in} = 4359093.885 \text{ lb/in}$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 12

TOPICS COVERED IN HOMEWORK 12

Chapter 8 Bolts Design: 8-29, 8-30, 8-31, 8-32, 8-33, 8-34

**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.6$ Mlbf/in and the stiffness of the members is $k_m = 16$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. Assume the bolts are preloaded to 75 percent of the proof load.

References

Section Break	Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Difficulty: Medium
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1.

Award: 5.00 points

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is .**Explanation:**From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2.6 \text{ Mlbf/in}}{(2.6 + 16) \text{ Mlbf/in}} = 0.1398$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32), we get

$$F_i = 0.75A_t S_p = 0.75(0.1419 \text{ in}^2)(120 \text{ ksi}) = 12.77 \text{ kips}$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80 \text{ kip}}{6} = 13.33 \text{ kip}$$

Therefore, from Eq. (8-28), the yielding factor of safety is

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{(120 \times 10^3 \text{ psi})(0.1419 \text{ in}^2)}{0.1398(13330 \text{ lbf}) + 12770 \text{ lbf}} = 1.1636$$

The yielding factor of safety is 1.1636.

References**Numeric Response**Problem 08.029.a -
Static bolt factor of
safety

Difficulty: Medium

2.

Award: 5.00 points

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is .

Explanation:

Therefore, from Eq. (8-29), the load factor of safety is

$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{(120 \times 10^3 \text{ psi})(0.1419 \text{ in}^2) - 12770 \text{ lbf}}{0.1398(13330 \text{ lbf})} = 2.2852$$

The load factor of safety is 2.2852.

References**Numeric Response**

Problem 08.029.b -
Static bolt factor of
safety

Difficulty: Medium

3.

Award: 5.00 points

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is $1.1137 \pm 2\%$.

Explanation:

From Eq. (8-30), the joint separation factor of safety is

$$n_O = \frac{F_i}{P(1-C)} = \frac{12770 \text{ lbf}}{13330 \text{ lbf}(1-0.1398)} = 1.1137$$

The factor of safety based on the joint separation is 1.1137.

References**Numeric
Response**

Problem 08.029.c -
Static bolt factor of
safety

Difficulty: Medium

Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.6$ Mlbf/in and the stiffness of the members is $k_m = 17$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. It is desired to find the range of torque that a mechanic could apply to initially preload the bolts without expecting failure once the joint is loaded. Assume a torque coefficient of $K = 0.2$.

References

Section Break	Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS	Difficulty: Medium
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4.

Award: 5.00 points

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **15259.7347 ± 2%** lbf.

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2.6 \text{ Mlbf/in}}{(2.6 + 17) \text{ Mlbf/in}} = 0.1327$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80000 \text{ lbf}}{6} = 13330 \text{ lbf}$$

The proof factor of safety, from Eq. (8-28), is used to find F_i :

$$F_{ia} = \frac{S_p A_t}{\eta_p} - CP = \frac{120 \times 10^3 \text{ psi} (0.1419 \text{ in}^2)}{1} - 0.1327(13330 \text{ lbf}) = 15259.7347 \text{ lbf}$$

The maximum bolt preload that can be applied is 15259.7347 lbf.

References**Numeric Response**

Problem 08.030.a -
Static bolt factor of
safety related to
preload

Difficulty: Medium

5.

Award: 5.00 points

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11561.7347 ± 2%** lbf.

Explanation:

The equation for the separation factor of safety is determined by Eq. (8-30):

$$F_{ib} = n_o P(1 - C) = 1(13330 \text{ lbf})(1 - 0.1327) = 11561.7347 \text{ lbf}$$

The minimum bolt preload that can be applied is 11561.7347 lbf.

References**Numeric
Response**

Problem 08.030.b -
Static bolt factor of
safety related to
preload

Difficulty: Medium

6.

Award: 5.00 points

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf·in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is 1341.0735 ± 2% lbf·in.

Explanation:

The torque can be determined by Eq. 8-27:

$$T = KF_i d$$

$$T = KF_i d = 0.2 \frac{F_{ia} + F_{ib}}{2} (0.5)$$

$$T = 0.2 \frac{15259.7347 \text{ lbf} + 11561.7347 \text{ lbf}}{2} (0.5 \text{ in}) = 1341.0735 \text{ lbf·in}$$

The torque that should be specified for preloading the bolt is 1341.0735 lbf·in.

References**Numeric Response**

Problem 08.030.c -
Static bolt factor of
safety related to
preload

Difficulty: Medium

Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 0.8$ MN/mm and the stiffness of the members is $k_m = 3.6$ MN/mm per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use M6 $\times$ 1 class 5.8 bolts with rolled threads.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS	Difficulty: Medium
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7.

Award: 7.00 points

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is $83952 \pm 2\%$ N.

Explanation:

From, Table 8–9, $S_p = 380$ MPa.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{0.8 \text{ MN/mm}}{(0.8 + 3.6) \text{ MN/mm}} = 0.1818$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75A_t S_p = 0.75(20.1 \text{ mm}^2)(380 \text{ MPa}) = 5730 \text{ N}$$

The proof factor of safety equation is given by Eq. (8-28):

$$P_{\max} = \left(\frac{S_p A_t}{n_p} - F_i \right) \frac{N}{C} = \left(\frac{380 \times 10^6 \text{ Pa} (2.01 \times 10^{-5} \text{ m}^2)}{1} - 5730 \text{ N} \right) \frac{8}{0.1818} = 83952 \text{ N}$$

The maximum external load for this case is 83952 N.

References**Numeric Response**

Problem 08.031.a -
Static bolt factor of
safety related to
preload

Difficulty: Medium

8.

Award: 7.00 points

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is $56026.6667 \pm 2\%$ N.

Explanation:

The equation for the separation factor of safety is given Eq. (8-30):

$$P_{max} = \frac{F_i}{n_o} \frac{1}{(1-C)} N = \frac{5730 \text{ N}}{1} \frac{1}{(1-0.1818)} (8) = 56026.6667 \text{ N}$$

The maximum external load for this case is 56026.6667 N.

References**Numeric Response**

Problem 08.031.b -
Static bolt factor of
safety related to
preload

Difficulty: Medium

Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS

For a bolted assembly, the stiffness of each bolt is $k_b = 4$ Mlbf/in and the stiffness of the members is $k_m = 14$ Mlbf/in per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. A fluctuating external load is applied to the entire joint with $P_{\max} = 80$ kips and $P_{\min} = 20$ kips. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS	Difficulty: Easy
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9.

Award: 8.00 points

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is $5 \pm 2\%$.

Explanation:

From Table 8–9,

$$S_p = 1.2(10^5) \text{ psi}$$

The stiffness constant is calculated as

$$C = \frac{k_b}{(k_b + k_m)}$$

$$C = \frac{4 \text{ Mlbf/in}}{(4 \text{ Mlbf/in} + 14 \text{ Mlbf/in})}$$

$$C = 0.2222$$

For a non-permanent connection, using Eqs. (8–31) and (8–32),

$$F_i = 0.75A_t S_p = 0.75(0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) = 12770 \text{ lbf}$$

The proof factor of safety is calculated as

$$N = \frac{CP_{\max}}{(S_p A_t - F_i)}$$

$$N = \frac{0.2222(80 \times 10^3 \text{ lbf})}{((0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) - 12770 \text{ lbf})}$$

$$N = 5$$

Minimum number of bolts necessary to avoid yielding of the bolts is 5.

References

**Numeric
Response**

Problem 08.032.a - Difficulty: Easy
Dynamic bolt factor of
safety related to
preload

10.

Award: 8.00 points

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is **5 ± 2%**.

Explanation:

The separation factor of safety is calculated as

$$N = \frac{P_{\max}(1 - C)}{F_i}$$

$$N = \frac{(80 \times 10^3 \text{ lbf})(1 - 0.2222)}{12770 \text{ lbf}}$$

$$N = 5$$

The minimum number of bolts necessary to avoid joint separation is 5.

References**Numeric
Response**

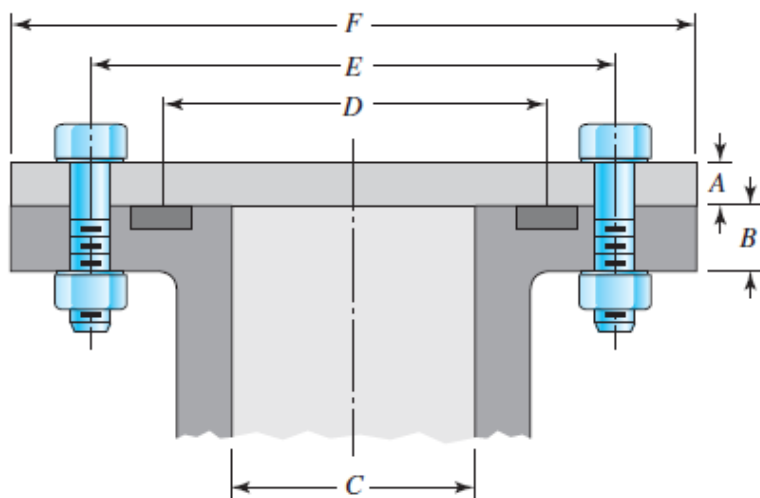
Problem 08.032.b - Difficulty: Easy
Dynamic bolt factor of
safety related to
preload

11.

Award: 20.00 points

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	7.35 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is $55 \pm 2\%$ mm.

The yielding factor of safety is $1.2887 \pm 2\%$.

The load factor is $9.0863 \pm 2\%$.

The joint separation factor is $9.6604 \pm 2\%$.

Explanation:

$$l = 20 \text{ mm} + 20 \text{ mm} = 40 \text{ mm}$$

From Table A-31, the nut height is $H = 10.8 \text{ mm}$.

$$L \geq l + H = 40 \text{ mm} + 10.8 \text{ mm} = 50.8 \text{ mm} \cong 55 \text{ mm}$$

From Eq. (8-14), we have

$$L_T = 2d + 6 \text{ mm} = 2(12 \text{ mm}) + 6 \text{ mm} = 30 \text{ mm}$$

From Table 8-7,

$$l_d = L - L_T = 55 \text{ mm} - 30 \text{ mm} = 25 \text{ mm}$$

$$l_t = l - l_d = 40 \text{ mm} - 25 \text{ mm} = 15 \text{ mm}$$

$$A_d = \frac{\pi (12 \text{ mm})^2}{4} = 113.1 \text{ mm}^2$$

From Table 8-1, we have $A_t = 84.3 \text{ mm}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(113.1 \text{ mm}^2)(84.3 \text{ mm}^2)207 \times 10^3 \text{ N/mm}^2}{(113.1 \text{ mm}^2)(15 \text{ mm}) + (84.3 \text{ mm}^2)(25 \text{ mm})} = 518.8 \times 10^3 \text{ N/mm} = 518.8 \text{ MN/m}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

$$k_1 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_1) + D - d)(D + d)}{(1.155(t_1) + D + d)(D - d)}\right)} = \frac{0.5774\pi(207 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 4470 \times 10^3 \text{ N/mm} = 4470 \text{ MN/m}$$

For cast iron, from Table 8-8, we have $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774\pi(100 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 2159 \times 10^3 \text{ N/mm} = 2159 \text{ MN/m}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2}\right)^{-1} = \left(\frac{1}{4470 \text{ MN/m}} + \frac{1}{2159 \text{ MN/m}}\right)^{-1} = 1456 \text{ MN/m}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{518.8 \text{ MN/m}}{518.8 \text{ MN/m} + 1456 \text{ MN/m}} = 0.263$$

From Table 8-11, we have $S_p = 650 \text{ MPa}$.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75A_t S_p = 0.75(84.3 \text{ mm}^2)(650 \text{ MPa}) \times 10^{-3} = 41.1 \text{ kN}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 100 mm. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(7.35 \text{ N/mm}^2) \pi (100 \text{ mm})^2}{4 \times 10} = 5772.6765 \text{ N/bolt} = 5.7727 \text{ kN/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{(650 \text{ N/mm}^2)(84.3 \text{ mm}^2)}{(0.263)(5772.6765 \text{ N}) + (41.10 \times 10^3 \text{ N})} = 1.2887$$

The overall factor of safety, from Eq. (8-29), is

$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{(650 \text{ N/mm}^2)(84.3 \text{ mm}^2) - (41.10 \times 10^3 \text{ N})}{(0.263)(5772.6765 \text{ N})} = 9.0863$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_i}{P(1-C)} = \frac{41.10 \text{ kN}}{5.7727 \text{ kN}(1-0.263)} = 9.6604$$

The suitable bolt length is 55 mm.

The yielding factor of safety is 1.2887.

The load factor is 9.0863.

The joint separation factor is 9.6604.

References

Numeric Response

Problem 08.033 -
Selection of a suitable
bolt length

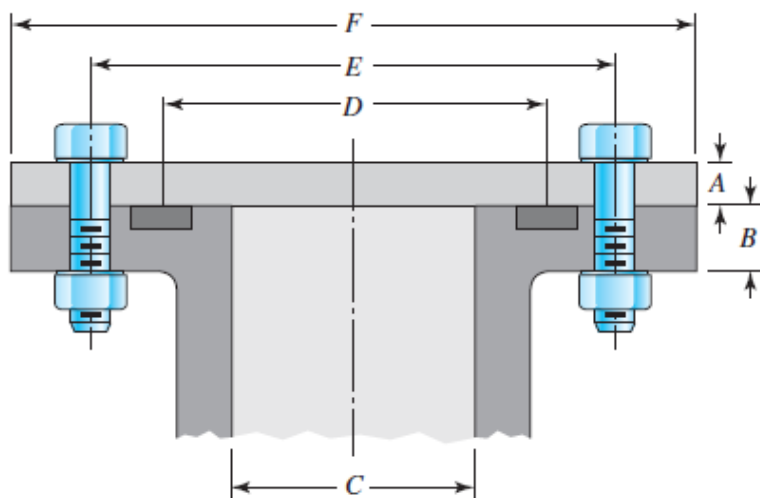
Difficulty: Hard

12.

Award: 20.00 points

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1430 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is $1.75 \pm 2\%$ in.

The yielding factor of safety is $1.2748 \pm 2\%$.

The load factor is $7.3207 \pm 2\%$.

The joint separation factor is $9.3836 \pm 2\%$.

Explanation:

From Table A-31, the nut height is $H = 0.4375$ in.

$$l = 0.5 \text{ in} + 0.625 \text{ in} = 1.125 \text{ in}$$

$$L \geq l + H = 1.125 \text{ in} + 0.4375 \text{ in} = 1.563 \text{ in} \cong 1.75 \text{ in}$$

From Eq. (8-13), we have

$$L_T = 2d + 0.25 \text{ in} = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8-7,

$$l_d = L - L_T = 1.75 \text{ in} - 1.25 \text{ in} = 0.5 \text{ in}$$

$$l_t = l - l_d = 1.125 \text{ in} - 0.5 \text{ in} = 0.625 \text{ in}$$

$$A_d = \frac{\pi (0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8-2, we have $A_t = 0.1419 \text{ in}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(0.1963 \text{ in}^2) (0.1419 \text{ in}^2) (30 \times 10^6 \text{ lbf/in}^2)}{(0.1963 \text{ in}^2) (0.625 \text{ in}) + (0.1419 \text{ in}^2) (0.5 \text{ in})} = 4.316 \times 10^6 \text{ lbf/in} = 4.316 \text{ Mlbf/in}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 0.5 \text{ in}$, $d = 0.5 \text{ in}$, and $D = 0.75 \text{ in}$.

$$k_1 = \frac{0.5774 \pi (E) d_1}{\ln \frac{[1.155 (t) + D_1 - d_1] (D_1 + d_1)}{[1.155 (t) + D_1 + d_1] (D_1 - d_1)}} = \frac{0.5774 \pi (30 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155 (0.5 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}) (0.75 \text{ in} + 0.5 \text{ in})}{(1.155 (0.5 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}) (0.75 \text{ in} - 0.5 \text{ in})}} = 33.30 \times 10^6 \text{ lbf/in} = 33.30 \text{ Mlbf/in}$$

Cast iron has two frusta. The midpoint of complete joint is at $\frac{1.125 \text{ in}}{2} = 0.5625 \text{ in}$.

