

SI VERSION



ENGINEERING MECHANICS
STATICS

| SIXTH EDITION |

J. L. MERIAM

L. G. KRAIGE

ENGINEERING MECHANICS

VOLUME 1

STATICS

SIXTH EDITION

SI VERSION

J. L. MERIAM

L. G. KRAIGE

*Virginia Polytechnic Institute
and State University*



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On the Cover: The Gateway Arch in St. Louis was first conceived by architect Eero Saarinen in the late 1940s. Later, a team of engineers led by John Dinkeloo devised further design details, and construction was completed in 1965. The shape of the 630-ft-high arch is that of a *weighted catenary*. This configuration could be formed by suspending a nonuniform flexible cable (symmetrically heavier near its ends) from two points on a horizontal line, “freezing” that shape, and inverting.

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FOREWORD

This series of textbooks was begun in 1951 by the late Dr. James L. Meriam. At that time, the books represented a revolutionary transformation in undergraduate mechanics education. They became the definitive textbooks for the decades that followed as well as models for other engineering mechanics texts that have subsequently appeared. Published under slightly different titles prior to the 1978 First Editions, this textbook series has always been characterized by logical organization, clear and rigorous presentation of the theory, instructive sample problems, and a rich collection of real-life problems, all with a high standard of illustration. In addition to the U.S. versions, the books have appeared in SI versions and have been translated into many foreign languages. These texts collectively represent an international standard for undergraduate texts in mechanics.

The innovations and contributions of Dr. Meriam (1917–2000) to the field of engineering mechanics cannot be overstated. He was one of the premier engineering educators of the second half of the twentieth century. Dr. Meriam earned his B.E., M. Eng., and Ph.D. degrees from Yale University. He had early industrial experience with Pratt and Whitney Aircraft and the General Electric Company. During the Second World War he served in the U.S. Coast Guard. He was a member of the faculty of the University of California–Berkeley, Dean of Engineering at Duke University, a faculty member at the California Polytechnic State University–San Luis Obispo, and visiting professor at the University of California–Santa Barbara, finally retiring in 1990. Professor Meriam always placed great emphasis on teaching, and this trait was recognized by his students wherever he taught. At Berkeley in 1963, he was the first recipient of the Outstanding Faculty Award of Tau Beta Pi, given primarily for excellence in teaching. In 1978, he received the Distinguished Educator Award for Outstanding Service to Engineering Mechanics Education from the American Society for Engineering Education, and in 1992 was the Society's recipient of the Benjamin Garver Lamme Award, which is ASSEE's highest annual national award.

Dr. L. Glenn Kraige, coauthor of the *Engineering Mechanics* series since the early 1980s, has also made significant contributions to mechanics education. Dr. Kraige earned his B.S., M.S., and Ph.D. degrees at the University of Virginia, principally in aerospace engi-

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neering, and he currently serves as Professor of Engineering Science and Mechanics at Virginia Polytechnic Institute and State University. During the mid 1970s, I had the singular pleasure of chairing Professor Kraige's graduate committee and take particular pride in the fact that he was the first of my three dozen Ph.D. graduates. Professor Kraige was invited by Professor Meriam to team with him and thereby ensure that the Meriam legacy of textbook authorship excellence was carried forward to future generations. For the past two and a half decades, this highly successful team of authors has made an enormous and global impact on the education of several generations of engineers.

In addition to his widely recognized research and publications in the field of spacecraft dynamics, Professor Kraige has devoted his attention to the teaching of mechanics at both introductory and advanced levels. His outstanding teaching has been widely recognized and has earned him teaching awards at the departmental, college, university, state, regional, and national levels. These include the Francis J. Maher Award for excellence in education in the Department of Engineering Science and Mechanics, the Wine Award for excellence in university teaching, and the Outstanding Educator Award from the State Council of Higher Education for the Commonwealth of Virginia. In 1996, the Mechanics Division of ASEE bestowed upon him the Archie Higdon Distinguished Educator Award. The Carnegie Foundation for the Advancement of Teaching and the Council for Advancement and Support of Education awarded him the distinction of Virginia Professor of the Year for 1997. In his teaching, Professor Kraige stresses the development of analytical capabilities along with the strengthening of physical insight and engineering judgment. Since the early 1980s, he has worked on personal-computer software designed to enhance the teaching/learning process in statics, dynamics, strength of materials, and higher-level areas of dynamics and vibrations.

The Sixth Edition of *Engineering Mechanics* continues the same high standards set by previous editions and adds new features of help and interest to students. It contains a vast collection of interesting and instructive problems. The faculty and students privileged to teach or study from Professors Meriam and Kraige's *Engineering Mechanics* will benefit from the several decades of investment by two highly accomplished educators. Following the pattern of the previous editions, this textbook stresses the application of theory to actual engineering situations, and at this important task it remains the best.



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PREFACE

Engineering mechanics is both a foundation and a framework for most of the branches of engineering. Many of the topics in such areas as civil, mechanical, aerospace, and agricultural engineering, and of course engineering mechanics itself, are based upon the subjects of statics and dynamics. Even in a discipline such as electrical engineering, practitioners, in the course of considering the electrical components of a robotic device or a manufacturing process, may find themselves first having to deal with the mechanics involved.

Thus, the engineering mechanics sequence is critical to the engineering curriculum. Not only is this sequence needed in itself, but courses in engineering mechanics also serve to solidify the student's understanding of other important subjects, including applied mathematics, physics, and graphics. In addition, these courses serve as excellent settings in which to strengthen problem-solving abilities.

PHILOSOPHY

The primary purpose of the study of engineering mechanics is to develop the capacity to predict the effects of force and motion while carrying out the creative design functions of engineering. This capacity requires more than a mere knowledge of the physical and mathematical principles of mechanics; also required is the ability to visualize physical configurations in terms of real materials, actual constraints, and the practical limitations which govern the behavior of machines and structures. One of the primary objectives in a mechanics course is to help the student develop this ability to visualize, which is so vital to problem formulation. Indeed, the construction of a meaningful mathematical model is often a more important experience than its solution. Maximum progress is made when the principles and their limitations are learned together within the context of engineering application.

There is a frequent tendency in the presentation of mechanics to use problems mainly as a vehicle to illustrate theory rather than to develop theory for the purpose of solving problems. When the first view is allowed to predominate, problems tend to become overly idealized and unrelated to engineering with the result that the exercise becomes dull, academic,

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and uninteresting. This approach deprives the student of valuable experience in formulating problems and thus of discovering the need for and meaning of theory. The second view provides by far the stronger motive for learning theory and leads to a better balance between theory and application. The crucial role played by interest and purpose in providing the strongest possible motive for learning cannot be overemphasized.

Furthermore, as mechanics educators, we should stress the understanding that, at best, theory can only approximate the real world of mechanics rather than the view that the real world approximates the theory. This difference in philosophy is indeed basic and distinguishes the *engineering* of mechanics from the *science* of mechanics.

Over the past several decades, several unfortunate tendencies have occurred in engineering education. First, emphasis on the geometric and physical meanings of prerequisite mathematics appears to have diminished. Second, there has been a significant reduction and even elimination of instruction in graphics, which in the past enhanced the visualization and representation of mechanics problems. Third, in advancing the mathematical level of our treatment of mechanics, there has been a tendency to allow the notational manipulation of vector operations to mask or replace geometric visualization. Mechanics is inherently a subject which depends on geometric and physical perception, and we should increase our efforts to develop this ability.

A special note on the use of computers is in order. The experience of formulating problems, where reason and judgment are developed, is vastly more important for the student than is the manipulative exercise in carrying out the solution. For this reason, computer usage must be carefully controlled. At present, constructing free-body diagrams and formulating governing equations are best done with pencil and paper. On the other hand, there are instances in which the *solution* to the governing equations can best be carried out and displayed using the computer. Computer-oriented problems should be genuine in the sense that there is a condition of design or criticality to be found, rather than “makework” problems in which some parameter is varied for no apparent reason other than to force artificial use of the computer. These thoughts have been kept in mind during the design of the computer-oriented problems in the Sixth Edition. To conserve adequate time for problem formulation, it is suggested that the student be assigned only a limited number of the computer-oriented problems.

As with previous editions, this Sixth Edition of *Engineering Mechanics* is written with the foregoing philosophy in mind. It is intended primarily for the first engineering course in mechanics, generally taught in the second year of study. *Engineering Mechanics* is written in a style which is both concise and friendly. The major emphasis is on basic principles and methods rather than on a multitude of special cases. Strong effort has been made to show both the cohesiveness of the relatively few fundamental ideas and the great variety of problems which these few ideas will solve.

PEDAGOGICAL FEATURES

The basic structure of this textbook consists of an article which rigorously treats the particular subject matter at hand, followed by one or more Sample Problems, followed by a group of Problems. There is a Chapter Review at the end of each chapter which summarizes the main points in that chapter, followed by a Review Problem set.

Problems

The 86 Sample Problems appear on specially colored pages by themselves. The solutions to typical statics problems are presented in detail. In addition, explanatory and cautionary notes (Helpful Hints) in blue type are number-keyed to the main presentation.

There are 1020 homework exercises, of which approximately 50 percent are new to the Sixth Edition. The problem sets are divided into *Introductory Problems* and *Representative Problems*. The first section consists of simple, uncomplicated problems designed to help students gain confidence with the new topic, while most of the problems in the second section are of average difficulty and length. The problems are generally arranged in order of increasing difficulty. More difficult exercises appear near the end of the *Representative Problems* and are marked with the symbol ►. *Computer-Oriented Problems*, marked with an asterisk, appear in a special section at the conclusion of the *Review Problems* at the end of each chapter. The answers to all odd-numbered problems and to all difficult problems have been provided.

SI units are used throughout the book, except in a limited number of introductory areas in which U.S. units are mentioned for purposes of completeness and contrast with SI units.

A notable feature of the Sixth Edition, as with all previous editions, is the wealth of interesting and important problems which apply to engineering design. Whether directly identified as such or not, virtually all of the problems deal with principles and procedures inherent in the design and analysis of engineering structures and mechanical systems.

Illustrations

In order to bring the greatest possible degree of realism and clarity to the illustrations, this textbook series continues to be produced in full color. It is important to note that color is used consistently for the identification of certain quantities:

- *red* for forces and moments,
- *green* for velocity and acceleration arrows,
- *orange dashes* for selected trajectories of moving points.

Subdued colors are used for those parts of an illustration which are not central to the problem at hand. Whenever possible, mechanisms or objects which commonly have a certain color will be portrayed in that color. All of the fundamental elements of technical illustration which have been an essential part of this *Engineering Mechanics* series of textbooks have been retained. The author wishes to restate the conviction that a high standard of illustration is critical to any written work in the field of mechanics.

Features New to this Edition

While retaining the hallmark features of all previous editions, we have incorporated these improvements:

- All theory portions have been reexamined in order to maximize rigor, clarity, readability, and level of friendliness.
- Key Concepts areas within the theory presentation have been specially marked and highlighted.
- The Chapter Reviews are highlighted and feature itemized summaries.
- Approximately 50 percent of the homework problems are new to this Sixth Edition. All new problems have been independently solved in order to ensure a high degree of accuracy.
- New Sample Problems have been added, including ones with computer-oriented solutions.

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- All Sample Problems are printed on specially colored pages for quick identification.
- Within-the-chapter photographs have been added in order to provide additional connection to actual situations in which statics has played a major role.

ORGANIZATION

In Chapter 1, the fundamental concepts necessary for the study of mechanics are established.

In Chapter 2, the properties of forces, moments, couples, and resultants are developed so that the student may proceed directly to the equilibrium of nonconcurrent force systems in Chapter 3 without unnecessarily belaboring the relatively trivial problem of the equilibrium of concurrent forces acting on a particle.

In both Chapters 2 and 3, analysis of two-dimensional problems is presented in Section A before three-dimensional problems are treated in Section B. With this arrangement, the instructor may cover all of Chapter 2 before beginning Chapter 3 on equilibrium, or the instructor may cover the two chapters in the order 2A, 3A, 2B, 3B. The latter order treats force systems and equilibrium in two dimensions and then treats these topics in three dimensions.

Application of equilibrium principles to simple trusses and to frames and machines is presented in Chapter 4 with primary attention given to two-dimensional systems. A sufficient number of three-dimensional examples are included, however, to enable students to exercise more general vector tools of analysis.

The concepts and categories of distributed forces are introduced at the beginning of Chapter 5, with the balance of the chapter divided into two main sections. Section A treats centroids and mass centers; detailed examples are presented to help students master early applications of calculus to physical and geometrical problems. Section B includes the special topics of beams, flexible cables, and fluid forces, which may be omitted without loss of continuity of basic concepts.

Chapter 6 on friction is divided into Section A on the phenomenon of dry friction and Section B on selected machine applications. Although Section B may be omitted if time is limited, this material does provide a valuable experience for the student in dealing with both concentrated and distributed friction forces.

Chapter 7 presents a consolidated introduction to virtual work with applications limited to single-degree-of-freedom systems. Special emphasis is placed on the advantage of the virtual-work and energy method for interconnected systems and stability determination. Virtual work provides an excellent opportunity to convince the student of the power of mathematical analysis in mechanics.

Moments and products of inertia of areas are presented in Appendix A. This topic helps to bridge the subjects of statics and solid mechanics. Appendix C contains a summary review of selected topics of elementary mathematics as well as several numerical techniques which the student should be prepared to use in computer-solved problems. Useful tables of physical constants, centroids, and moments of inertia are contained in Appendix D.

SUPPLEMENTS

The following items have been prepared to complement this textbook:

Instructor's Manual

Prepared by the authors and independently checked, fully worked solutions to all problems in the text are available to faculty by contacting their local Wiley representative.

Instructor Lecture Resources

(Available on the text website at www.wiley.com/college/meriam):

WileyPlus: A complete online learning system to help prepare and present lectures, assign and manage homework, keep track of student progress, and customize your course content and delivery. See the description in front of the book for more information, and the website for a demonstration. Talk to your Wiley representative for details on setting up your WileyPlus course.

Lecture software specifically designed to aid the lecturer, especially in larger classrooms. Written by the author and incorporating figures from the textbooks, this software is based on the Macromedia Flash[®] platform. Major use of animation, concise review of the theory, and numerous sample problems make this tool extremely useful for student self-review of the material.

Web-based simulations of representative applications in statics that allow for “what-if” analysis. Developed by Richard Stanley at Kettering University, these simulations allow an instructor to explore a problem from the text by changing variables and seeing the new results develop both visually and numerically. Available from the book website and as part of the WileyPlus package.

All **figures** in the text are available in electronic format for use in creating lecture presentations.

All **Sample Problems** are available as electronic files for display and discussion in the classroom.

Transparencies for over 40 solved problems, similar to those in the text, available in .pdf format for use in lecture or for self-study by students.

Extension sample problems build on sample problems from the text and show how computational tools can be used to investigate a variety of “what-if” scenarios. Available to both faculty and students, these were developed by Brian Harper at Ohio State University.

Solving Mechanics Problems with . . .

A series of booklets introduces the use of computational software in the solution of mechanics problems. Developed by Brian Harper at Ohio State University, the booklets are available for Matlab, MathCAD, and Maple. Please contact your local Wiley representative for more information, or visit the book website at www.wiley.com/college/meriam.

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Special recognition is due Dr. A. L. Hale, formerly of Bell Telephone Laboratories, for his continuing contribution in the form of invaluable suggestions and accurate checking of the manuscript. Dr. Hale has rendered similar service for all previous versions of this entire series of mechanics books, dating back to the 1950s. He reviews all aspects of the books, including all old and new text and figures. Dr. Hale carries out an independent solution to each new homework exercise and provides the author with suggestions and needed corrections to the solutions which appear in the *Instructor's Manual*. Dr. Hale is well known for being extremely accurate in his work, and his fine knowledge of the English language is a great asset which aids every user of this textbook. In addition to his normal contributions, Dr. Hale has been the prime proofreader for the Sixth Edition.

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I would like to thank the faculty members of the Department of Engineering Science and Mechanics at VPI&SU who regularly offer constructive suggestions. These include Scott L. Hendricks, Saad A. Ragab, Norman E. Dowling, Michael W. Hyer, and J. Wallace Grant. The contributions to the previous Fifth Edition by William J. Palm, III, of the University of Rhode Island are again gratefully acknowledged. In addition, my assistant of thirty years, Vanessa McCoy, is recognized for her long-term contribution to this textbook project.

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Finally, I wish to state the extremely significant contribution of my family. In addition to providing patience and support for this project, my wife Dale has managed the preparation of the manuscript for the Sixth Edition and has been a key individual in checking all stages of the proof. In addition, my son David has contributed problem ideas, illustrations, and solutions to a number of the problems.

I am extremely pleased to participate in extending the time duration of this textbook series well past the fifty-year mark. In the interest of providing you with the best possible educational materials over future years, I encourage and welcome all comments and suggestions. Please address your comments to kraige@vt.edu.

L. Glenn Kraige

Blacksburg, Virginia

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Structures which support large forces must be designed with the principles of mechanics foremost in mind. In this view of Pittsburgh, one can see a variety of such structures.

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INTRODUCTION
TO STATICS

CHAPTER OUTLINE

- 1/1 Mechanics
- 1/2 Basic Concepts
- 1/3 Scalars and Vectors
- 1/4 Newton's Laws
- 1/5 Units
- 1/6 Law of Gravitation
- 1/7 Accuracy, Limits, and Approximations
- 1/8 Problem Solving in Statics
- 1/9 Chapter Review

1/1 MECHANICS

Mechanics is the physical science which deals with the effects of forces on objects. No other subject plays a greater role in engineering analysis than mechanics. Although the principles of mechanics are few, they have wide application in engineering. The principles of mechanics are central to research and development in the fields of vibrations, stability and strength of structures and machines, robotics, rocket and spacecraft design, automatic control, engine performance, fluid flow, electrical machines and apparatus, and molecular, atomic, and subatomic behavior. A thorough understanding of this subject is an essential prerequisite for work in these and many other fields.

Mechanics is the oldest of the physical sciences. The early history of this subject is synonymous with the very beginnings of engineering. The earliest recorded writings in mechanics are those of Archimedes (287–212 B.C.) on the principle of the lever and the principle of buoyancy. Substantial progress came later with the formulation of the laws of vector combination of forces by Stevinus (1548–1620), who also formulated most of the principles of statics. The first investigation of a dynamics problem is credited to Galileo (1564–1642) for his experiments with falling stones. The accurate formulation of the laws of motion, as well as the law of gravitation, was made by Newton (1642–1727), who also conceived the idea of the infinitesimal in mathematical analysis. Substantial contributions to the development of mechanics were also made by da Vinci, Varignon, Euler, D'Alembert, Lagrange, Laplace, and others.

In this book we will be concerned with both the development of the principles of mechanics and their application. The principles of mechanics as a science are rigorously expressed by mathematics, and thus



Sir Isaac Newton

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4 Chapter 1 Introduction to Statics

mathematics plays an important role in the application of these principles to the solution of practical problems.

The subject of mechanics is logically divided into two parts: **statics**, which concerns the equilibrium of bodies under action of forces, and **dynamics**, which concerns the motion of bodies. *Engineering Mechanics* is divided into these two parts, *Vol. 1 Statics* and *Vol. 2 Dynamics*.

1/2 BASIC CONCEPTS

The following concepts and definitions are basic to the study of mechanics, and they should be understood at the outset.

Space is the geometric region occupied by bodies whose positions are described by linear and angular measurements relative to a coordinate system. For three-dimensional problems, three independent coordinates are needed. For two-dimensional problems, only two coordinates are required.

Time is the measure of the succession of events and is a basic quantity in dynamics. Time is not directly involved in the analysis of statics problems.

Mass is a measure of the inertia of a body, which is its resistance to a change of velocity. Mass can also be thought of as the quantity of matter in a body. The mass of a body affects the gravitational attraction force between it and other bodies. This force appears in many applications in statics.

Force is the action of one body on another. A force tends to move a body in the direction of its action. The action of a force is characterized by its *magnitude*, by the *direction* of its action, and by its *point of application*. Thus force is a vector quantity, and its properties are discussed in detail in Chapter 2.

A **particle** is a body of negligible dimensions. In the mathematical sense, a particle is a body whose dimensions are considered to be near zero so that we may analyze it as a mass concentrated at a point. We often choose a particle as a differential element of a body. We may treat a body as a particle when its dimensions are irrelevant to the description of its position or the action of forces applied to it.

Rigid body. A body is considered rigid when the change in distance between any two of its points is negligible for the purpose at hand. For instance, the calculation of the tension in the cable which supports the boom of a mobile crane under load is essentially unaffected by the small internal deformations in the structural members of the boom. For the purpose, then, of determining the external forces which act on the boom, we may treat it as a rigid body. Statics deals primarily with the calculation of external forces which act on rigid bodies in equilibrium. Determination of the internal deformations belongs to the study of the mechanics of deformable bodies, which normally follows statics in the curriculum.

1/3 SCALARS AND VECTORS

We use two kinds of quantities in mechanics—scalars and vectors. *Scalar quantities* are those with which only a magnitude is associated. Examples of scalar quantities are time, volume, density, speed, energy,

and mass. *Vector quantities*, on the other hand, possess direction as well as magnitude, and must obey the parallelogram law of addition as described later in this article. Examples of vector quantities are displacement, velocity, acceleration, force, moment, and momentum. Speed is a scalar. It is the magnitude of velocity, which is a vector. Thus velocity is specified by a direction as well as a speed.

Vectors representing physical quantities can be classified as free, sliding, or fixed.

A **free vector** is one whose action is not confined to or associated with a unique line in space. For example, if a body moves without rotation, then the movement or displacement of any point in the body may be taken as a vector. This vector describes equally well the direction and magnitude of the displacement of every point in the body. Thus, we may represent the displacement of such a body by a free vector.