For upper frustum,

$$d = 0.5 \text{ in}$$

$$t = 0.5625 \text{ in} - 0.5 \text{ in} = 0.0625 \text{ in}$$

$$D = 0.75 \text{ in} + 2(0.5 \text{ in}) \tan 30^\circ = 1.327 \text{ in}$$

From Eq. (8-20),

$$k_2 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_2) + D_2 - d] (D_2 + d)}{[1.155 (t_2) + D_2 + d] (D_2 - d)}} = \frac{0.5774 \pi (14.5 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155 (0.0625 \text{ in}) + 1.327 \text{ in} - 0.5 \text{ in}) (1.327 \text{ in} + 0.5 \text{ in})}{(1.155 (0.0625 \text{ in}) + 1.327 \text{ in} + 0.5 \text{ in}) (1.327 \text{ in} - 0.5 \text{ in})}} = 292.7 \times 10^6 \text{ lbf/in} = 292.7 \text{ Mlbf/in}$$

For lower frustum, $t = 0.5625 \text{ in}$, $d = 0.5 \text{ in}$, and $D = 0.75 \text{ in}$.

$$k_3 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_3) + D_3 - d] (D_3 + d)}{[1.155 (t_3) + D_3 + d] (D_3 - d)}} = \frac{0.5774 \pi (14.5 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155 (0.5625 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}) (0.75 \text{ in} + 0.5 \text{ in})}{(1.155 (0.5625 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}) (0.75 \text{ in} - 0.5 \text{ in})}} = 15.26 \times 10^6 \text{ lbf/in} = 15.26 \text{ Mlbf/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{33.30 \text{ Mlbf/in}} + \frac{1}{292.7 \text{ Mlbf/in}} + \frac{1}{15.26 \text{ Mlbf/in}} \right)^{-1} = 10.10 \text{ Mlbf/in}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{4.316 \text{ Mlbf/in}}{4.316 \text{ Mlbf/in} + 10.10 \text{ Mlbf/in}} = 0.299$$

From Table 8-9, we have $S_p = 85 \text{ kpsi}$.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (0.1419 \text{ in}^2) (85 \text{ kpsi}) = 9.05 \text{ kips}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 3.5 in. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(1430 \text{ lbf/in}^2) \pi (3.5 \text{ in})^2}{4 \times 10} = 1375.8212 \text{ lbf/bolt} = 1.3758 \text{ kips/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2)}{(0.299) (1375.8212 \text{ lbf}) + (9.05 \times 10^3 \text{ lbf})} = 1.2748$$

The overload factor of safety, from Eq. (8-29), is

$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2) - (9.05 \times 10^3 \text{ lbf})}{(0.299) (1375.8212 \text{ lbf})} = 7.3207$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_i}{P(1-C)} = \frac{9.05 \text{ kips}}{1.3758 \text{ kips}(1-0.299)} = 9.3836$$

The suitable bolt length is 1.75 in.

The yielding factor of safety is 1.2748.

The load factor is 7.3207.

The joint separation factor is 9.3836.

References

Numeric Response

Problem 08.034 - Difficulty: Hard
Selection of a suitable bolt length

ABOVE AVERAGE

Joseph Steele

Score: 100/100 Points 100 %

Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.4$ Mlbf/in and the stiffness of the members is $k_m = 16$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. Assume the bolts are preloaded to 75 percent of the proof load.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

1.

Award: 5 out of 5.00 points

 You received credit for this question in a previous attempt

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is **1.17** .

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.029.a - Static bolt factor of safety	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is **1.1736 ± 2%**.

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2.4 \text{ Mlbf/in}}{(2.4 + 16) \text{ Mlbf/in}} = 0.1304$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32), we get

$$F_i = 0.75A_tS_p = 0.75(0.1419 \text{ in}^2)(120 \text{ ksi}) = 12.77 \text{ kips}$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80 \text{ kip}}{6} = 13.33 \text{ kip}$$

Therefore, from Eq. (8-28), the yielding factor of safety is

$$n_p = \frac{S_pA_t}{CP + F_i} = \frac{(120 \times 10^3 \text{ psi})(0.1419 \text{ in}^2)}{0.1304(13330 \text{ lbf}) + 12770 \text{ lbf}} = 1.1736$$

2.

Award: 5 out of 5.00 points

 You received credit for this question in a previous attempt

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is **2.44** .

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.029.b - Static bolt factor of safety Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is **2.449 ± 2%**.

Explanation:

Therefore, from Eq. (8-29), the load factor of safety is

$$n_L = \frac{S_P A_t - F_i}{CP} = \frac{(120 \times 10^3 \text{ psi}) (0.1419 \text{ in}^2) - 12770 \text{ lbf}}{0.1304(13330 \text{ lbf})} = 2.449$$

3.

Award: 5 out of 5.00 points

 You received credit for this question in a previous attempt

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is **1.1** .

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.029.c - Static bolt factor of safety Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is **1.1017 ± 2%**.

Explanation:

From Eq. (8-30), the joint separation factor of safety is

$$n_O = \frac{F_i}{P(1-C)} = \frac{12770 \text{ lbf}}{13330 \text{ lbf}(1-0.1304)} = 1.1017$$

Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2$ Mlbf/in and the stiffness of the members is $k_m = 16$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. It is desired to find the range of torque that a mechanic could apply to initially preload the bolts without expecting failure once the joint is loaded. Assume a torque coefficient of $K = 0.2$.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

4. Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **15546.88**  lbf.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030.a - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **15546.8889 ± 2%** lbf.

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2 \text{ Mlbf/in}}{(2 + 16) \text{ Mlbf/in}} = 0.1111$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80000 \text{ lbf}}{6} = 13330 \text{ lbf}$$

The proof factor of safety, from Eq. (8-28), is used to find F_i :

$$F_{ia} = \frac{S_p A_t}{n_p} - CP = \frac{120 \times 10^3 \text{ psi} (0.1419 \text{ in}^2)}{1} - 0.1111(13330 \text{ lbf}) = 15546.8889 \text{ lbf}$$

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11848.88**  lbf.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.030.b - Static bolt factor of safety related to preload

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11848.8889 ± 2%** lbf.

Explanation:

The equation for the separation factor of safety is determined by Eq. (8-30):

$$F_{ib} = n_O P(1 - C) = 1(13330 \text{ lbf})(1 - 0.1111) = 11848.8889 \text{ lbf}$$

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf-in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is **1369.788**  lbf-in.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030.c - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf-in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is **1369.7889 ± 2%** lbf-in.

Explanation:

The torque can be determined by Eq. 8-27:

$$T = KF_id$$

$$T = KF_id = 0.2 \frac{F_{ia} + F_{ib}}{2} (0.5)$$

$$T = 0.2 \frac{15546.8889 \text{ lbf} + 11848.8889 \text{ lbf}}{2} (0.5 \text{ in}) = 1369.7889 \text{ lbf-in}$$

Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 0.9 \text{ MN/mm}$ and the stiffness of the members is $k_m = 3 \text{ MN/mm}$ per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use $M6 \times 1$ class 5.8 bolts with rolled threads.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is **66144**  N.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031.a - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is **66144 ± 2%** N.

Explanation:

From, Table 8–9, $S_p = 380$ MPa.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{0.9 \text{ MN/mm}}{(0.9 + 3) \text{ MN/mm}} = 0.2308$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75A_t S_p = 0.75(20.1 \text{ mm}^2)(380 \text{ MPa}) = 5730 \text{ N}$$

The proof factor of safety equation is given by Eq. (8-28):

$$P_{\max} = \left(\frac{S_p A_t}{n_p} - F_i \right) \frac{N}{C} = \left(\frac{380 \times 10^6 \text{ Pa} (2.01 \times 10^{-5} \text{ m}^2)}{1} - 5730 \text{ N} \right) \frac{8}{0.2308} = 66144 \text{ N}$$

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is **59571.03**  N.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031.b - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is **59592 ± 2%** N.

Explanation:

The equation for the separation factor of safety is given Eq. (8-30):

$$P_{\max} = \frac{F_t}{n_o} \frac{1}{(1-C)} N = \frac{5730 \text{ N}}{1} \frac{1}{(1-0.2308)} (8) = 59592 \text{ N}$$

Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS

For a bolted assembly, the stiffness of each bolt is $k_b = 4$ Mlbf/in and the stiffness of the members is $k_m = 14$ Mlbf/in per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. A fluctuating external load is applied to the entire joint with $P_{\max} = 80$ kips and $P_{\min} = 20$ kips. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS	Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is **5** .

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is **5 ± 2%**.

Explanation:

From Table 8–9,

$$S_p = 1.2(10^5) \text{ psi}$$

The stiffness constant is calculated as

$$C = \frac{k_b}{k_b + k_m}$$

$$C = \frac{4 \text{ Mlb/in}}{(4 \text{ Mlb/in} + 14 \text{ Mlb/in})}$$

$$C = 0.2222$$

For a non-permanent connection, using Eqs. (8–31) and (8–32),

$$F_i = 0.75A_tS_p = 0.75(0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) = 12770 \text{ lbf}$$

The proof factor of safety is calculated as

$$N = \frac{CP_{\max}}{(S_pA_t - F_i)}$$

$$N = \frac{0.2222(80 \times 10^3 \text{ lbf})}{((0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) - 12770 \text{ lbf})}$$

$$N = 5$$

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is **5** .

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is .

Explanation:

The separation factor of safety is calculated as

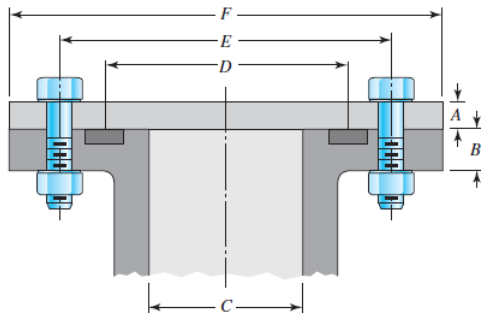
$$N = \frac{P_{\max}(1-C)}{F_i}$$

$$N = \frac{(80 \times 10^3 \text{ lbf})(1-0.2222)}{12770 \text{ lbf}}$$

$$N = 5$$

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	6 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is **55** mm.

The yielding factor of safety is **1.29** .

The load factor is **11.1** .

The joint separation factor is **11.8** .

References

Numeric Response

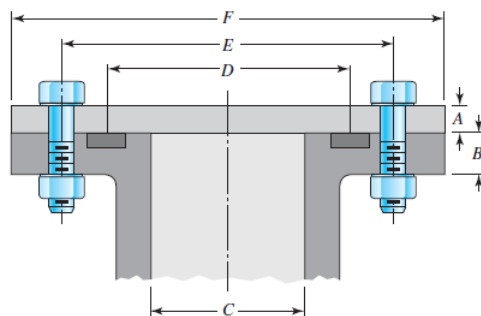
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.033 - Selection of a suitable bolt length

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	6 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is **55 $\pm$ 2%** mm.

The yielding factor of safety is $\boxed{1.2972 \pm 2\%}$.
 The load factor is $\boxed{11.1308 \pm 2\%}$.
 The joint separation factor is $\boxed{11.834 \pm 2\%}$.

Explanation:

$$l = 20 \text{ mm} + 20 \text{ mm} = 40 \text{ mm}$$

From Table A-31, the nut height is $H = 10.8 \text{ mm}$.

$$L \geq l + H = 40 \text{ mm} + 10.8 \text{ mm} = 50.8 \text{ mm} \approx 55 \text{ mm}$$

From Eq. (8-14), we have

$$L_T = 2d + 6 \text{ mm} = 2(12 \text{ mm}) + 6 \text{ mm} = 30 \text{ mm}$$

From Table 8-7,

$$l_d = L - L_T = 55 \text{ mm} - 30 \text{ mm} = 25 \text{ mm}$$

$$l_t = l - l_d = 40 \text{ mm} - 25 \text{ mm} = 15 \text{ mm}$$

$$A_d = \frac{\pi (12 \text{ mm})^2}{4} = 113.1 \text{ mm}^2$$

From Table 8-1, we have $A_t = 84.3 \text{ mm}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(113.1 \text{ mm}^2) (84.3 \text{ mm}^2) 207 \times 10^3 \text{ N/mm}^2}{(113.1 \text{ mm}^2) (15 \text{ mm}) + (84.3 \text{ mm}^2) (25 \text{ mm})} = 518.8 \times 10^3 \text{ N/mm} = 518.8 \text{ MN/m}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

$$k_1 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_1 + D - d)(D + d))}{(1.155(t_1 + D + d)(D - d))}\right)} = \frac{0.5774\pi(207 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 4470 \times 10^3 \text{ N/mm} = 4470 \text{ MN/m}$$

For cast iron, from Table 8-8, we have $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2 + D_2 - d)(D_2 + d))}{(1.155(t_2 + D_2 + d)(D_2 - d))}\right)} = \frac{0.5774\pi(100 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 2159 \times 10^3 \text{ N/mm} = 2159 \text{ MN/m}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2}\right)^{-1} = \left(\frac{1}{4470 \text{ MN/m}} + \frac{1}{2159 \text{ MN/m}}\right)^{-1} = 1456 \text{ MN/m}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{518.8 \text{ MN/m}}{518.8 \text{ MN/m} + 1456 \text{ MN/m}} = 0.263$$

From Table 8-11, we have $S_p = 650 \text{ MPa}$.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (84.3 \text{ mm}^2) (650 \text{ MPa}) \times 10^{-3} = 41.1 \text{ kN}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 100 mm. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(6 \text{ N/mm}^2) \pi (100 \text{ mm})^2}{4 \times 10} = 4712.3890 \text{ N/bolt} = 4.7124 \text{ kN/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{(650 \text{ N/mm}^2) (84.3 \text{ mm}^2)}{(0.263) (4712.3890 \text{ N}) + (41.10 \times 10^3 \text{ N})} = 1.2972$$

The overall factor of safety, from Eq. (8-29), is

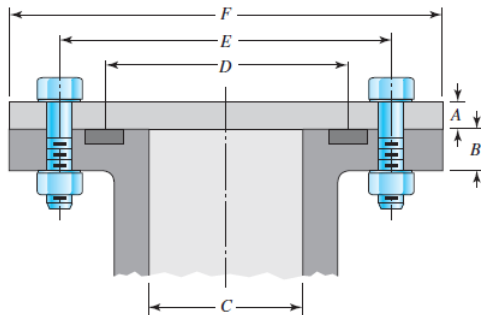
$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{(650 \text{ N/mm}^2) (84.3 \text{ mm}^2) - (41.10 \times 10^3 \text{ N})}{(0.263) (4712.3890 \text{ N})} = 11.1308$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_i}{P(1 - C)} = \frac{41.10 \text{ kN}}{4.7124 \text{ kN}(1 - 0.263)} = 11.8340$$

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1400 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is **1.75** in.
 The yielding factor of safety is **1.27**.
 The load factor is **7.477**.
 The joint separation factor is **9.5847**.

References

Numeric Response

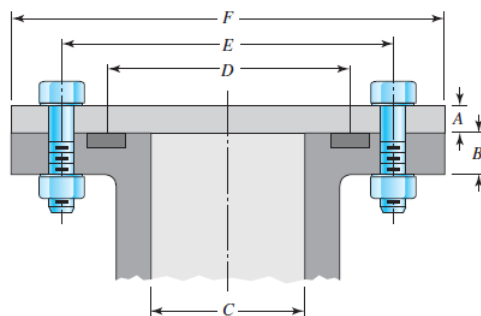
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.034 - Selection of a suitable bolt length

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1400 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is **1.75 ± 2%** in.

The yielding factor of safety is $\boxed{1.276 \pm 2\%}$.
 The load factor is $\boxed{7.4775 \pm 2\%}$.
 The joint separation factor is $\boxed{9.5847 \pm 2\%}$.

Explanation:

From Table A-31, the nut height is $H = 0.4375$ in.

$$l = 0.5 \text{ in} + 0.625 \text{ in} = 1.125 \text{ in}$$

$$L \geq l + H = 1.125 \text{ in} + 0.4375 \text{ in} = 1.563 \text{ in} \approx 1.75 \text{ in}$$

From Eq. (8-13), we have

$$L_T = 2d + 0.25 \text{ in} = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8-7,

$$l_d = L - L_T = 1.75 \text{ in} - 1.25 \text{ in} = 0.5 \text{ in}$$

$$l_t = l - l_d = 1.125 \text{ in} - 0.5 \text{ in} = 0.625 \text{ in}$$

$$A_d = \frac{\pi (0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8-2, we have $A_t = 0.1419 \text{ in}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(0.1963 \text{ in}^2) (0.1419 \text{ in}^2) (30 \times 10^6 \text{ lbf/in}^2)}{(0.1963 \text{ in}^2) (0.625 \text{ in}) + (0.1419 \text{ in}^2) (0.5 \text{ in})} = 4.316 \times 10^6 \text{ lbf/in} = 4.316 \text{ Mlbf/in}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 0.5$ in, $d = 0.5$ in, and $D = 0.75$ in.

$$k_1 = \frac{0.5774 \pi (E) d_1}{\ln \frac{[1.155 (t) + D_1 - d_1] (D_1 + d_1)}{[1.155 (t) + D_1 + d_1] (D_1 - d_1)}} = \frac{0.5774 \pi (30 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155 (0.5 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}) (0.75 \text{ in} + 0.5 \text{ in})}{(1.155 (0.5 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}) (0.75 \text{ in} - 0.5 \text{ in})}} = 33.30 \times 10^6 \text{ lbf/in} = 33.30 \text{ Mlbf/in}$$

Cast iron has two frusta. The midpoint of complete joint is at $\frac{1.125 \text{ in}}{2} = 0.5625$ in.

For upper frustum,

$$d = 0.5 \text{ in}$$

$$t = 0.5625 \text{ in} - 0.5 \text{ in} = 0.0625 \text{ in}$$

$$D = 0.75 \text{ in} + 2(0.5 \text{ in}) \tan 30^\circ = 1.327 \text{ in}$$

From Eq. (8-20),

$$k_2 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_2) + D_2 - d] (D_2 + d)}{[1.155 (t_2) + D_2 + d] (D_2 - d)}} = \frac{0.5774 \pi (14.5 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155 (0.0625 \text{ in}) + 1.327 \text{ in} - 0.5 \text{ in}) (1.327 \text{ in} + 0.5 \text{ in})}{(1.155 (0.0625 \text{ in}) + 1.327 \text{ in} + 0.5 \text{ in}) (1.327 \text{ in} - 0.5 \text{ in})}} = 292.7 \times 10^6 \text{ lbf/in} = 292.7 \text{ Mlbf/in}$$

For lower frustum, $t = 0.5625$ in, $d = 0.5$ in, and $D = 0.75$ in.

$$k_3 = \frac{0.5774 \pi E d}{\ln \frac{[1.155 (t_3) + D_3 - d] (D_3 + d)}{[1.155 (t_3) + D_3 + d] (D_3 - d)}} = \frac{0.5774 \pi (14.5 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155 (0.5625 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}) (0.75 \text{ in} + 0.5 \text{ in})}{(1.155 (0.5625 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}) (0.75 \text{ in} - 0.5 \text{ in})}} = 15.26 \times 10^6 \text{ lbf/in} = 15.26 \text{ Mlbf/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{33.30 \text{ Mlbf/in}} + \frac{1}{292.7 \text{ Mlbf/in}} + \frac{1}{15.26 \text{ Mlbf/in}} \right)^{-1} = 10.10 \text{ Mlbf/in}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{4.316 \text{ Mlbf/in}}{4.316 \text{ Mlbf/in} + 10.10 \text{ Mlbf/in}} = 0.299$$

From Table 8-9, we have $S_p = 85$ kpsi.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (0.1419 \text{ in}^2) (85 \text{ kpsi}) = 9.05 \text{ kips}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 3.5 in. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(1400 \text{ lbf/in}^2) \pi (3.5 \text{ in})^2}{4 \times 10} = 1346.9579 \text{ lbf/bolt} = 1.3470 \text{ kips/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{C P + F_i} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2)}{(0.299) (1346.9579 \text{ lbf}) + (9.05 \times 10^3 \text{ lbf})} = 1.2760$$

The overload factor of safety, from Eq. (8-29), is

$$n_L = \frac{S_y A_t - F_t}{CP} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2) - (9.05 \times 10^3 \text{ lbf})}{(0.299) (1346.9579 \text{ lbf})} = 7.4775$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_t}{P(1 - C)} = \frac{9.05 \text{ kips}}{1.3470 \text{ kips}(1 - 0.299)} = 9.5847$$

AVERAGE

Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.4$ Mlbf/in and the stiffness of the members is $k_m = 17$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. Assume the bolts are preloaded to 75 percent of the proof load.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

1.

Award: 5 out of 5.00 points

 You received credit for this question in a previous attempt

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is 1.2 .

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.029.a - Static bolt factor of safety	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is 1.1809 ± 2%.

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2.4 \text{ Mlbf/in}}{(2.4 + 17) \text{ Mlbf/in}} = 0.1237$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32), we get

$$F_i = 0.75 A_t S_p = 0.75 (0.1419 \text{ in}^2) (120 \text{ ksi}) = 12.77 \text{ kips}$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80 \text{ kip}}{6} = 13.33 \text{ kip}$$

Therefore, from Eq. (8-28), the yielding factor of safety is

$$n_p = \frac{S_p A_t}{C P + F_i} = \frac{(120 \times 10^3 \text{ psi}) (0.1419 \text{ in}^2)}{0.1237 (13330 \text{ lbf}) + 12770 \text{ lbf}} = 1.1809$$

2.

Award: 5 out of 5.00 points

You did not receive full credit for this question in a previous attempt

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is **2.58** ✓

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.029.b - Static bolt factor of safety Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is **2.5821 ± 2%**.

Explanation:

Therefore, from Eq. (8-29), the load factor of safety is

$$n_L = \frac{S_P A_t - F_i}{CP} = \frac{(120 \times 10^3 \text{ psi})(0.1419 \text{ in}^2) - 12770 \text{ lbf}}{0.1237(13330 \text{ lbf})} = 2.5821$$

3.

Award: 5 out of 5.00 points

You did not receive full credit for this question in a previous attempt

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is **1.09** ✓

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.029.c - Static bolt factor of safety Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is **1.0932 ± 2%**.

Explanation:

From Eq. (8-30), the joint separation factor of safety is

$$n_O = \frac{F_i}{P(1-C)} = \frac{12770 \text{ lbf}}{13330 \text{ lbf}(1-0.1237)} = 1.0932$$

Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.6$ Mlbf/in and the stiffness of the members is $k_m = 16$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. It is desired to find the range of torque that a mechanic could apply to initially preload the bolts without expecting failure once the joint is loaded. Assume a torque coefficient of $K = 0.2$.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

4. Award: 5 out of 5.00 points

 You did not receive full credit for this question in a previous attempt

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **15164.66**  lbf.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030.a - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **15164.6667 ± 2%** lbf.

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{k_b + k_m} = \frac{2.6 \text{ Mlbf/in}}{(2.6 + 16) \text{ Mlbf/in}} = 0.1398$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80000 \text{ lbf}}{6} = 13330 \text{ lbf}$$

The proof factor of safety, from Eq. (8-28), is used to find F_i :

$$F_{ia} = \frac{S_p A_t}{n_p} - CP = \frac{120 \times 10^3 \text{ psi} (0.1419 \text{ in}^2)}{1} - 0.1398(13330 \text{ lbf}) = 15164.6667 \text{ lbf}$$

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11466.66**  lbf.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.030.b - Static bolt factor of safety related to preload

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11466.6667 ± 2%** lbf.

Explanation:

The equation for the separation factor of safety is determined by Eq. (8-30):

$$F_{ib} = n_O P(1 - C) = 1(13330 \text{ lbf})(1 - 0.1398) = 11466.6667 \text{ lbf}$$

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf-in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is **1331.56** lbf-in.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030.c - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf-in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is **1331.5667 ± 2%** lbf-in.

Explanation:

The torque can be determined by Eq. 8-27:

$$T = KF_id$$

$$T = KF_id = 0.2 \frac{F_{ia} + F_{ib}}{2} (0.5)$$

$$T = 0.2 \frac{15164.6667 \text{ lbf} + 11466.6667 \text{ lbf}}{2} (0.5 \text{ in}) = 1331.5667 \text{ lbf-in}$$

Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 0.8 \text{ MN/mm}$ and the stiffness of the members is $k_m = 3.6 \text{ MN/mm}$ per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use $M6 \times 1$ class 5.8 bolts with rolled threads. NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is **83952**  N.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031.a - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is **83952 ± 2%** N.

Explanation:

From, Table 8–9, $S_p = 380$ MPa.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{0.8 \text{ MN/mm}}{(0.8 + 3.6) \text{ MN/mm}} = 0.1818$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75A_tS_p = 0.75(20.1 \text{ mm}^2)(380 \text{ MPa}) = 5730 \text{ N}$$

The proof factor of safety equation is given by Eq. (8-28):

$$P_{\max} = \left(\frac{S_p A_t}{n_p} - F_i \right) \frac{N}{C} = \left(\frac{380 \times 10^6 \text{ Pa} \left(2.01 \times 10^{-5} \text{ m}^2 \right)}{1} - 5730 \text{ N} \right) \frac{8}{0.1818} = 83952 \text{ N}$$

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is **54952**  N.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031.b - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is **56026.6667 ± 2%** N.

Explanation:

The equation for the separation factor of safety is given Eq. (8-30):

$$P_{\max} = \frac{F_t}{n_o} \frac{1}{(1-C)} N = \frac{5730 \text{ N}}{1} \frac{1}{(1-0.1818)} (8) = 56026.6667 \text{ N}$$

Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS

For a bolted assembly, the stiffness of each bolt is $k_b = 4 \text{ Mlbf/in}$ and the stiffness of the members is $k_m = 14 \text{ Mlbf/in}$ per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. A fluctuating external load is applied to the entire joint with $P_{\max} = 80 \text{ kips}$ and $P_{\min} = 20 \text{ kips}$. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads.
NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS	Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is **5** .

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is **5 ± 2%**.

Explanation:

From Table 8–9,

$$S_p = 1.2(10^5) \text{ psi}$$

The stiffness constant is calculated as

$$C = \frac{k_b}{k_b + k_m}$$

$$C = \frac{4 \text{ Mlb/in}}{(4 \text{ Mlb/in} + 14 \text{ Mlb/in})}$$

$$C = 0.2222$$

For a non-permanent connection, using Eqs. (8–31) and (8–32),

$$F_i = 0.75A_t S_p = 0.75(0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) = 12770 \text{ lbf}$$

The proof factor of safety is calculated as

$$N = \frac{CP_{\max}}{(S_p A_t - F_i)}$$

$$N = \frac{0.2222(80 \times 10^3 \text{ lbf})}{((0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) - 12770 \text{ lbf})}$$

$$N = 5$$

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is **5** .

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is .

Explanation:

The separation factor of safety is calculated as

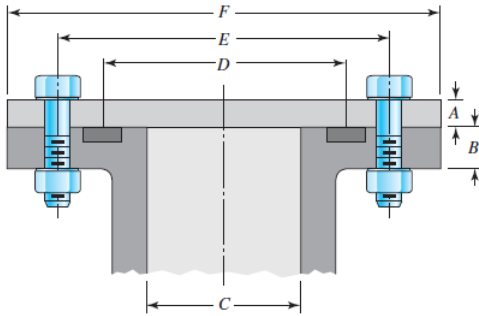
$$N = \frac{P_{\max}(1-C)}{F_i}$$

$$N = \frac{(80 \times 10^3 \text{ lbf})(1-0.2222)}{12770 \text{ lbf}}$$

$$N = 5$$

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	6.75 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is **55** mm.

The yielding factor of safety is **1.29**.

The load factor is **9.8**.

The joint separation factor is **10.5**.

References

Numeric Response

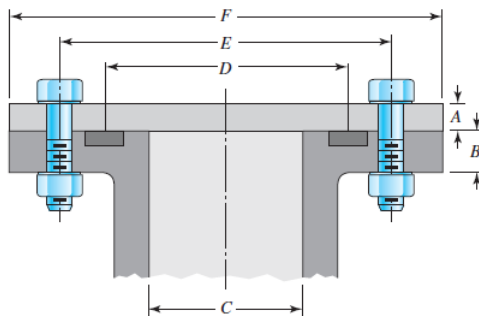
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.033 - Selection of a suitable bolt length

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	6.75 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is **55 $\pm$ 2%** mm.

The yielding factor of safety is $1.2925 \pm 2\%$.
 The load factor is $9.894 \pm 2\%$.
 The joint separation factor is $10.5192 \pm 2\%$.

Explanation:

$$l = 20 \text{ mm} + 20 \text{ mm} = 40 \text{ mm}$$

From Table A-31, the nut height is $H = 10.8 \text{ mm}$.

$$L \geq l + H = 40 \text{ mm} + 10.8 \text{ mm} = 50.8 \text{ mm} \approx 55 \text{ mm}$$

From Eq. (8-14), we have

$$L_T = 2d + 6 \text{ mm} = 2(12 \text{ mm}) + 6 \text{ mm} = 30 \text{ mm}$$

From Table 8-7,

$$l_d = L - L_T = 55 \text{ mm} - 30 \text{ mm} = 25 \text{ mm}$$

$$l_t = l - l_d = 40 \text{ mm} - 25 \text{ mm} = 15 \text{ mm}$$

$$A_d = \frac{\pi (12 \text{ mm})^2}{4} = 113.1 \text{ mm}^2$$

From Table 8-1, we have $A_t = 84.3 \text{ mm}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(113.1 \text{ mm}^2) (84.3 \text{ mm}^2) 207 \times 10^3 \text{ N/mm}^2}{(113.1 \text{ mm}^2) (15 \text{ mm}) + (84.3 \text{ mm}^2) (25 \text{ mm})} = 518.8 \times 10^3 \text{ N/mm} = 518.8 \text{ MN/m}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

$$k_1 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_1 + D - d)(D + d))}{(1.155(t_1 + D + d)(D - d))}\right)} = \frac{0.5774\pi(207 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 4470 \times 10^3 \text{ N/mm} = 4470 \text{ MN/m}$$

For cast iron, from Table 8-8, we have $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2 + D_2 - d)(D_2 + d))}{(1.155(t_2 + D_2 + d)(D_2 - d))}\right)} = \frac{0.5774\pi(100 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 2159 \times 10^3 \text{ N/mm} = 2159 \text{ MN/m}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2}\right)^{-1} = \left(\frac{1}{4470 \text{ MN/m}} + \frac{1}{2159 \text{ MN/m}}\right)^{-1} = 1456 \text{ MN/m}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{518.8 \text{ MN/m}}{518.8 \text{ MN/m} + 1456 \text{ MN/m}} = 0.263$$

From Table 8-11, we have $S_p = 650 \text{ MPa}$.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75A_t S_p = 0.75(84.3 \text{ mm}^2)(650 \text{ MPa}) \times 10^{-3} = 41.1 \text{ kN}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 100 mm. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(6.75 \text{ N/mm}^2) \pi (100 \text{ mm})^2}{4 \times 10} = 5301.4376 \text{ N/bolt} = 5.3014 \text{ kN/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{(650 \text{ N/mm}^2) (84.3 \text{ mm}^2)}{(0.263) (5301.4376 \text{ N}) + (41.10 \times 10^3 \text{ N})} = 1.2925$$

The overall factor of safety, from Eq. (8-29), is

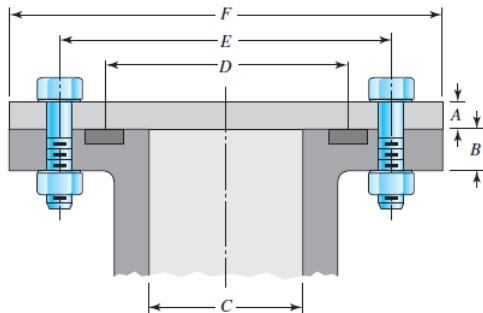
$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{(650 \text{ N/mm}^2) (84.3 \text{ mm}^2) - (41.10 \times 10^3 \text{ N})}{(0.263) (5301.4376 \text{ N})} = 9.8940$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_i}{P(1 - C)} = \frac{41.10 \text{ kN}}{5.3014 \text{ kN}(1 - 0.263)} = 10.5192$$

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1710 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is **1.75** ✓ in.

The yielding factor of safety is **1.3** ✗.

The load factor is **6.12** ✓.

The joint separation factor is **7.8** ✓.

References

Numeric Response

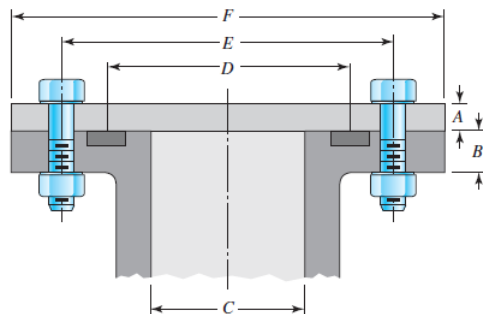
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.034 - Selection of a suitable bolt length

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1710 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is **1.75 ± 2%** in.

The yielding factor of safety is $\boxed{1.2641 \pm 2\%}$.
 The load factor is $\boxed{6.1219 \pm 2\%}$.
 The joint separation factor is $\boxed{7.8471 \pm 2\%}$.

Explanation:

From Table A-31, the nut height is $H = 0.4375$ in.

$$l = 0.5 \text{ in} + 0.625 \text{ in} = 1.125 \text{ in}$$

$$L \geq l + H = 1.125 \text{ in} + 0.4375 \text{ in} = 1.563 \text{ in} \approx 1.75 \text{ in}$$

From Eq. (8-13), we have

$$L_T = 2d + 0.25 \text{ in} = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8-7,

$$l_d = L - L_T = 1.75 \text{ in} - 1.25 \text{ in} = 0.5 \text{ in}$$

$$l_t = l - l_d = 1.125 \text{ in} - 0.5 \text{ in} = 0.625 \text{ in}$$

$$A_d = \frac{\pi (0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8-2, we have $A_t = 0.1419 \text{ in}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(0.1963 \text{ in}^2) (0.1419 \text{ in}^2) (30 \times 10^6 \text{ lbf/in}^2)}{(0.1963 \text{ in}^2) (0.625 \text{ in}) + (0.1419 \text{ in}^2) (0.5 \text{ in})} = 4.316 \times 10^6 \text{ lbf/in} = 4.316 \text{ Mlbf/in}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 0.5$ in, $d = 0.5$ in, and $D = 0.75$ in.

$$k_1 = \frac{0.5774 \pi (E) d_1}{\ln \frac{[1.155(t) + D_1 - d_1] (D_1 + d_1)}{[1.155(t) + D_1 + d_1] (D_1 - d_1)}} = \frac{0.5774 \pi (30 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155(0.5 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}) (0.75 \text{ in} + 0.5 \text{ in})}{(1.155(0.5 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}) (0.75 \text{ in} - 0.5 \text{ in})}} = 33.30 \times 10^6 \text{ lbf/in} = 33.30 \text{ Mlbf/in}$$

Cast iron has two frusta. The midpoint of complete joint is at $\frac{1.125 \text{ in}}{2} = 0.5625$ in.