A **sliding vector** has a unique line of action in space but not a unique point of application. For example, when an external force acts on a rigid body, the force can be applied at any point along its line of action without changing its effect on the body as a whole,* and thus it is a sliding vector.

A **fixed vector** is one for which a unique point of application is specified. The action of a force on a deformable or nonrigid body must be specified by a fixed vector at the point of application of the force. In this instance the forces and deformations within the body depend on the point of application of the force, as well as on its magnitude and line of action.

Conventions for Equations and Diagrams

A vector quantity \mathbf{V} is represented by a line segment, Fig. 1/1, having the direction of the vector and having an arrowhead to indicate the sense. The length of the directed line segment represents to some convenient scale the magnitude $|\mathbf{V}|$ of the vector, which is printed with lightface italic type V . For example, we may choose a scale such that an arrow one centimeter long represents a force of twenty newtons.

In scalar equations, and frequently on diagrams where only the magnitude of a vector is labeled, the symbol will appear in lightface italic type. Boldface type is used for vector quantities whenever the directional aspect of the vector is a part of its mathematical representation. When writing vector equations, *always* be certain to preserve the mathematical distinction between vectors and scalars. In handwritten work, use a distinguishing mark for each vector quantity, such as an underline, \underline{V} , or an arrow over the symbol, \vec{V} , to take the place of boldface type in print.

Working with Vectors

The direction of the vector \mathbf{V} may be measured by an angle θ from some known reference direction as shown in Fig. 1/1. The negative of \mathbf{V} is a vector $-\mathbf{V}$ having the same magnitude as \mathbf{V} but directed in the sense opposite to \mathbf{V} , as shown in Fig. 1/1.

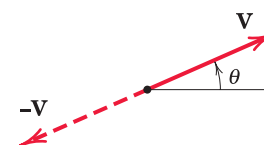


Figure 1/1

*This is the *principle of transmissibility*, which is discussed in Art. 2/2.

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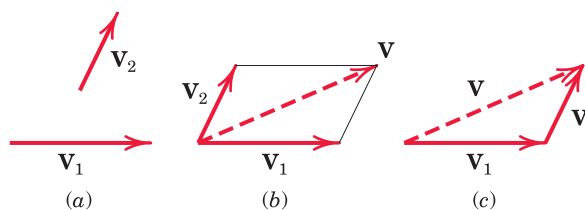


Figure 1/2

Vectors must obey the parallelogram law of combination. This law states that two vectors \mathbf{V}_1 and \mathbf{V}_2 , treated as free vectors, Fig. 1/2a, may be replaced by their equivalent vector \mathbf{V} , which is the diagonal of the parallelogram formed by \mathbf{V}_1 and \mathbf{V}_2 as its two sides, as shown in Fig. 1/2b. This combination is called the *vector sum*, and is represented by the vector equation

$$\mathbf{V} = \mathbf{V}_1 + \mathbf{V}_2$$

where the plus sign, when used with the vector quantities (in boldface type), means *vector* and not *scalar* addition. The scalar sum of the magnitudes of the two vectors is written in the usual way as $V_1 + V_2$. The geometry of the parallelogram shows that $V \neq V_1 + V_2$.

The two vectors \mathbf{V}_1 and \mathbf{V}_2 , again treated as free vectors, may also be added head-to-tail by the triangle law, as shown in Fig. 1/2c, to obtain the identical vector sum \mathbf{V} . We see from the diagram that the order of addition of the vectors does not affect their sum, so that $\mathbf{V}_1 + \mathbf{V}_2 = \mathbf{V}_2 + \mathbf{V}_1$.

The difference $\mathbf{V}_1 - \mathbf{V}_2$ between the two vectors is easily obtained by adding $-\mathbf{V}_2$ to \mathbf{V}_1 as shown in Fig. 1/3, where either the triangle or parallelogram procedure may be used. The difference \mathbf{V}' between the two vectors is expressed by the vector equation

$$\mathbf{V}' = \mathbf{V}_1 - \mathbf{V}_2$$

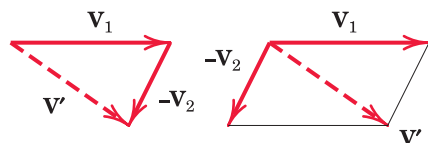


Figure 1/3

where the minus sign denotes *vector subtraction*.

Any two or more vectors whose sum equals a certain vector \mathbf{V} are said to be the *components* of that vector. Thus, the vectors \mathbf{V}_1 and \mathbf{V}_2 in Fig. 1/4a are the components of \mathbf{V} in the directions 1 and 2, respectively. It is usually most convenient to deal with vector components which are mutually perpendicular; these are called *rectangular components*. The

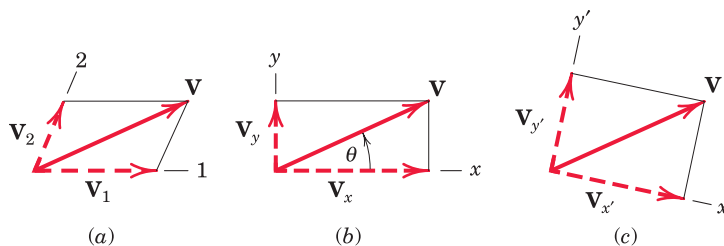


Figure 1/4

vectors \mathbf{V}_x and \mathbf{V}_y in Fig. 1/4b are the x - and y -components, respectively, of \mathbf{V} . Likewise, in Fig. 1/4c, $\mathbf{V}_{x'}$ and $\mathbf{V}_{y'}$ are the x' - and y' -components of \mathbf{V} . When expressed in rectangular components, the direction of the vector with respect to, say, the x -axis is clearly specified by the angle θ , where

$$\theta = \tan^{-1} \frac{V_y}{V_x}$$

A vector \mathbf{V} may be expressed mathematically by multiplying its magnitude V by a vector \mathbf{n} whose magnitude is one and whose direction coincides with that of \mathbf{V} . The vector \mathbf{n} is called a *unit vector*. Thus,

$$\mathbf{V} = V\mathbf{n}$$

In this way both the magnitude and direction of the vector are conveniently contained in one mathematical expression. In many problems, particularly three-dimensional ones, it is convenient to express the rectangular components of \mathbf{V} , Fig. 1/5, in terms of unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} , which are vectors in the x -, y -, and z -directions, respectively, with unit magnitudes. Because the vector \mathbf{V} is the vector sum of the components in the x -, y -, and z -directions, we can express \mathbf{V} as follows:

$$\mathbf{V} = V_x\mathbf{i} + V_y\mathbf{j} + V_z\mathbf{k}$$

We now make use of the *direction cosines* l , m , and n of \mathbf{V} , which are defined by

$$l = \cos \theta_x \quad m = \cos \theta_y \quad n = \cos \theta_z$$

Thus, we may write the magnitudes of the components of \mathbf{V} as

$$V_x = lV \quad V_y = mV \quad V_z = nV$$

where, from the Pythagorean theorem,

$$V^2 = V_x^2 + V_y^2 + V_z^2$$

Note that this relation implies that $l^2 + m^2 + n^2 = 1$.

1/4 NEWTON'S LAWS

Sir Isaac Newton was the first to state correctly the basic laws governing the motion of a particle and to demonstrate their validity.* Slightly reworded with modern terminology, these laws are:

Law 1. A particle remains at rest or continues to move with *uniform velocity* (in a straight line with a constant speed) if there is no unbalanced force acting on it.

*Newton's original formulations may be found in the translation of his *Principia* (1687) revised by F. Cajori, University of California Press, 1934.

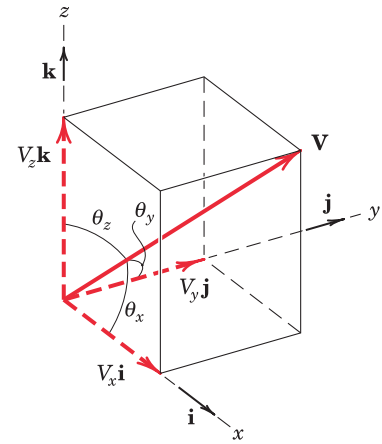


Figure 1/5

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Law II. The acceleration of a particle is proportional to the vector sum of forces acting on it, and is in the direction of this vector sum.

Law III. The forces of action and reaction between interacting bodies are equal in magnitude, opposite in direction, and *collinear* (they lie on the same line).

The correctness of these laws has been verified by innumerable accurate physical measurements. Newton's second law forms the basis for most of the analysis in dynamics. As applied to a particle of mass m , it may be stated as

$$\mathbf{F} = m\mathbf{a} \quad (1/1)$$

where \mathbf{F} is the vector sum of forces acting on the particle and \mathbf{a} is the resulting acceleration. This equation is a *vector* equation because the direction of \mathbf{F} must agree with the direction of \mathbf{a} , and the magnitudes of \mathbf{F} and $m\mathbf{a}$ must be equal.

Newton's first law contains the principle of the equilibrium of forces, which is the main topic of concern in statics. This law is actually a consequence of the second law, since there is no acceleration when the force is zero, and the particle either is at rest or is moving with a uniform velocity. The first law adds nothing new to the description of motion but is included here because it was part of Newton's classical statements.

The third law is basic to our understanding of force. It states that forces always occur in pairs of equal and opposite forces. Thus, the downward force exerted on the desk by the pencil is accompanied by an upward force of equal magnitude exerted on the pencil by the desk. This principle holds for all forces, variable or constant, regardless of their source, and holds at every instant of time during which the forces are applied. Lack of careful attention to this basic law is the cause of frequent error by the beginner.

In the analysis of bodies under the action of forces, it is absolutely necessary to be clear about which force of each action-reaction pair is being considered. It is necessary first of all to *isolate* the body under consideration and then to consider only the one force of the pair which acts *on* the body in question.

1/5 UNITS

In mechanics we use four fundamental quantities called *dimensions*. These are length, mass, force, and time. The units used to measure these quantities cannot all be chosen independently because they must be consistent with Newton's second law, Eq. 1/1. Although there are a number of different systems of units, only the two systems most commonly used in science and technology will be used in this text. The four fundamental dimensions and their units and symbols in the two systems are summarized in the following table.

QUANTITY	DIMENSIONAL SYMBOL	SI UNITS		U.S. CUSTOMARY UNITS	
		UNIT	SYMBOL	UNIT	SYMBOL
Mass	M	Base units	kilogram	slug	—
Length	L		meter	foot	ft
Time	T		second	second	sec
Force	F		newton	pound	lb

SI Units

The International System of Units, abbreviated SI (from the French, *Système International d'Unités*), is accepted in the United States and throughout the world, and is a modern version of the metric system. By international agreement, SI units will in time replace other systems. As shown in the table, in SI, the units kilogram (kg) for mass, meter (m) for length, and second (s) for time are selected as the base units, and the newton (N) for force is derived from the preceding three by Eq. 1/1. Thus, force (N) = mass (kg) \times acceleration (m/s^2) or

$$N = \text{kg} \cdot \text{m/s}^2$$

Thus, 1 newton is the force required to give a mass of 1 kg an acceleration of 1 m/s^2 .

Consider a body of mass m which is allowed to fall freely near the surface of the earth. With only the force of gravitation acting on the body, it falls with an acceleration g toward the center of the earth. This gravitational force is the *weight* W of the body, and is found from Eq. 1/1:

$$W (\text{N}) = m (\text{kg}) \times g (\text{m/s}^2)$$

U.S. Customary Units

The U.S. customary, or British system of units, also called the foot-pound-second (FPS) system, has been the common system in business and industry in English-speaking countries. Although this system will in time be replaced by SI units, for many more years engineers must be able to work with both SI units and FPS units.

As shown in the table, in the U.S. or FPS system, the units of feet (ft) for length, seconds (sec) for time, and pounds (lb) for force are selected as base units, and the slug for mass is derived from Eq. 1/1. Thus, force (lb) = mass (slugs) \times acceleration (ft/sec^2), or

$$\text{slug} = \frac{\text{lb} \cdot \text{sec}^2}{\text{ft}}$$

Therefore, 1 slug is the mass which is given an acceleration of 1 ft/sec^2 when acted on by a force of 1 lb. If W is the gravitational force or weight and g is the acceleration due to gravity, Eq. 1/1 gives

$$m(\text{slugs}) = \frac{W (\text{lb})}{g (\text{ft/sec}^2)}$$

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Note that seconds is abbreviated as s in SI units, and as sec in FPS units.

In U.S. units the pound is also used on occasion as a unit of mass, especially to specify thermal properties of liquids and gases. When distinction between the two units is necessary, the force unit is frequently written as lbf and the mass unit as lbm. In this book we use almost exclusively the force unit, which is written simply as lb. Other common units of force in the U.S. system are the *kilopound* (kip), which equals 1000 lb, and the *ton*, which equals 2000 lb.

The International System of Units (SI) is termed an *absolute* system because the measurement of the base quantity mass is independent of its environment. On the other hand, the U.S. system (FPS) is termed a *gravitational* system because its base quantity force is defined as the gravitational attraction (weight) acting on a standard mass under specified conditions (sea level and 45° latitude). A standard pound is also the force required to give a one-pound mass an acceleration of 32.1740 ft/sec^2 .

In SI units the kilogram is used *exclusively* as a unit of mass—*never* force. In the MKS (meter, kilogram, second) gravitational system, which has been used for many years in non-English-speaking countries, the kilogram, like the pound, has been used both as a unit of force and as a unit of mass.

Primary Standards

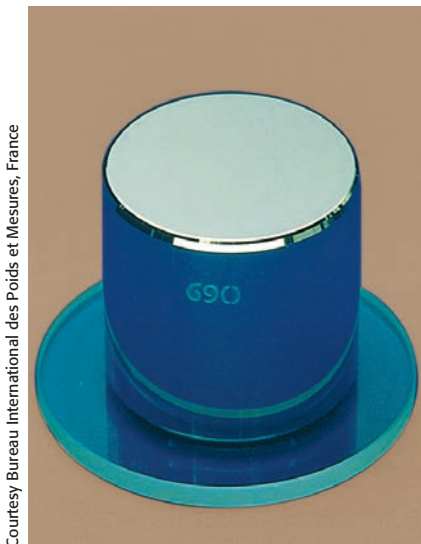
Primary standards for the measurements of mass, length, and time have been established by international agreement and are as follows:

Mass. The kilogram is defined as the mass of a specific platinum–iridium cylinder which is kept at the International Bureau of Weights and Measures near Paris, France. An accurate copy of this cylinder is kept in the United States at the National Institute of Standards and Technology (NIST), formerly the National Bureau of Standards, and serves as the standard of mass for the United States.

Length. The meter, originally defined as one ten-millionth of the distance from the pole to the equator along the meridian through Paris, was later defined as the length of a specific platinum–iridium bar kept at the International Bureau of Weights and Measures. The difficulty of accessing the bar and reproducing accurate measurements prompted the adoption of a more accurate and reproducible standard of length for the meter, which is now defined as 1 650 763.73 wavelengths of a specific radiation of the krypton-86 atom.

Time. The second was originally defined as the fraction $1/(86\,400)$ of the mean solar day. However, irregularities in the earth's rotation led to difficulties with this definition, and a more accurate and reproducible standard has been adopted. The second is now defined as the duration of 9 192 631 770 periods of the radiation of a specific state of the cesium-133 atom.

For most engineering work, and for our purpose in studying mechanics, the accuracy of these standards is considerably beyond our



The standard kilogram

needs. The standard value for gravitational acceleration g is its value at sea level and at a 45° latitude. In the two systems these values are

$$\text{SI units} \quad g = 9.806\,65 \text{ m/s}^2$$

$$\text{U.S. units} \quad g = 32.1740 \text{ ft/sec}^2$$

The approximate values of 9.81 m/s^2 and 32.2 ft/sec^2 , respectively, are sufficiently accurate for the vast majority of engineering calculations.

Unit Conversions

The characteristics of SI units are shown inside the front cover of this book, along with the numerical conversions between U.S. customary and SI units. In addition, charts giving the approximate conversions between selected quantities in the two systems appear inside the back cover for convenient reference. Although these charts are useful for obtaining a feel for the relative size of SI and U.S. units, in time engineers will find it essential to think directly in terms of SI units without converting from U.S. units. In statics we are primarily concerned with the units of length and force, with mass needed only when we compute gravitational force, as explained previously.

Figure 1/6 depicts examples of force, mass, and length in the two systems of units, to aid in visualizing their relative magnitudes.

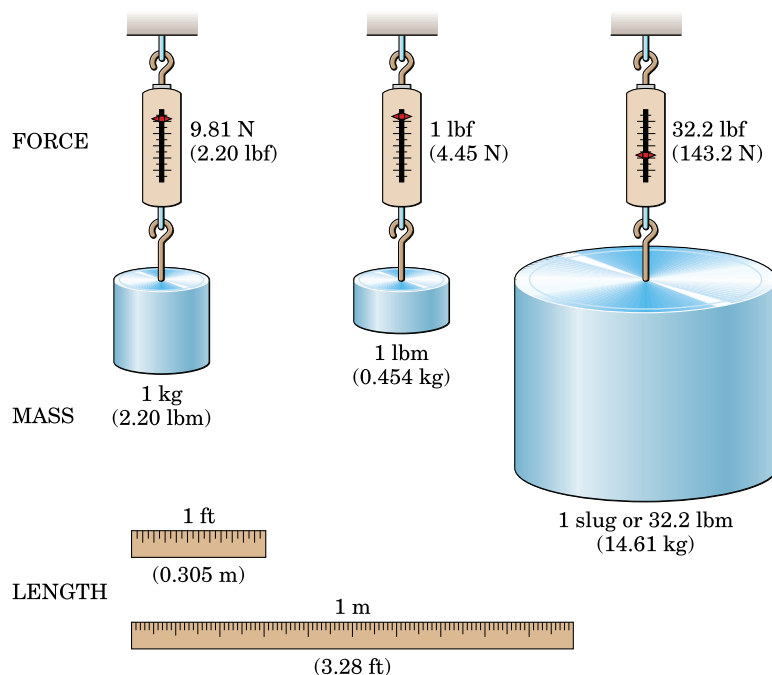


Figure 1/6

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1/6 LAW OF GRAVITATION

In statics as well as dynamics we often need to compute the weight of a body, which is the gravitational force acting on it. This computation depends on the *law of gravitation*, which was also formulated by Newton. The law of gravitation is expressed by the equation

$$F = G \frac{m_1 m_2}{r^2} \quad (1/2)$$

where F = the mutual force of attraction between two particles

G = a universal constant known as the *constant of gravitation*

m_1, m_2 = the masses of the two particles

r = the distance between the centers of the particles

The mutual forces F obey the law of action and reaction, since they are equal and opposite and are directed along the line joining the centers of the particles, as shown in Fig. 1/7. By experiment the gravitational constant is found to be $G = 6.673(10^{-11}) \text{ m}^3/(\text{kg} \cdot \text{s}^2)$.

Gravitational Attraction of the Earth

Gravitational forces exist between every pair of bodies. On the surface of the earth the only gravitational force of appreciable magnitude is the force due to the attraction of the earth. For example, each of two iron spheres 100 mm in diameter is attracted to the earth with a gravitational force of 37.1 N, which is its weight. On the other hand, the force of mutual attraction between the spheres if they are just touching is 0.000 000 095 1 N. This force is clearly negligible compared with the earth's attraction of 37.1 N. Consequently the gravitational attraction of the earth is the only gravitational force we need to consider for most engineering applications on the earth's surface.

The gravitational attraction of the earth on a body (its weight) exists whether the body is at rest or in motion. Because this attraction is a force, the weight of a body should be expressed in newtons (N) in SI units and in pounds (lb) in U.S. customary units. Unfortunately in common practice the mass unit kilogram (kg) has been frequently used as a measure of weight. This usage should disappear in time as SI units become more widely used, because in SI units the kilogram is used exclusively for mass and the newton is used for force, including weight.

For a body of mass m near the surface of the earth, the gravitational attraction F on the body is specified by Eq. 1/2. We usually denote the

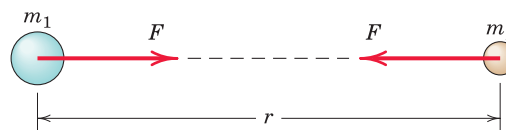


Figure 1/7

magnitude of this gravitational force or weight with the symbol W . Because the body falls with an acceleration g , Eq. 1/1 gives

$$W = mg \quad (1/3)$$

The weight W will be in newtons (N) when the mass m is in kilograms (kg) and the acceleration of gravity g is in meters per second squared (m/s^2). In U.S. customary units, the weight W will be in pounds (lb) when m is in slugs and g is in feet per second squared. The standard values for g of 9.81 m/s^2 and 32.2 ft/sec^2 will be sufficiently accurate for our calculations in statics.

The true weight (gravitational attraction) and the apparent weight (as measured by a spring scale) are slightly different. The difference, which is due to the rotation of the earth, is quite small and will be neglected. This effect will be discussed in *Vol. 2 Dynamics*.

1/7 ACCURACY, LIMITS, AND APPROXIMATIONS

The number of significant figures in an answer should be no greater than the number of figures justified by the accuracy of the given data. For example, suppose the 24-mm side of a square bar was measured to the nearest millimeter, so we know the side length to two significant figures. Squaring the side length gives an area of 576 mm^2 . However, according to our rule, we should write the area as 580 mm^2 , using only two significant figures.

When calculations involve small differences in large quantities, greater accuracy in the data is required to achieve a given accuracy in the results. Thus, for example, it is necessary to know the numbers 4.2503 and 4.2391 to an accuracy of five significant figures to express their difference 0.0112 to three-figure accuracy. It is often difficult in lengthy computations to know at the outset how many significant figures are needed in the original data to ensure a certain accuracy in the answer. Accuracy to three significant figures is considered satisfactory for most engineering calculations.