For upper frustum,

$$d = 0.5 \text{ in}$$

$$t = 0.5625 \text{ in} - 0.5 \text{ in} = 0.0625 \text{ in}$$

$$D = 0.75 \text{ in} + 2(0.5 \text{ in}) \tan 30^\circ = 1.327 \text{ in}$$

From Eq. (8-20),

$$k_2 = \frac{0.5774 \pi E d}{\ln \frac{[1.155(t_2) + D_2 - d] (D_2 + d)}{[1.155(t_2) + D_2 + d] (D_2 - d)}} = \frac{0.5774 \pi (14.5 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155(0.0625 \text{ in}) + 1.327 \text{ in} - 0.5 \text{ in}) (1.327 \text{ in} + 0.5 \text{ in})}{(1.155(0.0625 \text{ in}) + 1.327 \text{ in} + 0.5 \text{ in}) (1.327 \text{ in} - 0.5 \text{ in})}} = 292.7 \times 10^6 \text{ lbf/in} = 292.7 \text{ Mlbf/in}$$

For lower frustum, $t = 0.5625$ in, $d = 0.5$ in, and $D = 0.75$ in.

$$k_3 = \frac{0.5774 \pi E d}{\ln \frac{[1.155(t_3) + D_3 - d] (D_3 + d)}{[1.155(t_3) + D_3 + d] (D_3 - d)}} = \frac{0.5774 \pi (14.5 \times 10^6 \text{ lbf/in}^2) (0.5 \text{ in})}{\ln \frac{(1.155(0.5625 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in}) (0.75 \text{ in} + 0.5 \text{ in})}{(1.155(0.5625 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in}) (0.75 \text{ in} - 0.5 \text{ in})}} = 15.26 \times 10^6 \text{ lbf/in} = 15.26 \text{ Mlbf/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{33.30 \text{ Mlbf/in}} + \frac{1}{292.7 \text{ Mlbf/in}} + \frac{1}{15.26 \text{ Mlbf/in}} \right)^{-1} = 10.10 \text{ Mlbf/in}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{4.316 \text{ Mlbf/in}}{4.316 \text{ Mlbf/in} + 10.10 \text{ Mlbf/in}} = 0.299$$

From Table 8-9, we have $S_p = 85$ kpsi.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (0.1419 \text{ in}^2) (85 \text{ kpsi}) = 9.05 \text{ kips}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 3.5 in. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(1710 \text{ lbf/in}^2) \pi (3.5 \text{ in})^2}{4 \times 10} = 1645.2128 \text{ lbf/bolt} = 1.6452 \text{ kips/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{C P + F_i} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2)}{(0.299) (1645.2128 \text{ lbf}) + (9.05 \times 10^3 \text{ lbf})} = 1.2641$$

The overload factor of safety, from Eq. (8-29), is

$$n_L = \frac{S_y A_t - F_t}{CP} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2) - (9.05 \times 10^3 \text{ lbf})}{(0.299) (1645.2128 \text{ lbf})} = 6.1219$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_t}{P(1 - C)} = \frac{9.05 \text{ kips}}{1.6452 \text{ kips}(1 - 0.299)} = 7.8471$$

BELOW AVERAGE

Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.6$ Mlbf/in and the stiffness of the members is $k_m = 16$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. Assume the bolts are preloaded to 75 percent of the proof load.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.029 - Static bolt factor of safety - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

1. Award: 0 out of 5.00 points

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is **1.2** ❌.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.029.a - Static bolt factor of safety	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.a - Static bolt factor of safety

Determine the yielding factor of safety.

The yielding factor of safety is .

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2.6 \text{ Mlbf/in}}{(2.6 + 16) \text{ Mlbf/in}} = 0.1398$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32), we get

$$F_i = 0.75 A_t S_p = 0.75 (0.1419 \text{ in}^2) (120 \text{ ksi}) = 12.77 \text{ kips}$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80 \text{ kip}}{6} = 13.33 \text{ kip}$$

Therefore, from Eq. (8-28), the yielding factor of safety is

$$n_p = \frac{S_p A_t}{C P + F_i} = \frac{(120 \times 10^3 \text{ psi}) (0.1419 \text{ in}^2)}{0.1398 (13330 \text{ lbf}) + 12770 \text{ lbf}} = 1.1636$$

2.

Award: 5 out of 5.00 points

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is **2.28** ✓.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.029.b - Static bolt factor of safety

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.b - Static bolt factor of safety

Determine the load factor of safety.

The load factor of safety is **2.2852 ± 2%**.

Explanation:

Therefore, from Eq. (8-29), the load factor of safety is

$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{(120 \times 10^3 \text{ psi})(0.1419 \text{ in}^2) - 12770 \text{ lbf}}{0.1398(13330 \text{ lbf})} = 2.2852$$

3.

Award: 5 out of 5.00 points

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is **1.11** ✓.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.029.c - Static bolt factor of safety

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.029.c - Static bolt factor of safety

Determine the factor of safety based on the joint separation.

The factor of safety based on the joint separation is **1.1137 ± 2%**.

Explanation:

From Eq. (8-30), the joint separation factor of safety is

$$n_O = \frac{F_i}{P(1-C)} = \frac{12770 \text{ lbf}}{13330 \text{ lbf}(1-0.1398)} = 1.1137$$

Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS

For a bolted assembly with six bolts, the stiffness of each bolt is $k_b = 2.8$ Mlbf/in and the stiffness of the members is $k_m = 14$ Mlbf/in per bolt. An external load of 80 kips is applied to the entire joint. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads. It is desired to find the range of torque that a mechanic could apply to initially preload the bolts without expecting failure once the joint is loaded. Assume a torque coefficient of $K = 0.2$.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030 - Static bolt factor of safety related to preload - DEPENDENT MULTI-PART QUESTION - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

4. Award: 0 out of 5.00 points

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **17.028**  lbf.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030.a - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.a - Static bolt factor of safety related to preload

Determine the maximum bolt preload that can be applied without exceeding the proof strength of the bolts once the load is applied.

The maximum bolt preload that can be applied is **14806.3333 ± 2%** lbf.

Explanation:

From Table 8–9, $S_p = 120$ ksi.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{2.8 \text{ Mlbf/in}}{(2.8 + 14) \text{ Mlbf/in}} = 0.1667$$

The external load per bolt is

$$P = \frac{P_{\text{total}}}{N} = \frac{80000 \text{ lbf}}{6} = 13330 \text{ lbf}$$

The proof factor of safety, from Eq. (8-28), is used to find F_i :

$$F_{ia} = \frac{S_p A_t}{n_p} - CP = \frac{120 \times 10^3 \text{ psi} (0.1419 \text{ in}^2)}{1} - 0.1667 (13330 \text{ lbf}) = 14806.3333 \text{ lbf}$$

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11,464** ✖ lbf.

[References](#)**Numeric Response**

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.030.b - Static bolt factor of safety related to preload

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.b - Static bolt factor of safety related to preload

Determine the minimum bolt preload that can be applied while avoiding joint separation.

The minimum bolt preload that can be applied is **11108.3333 ± 2%** lbf.

Explanation:

The equation for the separation factor of safety is determined by Eq. (8-30):

$$F_{ib} = n_O P (1 - C) = 1 (13330 \text{ lbf}) (1 - 0.1667) = 11108.3333 \text{ lbf}$$

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf-in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is **118.72**  lbf-in.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.030.c - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.030.c - Static bolt factor of safety related to preload

Determine the value of torque in units of lbf-in that should be specified for preloading the bolts if it is desired to preload to the midpoint of the values found in parts (a) and (b). Use the average value of the previously calculated values of F_i .

The torque that should be specified for preloading the bolt is **1295.7333 ± 2%** lbf-in.

Explanation:

The torque can be determined by Eq. 8-27:

$$T = KF_id$$

$$T = KF_id = 0.2 \frac{F_{ia} + F_{ib}}{2} (0.5)$$

$$T = 0.2 \frac{14806.3333 \text{ lbf} + 11108.3333 \text{ lbf}}{2} (0.5 \text{ in}) = 1295.7333 \text{ lbf-in}$$

Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS

For a bolted assembly with eight bolts, the stiffness of each bolt is $k_b = 0.8 \text{ MN/mm}$ and the stiffness of the members is $k_m = 3.6 \text{ MN/mm}$ per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. Assume the external load is equally distributed to all the bolts. It has been determined to use $M6 \times 1$ class 5.8 bolts with rolled threads. NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031 - Static bolt factor of safety related to preload - DEPENDENT MULTI PART PROBLEM - ASSIGN ALL PARTS	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is **83.93** ✖ N.

References

Numeric Response Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.031.a - Static bolt factor of safety related to preload Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.a - Static bolt factor of safety related to preload

Determine the maximum external load $P_{\max}$ that can be applied to the entire joint without exceeding the proof strength of the bolts.

The maximum external load for this case is **83952 ± 2%** N.

Explanation:

From, Table 8–9, $S_p = 380$ MPa.

The stiffness constant is

$$C = \frac{k_b}{(k_b + k_m)} = \frac{0.8 \text{ MN/mm}}{(0.8 + 3.6) \text{ MN/mm}} = 0.1818$$

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (20.1 \text{ mm}^2) (380 \text{ MPa}) = 5730 \text{ N}$$

The proof factor of safety equation is given by Eq. (8-28):

$$P_{\max} = \left(\frac{S_p A_t}{n_p} - F_i \right) \frac{N}{C} = \left(\frac{380 \times 10^6 \text{ Pa} (2.01 \times 10^{-5} \text{ m}^2)}{1} - 5730 \text{ N} \right) \frac{8}{0.1818} = 83952 \text{ N}$$

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is **56.02** ✖ N.

References

Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.031.b - Static bolt factor of safety related to preload	Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.031.b - Static bolt factor of safety related to preload

Determine the maximum external load P_{max} that can be applied to the entire joint without causing the members to come out of compression.

The maximum external load for this case is **56026.6667 ± 2%** N.

Explanation:

The equation for the separation factor of safety is given Eq. (8-30):

$$P_{\max} = \frac{F_i}{n_o} \frac{1}{(1-C)} N = \frac{5730 \text{ N}}{1} \frac{1}{(1-0.1818)} (8) = 56026.6667 \text{ N}$$

Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS

For a bolted assembly, the stiffness of each bolt is $k_b = 2$ Mlbf/in and the stiffness of the members is $k_m = 14$ Mlbf/in per bolt. The joint is subject to occasional disassembly for maintenance and should be preloaded accordingly. A fluctuating external load is applied to the entire joint with $P_{\max} = 80$ kips and $P_{\min} = 20$ kips. Assume the load is equally distributed to all the bolts. It has been determined to use 1/2 in-13 UNC grade 8 bolts with rolled threads.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.032 - Dynamic bolt factor of safety related to preload - DEPENDENT MULTI-PART PROBLEM ASSIGN ALL PARTS	Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is **5** .

References	
Numeric Response	Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints
Problem 08.032.a - Dynamic bolt factor of safety related to preload	Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.a - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid yielding of the bolts.

The minimum number of bolts necessary to avoid yielding of the bolts is **3 ± 2%**.

Explanation:

From Table 8–9,

$S_p = 1.2(10^5) \text{ psi}$

The stiffness constant is calculated as

$$C = \frac{k_b}{(k_b + k_m)}$$

$$C = \frac{2 \text{ Mlbf/in}}{(2 \text{ Mlbf/in} + 14 \text{ Mlbf/in})}$$

$C = 0.125$

For a non-permanent connection, using Eqs. (8–31) and (8–32),

$F_i = 0.75A_tS_p = 0.75(0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) = 12770 \text{ lbf}$

The proof factor of safety is calculated as

$$N = \frac{CP_{\max}}{(S_pA_t - F_i)}$$

$$N = \frac{0.125(80 \times 10^3 \text{ lbf})}{((0.1419 \text{ in}^2)(120 \times 10^3 \text{ psi}) - 12770 \text{ lbf})}$$

$N = 3$

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is **5** ✖.

References

Numeric Response

Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Section: 08.11 Fatigue Loading of Tension Joints

Problem 08.032.b - Dynamic bolt factor of safety related to preload

Determine the minimum number of bolts necessary to avoid joint separation.

The minimum number of bolts necessary to avoid joint separation is .

Explanation:

The separation factor of safety is calculated as

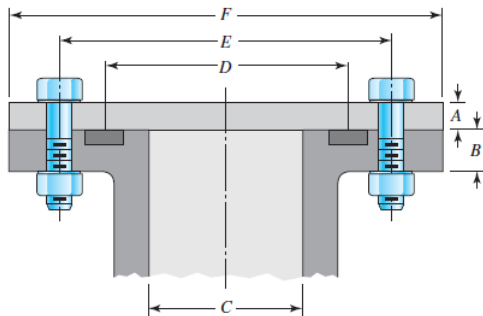
$$N = \frac{P_{\max}(1-C)}{F_i}$$

$$N = \frac{(80 \times 10^3 \text{ lbf})(1-0.125)}{12770 \text{ lbf}}$$

$$N = 6$$

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	6.15 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is **55** ✓ mm.
 The yielding factor of safety is **1.29** ✓.
 The load factor is **11.1** ✗.
 The joint separation factor is **11.8** ✗.

References

Numeric Response

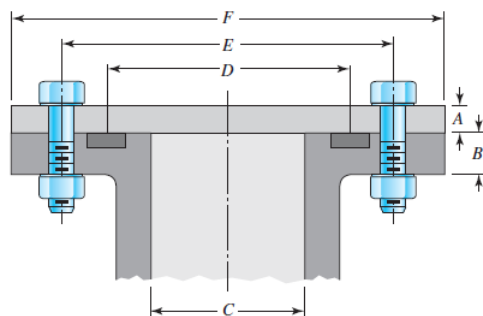
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.033 - Selection of a suitable bolt length

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.033 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 5 mm. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 207 GPa and for cast iron as 100 GPa.



Head thickness (A)	20 mm
Cylinder thickness (B)	20 mm
Cylinder ID (C)	60 mm
Gasket sealing diameter (D)	100 mm
Bolt circle diameter (E)	200 mm
Head outer diameter (F)	300 mm
Number of bolts (N)	10
Cylinder pressure (p_g)	6.15 MPa
Bolt grade (G)	ISO 9.8
Bolt spec.	M12 $\times$ 1.75

The suitable bolt length is **55 $\pm$ 2%** mm.

The yielding factor of safety is $\boxed{1.2963 \pm 2\%}$.

The load factor is $\boxed{10.8593 \pm 2\%}$.

The joint separation factor is $\boxed{11.5454 \pm 2\%}$.

Explanation:

$$l = 20 \text{ mm} + 20 \text{ mm} = 40 \text{ mm}$$

From Table A-31, the nut height is $H = 10.8 \text{ mm}$.

$$L \geq l + H = 40 \text{ mm} + 10.8 \text{ mm} = 50.8 \text{ mm} \cong 55 \text{ mm}$$

From Eq. (8-14), we have

$$L_T = 2d + 6 \text{ mm} = 2(12 \text{ mm}) + 6 \text{ mm} = 30 \text{ mm}$$

From Table 8-7,

$$l_d = L - L_T = 55 \text{ mm} - 30 \text{ mm} = 25 \text{ mm}$$

$$l_t = l - l_d = 40 \text{ mm} - 25 \text{ mm} = 15 \text{ mm}$$

$$A_d = \frac{\pi(12 \text{ mm})^2}{4} = 113.1 \text{ mm}^2$$

From Table 8-1, we have $A_t = 84.3 \text{ mm}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(113.1 \text{ mm}^2)(84.3 \text{ mm}^2)207 \times 10^3 \text{ N/mm}^2}{(113.1 \text{ mm}^2)(15 \text{ mm}) + (84.3 \text{ mm}^2)(25 \text{ mm})} = 518.8 \times 10^3 \text{ N/mm} = 518.8 \text{ MN/m}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

$$k_1 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_1) + D - d)(D + d)}{(1.155(t_1) + D + d)(D - d)}\right)} = \frac{0.5774\pi(207 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 4470 \times 10^3 \text{ N/mm} = 4470 \text{ MN/m}$$

For cast iron, from Table 8-8, we have $t = 20 \text{ mm}$, $d = 12 \text{ mm}$, and $D = 18 \text{ mm}$.