In this text, answers will generally be shown to three significant figures unless the answer begins with the digit 1, in which case the answer will be shown to four significant figures. For purposes of calculation, consider all data given in this book to be exact.

Differentials

The *order* of differential quantities frequently causes misunderstanding in the derivation of equations. Higher-order differentials may always be neglected compared with lower-order differentials when the mathematical limit is approached. For example, the element of volume ΔV of a right circular cone of altitude h and base radius r may be taken to be a circular slice a distance x from the vertex and of thickness Δx . The expression for the volume of the element is

$$\Delta V = \frac{\pi r^2}{h^2} [x^2 \Delta x + x(\Delta x)^2 + \frac{1}{3}(\Delta x)^3]$$

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Note that, when passing to the limit in going from ΔV to dV and from Δx to dx , the terms containing $(\Delta x)^2$ and $(\Delta x)^3$ drop out, leaving merely

$$dV = \frac{\pi r^2}{h^2} x^2 dx$$

which gives an exact expression when integrated.

Small-Angle Approximations

When dealing with small angles, we can usually make use of simplifying approximations. Consider the right triangle of Fig. 1/8 where the angle θ , expressed in radians, is relatively small. If the hypotenuse is unity, we see from the geometry of the figure that the arc length $1 \times \theta$ and $\sin \theta$ are very nearly the same. Also $\cos \theta$ is close to unity. Furthermore, $\sin \theta$ and $\tan \theta$ have almost the same values. Thus, for small angles we may write

$$\sin \theta \cong \tan \theta \cong \theta \quad \cos \theta \cong 1$$

provided that the angles are expressed in radians. These approximations may be obtained by retaining only the first terms in the series expansions for these three functions. As an example of these approximations, for an angle of 1°

$$\begin{aligned} 1^\circ &= 0.017453 \text{ rad} & \tan 1^\circ &= 0.017455 \\ \sin 1^\circ &= 0.017452 & \cos 1^\circ &= 0.999848 \end{aligned}$$

If a more accurate approximation is desired, the first two terms may be retained, and they are

$$\sin \theta \cong \theta - \theta^3/6 \quad \tan \theta \cong \theta + \theta^3/3 \quad \cos \theta \cong 1 - \theta^2/2$$

where the angles must be expressed in radians. (To convert degrees to radians, multiply the angle in degrees by $\pi/180^\circ$.) The error in replacing the sine by the angle for 1° (0.0175 rad) is only 0.005 percent. For 5° (0.0873 rad) the error is 0.13 percent, and for 10° (0.1745 rad), the error is still only 0.51 percent. As the angle θ approaches zero, the following relations are true in the mathematical limit:

$$\sin d\theta = \tan d\theta = d\theta \quad \cos d\theta = 1$$

where the differential angle $d\theta$ must be expressed in radians.

1/8 PROBLEM SOLVING IN STATICS

We study statics to obtain a quantitative description of forces which act on engineering structures in equilibrium. Mathematics establishes the relations between the various quantities involved and enables us to predict effects from these relations. We use a dual thought process in

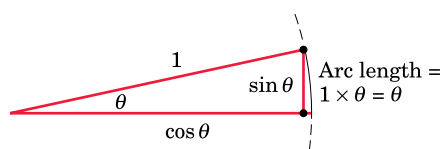


Figure 1/8

solving statics problems: We think about both the physical situation and the corresponding mathematical description. In the analysis of every problem, we make a transition between the physical and the mathematical. One of the most important goals for the student is to develop the ability to make this transition freely.

Making Appropriate Assumptions

We should recognize that the mathematical formulation of a physical problem represents an ideal description, or *model*, which approximates but never quite matches the actual physical situation. When we construct an idealized mathematical model for a given engineering problem, certain approximations will always be involved. Some of these approximations may be mathematical, whereas others will be physical.

For instance, it is often necessary to neglect small distances, angles, or forces compared with large distances, angles, or forces. Suppose a force is distributed over a small area of the body on which it acts. We may consider it to be a concentrated force if the dimensions of the area involved are small compared with other pertinent dimensions.

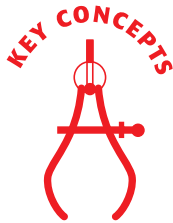
We may neglect the weight of a steel cable if the tension in the cable is many times greater than its total weight. However, if we must calculate the deflection or sag of a suspended cable under the action of its weight, we may not ignore the cable weight.

Thus, what we may assume depends on what information is desired and on the accuracy required. We must be constantly alert to the various assumptions called for in the formulation of real problems. The ability to understand and make use of the appropriate assumptions in the formulation and solution of engineering problems is certainly one of the most important characteristics of a successful engineer. One of the major aims of this book is to provide many opportunities to develop this ability through the formulation and analysis of many practical problems involving the principles of statics.

Using Graphics

Graphics is an important analytical tool for three reasons:

1. We use graphics to represent a physical system on paper with a sketch or diagram. Representing a problem geometrically helps us with its physical interpretation, especially when we must visualize three-dimensional problems.
2. We can often obtain a graphical solution to problems more easily than with a direct mathematical solution. Graphical solutions are both a practical way to obtain results, and an aid in our thought processes. Because graphics represents the physical situation and its mathematical expression simultaneously, graphics helps us make the transition between the two.
3. Charts or graphs are valuable aids for representing results in a form which is easy to understand.



Formulating Problems and Obtaining Solutions

In statics, as in all engineering problems, we need to use a precise and logical method for formulating problems and obtaining their solutions. We formulate each problem and develop its solution through the following sequence of steps.

1. Formulate the problem:
 - (a) State the given data.
 - (b) State the desired result.
 - (c) State your assumptions and approximations.
2. Develop the solution:
 - (a) Draw any diagrams you need to understand the relationships.
 - (b) State the governing principles to be applied to your solution.
 - (c) Make your calculations.
 - (d) Ensure that your calculations are consistent with the accuracy justified by the data.
 - (e) Be sure that you have used consistent units throughout your calculations.
 - (f) Ensure that your answers are reasonable in terms of magnitudes, directions, common sense, etc.
 - (g) Draw conclusions.

Keeping your work neat and orderly will help your thought process and enable others to understand your work. The discipline of doing orderly work will help you develop skill in formulation and analysis. Problems which seem complicated at first often become clear when you approach them with logic and discipline.

The Free-Body Diagram

The subject of statics is based on surprisingly few fundamental concepts and involves mainly the application of these basic relations to a variety of situations. In this application the *method* of analysis is all important. In solving a problem, it is essential that the laws which apply be carefully fixed in mind and that we apply these principles literally and exactly. In applying the principles of mechanics to analyze forces acting on a body, it is essential that we *isolate* the body in question from all other bodies so that a complete and accurate account of all forces acting on this body can be taken. This *isolation* should exist mentally and should be represented on paper. The diagram of such an isolated body with the representation of *all* external forces acting *on* it is called a *free-body diagram*.

The free-body-diagram method is the key to the understanding of mechanics. This is so because the *isolation* of a body is the tool by which

cause and *effect* are clearly separated, and by which our attention is clearly focused on the literal application of a principle of mechanics. The technique of drawing free-body diagrams is covered in Chapter 3, where they are first used.

Numerical Values versus Symbols

In applying the laws of statics, we may use numerical values to represent quantities, or we may use algebraic symbols, and leave the answer as a formula. When numerical values are used, the magnitude of each quantity expressed in its particular units is evident at each stage of the calculation. This is useful when we need to know the magnitude of each term.

The symbolic solution, however, has several advantages over the numerical solution. First, the use of symbols helps to focus our attention on the connection between the physical situation and its related mathematical description. Second, we can use a symbolic solution repeatedly for obtaining answers to the same type of problem, but having different units or numerical values. Third, a symbolic solution enables us to make a dimensional check at every step, which is more difficult to do when numerical values are used. In any equation representing a physical situation, the dimensions of every term on both sides of the equation must be the same. This property is called *dimensional homogeneity*.

Thus, facility with both numerical and symbolic forms of solution is essential.

Solution Methods

Solutions to the problems of statics may be obtained in one or more of the following ways.

1. Obtain mathematical solutions by hand, using either algebraic symbols or numerical values. We can solve most problems this way.
2. Obtain graphical solutions for certain problems.
3. Solve problems by computer. This is useful when a large number of equations must be solved, when a parameter variation must be studied, or when an intractable equation must be solved.

Many problems can be solved with two or more of these methods. The method utilized depends partly on the engineer's preference and partly on the type of problem to be solved. The choice of the most expedient method of solution is an important aspect of the experience to be gained from the problem work. There are a number of problems in *Vol. 1 Statics* which are designated as *Computer-Oriented Problems*. These problems appear at the end of the Review Problem sets and are selected to illustrate the type of problem for which solution by computer offers a distinct advantage.

18 Chapter 1 Introduction to Statics

1/9 CHAPTER REVIEW

This chapter has introduced the concepts, definitions, and units used in statics, and has given an overview of the procedure used to formulate and solve problems in statics. Now that you have finished this chapter, you should be able to do the following:

1. Express vectors in terms of unit vectors and perpendicular components, and perform vector addition and subtraction.
2. State Newton's laws of motion.
3. Perform calculations using SI and U.S. units, using appropriate accuracy.
4. Express the law of gravitation and calculate the weight of an object.
5. Apply simplifications based on differential and small-angle approximations.
6. Describe the methodology used to formulate and solve statics problems.

Sample Problem 1/1

Determine the weight in newtons of a car whose mass is 1400 kg. Convert the mass of the car to slugs and then determine its weight in pounds.

Solution. From relationship 1/3, we have

$$\textcircled{1} \quad W = mg = 1400(9.81) = 13\,730 \text{ N} \quad \text{Ans.}$$

From the table of conversion factors inside the front cover of the textbook, we see that 1 slug is equal to 14.594 kg. Thus, the mass of the car in slugs is

$$\textcircled{2} \quad m = 1400 \text{ kg} \left[\frac{1 \text{ slug}}{14.594 \text{ kg}} \right] = 95.9 \text{ slugs} \quad \text{Ans.}$$

Finally, its weight in pounds is

$$\textcircled{3} \quad W = mg = (95.9)(32.2) = 3090 \text{ lb} \quad \text{Ans.}$$

As another route to the last result, we can convert from kg to lbm. Again using the table inside the front cover, we have

$$m = 1400 \text{ kg} \left[\frac{1 \text{ lbm}}{0.45359 \text{ kg}} \right] = 3090 \text{ lbm}$$

The weight in pounds associated with the mass of 3090 lbm is 3090 lb, as calculated above. We recall that 1 lbm is the amount of mass which under standard conditions has a weight of 1 lb of force. We rarely refer to the U.S. mass unit lbm in this textbook series, but rather use the slug for mass. The sole use of slug, rather than the unnecessary use of two units for mass, will prove to be powerful and simple—especially in dynamics.

- $\textcircled{3}$ Note that we are using a previously calculated result (95.9 slugs). We must be sure that when a calculated number is needed in subsequent calculations, it is retained in the calculator to its full accuracy, (95.929834...) until it is needed. This may require storing it in a register upon its initial calculation and recalling it later. We must not merely punch 95.9 into our calculator and proceed to multiply by 32.2—this practice will result in loss of numerical accuracy. Some individuals like to place a small indication of the storage register used in the right margin of the work paper, directly beside the number stored.

Sample Problem 1/2

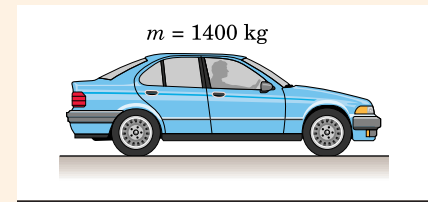
Use Newton's law of universal gravitation to calculate the weight of a 70-kg person standing on the surface of the earth. Then repeat the calculation by using $W = mg$ and compare your two results. Use Table D/2 as needed.

Solution. The two results are

$$\textcircled{1} \quad W = \frac{Gm_em}{R^2} = \frac{(6.673 \cdot 10^{-11})(5.976 \cdot 10^{24})(70)}{[6371 \cdot 10^3]^2} = 688 \text{ N} \quad \text{Ans.}$$

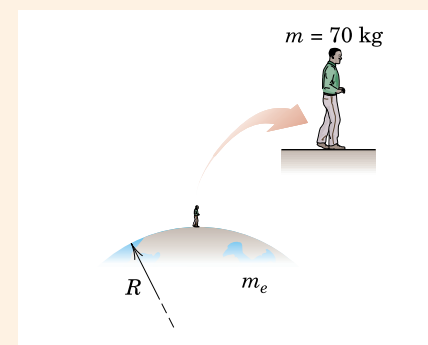
$$W = mg = 70(9.81) = 687 \text{ N} \quad \text{Ans.}$$

The discrepancy is due to the fact that Newton's universal gravitational law does not take into account the rotation of the earth. On the other hand, the value $g = 9.81 \text{ m/s}^2$ used in the second equation does account for the earth's rotation. Note that had we used the more accurate value $g = 9.80665 \text{ m/s}^2$ (which likewise accounts for the earth's rotation) in the second equation, the discrepancy would have been larger (686 N would have been the result).



Helpful Hints

- $\textcircled{1}$ Our calculator indicates a result of 13 734 N. Using the rules of significant-figure display used in this textbook, we round the written result to four significant figures, or 13 730 N. Had the number begun with any digit other than 1, we would have rounded to three significant figures.
- $\textcircled{2}$ A good practice with unit conversion is to multiply by a factor such as $\left[\frac{1 \text{ slug}}{14.594 \text{ kg}} \right]$, which has a value of 1, because the numerator and the denominator are equivalent. Make sure that cancellation of the units leaves the units desired; here the units of kg cancel, leaving the desired units of slug.



Helpful Hint

- $\textcircled{1}$ The effective distance between the mass centers of the two bodies involved is the radius of the earth.

20 Chapter 1 Introduction to Statics

Sample Problem 1/3

For the vectors \mathbf{V}_1 and \mathbf{V}_2 shown in the figure,

- determine the magnitude S of their vector sum $\mathbf{S} = \mathbf{V}_1 + \mathbf{V}_2$
- determine the angle α between \mathbf{S} and the positive x -axis
- write \mathbf{S} as a vector in terms of the unit vectors \mathbf{i} and \mathbf{j} and then write a unit vector \mathbf{n} along the vector sum \mathbf{S}
- determine the vector difference $\mathbf{D} = \mathbf{V}_1 - \mathbf{V}_2$

Solution (a) We construct to scale the parallelogram shown in Fig. *a* for adding \mathbf{V}_1 and \mathbf{V}_2 . Using the law of cosines, we have

$$S^2 = 3^2 + 4^2 - 2(3)(4) \cos 105^\circ$$

$$S = 5.59 \text{ units}$$

- ① (b) Using the law of sines for the lower triangle, we have

$$\frac{\sin 105^\circ}{5.59} = \frac{\sin(\alpha + 30^\circ)}{4}$$

$$\sin(\alpha + 30^\circ) = 0.692$$

$$(\alpha + 30^\circ) = 43.8^\circ \quad \alpha = 13.76^\circ$$

- (c) With knowledge of both S and α , we can write the vector \mathbf{S} as

$$\mathbf{S} = S[\mathbf{i} \cos \alpha + \mathbf{j} \sin \alpha]$$

$$= S[\mathbf{i} \cos 13.76^\circ + \mathbf{j} \sin 13.76^\circ] = 5.43\mathbf{i} + 1.328\mathbf{j} \text{ units}$$

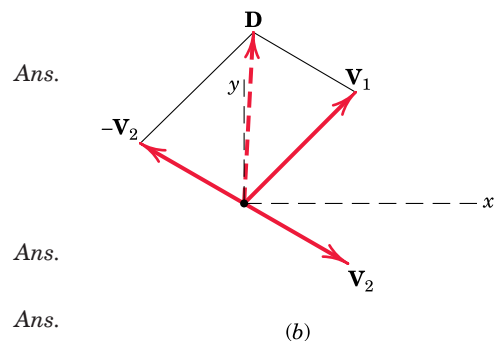
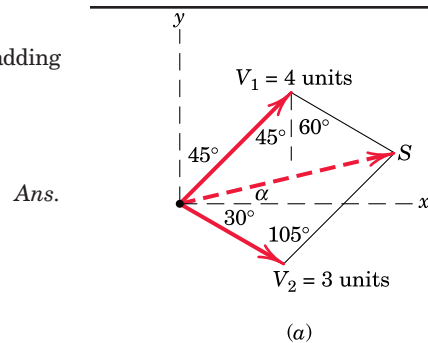
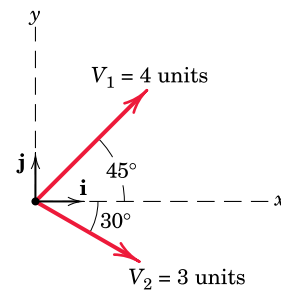
- ② Then $\mathbf{n} = \frac{\mathbf{S}}{S} = \frac{5.43\mathbf{i} + 1.328\mathbf{j}}{5.59} = 0.971\mathbf{i} + 0.238\mathbf{j}$

- (d) The vector difference \mathbf{D} is

$$\mathbf{D} = \mathbf{V}_1 - \mathbf{V}_2 = 4(\mathbf{i} \cos 45^\circ + \mathbf{j} \sin 45^\circ) - 3(\mathbf{i} \cos 30^\circ - \mathbf{j} \sin 30^\circ)$$

$$= 0.230\mathbf{i} + 4.33\mathbf{j} \text{ units}$$

The vector \mathbf{D} is shown in Fig. *b* as $\mathbf{D} = \mathbf{V}_1 + (-\mathbf{V}_2)$.



Helpful Hints

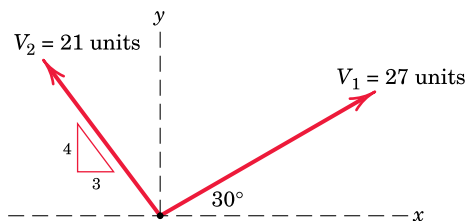
- You will frequently use the laws of cosines and sines in mechanics. See Art. C/6 of Appendix C for a review of these important geometric principles.
- A unit vector may always be formed by dividing a vector by its magnitude. Note that a unit vector is dimensionless.

PROBLEMS

- 1/1** Determine the angles made by the vector $\mathbf{V} = -36\mathbf{i} + 15\mathbf{j}$ with the positive x - and y -axes. Write the unit vector \mathbf{n} in the direction of \mathbf{V} .

$$\text{Ans. } \theta_x = 157.4^\circ, \theta_y = 67.4^\circ \\ \mathbf{n} = -0.923\mathbf{i} + 0.385\mathbf{j}$$

- 1/2** Determine the magnitude of the vector sum $\mathbf{V} = \mathbf{V}_1 + \mathbf{V}_2$ and the angle θ_x which \mathbf{V} makes with the positive x -axis. Complete both graphical and algebraic solutions.



Problem 1/2

- 1/3** For the given vectors \mathbf{V}_1 and \mathbf{V}_2 of Prob. 1/2, determine the magnitude of the vector difference $\mathbf{V}' = \mathbf{V}_2 - \mathbf{V}_1$ and the angle θ_x which \mathbf{V}' makes with the positive x -axis. Complete both graphical and algebraic solutions.

$$\text{Ans. } V' = 36.1 \text{ units}, \theta_x = 174.8^\circ$$

- 1/4** A force is specified by the vector $\mathbf{F} = 160\mathbf{i} + 80\mathbf{j} - 120\mathbf{k}$ N. Calculate the angles made by \mathbf{F} with the positive x -, y -, and z -axes.

- 1/5** What is the mass in both slugs and kilograms of a 1000-lb beam?

$$\text{Ans. } m = 31.1 \text{ slugs}, m = 454 \text{ kg}$$

- 1/6** From the gravitational law calculate the weight W (gravitational force with respect to the earth) of an 80-kg man in a spacecraft traveling in a circular orbit 250 km above the earth's surface. Express W in both newtons and pounds.

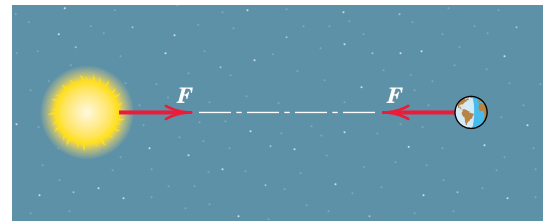
- 1/7** Determine the weight in newtons of a woman whose weight in pounds is 125. Also, find her mass in slugs and in kilograms. Determine your own weight in newtons.

$$\text{Ans. } W = 556 \text{ N}, m = 3.88 \text{ slugs}, m = 56.7 \text{ kg}$$

- 1/8** Suppose that two nondimensional quantities are exactly $A = 8.67$ and $B = 1.429$. Using the rules for significant figures as stated in this chapter, express the four quantities $(A + B)$, $(A - B)$, (AB) , and (A/B) .

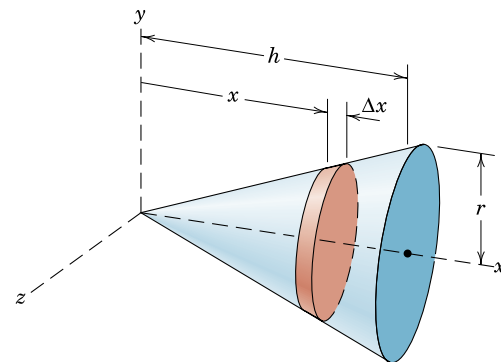
- 1/9** Compute the magnitude F of the force which the sun exerts on the earth. Perform the calculation first in newtons and then convert your result to pounds. Refer to Table D/2 for necessary physical quantities.

$$\text{Ans. } F = 3.55(10^{22}) \text{ N}, F = 7.97(10^{21}) \text{ lb}$$



Problem 1/9

- 1/10** The element of volume ΔV of the right circular cone of altitude h and base radius r is formed by slicing the cone at a distance x from the vertex. If the slice is of finite thickness Δx , show that its volume ΔV is $[\pi r^2/h^2] [x^2 \Delta x + x(\Delta x)^2 + \frac{1}{3}(\Delta x)^3]$. (Recall the formula for the volume of a cone.) Explain what happens to the second and third terms when Δx becomes the infinitesimal dx .



Problem 1/10

- 1/11** What is the percent error n in replacing the sine of 20° by the value of the angle in radians? Repeat for the tangent of 20° , and explain the qualitative difference in the two error percentages.

$$\text{Ans. } n = 2.06\%, n = 4.09\%$$



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The properties of force systems must be thoroughly understood by the engineers who design devices such as this crane and its lifting harness. Try to visualize the forces present in the various parts of this system.

2

FORCE SYSTEMS

CHAPTER OUTLINE

2/1 Introduction

2/2 Force

SECTION A TWO-DIMENSIONAL FORCE SYSTEMS

2/3 Rectangular Components

2/4 Moment

2/5 Couple

2/6 Resultants

SECTION B THREE-DIMENSIONAL FORCE SYSTEMS

2/7 Rectangular Components

2/8 Moment and Couple

2/9 Resultants

2/10 Chapter Review

2/1 INTRODUCTION

In this and the following chapters, we study the effects of forces which act on engineering structures and mechanisms. The experience gained here will help you in the study of mechanics and in other subjects such as stress analysis, design of structures and machines, and fluid flow. This chapter lays the foundation for a basic understanding not only of statics but also of the entire subject of mechanics, and you should master this material thoroughly.

2/2 FORCE

Before dealing with a group or *system* of forces, it is necessary to examine the properties of a single force in some detail. A force has been defined in Chapter 1 as an action of one body on another. In dynamics we will see that a force is defined as an action which tends to cause acceleration of a body. A force is a *vector quantity*, because its effect depends on the direction as well as on the magnitude of the action. Thus, forces may be combined according to the parallelogram law of vector addition.

The action of the cable tension on the bracket in Fig. 2/1a is represented in the side view, Fig. 2/1b, by the force vector \mathbf{P} of magnitude P . The effect of this action on the bracket depends on P , the angle θ , and the location of the point of application A . Changing any one of these three specifications will alter the effect on the bracket, such as the force

24 Chapter 2 Force Systems

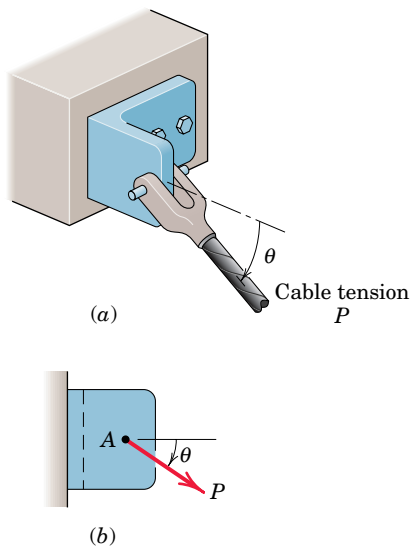


Figure 2/1

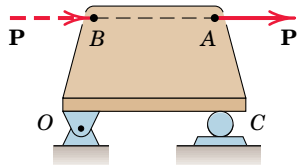


Figure 2/2



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The forces associated with this lifting rig must be carefully identified, classified, and analyzed in order to provide a safe and effective working environment.

in one of the bolts which secure the bracket to the base, or the internal force and deformation in the material of the bracket at any point. Thus, the complete specification of the action of a force must include its *magnitude*, *direction*, and *point of application*, and therefore we must treat it as a fixed vector.

External and Internal Effects

We can separate the action of a force on a body into two effects, *external* and *internal*. For the bracket of Fig. 2/1 the effects of \mathbf{P} external to the bracket are the reactive forces (not shown) exerted on the bracket by the foundation and bolts because of the action of \mathbf{P} . Forces external to a body can be either *applied* forces or *reactive* forces. The effects of \mathbf{P} internal to the bracket are the resulting internal forces and deformations distributed throughout the material of the bracket. The relation between internal forces and internal deformations depends on the material properties of the body and is studied in strength of materials, elasticity, and plasticity.

Principle of Transmissibility

When dealing with the mechanics of a rigid body, we ignore deformations in the body and concern ourselves with only the net external effects of external forces. In such cases, experience shows us that it is not necessary to restrict the action of an applied force to a given point. For example, the force \mathbf{P} acting on the rigid plate in Fig. 2/2 may be applied at A or at B or at any other point on its line of action, and the net external effects of \mathbf{P} on the bracket will not change. The external effects are the force exerted on the plate by the bearing support at O and the force exerted on the plate by the roller support at C.

This conclusion is summarized by the *principle of transmissibility*, which states that a force may be applied at any point on its given line of action without altering the resultant effects of the force *external* to the *rigid* body on which it acts. Thus, whenever we are interested in only the resultant external effects of a force, the force may be treated as a *sliding vector*, and we need specify only the *magnitude*, *direction*, and *line of action* of the force, and not its *point of application*. Because this book deals essentially with the mechanics of rigid bodies, we will treat almost all forces as sliding vectors for the rigid body on which they act.

Force Classification

Forces are classified as either *contact* or *body* forces. A contact force is produced by direct physical contact; an example is the force exerted on a body by a supporting surface. On the other hand, a body force is generated by virtue of the position of a body within a force field such as a gravitational, electric, or magnetic field. An example of a body force is your weight.

Forces may be further classified as either *concentrated* or *distributed*. Every contact force is actually applied over a finite area and is therefore really a distributed force. However, when the dimensions of the area are very small compared with the other dimensions of the

body, we may consider the force to be concentrated at a point with negligible loss of accuracy. Force can be distributed over an *area*, as in the case of mechanical contact, over a *volume* when a body force such as weight is acting, or over a *line*, as in the case of the weight of a suspended cable.

The *weight* of a body is the force of gravitational attraction distributed over its volume and may be taken as a concentrated force acting through the center of gravity. The position of the center of gravity is frequently obvious if the body is symmetric. If the position is not obvious, then a separate calculation, explained in Chapter 5, will be necessary to locate the center of gravity.

We can measure a force either by comparison with other known forces, using a mechanical balance, or by the calibrated movement of an elastic element. All such comparisons or calibrations have as their basis a primary standard. The standard unit of force in SI units is the newton (N) and in the U.S. customary system is the pound (lb), as defined in Art. 1/5.

Action and Reaction

According to Newton's third law, the *action* of a force is always accompanied by an *equal and opposite reaction*. It is essential to distinguish between the action and the reaction in a pair of forces. To do so, we first *isolate* the body in question and then identify the force exerted *on* that body (not the force exerted *by* the body). It is very easy to mistakenly use the wrong force of the pair unless we distinguish carefully between action and reaction.

Concurrent Forces

Two or more forces are said to be *concurrent at a point* if their lines of action intersect at that point. The forces \mathbf{F}_1 and \mathbf{F}_2 shown in Fig. 2/3a have a common point of application and are concurrent at the point A. Thus, they can be added using the parallelogram law in their common plane to obtain their sum or *resultant* \mathbf{R} , as shown in Fig. 2/3a. The resultant lies in the same plane as \mathbf{F}_1 and \mathbf{F}_2 .

Suppose the two concurrent forces lie in the same plane but are applied at two different points as in Fig. 2/3b. By the principle of transmissibility, we may move them along their lines of action and complete their vector sum \mathbf{R} at the point of concurrency A, as shown in Fig. 2/3b. We can replace \mathbf{F}_1 and \mathbf{F}_2 with the resultant \mathbf{R} without altering the external effects on the body upon which they act.

We can also use the triangle law to obtain \mathbf{R} , but we need to move the line of action of one of the forces, as shown in Fig. 2/3c. If we add the same two forces as shown in Fig. 2/3d, we correctly preserve the magnitude and direction of \mathbf{R} , but we lose the correct line of action, because \mathbf{R} obtained in this way does not pass through A. Therefore this type of combination should be avoided.

We can express the sum of the two forces mathematically by the vector equation

$$\mathbf{R} = \mathbf{F}_1 + \mathbf{F}_2$$

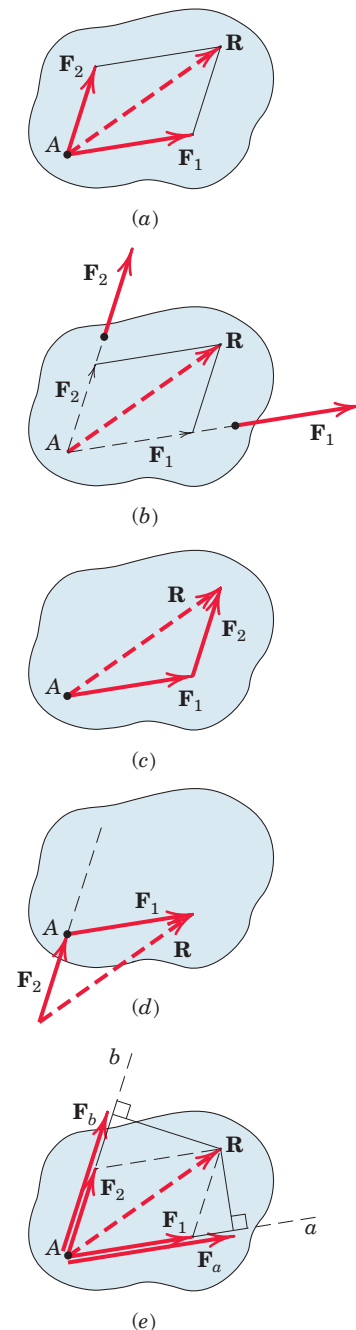


Figure 2/3

Vector Components

In addition to combining forces to obtain their resultant, we often need to replace a force by its *vector components* in directions which are convenient for a given application. The vector sum of the components must equal the original vector. Thus, the force \mathbf{R} in Fig. 2/3a may be replaced by, or *resolved* into, two vector components \mathbf{F}_1 and \mathbf{F}_2 with the specified directions by completing the parallelogram as shown to obtain the magnitudes of \mathbf{F}_1 and \mathbf{F}_2 .

The relationship between a force and its vector components along given axes must not be confused with the relationship between a force and its perpendicular* projections onto the same axes. Figure 2/3e shows the perpendicular projections \mathbf{F}_a and \mathbf{F}_b of the given force \mathbf{R} onto axes a and b , which are parallel to the vector components \mathbf{F}_1 and \mathbf{F}_2 of Fig. 2/3a. Figure 2/3e shows that the components of a vector are not necessarily equal to the projections of the vector onto the same axes. Furthermore, the vector sum of the projections \mathbf{F}_a and \mathbf{F}_b is not the vector \mathbf{R} , because the parallelogram law of vector addition must be used to form the sum. The components and projections of \mathbf{R} are equal only when the axes a and b are perpendicular.

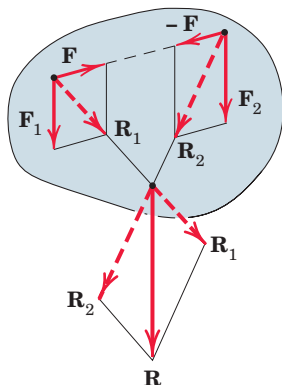


Figure 2/4

A Special Case of Vector Addition

To obtain the resultant when the two forces \mathbf{F}_1 and \mathbf{F}_2 are parallel as in Fig. 2/4, we use a special case of addition. The two vectors are combined by first adding two equal, opposite, and collinear forces \mathbf{F} and $-\mathbf{F}$ of convenient magnitude, which taken together produce no external effect on the body. Adding \mathbf{F}_1 and \mathbf{F} to produce \mathbf{R}_1 , and combining with the sum \mathbf{R}_2 of \mathbf{F}_2 and $-\mathbf{F}$ yield the resultant \mathbf{R} , which is correct in magnitude, direction, and line of action. This procedure is also useful for graphically combining two forces which have a remote and inconvenient point of concurrency because they are almost parallel.

It is usually helpful to master the analysis of force systems in two dimensions before undertaking three-dimensional analysis. Thus the remainder of Chapter 2 is subdivided into these two categories.

SECTION A TWO-DIMENSIONAL FORCE SYSTEMS

2/3 RECTANGULAR COMPONENTS

The most common two-dimensional resolution of a force vector is into rectangular components. It follows from the parallelogram rule that the vector \mathbf{F} of Fig. 2/5 may be written as

$$\mathbf{F} = \mathbf{F}_x + \mathbf{F}_y \quad (2/1)$$

where \mathbf{F}_x and \mathbf{F}_y are *vector components* of \mathbf{F} in the x - and y -directions. Each of the two vector components may be written as a scalar times the

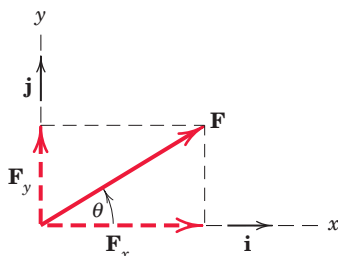


Figure 2/5

*Perpendicular projections are also called *orthogonal* projections.

appropriate unit vector. In terms of the unit vectors \mathbf{i} and \mathbf{j} of Fig. 2/5, $\mathbf{F}_x = F_x \mathbf{i}$ and $\mathbf{F}_y = F_y \mathbf{j}$, and thus we may write

$$\mathbf{F} = F_x \mathbf{i} + F_y \mathbf{j} \quad (2/2)$$

where the scalars F_x and F_y are the x and y scalar components of the vector \mathbf{F} .

The scalar components can be positive or negative, depending on the quadrant into which \mathbf{F} points. For the force vector of Fig. 2/5, the x and y scalar components are both positive and are related to the magnitude and direction of \mathbf{F} by

$$\begin{aligned} F_x &= F \cos \theta & F &= \sqrt{F_x^2 + F_y^2} \\ F_y &= F \sin \theta & \theta &= \tan^{-1} \frac{F_y}{F_x} \end{aligned} \quad (2/3)$$

Conventions for Describing Vector Components

We express the magnitude of a vector with lightface italic type in print; that is, $|\mathbf{F}|$ is indicated by F , a quantity which is always *nonnegative*. However, the scalar components, also denoted by lightface italic type, will include sign information. See Sample Problems 2/1 and 2/3 for numerical examples which involve both positive and negative scalar components.

When both a force and its vector components appear in a diagram, it is desirable to show the vector components of the force with dashed lines, as in Fig. 2/5, and show the force with a solid line, or vice versa. With either of these conventions it will always be clear that a force and its components are being represented, and not three separate forces, as would be implied by three solid-line vectors.

Actual problems do not come with reference axes, so their assignment is a matter of arbitrary convenience, and the choice is frequently up to the student. The logical choice is usually indicated by the way in which the geometry of the problem is specified. When the principal dimensions of a body are given in the horizontal and vertical directions, for example, you would typically assign reference axes in these directions.

Determining the Components of a Force

Dimensions are not always given in horizontal and vertical directions, angles need not be measured counterclockwise from the x -axis, and the origin of coordinates need not be on the line of action of a force. Therefore, it is essential that we be able to determine the correct components of a force no matter how the axes are oriented or how the angles are measured. Figure 2/6 suggests a few typical examples of vector resolution in two dimensions.

Memorization of Eqs. 2/3 is not a substitute for understanding the parallelogram law and for correctly projecting a vector onto a reference axis. A neatly drawn sketch always helps to clarify the geometry and avoid error.

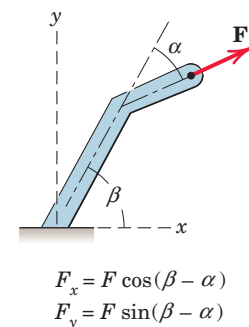
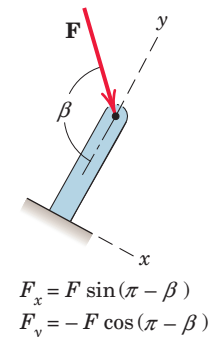
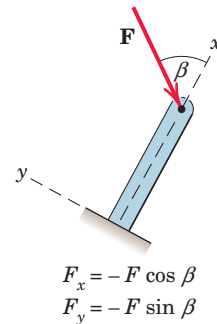
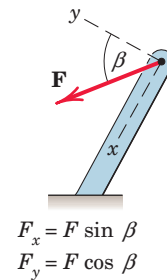


Figure 2/6

28 Chapter 2 Force Systems

Rectangular components are convenient for finding the sum or resultant \mathbf{R} of two forces which are concurrent. Consider two forces \mathbf{F}_1 and \mathbf{F}_2 which are originally concurrent at a point O . Figure 2/7 shows the line of action of \mathbf{F}_2 shifted from O to the tip of \mathbf{F}_1 according to the triangle rule of Fig. 2/3. In adding the force vectors \mathbf{F}_1 and \mathbf{F}_2 , we may write

$$\mathbf{R} = \mathbf{F}_1 + \mathbf{F}_2 = (F_{1x}\mathbf{i} + F_{1y}\mathbf{j}) + (F_{2x}\mathbf{i} + F_{2y}\mathbf{j})$$

or

$$R_x\mathbf{i} + R_y\mathbf{j} = (F_{1x} + F_{2x})\mathbf{i} + (F_{1y} + F_{2y})\mathbf{j}$$

from which we conclude that

$$\begin{aligned} R_x &= F_{1x} + F_{2x} = \Sigma F_x \\ R_y &= F_{1y} + F_{2y} = \Sigma F_y \end{aligned} \quad (2/4)$$

The term ΣF_x means “the algebraic sum of the x scalar components”. For the example shown in Fig. 2/7, note that the scalar component F_{2y} would be negative.



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The structural elements in the foreground transmit concentrated forces to the brackets at both ends.

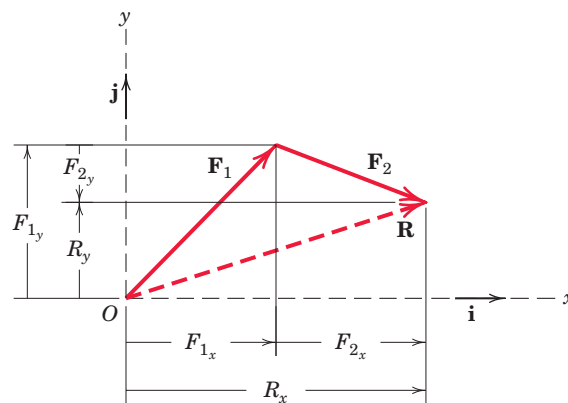


Figure 2/7

Sample Problem 2/1

The forces \mathbf{F}_1 , \mathbf{F}_2 , and \mathbf{F}_3 , all of which act on point A of the bracket, are specified in three different ways. Determine the x and y scalar components of each of the three forces.

Solution. The scalar components of \mathbf{F}_1 , from Fig. *a*, are

$$F_{1x} = 600 \cos 35^\circ = 491 \text{ N} \quad \text{Ans.}$$

$$F_{1y} = 600 \sin 35^\circ = 344 \text{ N} \quad \text{Ans.}$$

The scalar components of \mathbf{F}_2 , from Fig. *b*, are

$$F_{2x} = -500 \left(\frac{4}{5} \right) = -400 \text{ N} \quad \text{Ans.}$$

$$F_{2y} = 500 \left(\frac{3}{5} \right) = 300 \text{ N} \quad \text{Ans.}$$

Note that the angle which orients \mathbf{F}_2 to the x -axis is never calculated. The cosine and sine of the angle are available by inspection of the 3-4-5 triangle. Also note that the x scalar component of \mathbf{F}_2 is negative by inspection.

The scalar components of \mathbf{F}_3 can be obtained by first computing the angle α of Fig. *c*.

$$\alpha = \tan^{-1} \left[\frac{0.2}{0.4} \right] = 26.6^\circ$$

$$\textcircled{1} \quad \text{Then } F_{3x} = F_3 \sin \alpha = 800 \sin 26.6^\circ = 358 \text{ N} \quad \text{Ans.}$$

$$F_{3y} = -F_3 \cos \alpha = -800 \cos 26.6^\circ = -716 \text{ N} \quad \text{Ans.}$$

Alternatively, the scalar components of \mathbf{F}_3 can be obtained by writing \mathbf{F}_3 as a magnitude times a unit vector \mathbf{n}_{AB} in the direction of the line segment AB . Thus,

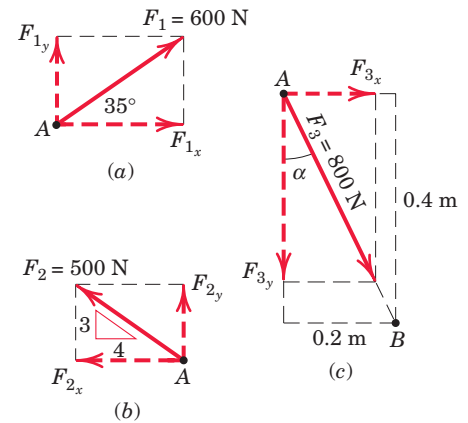
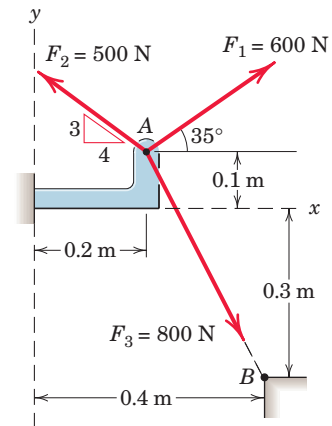
$$\begin{aligned} \textcircled{2} \quad \mathbf{F}_3 &= F_3 \mathbf{n}_{AB} = F_3 = \frac{\overrightarrow{AB}}{AB} = 800 \left[\frac{0.2\mathbf{i} - 0.4\mathbf{j}}{\sqrt{(0.2)^2 + (-0.4)^2}} \right] \\ &= 800 [0.447\mathbf{i} - 0.894\mathbf{j}] \\ &= 358\mathbf{i} - 716\mathbf{j} \text{ N} \end{aligned}$$

The required scalar components are then

$$F_{3x} = 358 \text{ N} \quad \text{Ans.}$$

$$F_{3y} = -716 \text{ N} \quad \text{Ans.}$$

which agree with our previous results.