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi(E)d}{\ln\left(\frac{(1.155(t_2) + D_2 - d)(D_2 + d)}{(1.155(t_2) + D_2 + d)(D_2 - d)}\right)} = \frac{0.5774\pi(100 \times 10^3 \text{ N/mm}^2)12 \text{ mm}}{\ln\left(\frac{(1.155(20 \text{ mm}) + 18 \text{ mm} - 12 \text{ mm})(18 \text{ mm} + 12 \text{ mm})}{(1.155(20 \text{ mm}) + 18 \text{ mm} + 12 \text{ mm})(18 \text{ mm} - 12 \text{ mm})}\right)} = 2159 \times 10^3 \text{ N/mm} = 2159 \text{ MN/m}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2}\right)^{-1} = \left(\frac{1}{4470 \text{ MN/m}} + \frac{1}{2159 \text{ MN/m}}\right)^{-1} = 1456 \text{ MN/m}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{518.8 \text{ MN/m}}{518.8 \text{ MN/m} + 1456 \text{ MN/m}} = 0.263$$

From Table 8-11, we have $S_p = 650 \text{ MPa}$.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (84.3 \text{ mm}^2) (650 \text{ MPa}) \times 10^{-3} = 41.1 \text{ kN}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 100 mm. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(6.15 \text{ N/mm}^2) \pi (100 \text{ mm})^2}{4 \times 10} = 4830.1987 \text{ N/bolt} = 4.8302 \text{ kN/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{C P + F_i} = \frac{(650 \text{ N/mm}^2) (84.3 \text{ mm}^2)}{(0.263)(4830.1987 \text{ N}) + (41.10 \times 10^3 \text{ N})} = 1.2963$$

The overall factor of safety, from Eq. (8-29), is

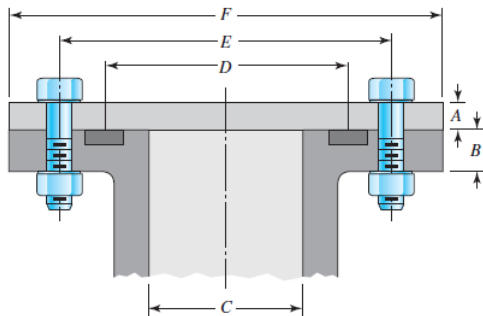
$$n_L = \frac{S_p A_t - F_i}{C P} = \frac{(650 \text{ N/mm}^2) (84.3 \text{ mm}^2) - (41.10 \times 10^3 \text{ N})}{(0.263)(4830.1987 \text{ N})} = 10.8593$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_i}{P(1-C)} = \frac{41.10 \text{ kN}}{4.8302 \text{ kN}(1-0.263)} = 11.5454$$

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1640 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is **1.75** ✓ in.
 The yielding factor of safety is **1.28** ✓.
 The load factor is **8.39** ✗.
 The joint separation factor is **15.78** ✗.

References

Numeric Response

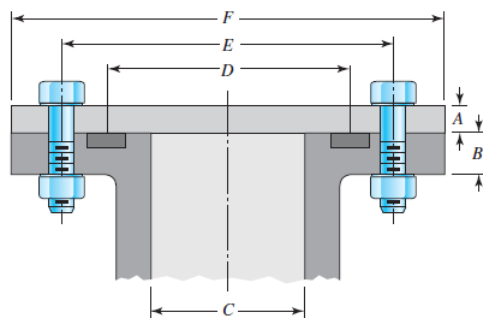
Chapter: 08 Screws, Fasteners, and the Design of Nonpermanent Joints

Problem 08.034 - Selection of a suitable bolt length

Section: 08.09 Statically Loaded Tension Joint with Preload

Problem 08.034 - Selection of a suitable bolt length

The figure illustrates the nonpermanent connection of a steel cylinder head to a grade 30 cast-iron pressure vessel using N bolts. A confined gasket seal has an effective sealing diameter D . The cylinder stores gas at a maximum pressure p_g . For the specifications given in the table for the specific problem assigned, select a suitable bolt length, assuming bolts are available in increments of 0.25 in. Then determine the yielding factor of safety n_p , the load factor n_L , and the joint separation factor n_0 . Take the value of E for steel as 30 Mpsi and for cast iron as 14.5 Mpsi.



Head thickness (A)	0.5 in
Cylinder thickness (B)	0.625 in
Cylinder ID (C)	2.75 in
Gasket sealing diameter (D)	3.5 in
Bolt circle diameter (E)	6 in
Head outer diameter (F)	8 in
Number of bolts (N)	10
Cylinder pressure (p_g)	1640 psi
Bolt grade (G)	SAE 5
Bolt spec.	0.5 in-13

The suitable bolt length is **1.75 ± 2%** in.

The yielding factor of safety is $\boxed{1.2667 \pm 2\%}$.
 The load factor is $\boxed{6.3833 \pm 2\%}$.
 The joint separation factor is $\boxed{8.182 \pm 2\%}$.

Explanation:

From Table A-31, the nut height is $H = 0.4375$ in.

$$l = 0.5 \text{ in} + 0.625 \text{ in} = 1.125 \text{ in}$$

$$L \geq l + H = 1.125 \text{ in} + 0.4375 \text{ in} = 1.563 \text{ in} \cong 1.75 \text{ in}$$

From Eq. (8-13), we have

$$L_T = 2d + 0.25 \text{ in} = 2(0.5 \text{ in}) + 0.25 \text{ in} = 1.25 \text{ in}$$

From Table 8-7,

$$l_d = L - L_T = 1.75 \text{ in} - 1.25 \text{ in} = 0.5 \text{ in}$$

$$l_t = l - l_d = 1.125 \text{ in} - 0.5 \text{ in} = 0.625 \text{ in}$$

$$A_d = \frac{\pi(0.5 \text{ in})^2}{4} = 0.1963 \text{ in}^2$$

From Table 8-2, we have $A_t = 0.1419 \text{ in}^2$.

From Eq. (8-17), we have

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{(0.1963 \text{ in}^2)(0.1419 \text{ in}^2)(30 \times 10^6 \text{ lbf/in}^2)}{(0.1963 \text{ in}^2)(0.625 \text{ in}) + (0.1419 \text{ in}^2)(0.5 \text{ in})} = 4.316 \times 10^6 \text{ lbf/in} = 4.316 \text{ Mlbf/in}$$

Now calculate the member stiffness.

For steel cylinder head, $t = 0.5$ in, $d = 0.5$ in, and $D = 0.75$ in.

$$k_1 = \frac{0.5774\pi(E)d_1}{\ln \frac{[1.155(t) + D_1 - d_1](D_1 + d_1)}{[1.155(t) + D_1 + d_1](D_1 - d_1)}} = \frac{0.5774\pi(30 \times 10^6 \text{ lbf/in}^2)(0.5 \text{ in})}{\ln \frac{(1.155(0.5 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{(1.155(0.5 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}} = 33.30 \times 10^6 \text{ lbf/in} = 33.30 \text{ Mlbf/in}$$

Cast iron has two frusta. The midpoint of complete joint is at $\frac{1.125 \text{ in}}{2} = 0.5625$ in.

For upper frustum,

$$d = 0.5 \text{ in}$$

$$t = 0.5625 \text{ in} - 0.5 \text{ in} = 0.0625 \text{ in}$$

$$D = 0.75 \text{ in} + 2(0.5 \text{ in})\tan 30^\circ = 1.327 \text{ in}$$

From Eq. (8-20),

$$k_2 = \frac{0.5774\pi E d}{\ln \frac{[1.155(t_2) + D_2 - d](D_2 + d)}{[1.155(t_2) + D_2 + d](D_2 - d)}} = \frac{0.5774\pi(14.5 \times 10^6 \text{ lbf/in}^2)(0.5 \text{ in})}{\ln \frac{(1.155(0.0625 \text{ in}) + 1.327 \text{ in} - 0.5 \text{ in})(1.327 \text{ in} + 0.5 \text{ in})}{(1.155(0.0625 \text{ in}) + 1.327 \text{ in} + 0.5 \text{ in})(1.327 \text{ in} - 0.5 \text{ in})}} = 292.7 \times 10^6 \text{ lbf/in} = 292.7 \text{ Mlbf/in}$$

For lower frustum, $t = 0.5625$ in, $d = 0.5$ in, and $D = 0.75$ in.

$$k_3 = \frac{0.5774\pi E d}{\ln \frac{[1.155(t_3) + D_3 - d](D_3 + d)}{[1.155(t_3) + D_3 + d](D_3 - d)}} = \frac{0.5774\pi(14.5 \times 10^6 \text{ lbf/in}^2)(0.5 \text{ in})}{\ln \frac{(1.155(0.5625 \text{ in}) + 0.75 \text{ in} - 0.5 \text{ in})(0.75 \text{ in} + 0.5 \text{ in})}{(1.155(0.5625 \text{ in}) + 0.75 \text{ in} + 0.5 \text{ in})(0.75 \text{ in} - 0.5 \text{ in})}} = 15.26 \times 10^6 \text{ lbf/in} = 15.26 \text{ Mlbf/in}$$

From Eq. (8-18),

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3} \right)^{-1} = \left(\frac{1}{33.30 \text{ Mlbf/in}} + \frac{1}{292.7 \text{ Mlbf/in}} + \frac{1}{15.26 \text{ Mlbf/in}} \right)^{-1} = 10.10 \text{ Mlbf/in}$$

$$C = \frac{k_b}{(k_b + k_m)} = \frac{4.316 \text{ Mlbf/in}}{4.316 \text{ Mlbf/in} + 10.10 \text{ Mlbf/in}} = 0.299$$

From Table 8-9, we have $S_p = 85$ kpsi.

For a nonpermanent connection, using Eqs. (8-31) and (8-32),

$$F_i = 0.75 A_t S_p = 0.75 (0.1419 \text{ in}^2) (85 \text{ kpsi}) = 9.05 \text{ kips}$$

The total external load is $P_{\text{total}} = p_g A_g$, where A_g is the effective area of the cylinder, based on the effective sealing diameter of 3.5 in. The external load per bolt is $P = P_{\text{total}}/N$. Thus,

$$P = \frac{p_g A_g}{N} = \frac{(1640 \text{ lbf/in}^2) \pi (3.5 \text{ in})^2}{4 \times 10} = 1577.8649 \text{ lbf/bolt} = 1.5779 \text{ kips/bolt}$$

The yielding factor of safety, from Eq. (8-28), is

$$n_p = \frac{S_p A_t}{C P + F_i} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2)}{(0.299)(1577.8649 \text{ lbf}) + (9.05 \times 10^3 \text{ lbf})} = 1.2667$$

The overload factor of safety, from Eq. (8-29), is

$$n_L = \frac{S_p A_t - F_i}{C P} = \frac{(85 \times 10^3 \text{ lbf/in}^2) (0.1419 \text{ in}^2) - (9.05 \times 10^3 \text{ lbf})}{(0.299)(1577.8649 \text{ lbf})} = 6.3833$$

The separation factor of safety, from Eq. (8-30), is

$$n_0 = \frac{F_i}{P(1-C)} = \frac{9.05 \text{ kips}}{1.5779 \text{ kips}(1-0.299)} = 8.1820$$

**MACHINE DESIGN I
FALL 2021**

HOMEWORK 13

TOPICS COVERED IN HOMEWORK 13

Chapter 11 Bearing: 11-9, 11-10, 11-11, 11-12, 11-22, 11-23

**MACHINE DESIGN I
FALL 2021**

**SAMPLE HOMEWORK QUESTIONS,
SOLUTION, AND GRADING SCHEME**

1. Award: 15.00 points

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.18 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 6.6 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing **A** can carry the larger load.

Explanation:

Both bearings need to be rated in terms of the same catalog rating system to compare them. Using a rating life of one million revolutions, both bearings can be rated in terms of a Basic Load Rating.

From equation 11-3, the catalog load rating as a function of the desired load, desired life, and catalog rating life is given as follows:

$$C_A = F_A \left(\frac{L_A}{L_R} \right)^{1/a} = F_A \left(\frac{L_A n_A 60}{L_R} \right)^{1/a}$$

(a = 3 for ball bearings)

$$C_A = 2.18 \left(\frac{(3000 \text{ h})(500 \text{ rev/min})(60)}{10^6 \text{ cycles}} \right)^{1/3}$$

$$C_A = 9.77 \text{ kN}$$

The catalog load rating of Bearing A based on one million revolutions is 9.77 kN.

Bearing B is already rated at one million revolutions, so $C_B = 6.6 \text{ kN}$.

Since $C_A > C_B$, bearing A can carry the larger load.

[References](#)

Multiple Choice

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Difficulty: Easy

2.

Award: 15.00 points

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.17 kN	10^9 rev	93%

The Basic Load Rating for the ball bearing is $23.6 \pm 2\%$ kN.

Explanation:

Given, $F_D = 2.17$ kN, $D_D = 10^9$ rev, and $R = 0.93$, $a = 3$.

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 10^6$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$x_D = \frac{L_D}{L_R}$$

$$x_D = \frac{10^9 \text{ rev}}{10^6 \text{ rev}}$$

$$x_D = 10^3$$

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 2.17 \text{ kN} \left(\frac{10^3}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.93} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 23.6 \text{ kN}$$

The Basic Load Rating for the ball bearing is 23.6 kN.

References**Numeric Response**

Problem 11.010 - Determination of Basic Load Rating for a ball bearing Difficulty: Easy

3.

Award: 15.00 points

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)

Radial Load	Design Life	Desired Reliability
820 lbf	12300 h, 315 rev/min	92%

The Basic Load Rating for the ball bearing is $5319.98 \pm 2\%$ lbf.

Explanation:

Given, $F_D = 820$ lbf, $L_D = 12300$ hours, $n_D = 315$ rev/min, and $R = 0.92$.

$$x_D = \frac{L_D n_D 60}{L_R}$$

$$x_D = \frac{12300 \text{ h} \times 315 \text{ rev/min} \times 60}{10^6 \text{ rev}}$$

$$x_D = 232.47$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 10^6$ revolutions, $b = 1.483$, and $\theta = 4.459$.

Therefore, we have

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 820 \text{ lbf} \left(\frac{232.47}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.92} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 5319.98 \text{ lbf.}$$

The Basic Load Rating for the ball bearing is 5319.98 lbf.

References**Numeric Response**

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

Difficulty: Easy

4.

Award: 15.00 points

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)

Radial Load	Design Life	Desired Reliability
4.05 kN	8.4 kh, 500 rev/min	98%

The Basic Load Rating for the ball bearing is $36.67 \pm 2\%$ kN.

Explanation:

Given, $F_D = 4.05$ kN, $LD = 8400$ hours, $n_D = 500$ rev/min, and $R = 0.98$.

From equation 11–3 the catalog load rating is given as follows:

$$x_D = \frac{L_D n_D 60}{L_R}$$

$$x_D = \frac{(8.4 \times 1000) \ln(500 \text{ rev/min}) (60)}{10^6 \text{ rev}}$$

$$x_D = 252$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 4.05 \text{ kN} \left(\frac{252}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.98} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 36.67 \text{ kN}$$

The Basic Load Rating for the ball bearing is 36.67 kN.

References**Numeric Response**

Problem 11.012 - Determination of Basic Load Rating for a ball bearing Difficulty: Easy

Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 65-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 3-kN axial load and a 7-kN radial load. The outer ring rotates at 400 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References**Section Break**

Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS Difficulty: Medium

5. Award: 20.00 points

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i taken from Table 11-1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o for a 65-mm bore, from Table 11-2 gives:

$$e = \frac{3.0 \text{ kN}}{34.0 \text{ kN}}$$

$$e = 0.088$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{3.0 \text{ kN}}{1.2 \times (7.0 \text{ kN})} = 0.357$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11-1.

The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56) (1.2) (7.0 \text{ kN}) + (1.53) (3.0 \text{ kN})$$

$$F_e = 9.29 \text{ kN}$$

The equivalent radial load is 9.29 kN.

References

Numeric Response

Problem 11.022.a - Calculation of Effective Radial Load

Difficulty: Medium

Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 30-mm bore (see Tables 11-1 and 11-2 for specifications) is loaded with a 2-kN axial load and a 5-kN radial load. The inner ring rotates at 550 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break

Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

Difficulty: Medium

6.

Award: 20.00 points

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .The equivalent radial load, F_e , is kN.**Explanation:**

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i , taken from Table 11-1.First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_0}$ is determined.Taking the value of C_0 for a 30-mm bore, from Table 11-2 gives:

$$C_0 = 10$$

Interpolate the $\frac{F_a}{C_0}$ value between 0.17 and 0.28 to get the value of e at $\frac{F_a}{C_0} = 0.2$.

$$e = \frac{(0.2 - 0.17)(0.38 - 0.34)}{(0.28 - 0.17)} + 0.34$$

$$e = 0.35$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{2.0 \text{ kN}}{1.0 \times (5.0 \text{ kN})} = 0.4$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11-1.The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56) \times (1) \times (5.0 \text{ kN}) + (1.27) \times (2.0 \text{ kN})$$

$$F_e = 5.34 \text{ kN}$$

The equivalent radial load, F_e , is 5.34 kN.**References****Numeric Response**

Problem 11.023.a - Calculation of Effective Radial Load

Difficulty: Medium

ABOVE AVERAGE

1.