Helpful Hints

- ① You should carefully examine the geometry of each component determination problem and not rely on the blind use of such formulas as $F_x = F \cos \theta$ and $F_y = F \sin \theta$.
- ② A unit vector can be formed by dividing any vector, such as the geometric position vector \overrightarrow{AB} , by its length or magnitude. Here we use the overarrow to denote the vector which runs from A to B and the overbar to determine the distance between A and B.

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Sample Problem 2/2

Combine the two forces **P** and **T**, which act on the fixed structure at *B*, into a single equivalent force **R**.

- Graphical solution.** The parallelogram for the vector addition of forces **T** and **P** is constructed as shown in Fig. *a*. The scale used here is 1 cm = 400 N; a scale of 1 cm = 100 N would be more suitable for regular-size paper and would give greater accuracy. Note that the angle α must be determined prior to construction of the parallelogram. From the given figure

$$\tan \alpha = \frac{BD}{AD} = \frac{6 \sin 60^\circ}{3 + 6 \cos 60^\circ} = 0.866 \quad \alpha = 40.9^\circ$$

Measurement of the length *R* and direction θ of the resultant force **R** yields the approximate results

$$R = 525 \text{ N} \quad \theta = 49^\circ \quad \text{Ans.}$$

- Geometric solution.** The triangle for the vector addition of **T** and **P** is shown in Fig. *b*. The angle α is calculated as above. The law of cosines gives

$$R^2 = (600)^2 + (800)^2 - 2(600)(800) \cos 40.9^\circ = 274,300$$

$$R = 524 \text{ N} \quad \text{Ans.}$$

From the law of sines, we may determine the angle θ which orients **R**. Thus,

$$\frac{600}{\sin \theta} = \frac{524}{\sin 40.9^\circ} \quad \sin \theta = 0.750 \quad \theta = 48.6^\circ \quad \text{Ans.}$$

Algebraic solution. By using the *x-y* coordinate system on the given figure, we may write

$$R_x = \Sigma F_x = 800 - 600 \cos 40.9^\circ = 346 \text{ N}$$

$$R_y = \Sigma F_y = -600 \sin 40.9^\circ = -393 \text{ N}$$

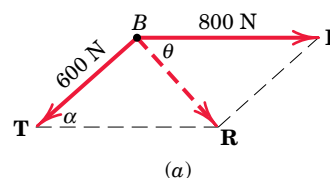
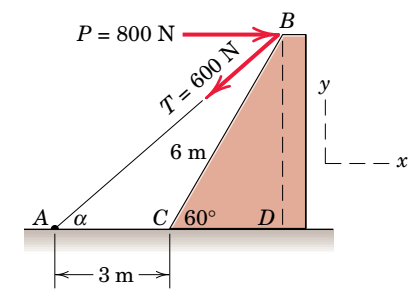
The magnitude and direction of the resultant force **R** as shown in Fig. *c* are then

$$R = \sqrt{R_x^2 + R_y^2} = \sqrt{(346)^2 + (-393)^2} = 524 \text{ N} \quad \text{Ans.}$$

$$\theta = \tan^{-1} \frac{|R_y|}{|R_x|} = \tan^{-1} \frac{393}{346} = 48.6^\circ \quad \text{Ans.}$$

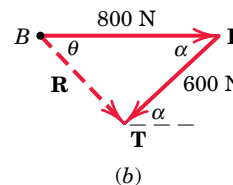
The resultant **R** may also be written in vector notation as

$$\mathbf{R} = R_x \mathbf{i} + R_y \mathbf{j} = 346 \mathbf{i} - 393 \mathbf{j} \text{ N} \quad \text{Ans.}$$

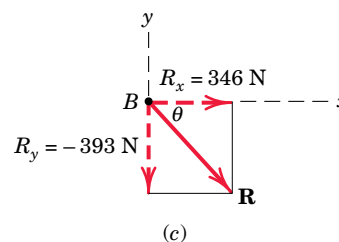


Helpful Hints

- ① Note the repositioning of **P** to permit parallelogram addition at *B*.



- ② Note the repositioning of **F** so as to preserve the correct line of action of the resultant **R**.



Sample Problem 2/3

The 500-N force \mathbf{F} is applied to the vertical pole as shown. (1) Write \mathbf{F} in terms of the unit vectors \mathbf{i} and \mathbf{j} and identify both its vector and scalar components. (2) Determine the scalar components of the force vector \mathbf{F} along the x' - and y' -axes. (3) Determine the scalar components of \mathbf{F} along the x - and y' -axes.

Solution. **Part (1).** From Fig. *a* we may write \mathbf{F} as

$$\begin{aligned}\mathbf{F} &= (F \cos \theta)\mathbf{i} - (F \sin \theta)\mathbf{j} \\ &= (500 \cos 60^\circ)\mathbf{i} - (500 \sin 60^\circ)\mathbf{j} \\ &= (250\mathbf{i} - 433\mathbf{j}) \text{ N} \quad \text{Ans.}\end{aligned}$$

The scalar components are $F_x = 250$ N and $F_y = -433$ N. The vector components are $\mathbf{F}_x = 250\mathbf{i}$ N and $\mathbf{F}_y = -433\mathbf{j}$ N.

Part (2). From Fig. *b* we may write \mathbf{F} as $\mathbf{F} = 500\mathbf{i}'$ N, so that the required scalar components are

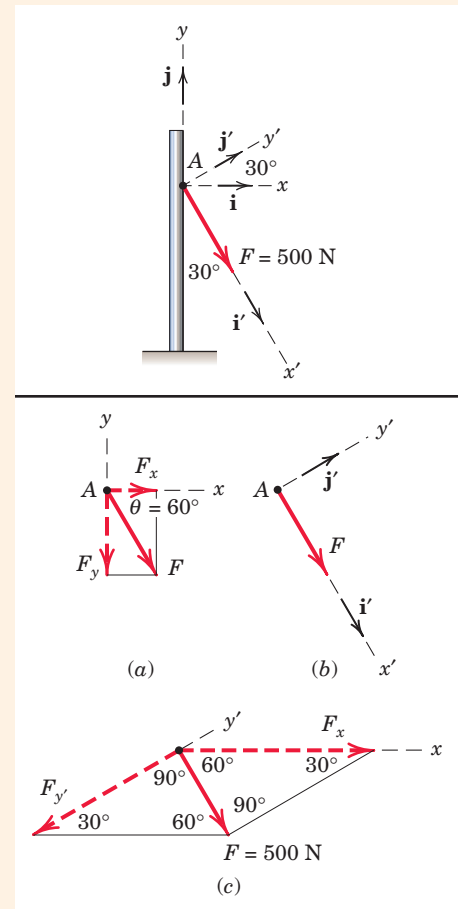
$$F_{x'} = 500 \text{ N} \quad F_{y'} = 0 \quad \text{Ans.}$$

Part (3). The components of \mathbf{F} in the x - and y' -directions are nonrectangular and are obtained by completing the parallelogram as shown in Fig. *c*. The magnitudes of the components may be calculated by the law of sines. Thus,

$$\begin{aligned}\frac{|F_x|}{\sin 90^\circ} &= \frac{500}{\sin 30^\circ} & |F_x| &= 1000 \text{ N} \\ \frac{|F_{y'}|}{\sin 60^\circ} &= \frac{500}{\sin 30^\circ} & |F_{y'}| &= 866 \text{ N}\end{aligned}$$

The required scalar components are then

$$F_x = 1000 \text{ N} \quad F_{y'} = -866 \text{ N} \quad \text{Ans.}$$



Helpful Hint

① Obtain F_x and $F_{y'}$ graphically and compare your results with the calculated values.

Sample Problem 2/4

Forces \mathbf{F}_1 and \mathbf{F}_2 act on the bracket as shown. Determine the projection F_b of their resultant \mathbf{R} onto the b -axis.

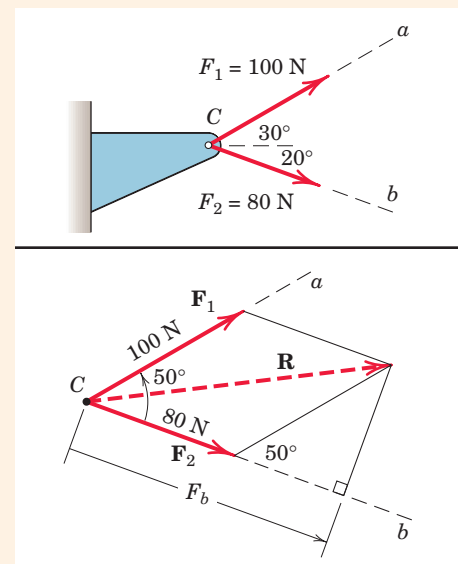
Solution. The parallelogram addition of \mathbf{F}_1 and \mathbf{F}_2 is shown in the figure. Using the law of cosines gives us

$$R^2 = (80)^2 + (100)^2 - 2(80)(100) \cos 130^\circ \quad R = 163.4 \text{ N}$$

The figure also shows the orthogonal projection F_b of \mathbf{R} onto the b -axis. Its length is

$$F_b = 80 + 100 \cos 50^\circ = 144.3 \text{ N} \quad \text{Ans.}$$

Note that the components of a vector are in general not equal to the projections of the vector onto the same axes. If the a -axis had been perpendicular to the b -axis, then the projections and components of \mathbf{R} would have been equal.



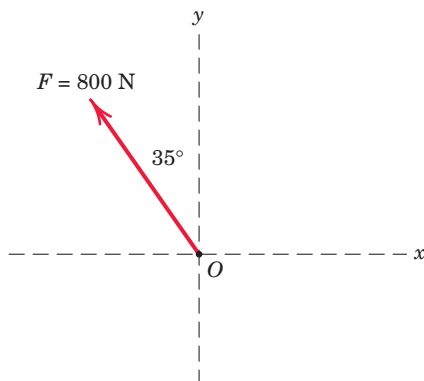
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PROBLEMS

Introductory Problems

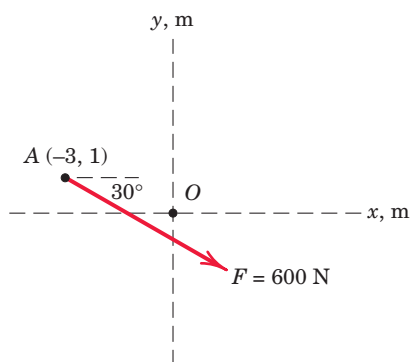
- 2/1** The force \mathbf{F} has a magnitude of 800 N. Express \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} and \mathbf{j} . Identify the x and y scalar components of \mathbf{F} .

Ans. $\mathbf{F} = -459\mathbf{i} + 655\mathbf{j}$ N
 $F_x = -459$ N, $F_y = 655$ N



Problem 2/1

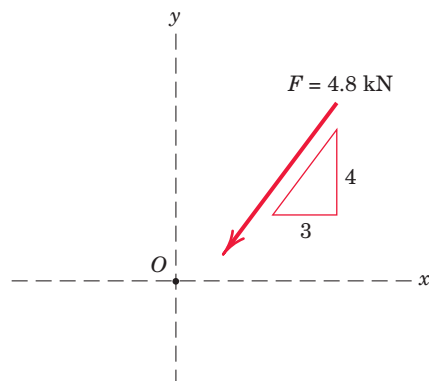
- 2/2** The magnitude of the force \mathbf{F} is 600 N. Express \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} and \mathbf{j} . Identify both the scalar and vector components of \mathbf{F} .



Problem 2/2

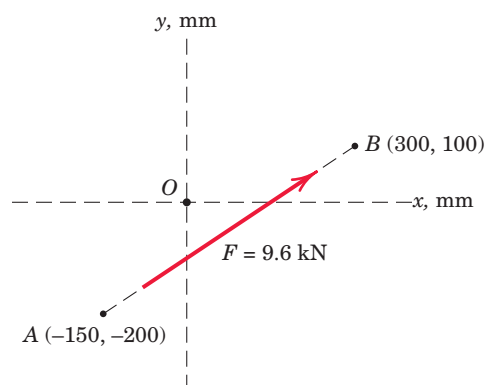
- 2/3** The slope of the 4.8-kN force \mathbf{F} is specified as shown in the figure. Express \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} and \mathbf{j} .

Ans. $\mathbf{F} = -2.88\mathbf{i} - 3.84\mathbf{j}$ kN



Problem 2/3

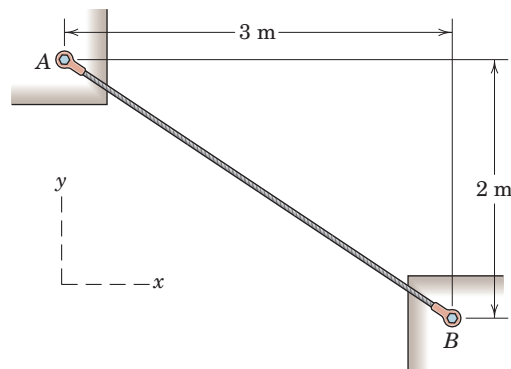
- 2/4** The line of action of the 9.6-kN force \mathbf{F} runs through the points A and B as shown in the figure. Determine the x and y scalar components of \mathbf{F} .



Problem 2/4

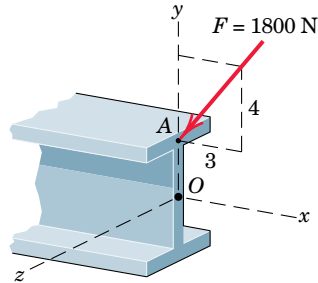
- 2/5** A cable stretched between the fixed supports A and B is under a tension T of 900 N. Express the tension as a vector using the unit vectors \mathbf{i} and \mathbf{j} , first, as a force \mathbf{T}_A acting on A and second, as a force \mathbf{T}_B acting on B .

Ans. $\mathbf{T}_A = 749\mathbf{i} - 499\mathbf{j}$ N, $\mathbf{T}_B = -749\mathbf{i} + 499\mathbf{j}$ N



Problem 2/5

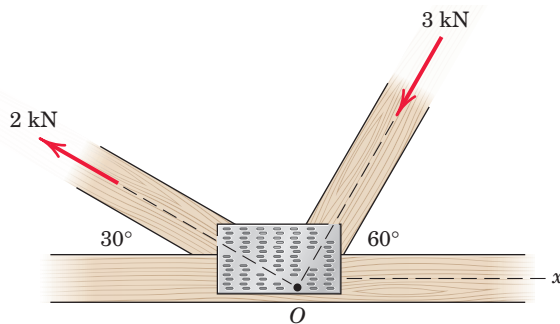
- 2/6** The 1800-N force \mathbf{F} is applied to the end of the I-beam. Express \mathbf{F} as a vector using the unit vectors \mathbf{i} and \mathbf{j} .



Problem 2/6

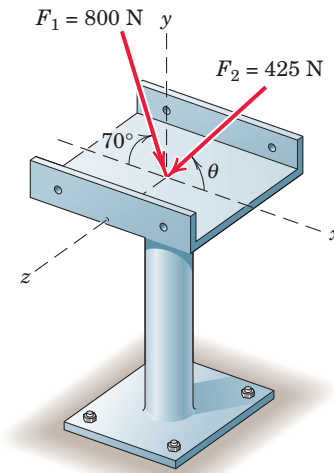
- 2/7** The two structural members, one of which is in tension and the other in compression, exert the indicated forces on joint O . Determine the magnitude of the resultant \mathbf{R} of the two forces and the angle θ which \mathbf{R} makes with the positive x -axis.

Ans. $R = 3.61 \text{ kN}$, $\theta = 206^\circ$



Problem 2/7

- 2/8** Two forces are applied to the construction bracket as shown. Determine the angle θ which makes the resultant of the two forces vertical. Determine the magnitude R of the resultant.

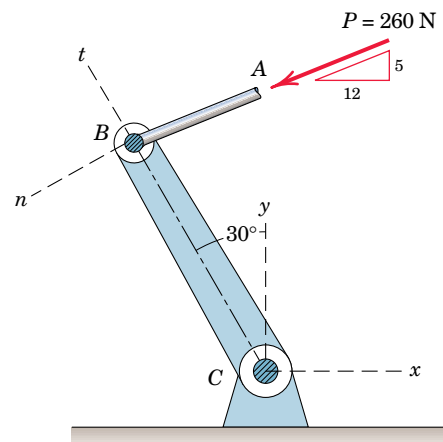


Problem 2/8

Representative Problems

- 2/9** In the design of a control mechanism, it is determined that rod AB transmits a 260-N force \mathbf{P} to the crank BC . Determine the x and y scalar components of \mathbf{P} .

Ans. $P_x = -240 \text{ N}$
 $P_y = -100 \text{ N}$



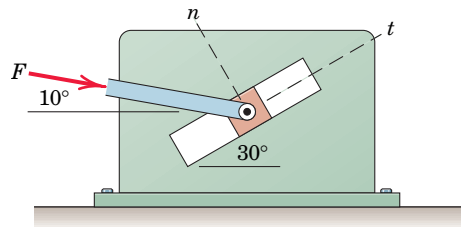
Problem 2/9

- 2/10** For the mechanism of Prob. 2/9, determine the scalar components P_t and P_n of \mathbf{P} which are tangent and normal, respectively, to crank BC .

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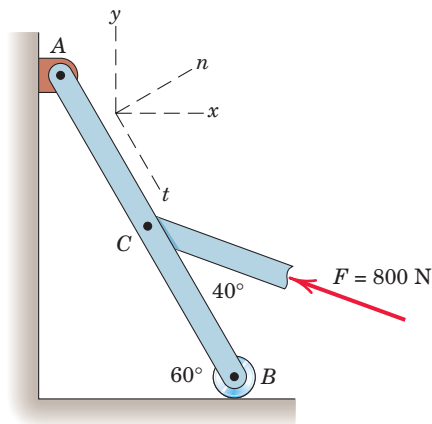
- 2/11** The t -component of the force \mathbf{F} is known to be 75 N. Determine the n -component and the magnitude of \mathbf{F} .

Ans. $F_n = -62.9 \text{ N}$, $F = 97.9 \text{ N}$



Problem 2/11

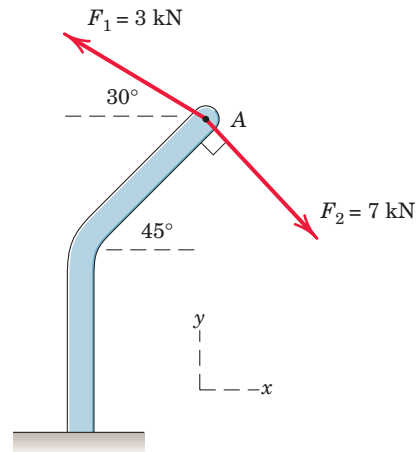
- 2/12** A force \mathbf{F} of magnitude 800 N is applied to point C of the bar AB as shown. Determine both the x - y and the n - t components of \mathbf{F} .



Problem 2/12

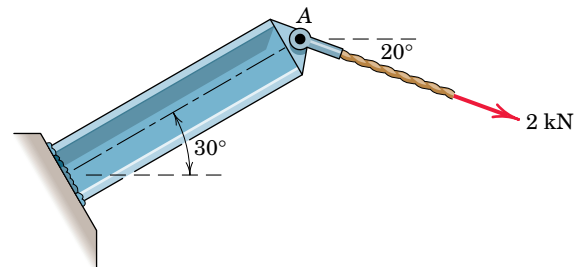
- 2/13** The two forces shown act at point A of the bent bar. Determine the resultant \mathbf{R} of the two forces.

Ans. $\mathbf{R} = 2.35\mathbf{i} - 3.45\mathbf{j} \text{ kN}$



Problem 2/13

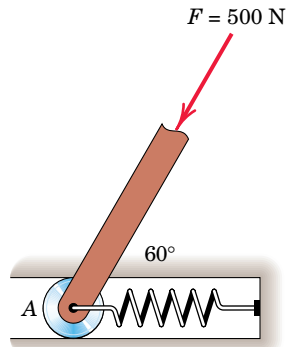
- 2/14** To satisfy design limitations it is necessary to determine the effect of the 2-kN tension in the cable on the shear, tension, and bending of the fixed I-beam. For this purpose replace this force by its equivalent of two forces at A , F_t parallel and F_n perpendicular to the beam. Determine F_t and F_n .



Problem 2/14

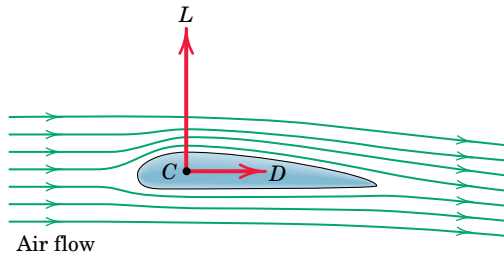
- 2/15** Determine the magnitude F_s of the tensile spring force in order that the resultant of \mathbf{F}_s and \mathbf{F} is a vertical force. Determine the magnitude R of this vertical resultant force.

Ans. $F_s = 250 \text{ N}$, $R = 433 \text{ N}$



Problem 2/15

- 2/16** The ratio of the lift force L to the drag force D for the simple airfoil is $L/D = 10$. If the lift force on a short section of the airfoil is 200 N, compute the magnitude of the resultant force \mathbf{R} and the angle θ which it makes with the horizontal.

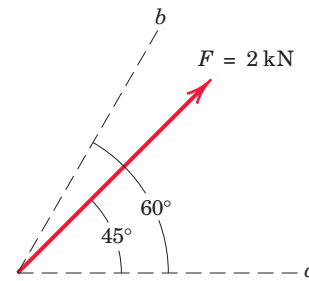


Problem 2/16

- 2/17** Determine the components of the 2-kN force along the oblique axes a and b . Determine the projections of \mathbf{F} onto the a - and b -axes.

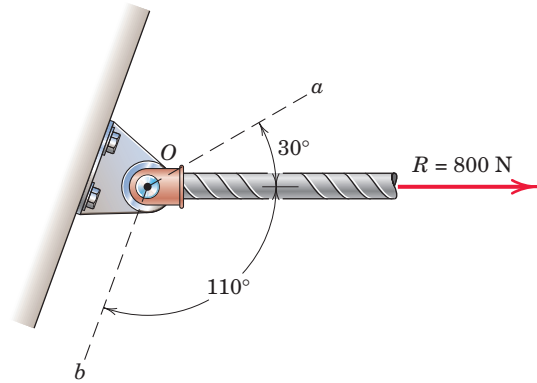
Ans. $F_a = 0.598 \text{ kN}$, $F_b = 1.633 \text{ kN}$

$P_a = 1.414 \text{ kN}$, $P_b = 1.932 \text{ kN}$



Problem 2/17

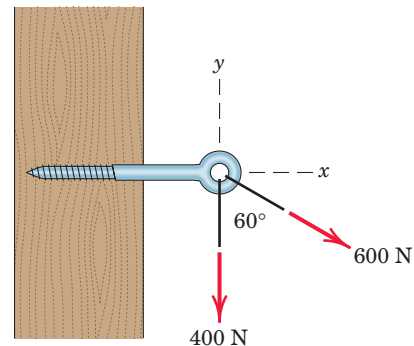
- 2/18** Determine the scalar components R_a and R_b of the force \mathbf{R} along the nonrectangular axes a and b . Also determine the orthogonal projection P_a of \mathbf{R} onto axis a .



Problem 2/18

- 2/19** Determine the resultant \mathbf{R} of the two forces shown by (a) applying the parallelogram rule for vector addition and (b) summing scalar components.

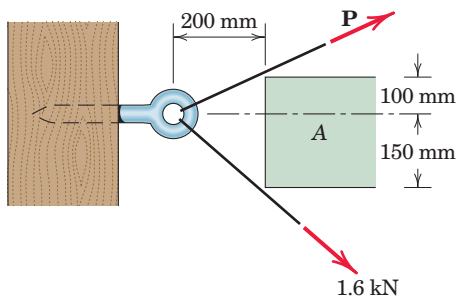
Ans. $\mathbf{R} = 520\mathbf{i} - 700\mathbf{j} \text{ N}$



Problem 2/19

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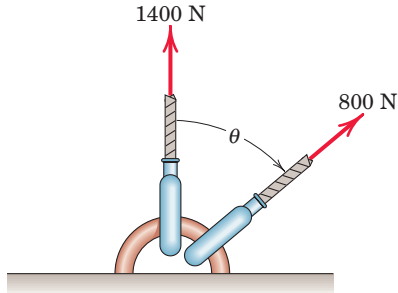
- 2/20** It is desired to remove the spike from the timber by applying force along its horizontal axis. An obstruction A prevents direct access, so that two forces, one 1.6 kN and the other \mathbf{P} , are applied by cables as shown. Compute the magnitude of \mathbf{P} necessary to ensure a resultant \mathbf{T} directed along the spike. Also find T .



Problem 2/20

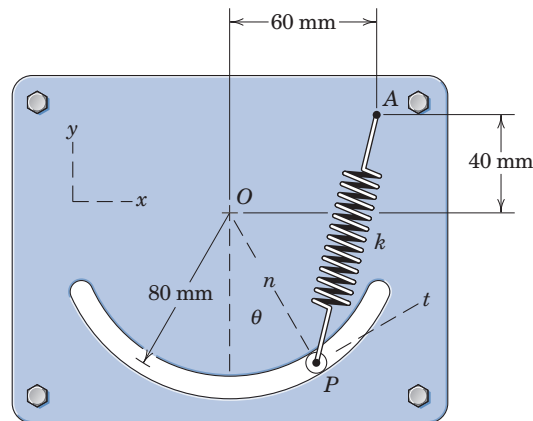
- 2/21** At what angle θ must the 800-N force be applied in order that the resultant \mathbf{R} of the two forces has a magnitude of 2000 N? For this condition, determine the angle β between \mathbf{R} and the vertical.

Ans. $\theta = 51.3^\circ$, $\beta = 18.19^\circ$



Problem 2/21

- 2/22** The unstretched length of the spring of modulus $k = 1.2$ kN/m is 100 mm. When pin P is in the position $\theta = 30^\circ$, determine the x - and y -components of the force which the spring exerts on the pin. (The force in a spring is given by $F = kx$, where x is the deflection from the unstretched length.)

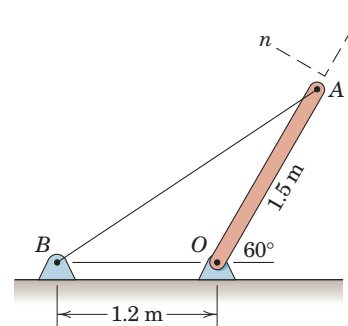


Problem 2/22

- 2/23** Refer to the statement and figure of Prob. 2/22. When pin P is in the position $\theta = 20^\circ$, determine the n - and t -components of the force F which the spring of modulus $k = 1.2$ kN/m exerts on the pin.

Ans. $F_n = 19.18$ N, $F_t = 13.84$ N

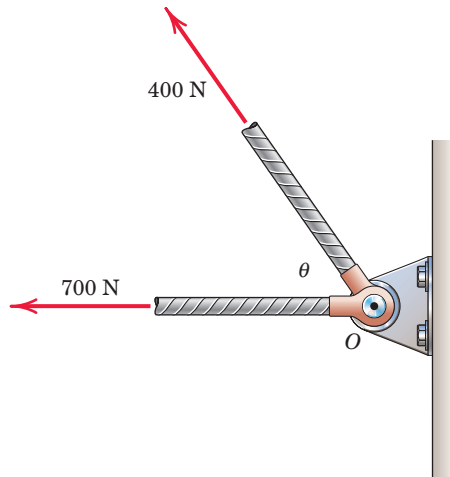
- 2/24** The cable AB prevents bar OA from rotating clockwise about the pivot O . If the cable tension is 750 N, determine the n - and t -components of this force acting on point A of the bar.



Problem 2/24

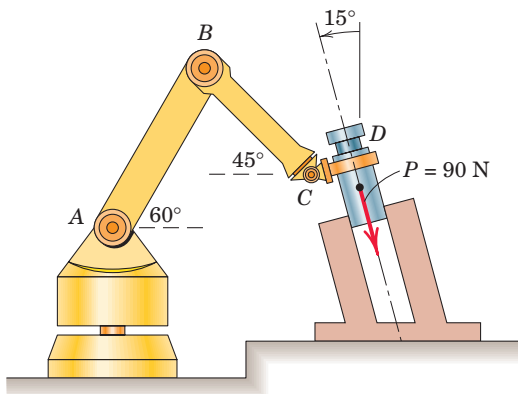
- 2/25** At what angle θ must the 400-N force be applied in order that the resultant \mathbf{R} of the two forces have a magnitude of 1000 N? For this condition what will be the angle β between \mathbf{R} and the horizontal?

Ans. $\theta = 51.3^\circ$, $\beta = 18.19^\circ$



Problem 2/25

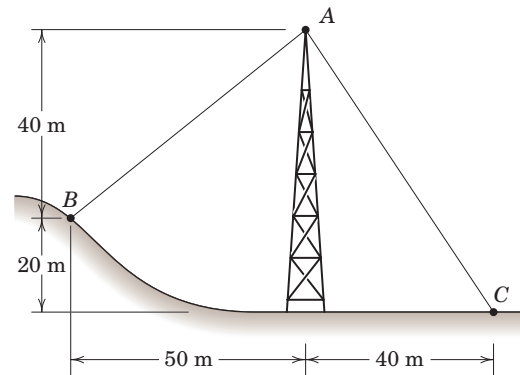
- 2/26** In the design of the robot to insert the small cylindrical part into a close-fitting circular hole, the robot arm must exert a 90-N force P on the part parallel to the axis of the hole as shown. Determine the components of the force which the part exerts on the robot along axes (a) parallel and perpendicular to the arm AB , and (b) parallel and perpendicular to the arm BC .



Problem 2/26

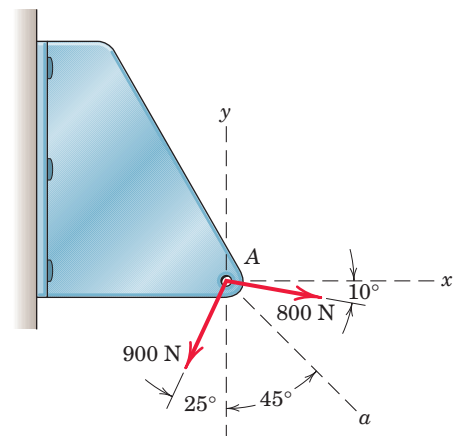
- 2/27** The guy cables AB and AC are attached to the top of the transmission tower. The tension in cable AC is 8 kN. Determine the required tension T in cable AB such that the net effect of the two cable tensions is a downward force at point A . Determine the magnitude R of this downward force.

Ans. $T = 5.68$ kN, $R = 10.21$ kN



Problem 2/27

- 2/28** The gusset plate is subjected to the two forces shown. Replace them by two equivalent forces, F_x in the x -direction and F_a in the a -direction. Determine the magnitudes of F_x and F_a . Solve geometrically or graphically.



Problem 2/28

2/4 MOMENT

In addition to the tendency to move a body in the direction of its application, a force can also tend to rotate a body about an axis. The axis may be any line which neither intersects nor is parallel to the line of action of the force. This rotational tendency is known as the *moment* \mathbf{M} of the force. Moment is also referred to as *torque*.

As a familiar example of the concept of moment, consider the pipe wrench of Fig. 2/8a. One effect of the force applied perpendicular to the handle of the wrench is the tendency to rotate the pipe about its vertical axis. The magnitude of this tendency depends on both the magnitude F of the force and the effective length d of the wrench handle. Common experience shows that a pull which is not perpendicular to the wrench handle is less effective than the right-angle pull shown.

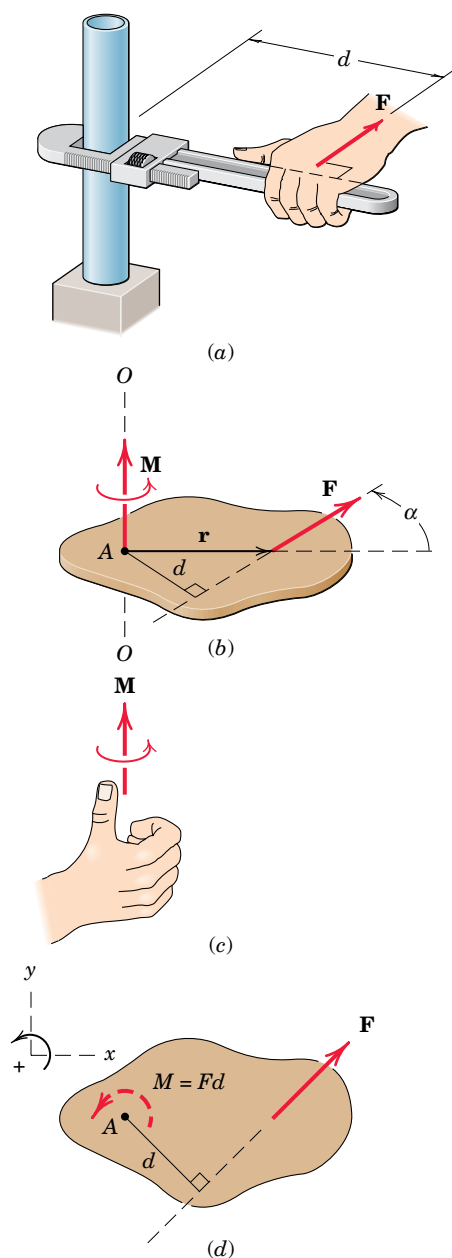


Figure 2/8

Moment about a Point

Figure 2/8b shows a two-dimensional body acted on by a force \mathbf{F} in its plane. The magnitude of the moment or tendency of the force to rotate the body about the axis $O-O$ perpendicular to the plane of the body is proportional both to the magnitude of the force and to the *moment arm* d , which is the perpendicular distance from the axis to the line of action of the force. Therefore, the magnitude of the moment is defined as

$$M = Fd \quad (2/5)$$

The moment is a vector \mathbf{M} perpendicular to the plane of the body. The sense of \mathbf{M} depends on the direction in which \mathbf{F} tends to rotate the body. The right-hand rule, Fig. 2/8c, is used to identify this sense. We represent the moment of \mathbf{F} about $O-O$ as a vector pointing in the direction of the thumb, with the fingers curled in the direction of the rotational tendency.

The moment \mathbf{M} obeys all the rules of vector combination and may be considered a sliding vector with a line of action coinciding with the moment axis. The basic units of moment in SI units are newton-meters ($\text{N}\cdot\text{m}$), and in the U.S. customary system are pound-feet ($\text{lb}\cdot\text{ft}$).

When dealing with forces which all act in a given plane, we customarily speak of the moment *about a point*. By this we mean the moment with respect to an axis normal to the plane and passing through the point. Thus, the moment of force \mathbf{F} about point A in Fig. 2/8d has the magnitude $M = Fd$ and is counterclockwise.

Moment directions may be accounted for by using a stated sign convention, such as a plus sign (+) for counterclockwise moments and a minus sign (−) for clockwise moments, or vice versa. Sign consistency within a given problem is essential. For the sign convention of Fig. 2/8d, the moment of \mathbf{F} about point A (or about the z -axis passing through point A) is positive. The curved arrow of the figure is a convenient way to represent moments in two-dimensional analysis.

The Cross Product

In some two-dimensional and many of the three-dimensional problems to follow, it is convenient to use a vector approach for moment calculations. The moment of \mathbf{F} about point A of Fig. 2/8b may be represented by the cross-product expression

$$\mathbf{M} = \mathbf{r} \times \mathbf{F} \quad (2/6)$$

where \mathbf{r} is a position vector which runs from the moment reference point A to *any* point on the line of action of \mathbf{F} . The magnitude of this expression is given by*

$$M = Fr \sin \alpha = Fd \quad (2/7)$$

which agrees with the moment magnitude as given by Eq. 2/5. Note that the moment arm $d = r \sin \alpha$ does not depend on the particular point on the line of action of \mathbf{F} to which the vector \mathbf{r} is directed. We establish the direction and sense of \mathbf{M} by applying the right-hand rule to the sequence $\mathbf{r} \times \mathbf{F}$. If the fingers of the right hand are curled in the direction of rotation from the positive sense of \mathbf{r} to the positive sense of \mathbf{F} , then the thumb points in the positive sense of \mathbf{M} .

We must maintain the sequence $\mathbf{r} \times \mathbf{F}$, because the sequence $\mathbf{F} \times \mathbf{r}$ would produce a vector with a sense opposite to that of the correct moment. As was the case with the scalar approach, the moment \mathbf{M} may be thought of as the moment about point A or as the moment about the line $O-O$ which passes through point A and is perpendicular to the plane containing the vectors \mathbf{r} and \mathbf{F} . When we evaluate the moment of a force about a given point, the choice between using the vector cross product or the scalar expression depends on how the geometry of the problem is specified. If we know or can easily determine the perpendicular distance between the line of action of the force and the moment center, then the scalar approach is generally simpler. If, however, \mathbf{F} and \mathbf{r} are not perpendicular and are easily expressible in vector notation, then the cross-product expression is often preferable.

In Section B of this chapter, we will see how the vector formulation of the moment of a force is especially useful for determining the moment of a force about a point in three-dimensional situations.

Varignon's Theorem

One of the most useful principles of mechanics is *Varignon's theorem*, which states that the moment of a force about any point is equal to the sum of the moments of the components of the force about the same point.

*See item 7 in Art. C/7 of Appendix C for additional information concerning the cross product.

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To prove this theorem, consider the force \mathbf{R} acting in the plane of the body shown in Fig. 2/9a. The forces \mathbf{P} and \mathbf{Q} represent any two non-rectangular components of \mathbf{R} . The moment of \mathbf{R} about point O is

$$\mathbf{M}_O = \mathbf{r} \times \mathbf{R}$$

Because $\mathbf{R} = \mathbf{P} + \mathbf{Q}$, we may write

$$\mathbf{r} \times \mathbf{R} = \mathbf{r} \times (\mathbf{P} + \mathbf{Q})$$

Using the distributive law for cross products, we have

$$\mathbf{M}_O = \mathbf{r} \times \mathbf{R} = \mathbf{r} \times \mathbf{P} + \mathbf{r} \times \mathbf{Q} \quad (2/8)$$

which says that the moment of \mathbf{R} about O equals the sum of the moments about O of its components \mathbf{P} and \mathbf{Q} . This proves the theorem.

Varignon's theorem need not be restricted to the case of two components, but it applies equally well to three or more. Thus we could have used any number of concurrent components of \mathbf{R} in the foregoing proof.*

Figure 2/9b illustrates the usefulness of Varignon's theorem. The moment of \mathbf{R} about point O is Rd . However, if d is more difficult to determine than p and q , we can resolve \mathbf{R} into the components \mathbf{P} and \mathbf{Q} , and compute the moment as

$$M_O = Rd = -pP + qQ$$

where we take the clockwise moment sense to be positive.

Sample Problem 2/5 shows how Varignon's theorem can help us to calculate moments.

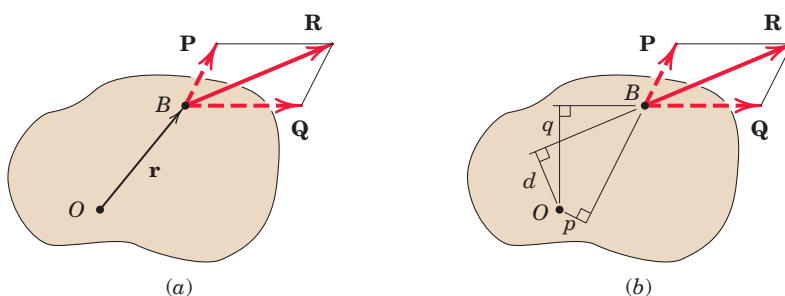


Figure 2/9

*As originally stated, Varignon's theorem was limited to the case of two concurrent components of a given force. See *The Science of Mechanics*, by Ernst Mach, originally published in 1883.

Sample Problem 2/5

Calculate the magnitude of the moment about the base point O of the 600-N force in five different ways.

Solution. (I) The moment arm to the 600-N force is

$$d = 4 \cos 40^\circ + 2 \sin 40^\circ = 4.35 \text{ m}$$

- ① By $M = Fd$ the moment is clockwise and has the magnitude

$$M_O = 600(4.35) = 2610 \text{ N}\cdot\text{m}$$

(II) Replace the force by its rectangular components at A

$$F_1 = 600 \cos 40^\circ = 460 \text{ N}, \quad F_2 = 600 \sin 40^\circ = 386 \text{ N}$$

By Varignon's theorem, the moment becomes

- ② $M_O = 460(4) + 386(2) = 2610 \text{ N}\cdot\text{m}$

(III) By the principle of transmissibility, move the 600-N force along its line of action to point B , which eliminates the moment of the component F_2 . The moment arm of F_1 becomes

$$d_1 = 4 + 2 \tan 40^\circ = 5.68 \text{ m}$$

and the moment is

$$M_O = 460(5.68) = 2610 \text{ N}\cdot\text{m}$$

- ③ (IV) Moving the force to point C eliminates the moment of the component F_1 . The moment arm of F_2 becomes

$$d_2 = 2 + 4 \cot 40^\circ = 6.77 \text{ m}$$

and the moment is

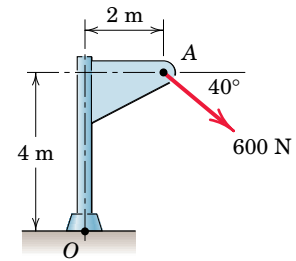
$$M_O = 386(6.77) = 2610 \text{ N}\cdot\text{m}$$

(V) By the vector expression for a moment, and by using the coordinate system indicated on the figure together with the procedures for evaluating cross products, we have

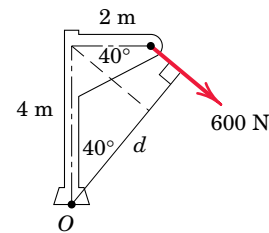
- ④
$$\begin{aligned} \mathbf{M}_O &= \mathbf{r} \times \mathbf{F} = (2\mathbf{i} + 4\mathbf{j}) \times 600(\mathbf{i} \cos 40^\circ - \mathbf{j} \sin 40^\circ) \\ &= -2610\mathbf{k} \text{ N}\cdot\text{m} \end{aligned}$$

The minus sign indicates that the vector is in the negative z -direction. The magnitude of the vector expression is

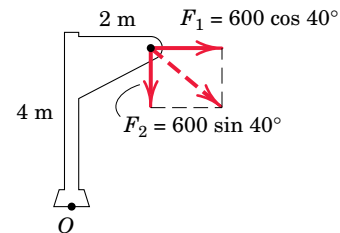
$$M_O = 2610 \text{ N}\cdot\text{m}$$



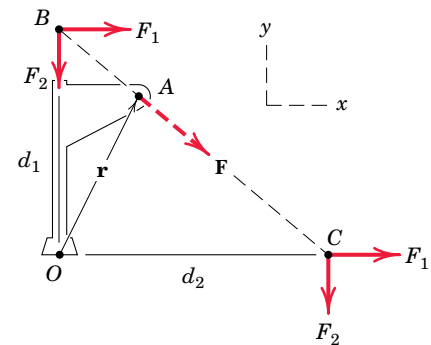
Ans.