Award: 15 out of 15.00 points

 You did not receive full credit for this question in a previous attempt

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.08 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 7 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing **A**  can carry the larger load.

References

Multiple Choice

Chapter: 11 Rolling-Contact Bearings

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Section: 11.05 Relating Load, Life, and Reliability

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.08 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 7 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing **A**  can carry the larger load.

Explanation:

Both bearings need to be rated in terms of the same catalog rating system to compare them. Using a rating life of one million revolutions, both bearings can be rated in terms of a Basic Load Rating.

From equation 11–3, the catalog load rating as a function of the desired load, desired life, and catalog rating life is given as follows:

$$C_A = F_A \left(\frac{L_A}{L_R} \right)^{1/a} = F_A \left(\frac{\mathcal{L}_A n_A 60}{L_R} \right)^{1/a}$$

(a = 3 for ball bearings)

$$C_A = 2.08 \left(\frac{(3000 \text{ h})(500 \text{ rev/min})(60)}{10^6 \text{ cycles}} \right)^{1/3}$$

$$C_A = 9.32 \text{ kN}$$

The catalog load rating of Bearing A based on one million revolutions is 9.32 kN.

Bearing B is already rated at one million revolutions, so $C_B = 7 \text{ kN}$.

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.02 kN	10 ⁹ rev	91%

The Basic Load Rating for the ball bearing is **20.74**  kN.

References

Numeric Response Chapter: 11 Rolling-Contact Bearings

Problem 11.010 - Determination of Basic Load Rating for a ball bearing Section: 11.05 Relating Load, Life, and Reliability

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.02 kN	10 ⁹ rev	91%

The Basic Load Rating for the ball bearing is **20.74 ± 2%** kN.

Explanation:

Given, $F_D = 2.02$ kN, $D_D = 10^9$ rev, and $R = 0.91$, $a = 3$.

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 10^6$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$x_D = \frac{L_D}{L_R}$$

$$x_D = \frac{10^9 \text{ rev}}{10^6 \text{ rev}}$$

$$x_D = 10^3$$

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 2.02 \text{ kN} \left(\frac{10^3}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.91} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 20.74 \text{ kN}$$

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)

Radial Load	Design Life	Desired Reliability
800 lbf	13000 h, 330 rev/min	91%

The Basic Load Rating for the ball bearing is **5226.15**  lbf.

References

Numeric Response

Chapter: 11 Rolling-Contact Bearings

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

Section: 11.05 Relating Load, Life, and Reliability

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)

Radial Load	Design Life	Desired Reliability
800 lbf	13000 h, 330 rev/min	91%

The Basic Load Rating for the ball bearing is **5226.15 ± 2%** lbf.

Explanation:

Given, $F_D = 800$ lbf, $LD = 13000$ hours, $n_D = 330$ rev/min, and $R = 0.91$.

$$x_D = \frac{\mathcal{L}_D n_D 60}{L_R}$$

$$x_D = \frac{13000 \text{ h} \times 330 \text{ rev/min} \times 60}{10^6 \text{ rev}}$$

$$x_D = 257.4$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $LR = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

Therefore, we have

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 800 \text{ lbf} \left(\frac{257.4}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.91} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 5226.15 \text{ lbf.}$$

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
4.15 kN	8 kh, 550 rev/min	91%

The Basic Load Rating for the ball bearing is **27.34** ✓ kN.

References

Numeric Response Chapter: 11 Rolling-Contact Bearings

Problem 11.012 - Determination of Basic Load Rating for a ball bearing Section: 11.05 Relating Load, Life, and Reliability

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
4.15 kN	8 kh, 550 rev/min	91%

The Basic Load Rating for the ball bearing is **27.34 ± 2%** kN.

Explanation:

Given, $F_D = 4.15\text{ kN}$, $L_D = 8000\text{ hours}$, $n_D = 550\text{ rev/min}$, and $R = 0.91$.

From equation 11–3 the catalog load rating is given as follows:

$$x_D = \frac{L_D n_D 60}{L_R}$$
$$x_D = \frac{(8 \times 1000) \text{ h}(550 \text{ rev/min}) (60)}{10^6 \text{ rev}}$$
$$x_D = 264$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$
$$C_{10} = 1 \times 4.15 \text{ kN} \left(\frac{264}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.91} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$
$$C_{10} = 27.34 \text{ kN}$$

Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 65-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 3-kN axial load and a 7-kN radial load. The outer ring rotates at 700 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 11 Rolling-Contact Bearings
Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is 9.29  kN.

References

Numeric Response	Chapter: 11 Rolling-Contact Bearings
Problem 11.022.a - Calculation of Effective Radial Load	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is 9.29 ± 2% kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i taken from Table 11–1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o for a 65-mm bore, from Table 11–2 gives:

$$e = \frac{3.0 \text{ kN}}{34.0 \text{ kN}}$$

$$e = 0.088$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{3.0 \text{ kN}}{1.2 \times (7.0 \text{ kN})} = 0.357$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11–1.

The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56) (1.2) (7.0 \text{ kN}) + (1.53) (3.0 \text{ kN})$$

$$F_e = 9.29 \text{ kN}$$

Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 30-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 2-kN axial load and a 5-kN radial load. The inner ring rotates at 450 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 11 Rolling-Contact Bearings
Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 11.05 Relating Load, Life, and Reliability

6.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .

The equivalent radial load, F_e , is **5.34**  kN.

References

Numeric Response	Chapter: 11 Rolling-Contact Bearings
Problem 11.023.a - Calculation of Effective Radial Load	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .

The equivalent radial load, F_e , is kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i , taken from Table 11–1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o , for a 30-mm bore, from Table 11–2 gives:

$$C_o = 10$$

Interpolate the $\frac{F_a}{C_o}$ value between 0.17 and 0.28 to get the value of e at $\frac{F_a}{C_o} = 0.2$.

$$e = \frac{(0.2 - 0.17)(0.38 - 0.34)}{(0.28 - 0.17)} + 0.34$$

$$e = 0.35$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{2.0 \text{ kN}}{1.0 \times (5.0 \text{ kN})} = 0.4$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11–1.

The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56) \times (1) \times (5.0 \text{ kN}) + (1.27) \times (2.0 \text{ kN})$$

$$F_e = 5.34 \text{ kN}$$

AVERAGE

1.

Award: 15 out of 15.00 points

 You received credit for this question in a previous attempt

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.08 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 7.1 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing **A**  can carry the larger load.

References

Multiple Choice

Chapter: 11 Rolling-Contact Bearings

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Section: 11.05 Relating Load, Life, and Reliability

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.08 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 7.1 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing A  can carry the larger load.

Explanation:

Both bearings need to be rated in terms of the same catalog rating system to compare them. Using a rating life of one million revolutions, both bearings can be rated in terms of a Basic Load Rating.

From equation 11–3, the catalog load rating as a function of the desired load, desired life, and catalog rating life is given as follows:

$$C_A = F_A \left(\frac{L_A}{L_R} \right)^{1/a} = F_A \left(\frac{\mathcal{L}_A n_A 60}{L_R} \right)^{1/a}$$

(a = 3 for ball bearings)

$$C_A = 2.08 \left(\frac{(3000 \text{ h})(500 \text{ rev/min})(60)}{10^6 \text{ cycles}} \right)^{1/3}$$

$$C_A = 9.32 \text{ kN}$$

The catalog load rating of Bearing A based on one million revolutions is 9.32 kN.

Bearing B is already rated at one million revolutions, so $C_B = 7.1 \text{ kN}$.

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.16 kN	10^9 rev	96%

The Basic Load Rating for the ball bearing is **26.63**  kN.

References

Numeric Response

Chapter: 11 Rolling-Contact Bearings

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

Section: 11.05 Relating Load, Life, and Reliability

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.16 kN	10^9 rev	96%

The Basic Load Rating for the ball bearing is **26.63 ± 2%** kN.

Explanation:

Given, $F_D = 2.16$ kN, $D_D = 10^9$ rev, and $R = 0.96$, $a = 3$.

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 10^6$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$x_D = \frac{L_D}{L_R}$$

$$x_D = \frac{10^9 \text{ rev}}{10^6 \text{ rev}}$$

$$x_D = 10^3$$

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 2.16 \text{ kN} \left(\frac{10^3}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.96} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 26.63 \text{ kN}$$

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)

Radial Load	Design Life	Desired Reliability
800 lbf	12800 h, 330 rev/min	95%

The Basic Load Rating for the ball bearing is **5940.25** ✓ lbf.

References

Numeric Response

Chapter: 11 Rolling-Contact Bearings

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

Section: 11.05 Relating Load, Life, and Reliability

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)

Radial Load	Design Life	Desired Reliability
800 lbf	12800 h, 330 rev/min	95%

The Basic Load Rating for the ball bearing is **5940.25 ± 2%** lbf.

Explanation:

Given, $F_D = 800$ lbf, $LD = 12800$ hours, $n_D = 330$ rev/min, and $R = 0.95$.

$$x_D = \frac{\mathcal{L}_D n_D 60}{L_R}$$

$$x_D = \frac{12800 \text{ h} \times 330 \text{ rev/min} \times 60}{10^6 \text{ rev}}$$

$$x_D = 253.44$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $LR = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

Therefore, we have

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 800 \text{ lbf} \left(\frac{253.44}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.95} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 5940.25 \text{ lbf.}$$

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
4.1 kN	8.3 kh, 490 rev/min	90%

The Basic Load Rating for the ball bearing is **25.08** ✖ kN.

References

Numeric Response Chapter: 11 Rolling-Contact Bearings

Problem 11.012 - Determination of Basic Load Rating for a ball bearing Section: 11.05 Relating Load, Life, and Reliability

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
4.1 kN	8.3 kh, 490 rev/min	90%

The Basic Load Rating for the ball bearing is **25.68 ± 2%** kN.

Explanation:

Given, $F_D = 4.1$ kN, $L_D = 8300$ hours, $n_D = 490$ rev/min, and $R = 0.9$.

From equation 11–3 the catalog load rating is given as follows:

$$x_D = \frac{\mathcal{L}_D n_D 60}{L_R}$$
$$x_D = \frac{(8.3 \times 1000) \text{ h}(490 \text{ rev/min}) (60)}{10^6 \text{ rev}}$$

$$x_D = 244.02$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$
$$C_{10} = 1 \times 4.1 \text{ kN} \left(\frac{244.02}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.9} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 25.68 \text{ kN}$$

Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 65-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 3-kN axial load and a 7-kN radial load. The outer ring rotates at 500 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 11 Rolling-Contact Bearings
Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is 9.29  kN.

References

Numeric Response	Chapter: 11 Rolling-Contact Bearings
Problem 11.022.a - Calculation of Effective Radial Load	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is 9.29 ± 2% kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i taken from Table 11–1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o for a 65-mm bore, from Table 11–2 gives:

$$e = \frac{3.0 \text{ kN}}{34.0 \text{ kN}}$$

$$e = 0.088$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{3.0 \text{ kN}}{1.2 \times (7.0 \text{ kN})} = 0.357$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11–1.

The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56)(1.2)(7.0 \text{ kN}) + (1.53)(3.0 \text{ kN})$$

$$F_e = 9.29 \text{ kN}$$

Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 30-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 2-kN axial load and a 5-kN radial load. The inner ring rotates at 500 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 11 Rolling-Contact Bearings
Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 11.05 Relating Load, Life, and Reliability

6.

Award: 20 out of 20.00 points



You did not receive full credit for this question in a previous attempt

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .

The equivalent radial load, F_e , is **5.34**  kN.

References

Numeric Response	Chapter: 11 Rolling-Contact Bearings
Problem 11.023.a - Calculation of Effective Radial Load	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .

The equivalent radial load, F_e , is kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i , taken from Table 11–1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o , for a 30-mm bore, from Table 11–2 gives:

$$C_o = 10$$

Interpolate the $\frac{F_a}{C_o}$ value between 0.17 and 0.28 to get the value of e at $\frac{F_a}{C_o} = 0.2$.

$$e = \frac{(0.2 - 0.17)(0.38 - 0.34)}{(0.28 - 0.17)} + 0.34$$

$$e = 0.35$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{2.0 \text{ kN}}{1.0 \times (5.0 \text{ kN})} = 0.4$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11–1.

The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56) \times (1) \times (5.0 \text{ kN}) + (1.27) \times (2.0 \text{ kN})$$

$$F_e = 5.34 \text{ kN}$$

BELOW AVERAGE

1.

Award: 0 out of 15.00 points

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.08 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 6.7 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing **B** ❌ can carry the larger load.

References

Multiple Choice

Chapter: 11 Rolling-Contact Bearings

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Section: 11.05 Relating Load, Life, and Reliability

Problem 11.009 - Selection of bearing based on ratings from different manufacturers

Two ball bearings from different manufacturers are being considered for a certain application. Bearing A has a catalog rating of 2.08 kN based on a catalog rating system of 3000 hours at 500 rev/min. Bearing B has a catalog rating of 6.7 kN based on a catalog that rates at 10^6 cycles. For a given application, determine which bearing can carry the larger load.

Bearing **A** ✅ can carry the larger load.

Explanation:

Both bearings need to be rated in terms of the same catalog rating system to compare them. Using a rating life of one million revolutions, both bearings can be rated in terms of a Basic Load Rating.

From equation 11–3, the catalog load rating as a function of the desired load, desired life, and catalog rating life is given as follows:

$$C_A = F_A \left(\frac{L_A}{L_R} \right)^{1/a} = F_A \left(\frac{\mathcal{L}_A n_A 60}{L_R} \right)^{1/a}$$

(a = 3 for ball bearings)

$$C_A = 2.08 \left(\frac{(3000 \text{ h})(500 \text{ rev/min})(60)}{10^6 \text{ cycles}} \right)^{1/3}$$

$$C_A = 9.32 \text{ kN}$$

The catalog load rating of Bearing A based on one million revolutions is 9.32 kN.

Bearing B is already rated at one million revolutions, so $C_B = 6.7 \text{ kN}$.

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.09 kN	10 ⁹ rev	94%

The Basic Load Rating for the ball bearing is **4491.8** ❌ kN.

References

Numeric Response Chapter: 11 Rolling-Contact Bearings

Problem 11.010 - Determination of Basic Load Rating for a ball bearing Section: 11.05 Relating Load, Life, and Reliability

Problem 11.010 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications):

Radial Load	Design Life	Desired Reliability
2.09 kN	10 ⁹ rev	94%

The Basic Load Rating for the ball bearing is

23.54 ± 2%

 kN.

Explanation:

Given, $F_D = 2.09$ kN, $D_D = 10^9$ rev, and $R = 0.94$, $a = 3$.

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 10^6$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$x_D = \frac{L_D}{L_R}$$

$$x_D = \frac{10^9 \text{ rev}}{10^6 \text{ rev}}$$

$$x_D = 10^3$$

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 2.09 \text{ kN} \left(\frac{10^3}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.94} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 23.54 \text{ kN}$$

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
840 lbf	12300 h, 330 rev/min	92%

The Basic Load Rating for the ball bearing is **6481.3** ❌ lbf.

References

Numeric Response Chapter: 11 Rolling-Contact Bearings

Problem 11.011 - Determination of Basic Load Rating for a ball bearing Section: 11.05 Relating Load, Life, and Reliability

Problem 11.011 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
840 lbf	12300 h, 330 rev/min	92%

The Basic Load Rating for the ball bearing is **5534.9 ± 2%** lbf.

Explanation:

Given, $F_D = 840$ lbf, $L_D = 12300$ hours, $n_D = 330$ rev/min, and $R = 0.92$.

$$x_D = \frac{L_D n_D 60}{L_R}$$

$$x_D = \frac{12300 \text{ h} \times 330 \text{ rev/min} \times 60}{10^6 \text{ rev}}$$

$$x_D = 243.54$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

Therefore, we have

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$

$$C_{10} = 1 \times 840 \text{ lbf} \left(\frac{243.54}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.92} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$$C_{10} = 5534.9 \text{ lbf.}$$

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
4.15 kN	7.9 kh, 470 rev/min	91%

The Basic Load Rating for the ball bearing is 40.23 ✖ kN.

References

Numeric Response Chapter: 11 Rolling-Contact Bearings

Problem 11.012 - Determination of Basic Load Rating for a ball bearing Section: 11.05 Relating Load, Life, and Reliability

Problem 11.012 - Determination of Basic Load Rating for a ball bearing

For the bearing application specifications given in Table A, determine the Basic Load Rating for a ball bearing with which to enter a bearing catalog of manufacturer 2 in Table 11–6. Assume an application factor of one.