Ans.



Ans.



Helpful Hints

- ① The required geometry here and in similar problems should not cause difficulty if the sketch is carefully drawn.
- ② This procedure is frequently the shortest approach.
- ③ The fact that points B and C are not on the body proper should not cause concern, as the mathematical calculation of the moment of a force does not require that the force be on the body.
- ④ Alternative choices for the position vector \mathbf{r} are $\mathbf{r} = d_1\mathbf{j} = 5.68\mathbf{j} \text{ m}$ and $\mathbf{r} = d_2\mathbf{i} = 6.77\mathbf{i} \text{ m}$.

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Sample Problem 2/6

The trap door OA is raised by the cable AB , which passes over the small frictionless guide pulleys at B . The tension everywhere in the cable is T , and this tension applied at A causes a moment M_O about the hinge at O . Plot the quantity M_O/T as a function of the door elevation angle θ over the range $0 \leq \theta \leq 90^\circ$ and note minimum and maximum values. What is the physical significance of this ratio?

Solution. We begin by constructing a figure which shows the tension force \mathbf{T} acting directly on the door, which is shown in an arbitrary angular position θ . It should be clear that the direction of \mathbf{T} will vary as θ varies. In order to deal with this variation, we write a unit vector \mathbf{n}_{AB} which “aims” \mathbf{T} :

$$\textcircled{1} \quad \mathbf{n}_{AB} = \frac{\mathbf{r}_{AB}}{r_{AB}} = \frac{\mathbf{r}_{OB} - \mathbf{r}_{OA}}{r_{AB}}$$

Using the x - y coordinates of our figure, we can write

$$\textcircled{2} \quad \mathbf{r}_{OB} = 0.4\mathbf{j} \text{ m and } \mathbf{r}_{OA} = 0.5(\cos \theta \mathbf{i} + \sin \theta \mathbf{j}) \text{ m}$$

So

$$\begin{aligned} \mathbf{r}_{AB} &= \mathbf{r}_{OB} - \mathbf{r}_{OA} = 0.4\mathbf{j} - (0.5)(\cos \theta \mathbf{i} + \sin \theta \mathbf{j}) \\ &= -0.5 \cos \theta \mathbf{i} + (0.4 - 0.5 \sin \theta)\mathbf{j} \text{ m} \end{aligned}$$

and

$$\begin{aligned} r_{AB} &= \sqrt{(0.5 \cos \theta)^2 + (0.4 - 0.5 \sin \theta)^2} \\ &= \sqrt{0.41 - 0.4 \sin \theta} \text{ m} \end{aligned}$$

The desired unit vector is

$$\mathbf{n}_{AB} = \frac{\mathbf{r}_{AB}}{r_{AB}} = \frac{-0.5 \cos \theta \mathbf{i} + (0.4 - 0.5 \sin \theta)\mathbf{j}}{\sqrt{0.41 - 0.4 \sin \theta}}$$

Our tension vector can now be written as

$$\mathbf{T} = T\mathbf{n}_{AB} = T \left[\frac{-0.5 \cos \theta \mathbf{i} + (0.4 - 0.5 \sin \theta)\mathbf{j}}{\sqrt{0.41 - 0.4 \sin \theta}} \right]$$

$\textcircled{3}$ The moment of \mathbf{T} about point O , as a vector, is $\mathbf{M}_O = \mathbf{r}_{OB} \times \mathbf{T}$, where $\mathbf{r}_{OB} = 0.4\mathbf{j}$ m, or

$$\begin{aligned} \mathbf{M}_O &= 0.4\mathbf{j} \times T \left[\frac{-0.5 \cos \theta \mathbf{i} + (0.4 - 0.5 \sin \theta)\mathbf{j}}{\sqrt{0.41 - 0.4 \sin \theta}} \right] \\ &= \frac{0.2T \cos \theta}{\sqrt{0.41 - 0.4 \sin \theta}} \mathbf{k} \end{aligned}$$

The magnitude of \mathbf{M}_O is

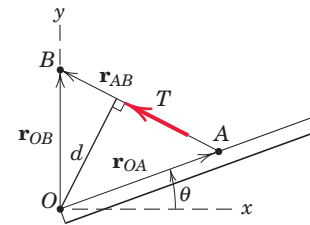
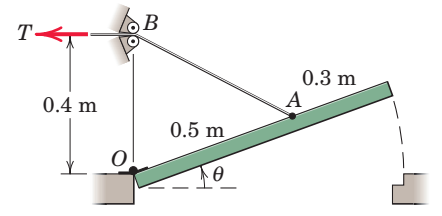
$$M_O = \frac{0.2T \cos \theta}{\sqrt{0.41 - 0.4 \sin \theta}}$$

and the requested ratio is

$$\frac{M_O}{T} = \frac{0.2 \cos \theta}{\sqrt{0.41 - 0.4 \sin \theta}} \quad \text{Ans.}$$

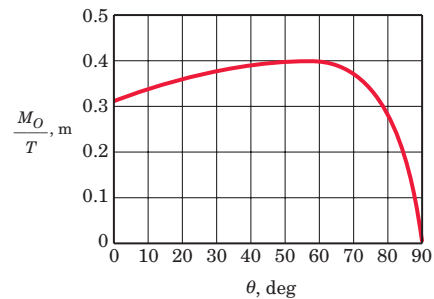
which is plotted in the accompanying graph. The expression M_O/T is the moment arm d (in meters) which runs from O to the line of action of \mathbf{T} . It has a maximum value of 0.4 m at $\theta = 53.1^\circ$ (at which point \mathbf{T} is horizontal) and a minimum value of 0 at $\theta = 90^\circ$ (at which point \mathbf{T} is vertical). The expression is valid even if T varies.

This sample problem treats moments in two-dimensional force systems, and it also points out the advantages of carrying out a solution for an arbitrary position, so that behavior over a range of positions can be examined.



Helpful Hints

$\textcircled{1}$ Recall that any unit vector can be written as a vector divided by its magnitude. In this case the vector in the numerator is a position vector.



$\textcircled{2}$ Recall that any vector may be written as a magnitude times an “aiming” unit vector.

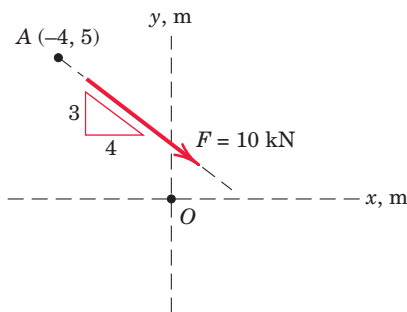
$\textcircled{3}$ In the expression $\mathbf{M} = \mathbf{r} \times \mathbf{F}$, the position vector \mathbf{r} runs from the moment center to any point on the line of action of \mathbf{F} . Here, \mathbf{r}_{OB} is more convenient than \mathbf{r}_{OA} .

PROBLEMS

Introductory Problems

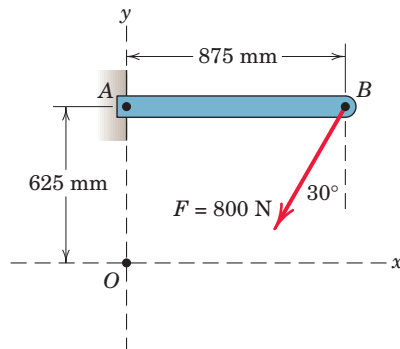
- 2/29** The 10-kN force is applied at point A. Determine the moment of \mathbf{F} about point O. Determine the points on the x - and y -axes about which the moment of \mathbf{F} is zero.

Ans. $M_O = 16 \text{ kN} \cdot \text{m}$ CW
 $(x, y) = (2.67, 0) \text{ m}$ and $(0, 2) \text{ m}$



Problem 2/29

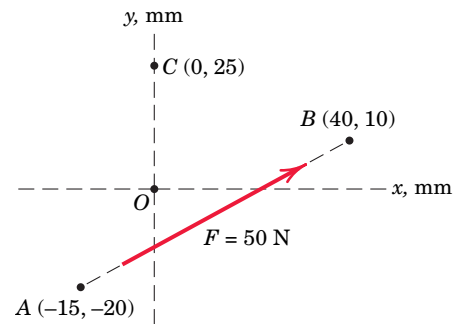
- 2/30** Determine the moment of the 800-N force about point A and about point O.



Problem 2/30

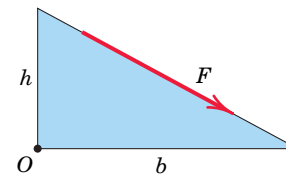
- 2/31** Determine the moment of the 50-N force (a) about point O by Varignon's theorem and (b) about point C by a vector approach.

Ans. $M_O = 519 \text{ N} \cdot \text{mm}$ CCW, $M_C = 1616 \text{ N} \cdot \text{mm}$



Problem 2/31

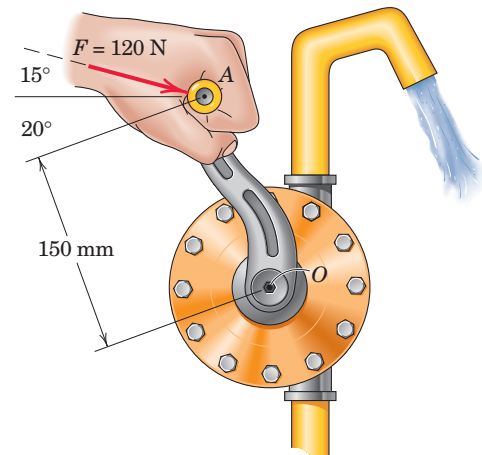
- 2/32** The force of magnitude F acts along the edge of the triangular plate. Determine the moment of \mathbf{F} about point O.



Problem 2/32

- 2/33** In steadily turning the water pump, a person exerts the 120-N force on the handle as shown. Determine the moment of this force about point O.

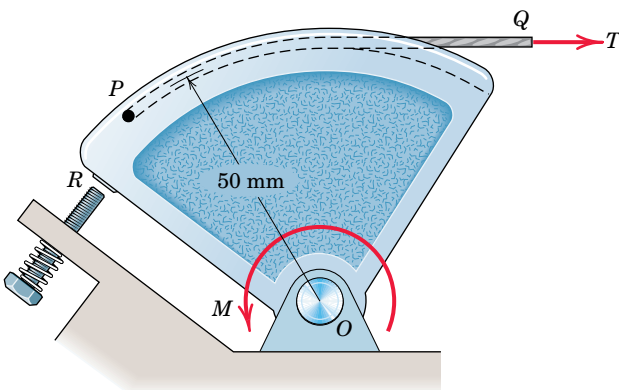
Ans. $M_O = 14.74 \text{ N} \cdot \text{m}$ CW



Problem 2/33

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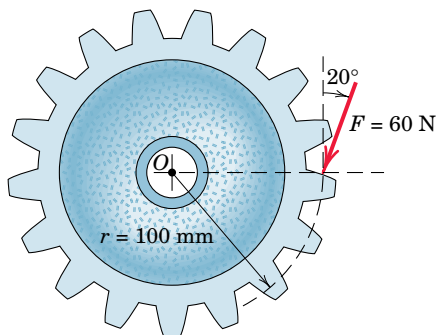
- 2/34** The throttle-control sector pivots freely at O . If an internal torsional spring exerts a return moment $M = 1.8 \text{ N}\cdot\text{m}$ on the sector when in the position shown, for design purposes determine the necessary throttle-cable tension T so that the net moment about O is zero. Note that when T is zero, the sector rests against the idle-control adjustment screw at R .



Problem 2/34

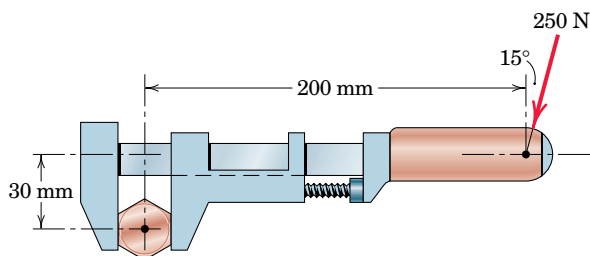
- 2/35** A force \mathbf{F} of magnitude 60 N is applied to the gear. Determine the moment of \mathbf{F} about point O .

Ans. $M_O = 5.64 \text{ N}\cdot\text{m}$ CW



Problem 2/35

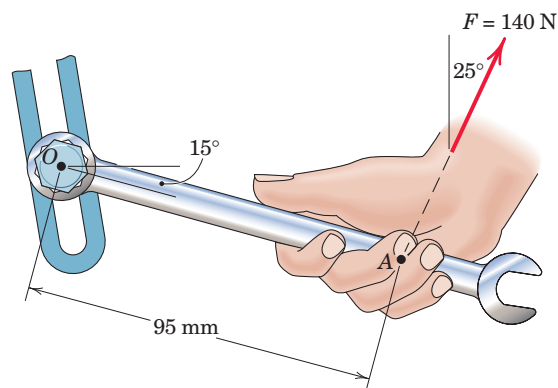
- 2/36** Calculate the moment of the 250-N force on the handle of the monkey wrench about the center of the bolt.



Problem 2/36

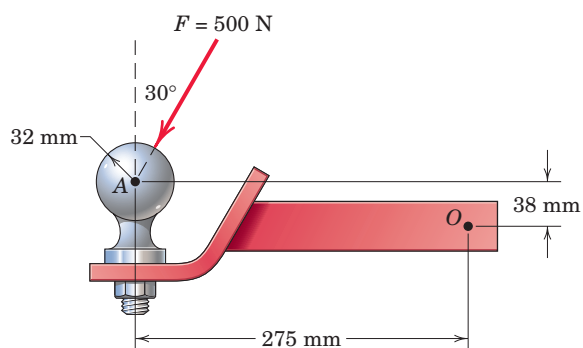
- 2/37** A mechanic pulls on the 13-mm combination wrench with the 140-N force shown. Determine the moment of this force about the bolt center O .

Ans. $M_O = 13.10 \text{ N}\cdot\text{m}$ CCW



Problem 2/37

- 2/38** As a trailer is towed in the forward direction, the force $F = 500 \text{ N}$ is applied as shown to the ball of the trailer hitch. Determine the moment of this force about point O .

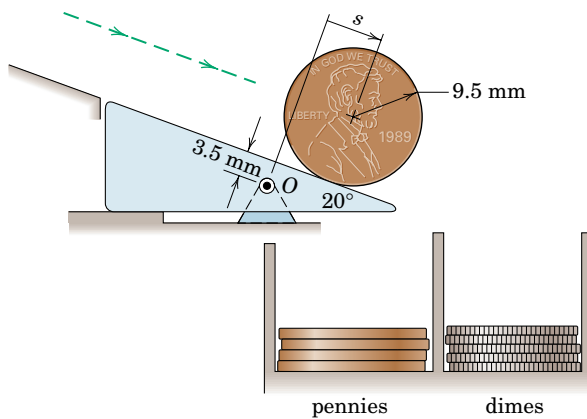


Problem 2/38

Representative Problems

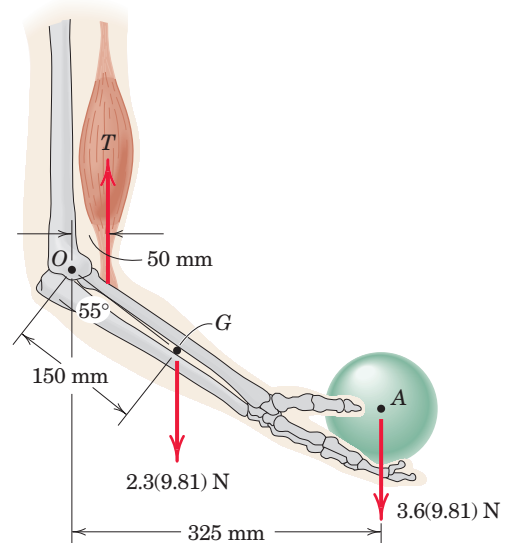
- 2/39** A portion of a mechanical coin sorter works as follows: Pennies and dimes roll down the 20° incline, the last triangular portion of which pivots freely about a horizontal axis through O . Dimes are light enough (2.28 grams each) so that the triangular portion remains stationary, and the dimes roll into the right collection column. Pennies, on the other hand, are heavy enough (3.06 grams each) so that the triangular portion pivots clockwise, and the pennies roll into the left collection column. Determine the moment about O of the weight of the penny in terms of the slant distance s in millimeters.

Ans. $M_O = 0.1335 + 0.0282s \text{ N}\cdot\text{mm}$ (s in mm)



Problem 2/39

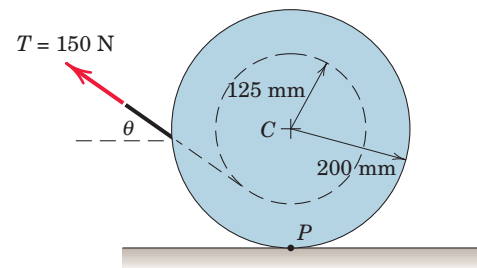
- 2/40** Elements of the lower arm are shown in the figure. The mass of the forearm is 2.3 kg with mass center at G . Determine the combined moment about the elbow pivot O of the weights of the forearm and the 3.6-kg homogeneous sphere. What must the biceps tension force be so that the overall moment about O is zero?



Problem 2/40

- 2/41** A 150-N pull T is applied to a cord, which is wound securely around the inner hub of the drum. Determine the moment of T about the drum center C . At what angle θ should T be applied so that the moment about the contact point P is zero?

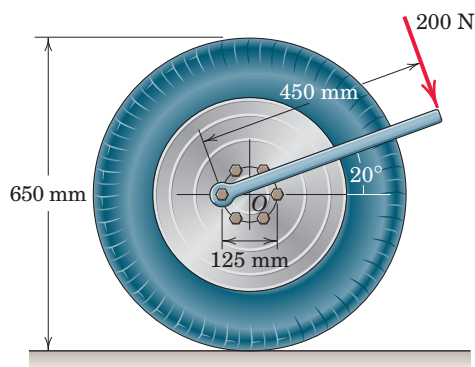
Ans. $M_C = 18.75 \text{ N}\cdot\text{m}$ CW, $\theta = 51.3^\circ$



Problem 2/41

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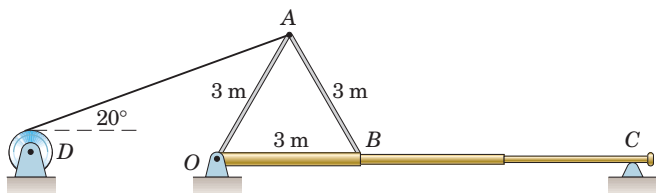
- 2/42** A force of 200 N is applied to the end of the wrench to tighten a flange bolt which holds the wheel to the axle. Determine the moment M produced by this force about the center O of the wheel for the position of the wrench shown.



Problem 2/42

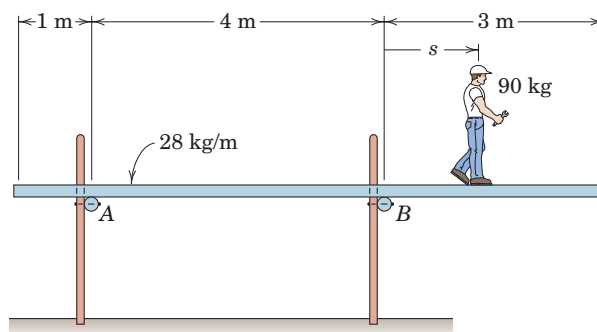
- 2/43** In order to raise the flagpole OC , a light frame OAB is attached to the pole and a tension of 3.2 kN is developed in the hoisting cable by the power winch D . Calculate the moment M_O of this tension about the hinge point O .

Ans. $M_O = 6.17 \text{ kN} \cdot \text{m}$ CCW



Problem 2/43

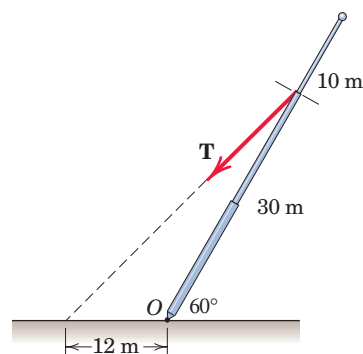
- 2/44** The uniform work platform, which has a mass per unit length of 28 kg/m, is simply supported by cross rods A and B . The 90-kg construction worker starts from point B and walks to the right. At what location s will the combined moment of the weights of the man and platform about point B be zero?



Problem 2/44

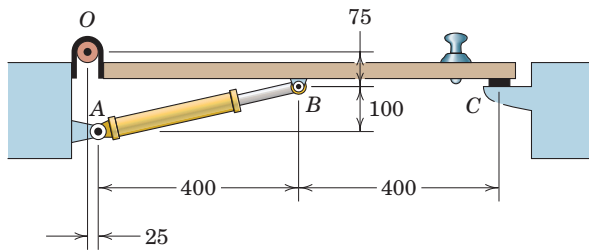
- 2/45** In raising the pole from the position shown, the tension T in the cable must supply a moment about O of $72 \text{ kN} \cdot \text{m}$. Determine T .

Ans. $T = 8.65 \text{ kN}$



Problem 2/45

- 2/46** The force exerted by the plunger of cylinder AB on the door is 40 N directed along the line AB , and this force tends to keep the door closed. Compute the moment of this force about the hinge O . What force F_C normal to the plane of the door must the door stop at C exert on the door so that the combined moment about O of the two forces is zero?