Table A (bearing specifications)		
Radial Load	Design Life	Desired Reliability
4.15 kN	7.9 kh, 470 rev/min	91%

The Basic Load Rating for the ball bearing is 25.84 ± 2% kN.

Explanation:

Given, $F_D = 4.15\text{ kN}$, $L_D = 7900\text{ hours}$, $n_D = 470\text{ rev/min}$, and $R = 0.91$.

From equation 11–3 the catalog load rating is given as follows:

$$x_D = \frac{\mathcal{L}_D n_D 60}{L_R}$$
$$x_D = \frac{(7.9 \times 1000) \text{ h}(470 \text{ rev/min}) (60)}{10^6 \text{ rev}}$$
$$x_D = 222.78$$

$a = 3$ for ball bearings.

From table 11–6, $x_0 = 0.02$, $L_R = 106$ revolutions, $b = 1.483$, and $\theta = 4.459$.

$$C_{10} = a_f F_D \left(\frac{x_D}{x_0 + (\theta - x_0) \left(\ln \left(\frac{1}{R_D} \right) \right)^{\frac{1}{b}}} \right)^{\frac{1}{a}}$$
$$C_{10} = 1 \times 4.15 \text{ kN} \left(\frac{222.78}{0.02 + (4.459 - 0.02) \left(\ln \left(\frac{1}{0.91} \right) \right)^{\frac{1}{1.483}}} \right)^{\frac{1}{3}}$$

$C_{10} = 25.84 \text{ kN}$

Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 65-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 3-kN axial load and a 7-kN radial load. The outer ring rotates at 400 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 11 Rolling-Contact Bearings
Problem 11.022 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is **8.57** ✖ kN.

References

Numeric Response	Chapter: 11 Rolling-Contact Bearings
Problem 11.022.a - Calculation of Effective Radial Load	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.022.a - Calculation of Effective Radial Load

What is the equivalent radial load, F_e ?

The equivalent radial load is kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i taken from Table 11–1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o for a 65-mm bore, from Table 11–2 gives:

$$e = \frac{3.0 \text{ kN}}{34.0 \text{ kN}}$$

$$e = 0.088$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{3.0 \text{ kN}}{1.2 \times (7.0 \text{ kN})} = 0.357$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11–1.

The value of $X_i \approx 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56)(1.2)(7.0 \text{ kN}) + (1.53)(3.0 \text{ kN})$$

$$F_e = 9.29 \text{ kN}$$

Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS

A 02-series single-row deep-groove ball bearing with a 30-mm bore (see Tables 11–1 and 11–2 for specifications) is loaded with a 2-kN axial load and a 5-kN radial load. The inner ring rotates at 350 rev/min.

NOTE: This is a multi-part question. Once an answer is submitted, you will be unable to return to this part.

References

Section Break	Chapter: 11 Rolling-Contact Bearings
Problem 11.023 - Calculation of Effective Radial Load - DEPENDENT MULTI-PART PROBLEM - ASSIGN ALL PARTS	Section: 11.05 Relating Load, Life, and Reliability

6. Award: 20 out of 20.00 points

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .

The equivalent radial load, F_e , is **5.34**  kN.

References

Numeric Response	Chapter: 11 Rolling-Contact Bearings
Problem 11.023.a - Calculation of Effective Radial Load	Section: 11.05 Relating Load, Life, and Reliability

Problem 11.023.a - Calculation of Effective Radial Load

Determine the equivalent radial load, F_e .

The equivalent radial load, F_e , is **5.34 ± 2%** kN.

Explanation:

The following equation is used to determine the equivalent radial load:

$$F_e = X_i V F_r + Y_i F_a$$

To determine if the axial load is sufficiently large enough to impact the required capacity of the bearing, a two-step process is used to determine the values of X_i and Y_i , taken from Table 11–1.

First, the ratio of the axial load to the static load rating of the bearing $\frac{F_a}{C_o}$ is determined.

Taking the value of C_o , for a 30-mm bore, from Table 11–2 gives:

$$C_o = 10$$

Interpolate the $\frac{F_a}{C_o}$ value between 0.17 and 0.28 to get the value of e at $\frac{F_a}{C_o} = 0.2$.

$$e = \frac{(0.2 - 0.17)(0.38 - 0.34)}{(0.28 - 0.17)} + 0.34$$

$$e = 0.35$$

Secondly, the ratio of $\frac{F_a}{V F_r}$ is calculated.

$$\frac{2.0 \text{ kN}}{1.0 \times (5.0 \text{ kN})} = 0.4$$

Since $e < \frac{F_a}{V F_r}$, X_i is not 1.0 and Y_i is not 0.0 in Table 11–1.

The value of $X_i = 0.56$; however, it is necessary to interpolate for Y_i to obtain an equivalent radial load of

$$F_e = (0.56) \times (1) \times (5.0 \text{ kN}) + (1.27) \times (2.0 \text{ kN})$$

$$F_e = 5.34 \text{ kN}$$

**MACHINE DESIGN I
FALL 2021**

Project # 1

Safety Factor of Bolt in Real-life Engineering Application

**MACHINE DESIGN I
FALL 2021**

SAMPLE PROJECT STATEMENT AND GRADING SCHEME

MCEG3043: Machine Design I

Fall 2021

Project: Safety Factor of Bolt in Real-life Engineering Application

Assigned on 11/15/2021; Due on 12/1/2021

Total points: 100

1. Introduction

This is an open-ended design project related to the application of bolt. You need to choose one real-life engineering application of bolts and determine the number of bolts required with consideration of safety. The design tasks are defined in Section 2.

2. Design Tasks

In this project, you need to complete the following design tasks.

- (1) Choose one real-life engineering application of bolts with extension loads. The steel cylinder head is shown in Figure 1 as an example. You CAN NOT use steel cylinder head or pressure vessel in Example 8-4 as your design application.

You need to describe (a) what the application is; (b) how the bolts are used in the application; (c) how the extension loads are generated; (d) show a schematic diagram (picture or hand sketch) of the application similar to Figure 1.

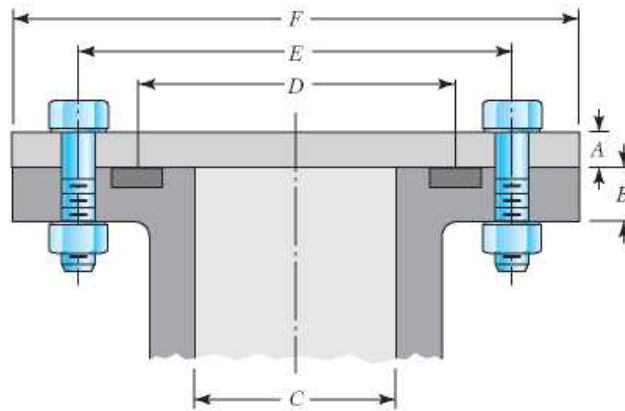


Figure 1. Skematic diagram of steel cylinder head

- (2) Define the given parameters. Table 1 is shown as example for steel cylinder head application. Be aware that your given parameters may be different from Table 1 because your application will be different.

Table 1. Given parameters of steel cylinder head

Parameters	A	B	C	D	p_g	Bolt grade	Bolt spec.
Value	$\frac{1}{2}$ in	$\frac{5}{8}$ in	2.75 in	3.5 in	1500 psi	SAE 5	$\frac{1}{2}$ in-13

You need to define: (a) the bolts specifications, suitable length of the bolt, and bolt grade for the bolt used in the application; (b) the thickness of members and materials of members; (c) the parameters needed to calculate the extension load. All the parameters you defined need to be realistic and need to be justified.

- (3) Calculate k_b , k_m , and C . You need to type detail calculation of the bolt stiffness k_b , stiffness of members k_m , and stiffness constant C . The calculation must be typed in **MS Word using “Equation Editor”**.
- (4) Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart. You need to type the detail calculation in **MS Word using “Equation Editor”**.
- (5) With the number of bolts obtained in part (4), determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation.

3. Project Submission

You need to submit soft copy (word file) of your report on **Taskstream** (<http://ecourses.pvamu.edu>). Submit your report to **“Assignment (Outcome 2)”**.

4. Report Format

The report should at least include the following sections:

- a. Cover page;
- b. Abstract;
- c. Problem Description;
- d. Detail Description of Design Tasks; (All calculation should be typed in **MS Word using “Equation Editor”**)
- e. Summary of Results;

5. Grading Scheme

Item	Description	Maximum Points	Your Score
1	Choose one real-life engineering application of bolts with extension loads and describe details of the applications.	15	
2	Define the given parameters and justify your selection.	20	
3	Calculate k_b , k_m , and C .	25	
4	Find the number of bolts required.	10	
5	Determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation	10	
6	Overall quality of the report.	20	
	Total	100	

Reference:

1. Example 8-4 and Problem 8-34 in the textbook.

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MECHANICAL ENGINEERING DEPARTMENT PRAIRIE VIEW A&M UNIVERSITY



MCEG 3304 MACHINE DESIGN 1

Fall 2021 Section P01

PROJECT REPORT ON

Project: Safety Factor of Bolt in Real-life Engineering Application

Assigned on 11/15/2021; Due on 12/1/2021

Submitted By

Joseph Steele

To

Dr. Xiaobo Peng

No.	First Name	Application(15)	Parameters(20)	kb, km, C(25)	N(10)	nL, np, no (10)	report (20)	total
21	Steele, Joseph	15.0	19.0	25.0	10.0	10.0	18.0	97.0

ABSTRACT

Application of choice: In engineering, there is heavy exponential growth in influence in regards to technological advancements that assist with our daily task. In most workplaces, it is common to see that they are densely populated with computer desktops. Desktops have become significantly faster over decades which have enabled many experts to evaluate many theoretical situations that incorporate intense mathematical applications. Specifically speaking, the innovative advancements of computer components, such as motherboards and central processing units, has led to the increase of allotted storage of inherent engineering information to perform major task.

Bolts in the Application: When discussing the assembly process of technologies concerning desktops, we must address the main components such as the RAM (Random Access Memory), CPU (Central Processing Unit), GPU (Graphics Processing Unit), Power supply, as well as the motherboard. In this project I am using the application of bolts and nuts that are used to fasten the copper material from the motherboard to the back of the steel motherboard plates found on the desktop case that support the loads from the installation of the CPU and the GPU.

How the extension loads are Generated: Once again, when the bolts are fastened, it supports its purpose by supporting the loads of the GPU by keeping the motherboard in place within the desktop casing. This is essential as the Graphics card or GPU, is typically the largest and most dense component in the assembly process. If the bolt cannot suffice the tension received from the GPU, the motherboard would fall out of place and would be unable to function which could also negatively affect other components within the system.

PROBLEM DESCRIPTION

Given the parameters of the members and the bolts used within this application, we must find the suitable length of the bolt, evaluate the thickness of the members, the bolt stiffness, the material stiffness, the yielding factor of safety, load factor of safety, and the safety based on the joint separation.

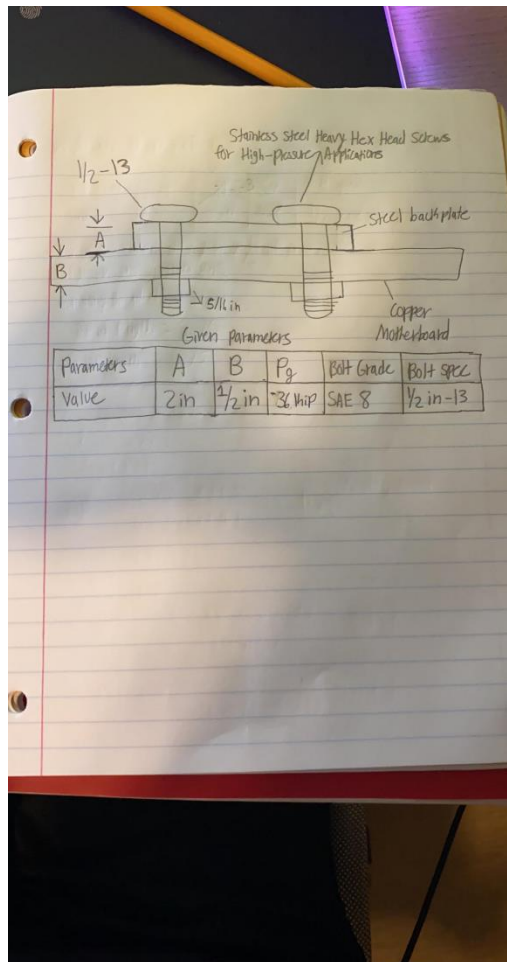
(Copper Motherboard installed inside the Desktop Casing)



(Steel Back Plate Motherboard Attachment)



DETAILED DESCRIPTION OF DESIGN TASK



Parameters	A (Steel member)	B(copper Member)	Pg	BOLT GRADE	BOLT SPEC
Values	2 in	½ in	36 kip	SAE 8	½ IN-13

Explain why Pg=36kip

- 1.) Length of the bolt

$$l = l_1 + l_2$$

$$l = 2 + \frac{1}{2} = 2.5in$$

Height of the nut

Table A-31

$$h = \frac{7}{16} = 0.4375in$$

$$L = l + h$$

$$L = 2.5 + 0.4375 = 2.9375in$$

Round to the nearest quarter inch= 3in

- 2.) Find Kb, bolt stiffness

Table 8-7

$$L_t = 2d + \frac{1}{4}in$$

$$L_t = 2(0.5) + \frac{1}{4} = 1.25in$$

$$l_d = L - L_t$$

$$l_d = 3 - 1.25 = 1.75in$$

$$l_t = l - l_d$$

$$l_t = 2.5 - 1.75 = 0.75in$$

Table 8-2

$$A_t = 0.1419$$

$$A_d = \frac{\pi d^2}{4}$$

$$A_d = \frac{\pi (.5)^2}{4} = .1963in$$

$$k_b = \frac{A_t A_d E}{A_d l_t + A_t l_d}$$

$$k_b = \frac{0.1419 * 0.1963 * 30}{(0.1963 * 0.75) + (0.1419 * 1.75)} = \frac{2.11Mlbf}{in}$$

- 3.) Find km, member stiffness

$$D = 1.5d$$

$$D = 1.5(0.5) = .75$$

$$D1 = .75$$

$$D2 = .75 + 2(2.5 - 2 * \tan 30) = 3.4405$$

$$D3 = .75$$

$$t1 = \frac{l}{2} = 1.25$$

$$t2 = A - t1 = .75$$

$$t3 = .5$$

$$K = \frac{.5774\pi(E)d_1}{\ln\left(\frac{1.155(\omega)+D_1-d_1}{1.155(\omega)+D_1+d_1} \cdot \frac{D_1+d_1}{D_1-d_1}\right)}$$

$$k_1 = \frac{.5774\pi(30)*.5}{\ln\left(\frac{1.155(1.25)+.75-.5}{1.155(1.25)+.75+.5} \cdot \frac{.75+.5}{.75-.5}\right)} = 23.75 \text{ Mlb/in}$$

$$k_2 = \frac{.5774\pi(30)*.5}{\ln\left(\frac{1.155(.75)+3.4405-.5}{1.155(.75)+3.4405+.5} \cdot \frac{3.4405+.5}{3.4405-.5}\right)} = 457.33 \text{ Mlb/in}$$

$$k_3 = \frac{.5774\pi(30)*.5}{\ln\left(\frac{1.155(.5)+.75-.5}{1.155(.5)+.75+.5} \cdot \frac{.75+.5}{.75-.5}\right)} = 19.201 \text{ Mlb/in}$$

$$k_m = \left(\frac{1}{k_1} + \frac{1}{k_2} + \frac{1}{k_3}\right)^{-1} = \left(\frac{1}{23.751} + \frac{1}{457.433} + \frac{1}{19.201}\right)^{-1} = 10.376 \frac{\text{Mlb}}{\text{in}}$$

4.) Calculate Stiffness constant

$$C = \frac{k_b}{k_b + k_m}$$

$$C = \frac{2.11}{2.11 + 10.376} = \frac{0.1689 \text{ Mlb}}{\text{in}}$$

5.) Find number of bolts required for a load factor of 2

$$\text{Load Factor} = nL = 2$$

$$nL = \frac{S_{pA_t} - Fi}{C\left(\frac{P_{total}}{N}\right)}$$

$$N = \frac{(36) * (2) * 0.1689}{120 * (0.1419) - 12.771} = 2.8566 = 3 \text{ bolts obtained}$$

6.) Find the realized load factor for overload

$$nL = \frac{S_{pA_t} - Fi}{C\left(\frac{P_{total}}{N}\right)}$$

$$nL = \frac{120 * (0.1419) - 12.771}{0.1689 * \left(\frac{36}{3}\right)} = 2.10$$

7.) Find the yielding factor of safety

$$n_p = \frac{S_{pA_t}}{C\left(\frac{P_{total}}{N}\right) + Fi}$$

$$n_p = \frac{120 * (0.1419)}{0.1689 * \left(\frac{36}{3}\right) + 12.771} = 1.15$$

8.) Find the load factor for joint separation

$$n_0 = \frac{Fi}{\left(\frac{P_{total}}{N}\right) + (1 - C)}$$

$$n_0 = \frac{12.771}{\left(\frac{36}{3}\right) + (1 - 0.1689)} = 1.28$$

SUMMARY OF RESULTS

The analyzation of the bolts and the member specifications given by performing statistical analysis of various stiffnesses and load factors portray how the assembly needs at least 3 bolts to sufficiently support the weight of load caused by the installation of the other components. This is an inherent finding, as anything less than three bolts of the same graded material would cause a mounting failure within the desktop casing. This could lead to further damage within the system as the detachment of the motherboard due to the insufficient mounting failure could damage other components in the system. On the other hand, if the assembly process uses 4 bolts, then the assembly process will be safe as it will support the load applied on the extension of the bolts when fastened and assembled. Overall, this project allowed me to receive a further understanding of how extension loads on bolts interact in regards to clamping to materials members together in a real-life application.

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MECHANICAL ENGINEERING DEPARTMENT
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MCEG 3304 MACHINE DESIGN 1

Fall 2021 Section P01

Project Report

“Safety Factor of Bolt in Real Life Engineering Application”

Submitted by:

Harold Rodriguez

On

December 1st 2021

To

Dr. Peng

No.	First Name	Application(15)	Parameters(20)	kb, km, C(25)	N(10)	nL, np, no (10)	report (20)	total
17	Rodriguez, Harold	15.0	18.0	25.0	0.0	10.0	16.0	84.0

ABSTRACT

This project focus on real-life engineering application of bolts with consideration of safety. In this project we going to complete several task which consider of: What Application is, how the bolts are used in the application, and how the extension loads are generated. It would include a problem description with all the calculation.

Problem Description

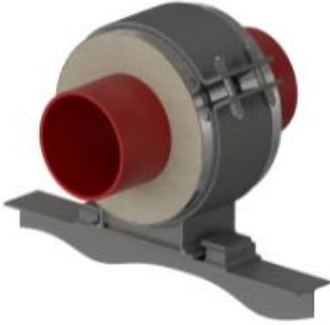
This is an open-ended design project related to the application of bolt. You need to choose one real-life engineering application of bolts and determine the number of bolts required with consideration of safety.

In this project, you need to complete the following design tasks. (1) Choose one real-life engineering application of bolts with extension loads. The steel cylinder head is shown in Figure 1 as an example. You CAN NOT use steel cylinder head or pressure vessel in Example 8-4 as your design application. You need to describe (a) what the application is; (b) how the bolts are used in the application; (c) how the extension loads are generated; (d) show a schematic diagram (picture or hand sketch). You need to define: (a) the bolts specifications, suitable length of the bolt, and bolt grade for the bolt used in the application; (b) the thickness of members and materials of members; (c) the parameters needed to calculate the extension load. All the parameters you defined need to be realistic and need to be justified.

(1) Calculate k_b , k_m , and C . You need to type detail calculation of the bolt stiffness k_b , stiffness of members k_m , and stiffness constant C . The calculation must be typed in MS Word using "Equation Editor".

(2) Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart. You need to type the detail calculation in MS Word using "Equation Editor".

(3) With the number of bolts obtained in part (2), determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation.



Pipe Shields to support the pipes to hold it steady against pressure.

A	B	C	D	Pg	Bolt Grade	Bolt Spec.
7/16	5/8	2.25	3	1400psi	SAE 5	3/8in-13 UNC

Explain the variables and indicate them in the diagram.

$$\text{Grip, } l = \frac{7}{16} + \frac{5}{8} = 1.06, \text{ From A-31 H} = 0.328$$

$$L = 1.06 + .328 = 1.388 \text{ in}$$

$$\text{Eq(8-13) } Lt = 2d + .25 = 2(.375) + .25 = 1$$

$$ld = l - Lt = 1.388 - 1 = .388$$

$$lt = l - ld = 1.06 - .388 = .672$$

$$At = \text{From Table 8-2} = .0775$$

$$Ad = \frac{\pi d^2}{4} = \frac{\pi (.375)^2}{4} = .1104 \text{ in}^2$$

$$E = 30 \text{ Mpsi}$$

Calculation of the Bolt Stiffness K_b

$$(\text{Eq 8-17}) K_b = \frac{AdAtE}{Adlt + Atld} = \frac{.1104 * .0775 * 30 \times 10^6}{.1104 * .672 + .0775 * .388} = 2.567 \frac{\text{Mlbf}}{\text{in}}$$

Aluminum Plate.

Frustrum 1

$$t = 1.5 \quad d = .375 \quad D = 1.5 \times .375 = .5625 \quad E = 10.3 \times 10^6$$

$$K1 = \frac{.5774\pi Ed}{\ln \frac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

$$K_1 = \frac{.5774\pi E d}{\ln \frac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

$$K_1 = \frac{.5774\pi \times 10.3(10^6) \times .375}{\ln \frac{(1.155(1.5) + .5625 - .375)(1.5 + .5625)}{(1.155(1.5) + .5625 + .375)(1.5 - .5625)}} = 21.26 \text{ MPA/m}$$

Frustrum 2

$$t = .5 d = .375 D = D_3 + 2 \times t_3 \times \tan 30^\circ \quad D = .5625 + 2 \times 1 \times \tan 30^\circ = .6485 \quad E = 10.3 \times 10^6$$

$$K_2 = \frac{.5774\pi E d}{\ln \frac{(1.155t + D - d)(D + d)}{(1.155t + D + d)(D - d)}}$$

$$K_2 = \frac{.5774\pi \times 10.3 \times 10^6 \times .375}{\ln \frac{(1.155(.5) + .6485 - .375)(.6485 + .375)}{(1.155(.5) + .6485 + .375)(.6485 - .375)}} = 10.22 \text{ MPA/m}$$

Cast Iron

Frustrum

$$t_3 = 1 \quad d = .375 \quad D_3 = .5625 \quad E = 14.5 \text{ Mpsi}$$

$$K3 = \frac{.5774\pi E d}{\ln \frac{(1.155t+D-d)(D+d)}{(1.155t+D+d)(D-d)}}$$

$$C.I \quad t_3 = 1, \quad d = .375 \quad D = .5625 \quad E = 14.5 \text{ mpsi}$$

$$K3 = \frac{.5774 \pi (14.5)(10^6) \times .375}{\ln \frac{(1.155(1) + .5625 - .375)(.5625 + .375)}{(1.155(1) + .5625 + .375)(.5625 - .375)}} = 8.46 \text{ Mlbf/in}$$

$$(Eq 8-18) \quad Km = \left(\frac{1}{K1} + \frac{1}{K2} + \frac{1}{K3} \right)^{-1}$$

$$Km = \left(\frac{1}{21.26} + \frac{1}{10.22} + \frac{1}{8.46} \right)^{-1} = \frac{3.80 \text{ Mlbf}}{\text{in}}$$

$$C = \frac{kb}{kb + Km} \rightarrow \frac{2.567}{2.567 + 3.80} = .4032$$

From Table 8.9 Sp(120 Kpsi) Grade 5

$$Fi = 0.75AtSp ==> .75 \times .0775 \times 120 = 6.975 \text{ kips}$$

$$\sigma_i = \frac{fi}{At} = \frac{6.975}{.0775} = 90 < 120(Sp) \text{ Safe}$$

$$P_{total} = Pg - Ac$$

$$P_{total} = \left[\frac{1400\pi(3^2)}{4} \right] \left(\frac{10^{-3}}{8} \right) = \frac{1.24 \text{ kips}}{\text{bolt}} \quad N=?$$

Yielding Factor of Safety (8-28)

$$np = \frac{SpAt}{Cp + fi} = \frac{120 \times .0775}{.4032(1.24) + 6.975} = 1.24 > \text{Safe}$$

Overload Factor of Safety (8-29)

$$nl = \frac{SpAt - fi}{Cp} = \frac{120 \times .0775 - 6.975}{.4032(1.24)} = 4.65 > \text{Safe}$$

Separation Factor of Safety (8-30)

$$no = \frac{fi}{P(1-C)} = \frac{6.975}{1.24(1-.4032)} = 9.425 > \text{Safe}$$

SUMMARY RESULTS

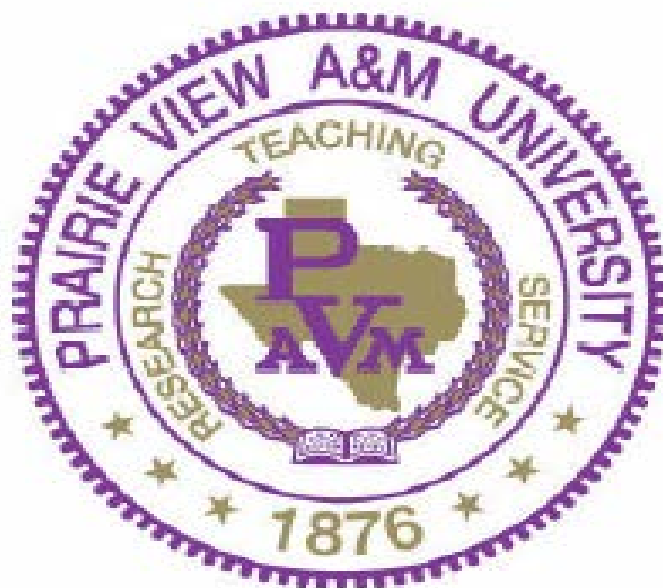
By doing all the calculations with the correct given equations we can conclude that the Safety Factor (1.24), Over Load Factor (4.65), and Separation Factor of Safety (9.425) it's all safe and we can know that there is no leaking or separation in the bolts with the

BELOW AVERAGE

First Name	Application(15)	Parameters(20)	kb, km, C(25)	N(10)	nL, np, no (10)	report (20)	total
Guerrero, Edward	9.0	20.0	25.0	0.0	9.0	12.0	75.0

MCEG3043: Machine Design I

Fall 2021



Project: Safety Factor of Bolt in Real-life Engineering Application

Assigned on 11/15/2021; Due on 12/1/2021

Monday - Wednesday 8:00 – 9:50am

Dr. Xioabo Peng

Edward Guerrero

Abstract

This project focuses on the application and analysis of a bolt, the design tasks are defined in Section 2. 2.

Design Tasks In this project, you need to complete the following design tasks. Choose one real-life engineering application of bolts with extension loads. We cannot use steel cylinder head or pressure vessel in Example 8-4 as our design application. We need to describe: what the application is, how the bolts are used in the application, how the extension loads are generated and show a schematic diagram (picture or hand sketch) of the application.

Problem Description

In this project, you need to complete the following design tasks. Choose one real-life engineering application of bolts with extension loads. The steel cylinder head is shown in Figure as an example. You CAN NOT use steel cylinder head or pressure vessel in Example 8-4 as your design application. You need to describe (a) what the application is; (b) how the bolts are used in the application; (c) how the extension loads are generated; (d) show a schematic diagram (picture or hand sketch) of the application similar to the figure shown in class.

Define the given parameters. Table 1 is shown as example for steel cylinder head application. Be aware that your given parameters may be different from Table 1 because your application will be different.

You need to define: (a) the bolts specifications, suitable length of the bolt, and bolt grade for the bolt used in the application; (b) the thickness of members and materials of members; (c) the parameters needed to calculate the extension load. All the parameters you defined need to be realistic and need to be justified.

Calculate k_b , k_m , and C . You need to type detail calculation of the bolt stiffness k_b , stiffness of members k_m , and stiffness constant C . The calculation must be typed in MS Word using "Equation Editor". Find the number of bolts required for a load factor of 2 where the bolts may be reused when the joint is taken apart. You need to type the detail calculation in MS Word using "Equation Editor". With the number of bolts obtained in part, determine the realized load factor for overload, the yielding factor of safety, and the load factor for joint separation.

Design Tasks



A	B	C	D	E	F	N	Pg	Grade	Spec.
0.375	0.5	2.75	3.25	5.5	7	8	1200	SAE 8	7/16 in-14

$$l_t = l - l_d = 0.875 - 0.125 = 0.75 \text{ in}$$

$$A_d = \pi (7/16) / 4 = 0.1503 \text{ in}^2$$

$$\text{Table 8-2: } A_t = 0.1063 \text{ in}^2$$

Eq. (8-17)

$$k_b = \frac{A_d A_t E}{A_d l_t + A_t l_d} = \frac{0.1503(0.1063)30}{0.1503(0.75) + 0.1063(0.125)} = 3.804 \text{ Mlbf/in}$$

Steel Cylinder head:

$$t = 0.375 \text{ in, } d = 0.4375 \text{ in, } D = 0.65625 \text{ in, } E = 30 \text{ Mpsi}$$

$$k_1 = \frac{0.5774\pi(30)0.4375}{\ln \left[\frac{1.155(0.375) + 0.65625 - 0.4375}{1.155(0.375) + 0.65625 + 0.4375} \right] \frac{(0.65625 + 0.4375)}{(0.65625 - 0.4375)}} = 31.40 \text{ Mlbf/in}$$

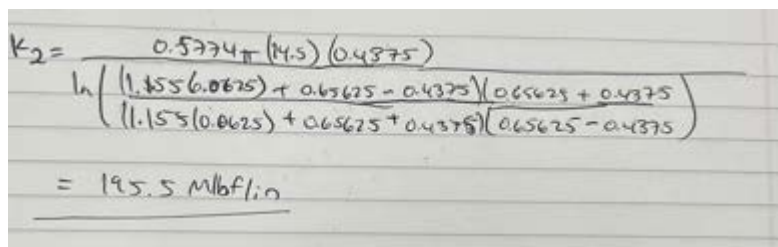
Cast iron has 2 frustrum.

$$\text{Midpoint of the complete joint is } (3/8 + 1/2)/2 = 0.4375 \text{ in}$$

Upper frustrum,

$$t = 0.4375 - 0.375 = 0.0625 \text{ in, } d = 0.4375 \text{ in, } D = 0.65625 + 2(0.375) \tan(30^\circ) = 1.089 \text{ in}$$

$$E = 14.5 \text{ Mpsi}$$

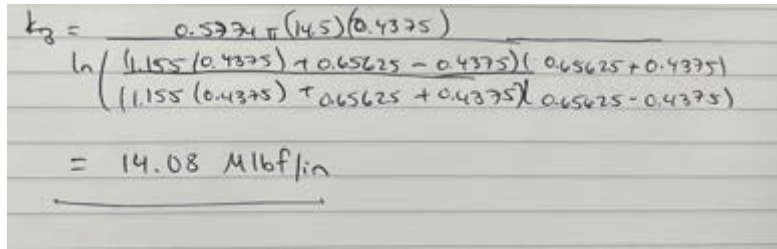


Handwritten calculation for k_2 :

$$k_2 = \frac{0.5774\pi(14.5)(0.4375)}{\ln \left(\frac{1.155(0.0625) + 0.65625 - 0.4375}{1.155(0.0625) + 0.65625 + 0.4375} \right) \frac{(0.65625 + 0.4375)}{(0.65625 - 0.4375)}} = 195.5 \text{ Mlbf/in}$$

Lower frustrum

$t = 0.4375$ in, $d = 0.4375$ in, $D = 0.65625$ in, $E = 14.5$ Mpsi



$$k_3 = \frac{0.5374 \pi (14.5) (0.4375)}{\ln \left(\frac{(1.155 / 0.4375) + 0.65625 - 0.4375}{(1.155 (0.4375) + 0.65625 + 0.4375)} \right) (0.65625 + 0.4375)}$$

$$= 14.08 \text{ Mlbf/in}$$

$$k_m = (1/31.40 + 1/195.5 + 1/14.08)^{-1} = 9.261 \text{ Mlbf/in}$$

$$C = k_b / (k_b + k_m) = 3.804 / (3.804 + 9.261) = 0.291$$

$$S_p = 120 \text{ kpsi}$$

$$F_i = 0.75 A_t S_p = 0.75(0.1063) (120) = 9.57 \text{ kips}$$

$$P_{\text{total}} = p_g A_c$$

$$P = P_{\text{total}} / N$$

N=?

$$P = [1200\pi (3.25^2)/4] (10^{-3}) / 8 = 1.244 \text{ kips/bolt}$$

Yielding factor of safety

$$n_p = \frac{S_p A_t}{CP + F_i} = \frac{120(0.1063)}{0.291(1.244) + 9.57} = 1.28$$

Overload Factor of safety

$$n_L = \frac{S_p A_t - F_i}{CP} = \frac{120(0.1063) - 9.57}{0.291(1.244)} = 8.80$$

Separation factor of safety

$$n_0 = \frac{F_i}{P(1-C)} = \frac{9.57}{1.244(1-0.291)} = 10.9$$

Summary Results

After all the calculations we know that the yielding factor of safety is safe since is over 1, load factor is 8.8 kips, and joint separation 10.9 and we know that it will not have leaks and is safe due of the value being higher than the minimum.