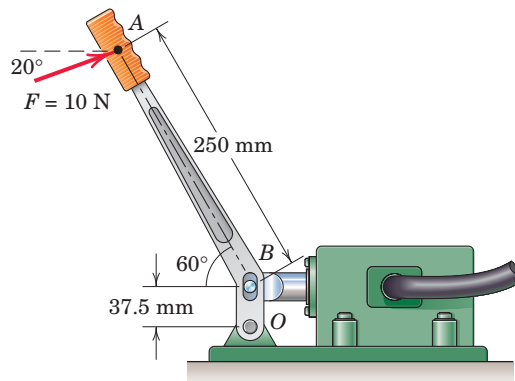


Dimensions in millimeters

Problem 2/46

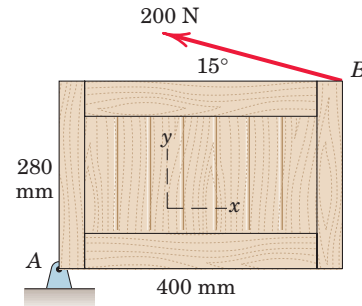
- 2/47** The 10-N force is applied to the handle of the hydraulic control valve as shown. Calculate the moment of this force about point O .

Ans. $M_O = 2.81 \text{ N} \cdot \text{m}$ CW



Problem 2/47

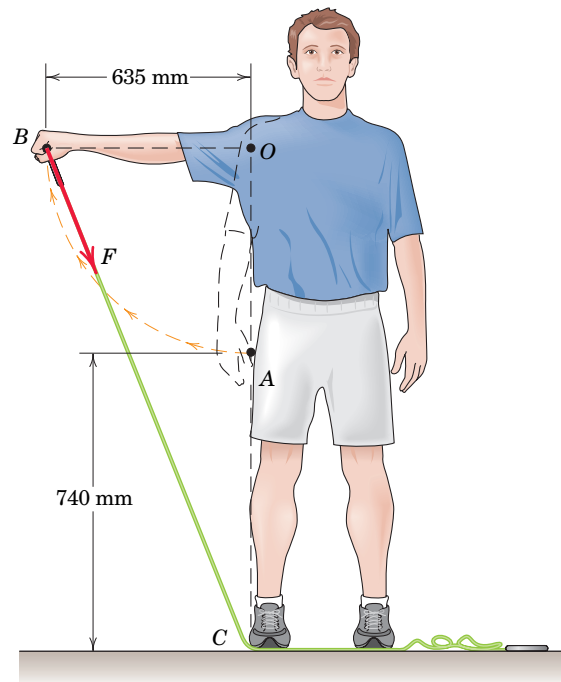
- 2/48** Calculate the moment M_A of the 200-N force about point A by using three scalar methods and one vector method.



Problem 2/48

- 2/49** An exerciser begins with his arm in the relaxed vertical position OA , at which the elastic band is unstretched. He then rotates his arm to the horizontal position OB . The elastic modulus of the band is $k = 60 \text{ N/m}$ —that is, 60 N of force is required to stretch the band each additional meter of elongation. Determine the moment about O of the force which the band exerts on the hand B .

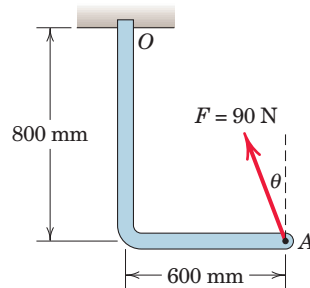
Ans. $M_O = 26.8 \text{ N} \cdot \text{m}$ CCW



Problem 2/49

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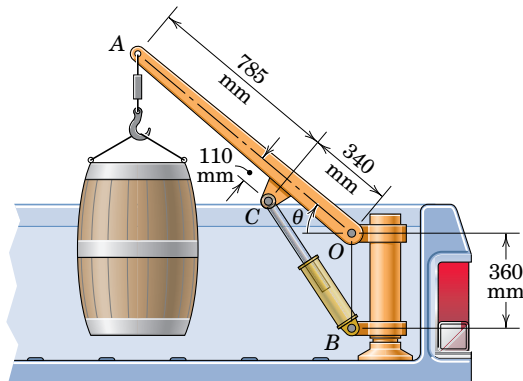
- 2/50** (a) Calculate the moment of the 90-N force about point O for the condition $\theta = 15^\circ$. Also, determine the value of θ for which the moment about O is (b) zero and (c) a maximum.



Problem 2/50

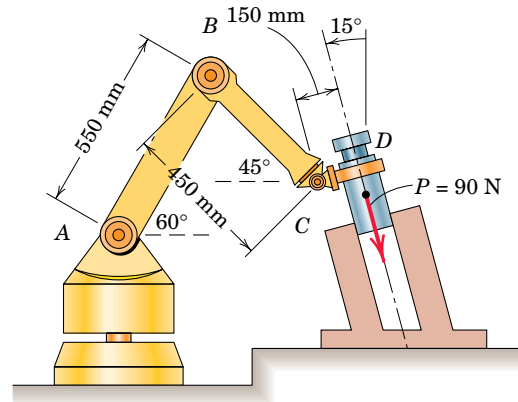
- 2/51** The small crane is mounted along the side of a pickup bed and facilitates the handling of heavy loads. When the boom elevation angle is $\theta = 40^\circ$, the force in the hydraulic cylinder BC is 4.5 kN, and this force applied at point C is in the direction from B to C (the cylinder is in compression). Determine the moment of this 4.5-kN force about the boom pivot point O .

Ans. $M_O = 0.902 \text{ kN} \cdot \text{m CW}$



Problem 2/51

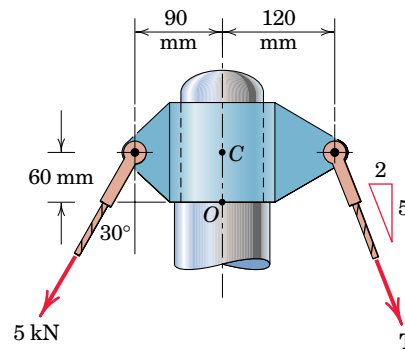
- 2/52** Design criteria require that the robot exert the 90-N force on the part as shown while inserting a cylindrical part into the circular hole. Determine the moment about points A , B , and C of the force which the part exerts on the robot.



Problem 2/52

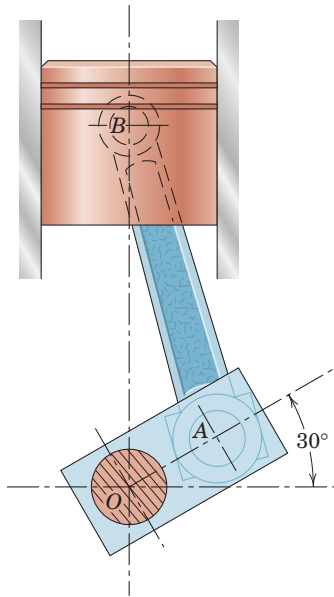
- 2/53** The masthead fitting supports the two forces shown. Determine the magnitude of T which will cause no bending of the mast (zero moment) at point O .

Ans. $T = 4.04 \text{ kN}$



Problem 2/53

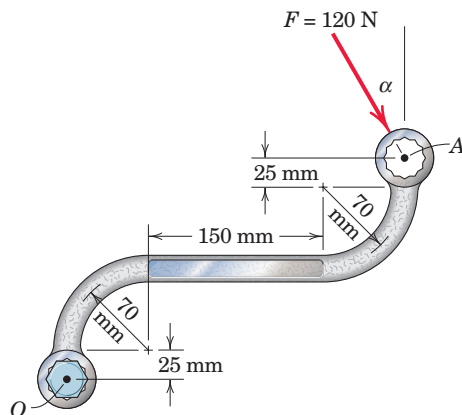
- 2/54** The piston, connecting rod, and crankshaft of a diesel engine are shown in the figure. The crank throw OA is half the stroke of 200 mm, and the length AB of the rod is 350 mm. For the position indicated, the rod is under a compression along AB of 16 kN. Determine the moment M of this force about the crankshaft axis O .



Problem 2/54

- 2/55** The 120-N force is applied as shown to one end of the curved wrench. If $\alpha = 30^\circ$, calculate the moment of F about the center O of the bolt. Determine the value of α which would maximize the moment about O ; state the value of this maximum moment.

Ans. $M_O = 41.5 \text{ N}\cdot\text{m}$ CW
 $\alpha = 33.2^\circ$, $(M_O)_{\max} = 41.6 \text{ N}\cdot\text{m}$ CW

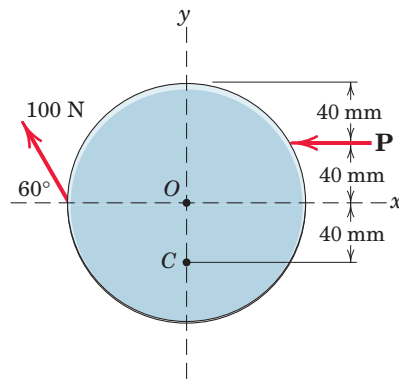


Problem 2/55

- 2/56** If the combined moment of the two forces about point C is zero, determine

- the magnitude of the force \mathbf{P}
- the magnitude R of the resultant of the two forces
- the coordinates x and y of the point A on the rim of the wheel about which the combined moment of the two forces is a maximum
- the combined moment M_A of the two forces about A .

Ans. (a) $P = 61.6 \text{ N}$
 (b) $R = 141.3 \text{ N}$
 (c) $x = 49.0 \text{ mm}$, $y = 63.2 \text{ mm}$
 (d) $M_A = 15.77 \text{ N}\cdot\text{m}$ CW



Problem 2/56

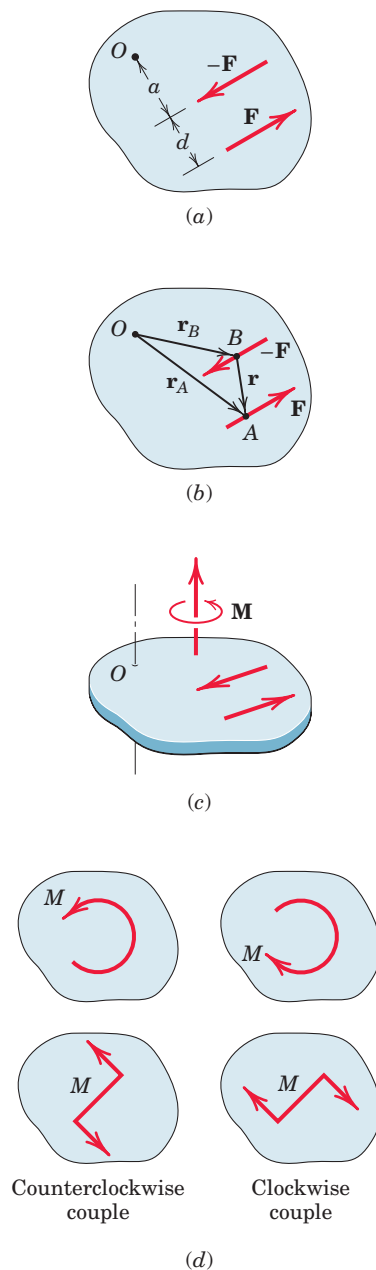


Figure 2/10

2/5 COUPLE

The moment produced by two equal, opposite, and noncollinear forces is called a *couple*. Couples have certain unique properties and have important applications in mechanics.

Consider the action of two equal and opposite forces \mathbf{F} and $-\mathbf{F}$ a distance d apart, as shown in Fig. 2/10a. These two forces cannot be combined into a single force because their sum in every direction is zero. Their only effect is to produce a tendency of rotation. The combined moment of the two forces about an axis normal to their plane and passing through any point such as O in their plane is the couple \mathbf{M} . This couple has a magnitude

$$M = F(a + d) - Fa$$

or

$$M = Fd$$

Its direction is counterclockwise when viewed from above for the case illustrated. Note especially that the magnitude of the couple is independent of the distance a which locates the forces with respect to the moment center O . It follows that the moment of a couple has the same value for all moment centers.

Vector Algebra Method

We may also express the moment of a couple by using vector algebra. With the cross-product notation of Eq. 2/6, the combined moment about point O of the forces forming the couple of Fig. 2/10b is

$$\mathbf{M} = \mathbf{r}_A \times \mathbf{F} + \mathbf{r}_B \times (-\mathbf{F}) = (\mathbf{r}_A - \mathbf{r}_B) \times \mathbf{F}$$

where \mathbf{r}_A and \mathbf{r}_B are position vectors which run from point O to arbitrary points A and B on the lines of action of \mathbf{F} and $-\mathbf{F}$, respectively. Because $\mathbf{r}_A - \mathbf{r}_B = \mathbf{r}$, we can express \mathbf{M} as

$$\mathbf{M} = \mathbf{r} \times \mathbf{F}$$

Here again, the moment expression contains no reference to the moment center O and, therefore, is the same for all moment centers. Thus, we may represent \mathbf{M} by a free vector, as shown in Fig. 2/10c, where the direction of \mathbf{M} is normal to the plane of the couple and the sense of \mathbf{M} is established by the right-hand rule.

Because the couple vector \mathbf{M} is always perpendicular to the plane of the forces which constitute the couple, in two-dimensional analysis we can represent the sense of a couple vector as clockwise or counterclockwise by one of the conventions shown in Fig. 2/10d. Later, when we deal with couple vectors in three-dimensional problems, we will make full use of vector notation to represent them, and the mathematics will automatically account for their sense.

Equivalent Couples

Changing the values of F and d does not change a given couple as long as the product Fd remains the same. Likewise, a couple is not affected if the forces act in a different but parallel plane. Figure 2/11

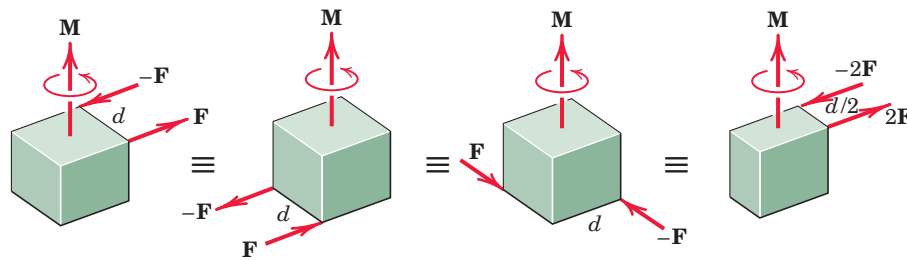


Figure 2/11

shows four different configurations of the same couple \mathbf{M} . In each of the four cases, the couples are equivalent and are described by the same free vector which represents the identical tendencies to rotate the bodies.

Force–Couple Systems

The effect of a force acting on a body is the tendency to push or pull the body in the direction of the force, and to rotate the body about any fixed axis which does not intersect the line of the force. We can represent this dual effect more easily by replacing the given force by an equal parallel force and a couple to compensate for the change in the moment of the force.

The replacement of a force by a force and a couple is illustrated in Fig. 2/12, where the given force \mathbf{F} acting at point A is replaced by an equal force \mathbf{F} at some point B and the counterclockwise couple $M = Fd$. The transfer is seen in the middle figure, where the equal and opposite forces \mathbf{F} and $-\mathbf{F}$ are added at point B without introducing any net external effects on the body. We now see that the original force at A and the equal and opposite one at B constitute the couple $M = Fd$, which is counterclockwise for the sample chosen, as shown in the right-hand part of the figure. Thus, we have replaced the original force at A by the same force acting at a different point B and a couple, without altering the external effects of the original force on the body. The combination of the force and couple in the right-hand part of Fig. 2/12 is referred to as a *force–couple system*.

By reversing this process, we can combine a given couple and a force which lies in the plane of the couple (normal to the couple vector) to produce a single, equivalent force. Replacement of a force by an equivalent force–couple system, and the reverse procedure, have many applications in mechanics and should be mastered.

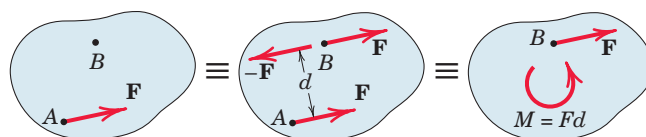


Figure 2/12

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Sample Problem 2/7

The rigid structural member is subjected to a couple consisting of the two 100-N forces. Replace this couple by an equivalent couple consisting of the two forces \mathbf{P} and $-\mathbf{P}$, each of which has a magnitude of 400 N. Determine the proper angle θ .

Solution. The original couple is counterclockwise when the plane of the forces is viewed from above, and its magnitude is

$$[M = Fd] \quad M = 100(0.1) = 10 \text{ N}\cdot\text{m}$$

The forces \mathbf{P} and $-\mathbf{P}$ produce a counterclockwise couple

$$M = 400(0.040) \cos \theta$$

- ① Equating the two expressions gives

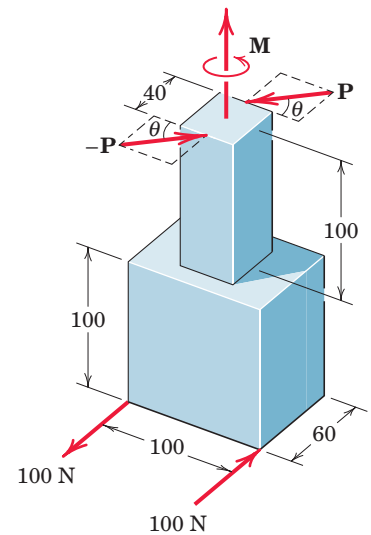
$$10 = (400)(0.040) \cos \theta$$

$$\theta = \cos^{-1} \frac{10}{16} = 51.3^\circ$$

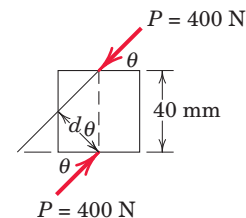
Ans.

Helpful Hint

- ① Since the two equal couples are parallel free vectors, the only dimensions which are relevant are those which give the perpendicular distances between the forces of the couples.



Dimensions in millimeters



Sample Problem 2/8

Replace the horizontal 400-N force acting on the lever by an equivalent system consisting of a force at O and a couple.

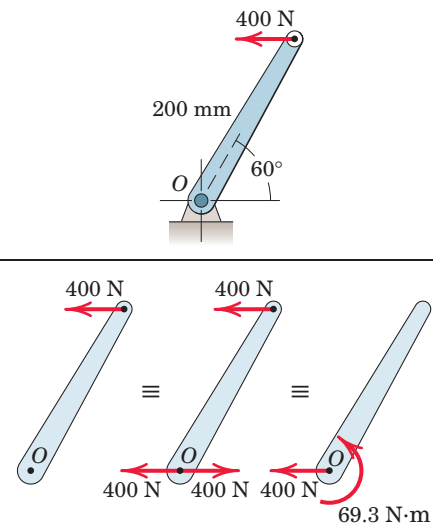
Solution. We apply two equal and opposite 400-N forces at O and identify the counterclockwise couple

$$[M = Fd] \quad M = 400(0.200 \sin 60^\circ) = 69.3 \text{ N}\cdot\text{m} \quad \text{Ans.}$$

- ① Thus, the original force is equivalent to the 400-N force at O and the 69.3-N·m couple as shown in the third of the three equivalent figures.

Helpful Hint

- ① The reverse of this problem is often encountered, namely, the replacement of a force and a couple by a single force. Proceeding in reverse is the same as replacing the couple by two forces, one of which is equal and opposite to the 400-N force at O . The moment arm to the second force would be $M/F = 69.3/400 = 0.1732 \text{ m}$, which is $0.2 \sin 60^\circ$, thus determining the line of action of the single resultant force of 400 N.

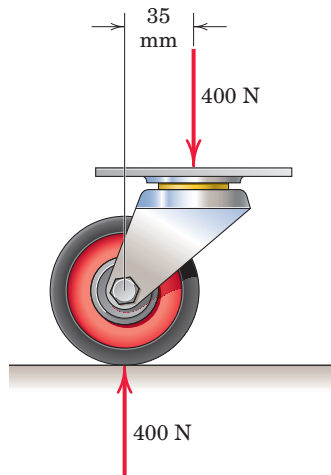


PROBLEMS

Introductory Problems

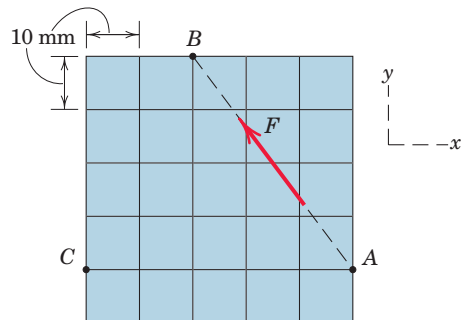
- 2/57** The caster unit is subjected to the pair of 400-N forces shown. Determine the moment associated with these forces.

Ans. $M = 14 \text{ N}\cdot\text{m}$ CW



Problem 2/57

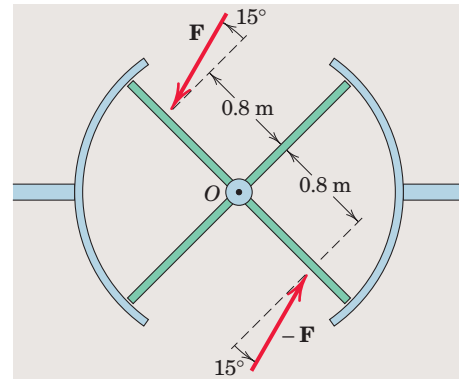
- 2/58** A force $F = 60 \text{ N}$ acts along the line AB . Determine the equivalent force-couple system at point C .



Problem 2/58

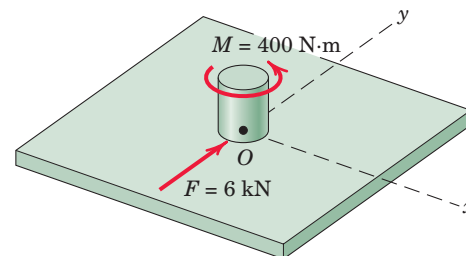
- 2/59** The top view of a revolving entrance door is shown. Two persons simultaneously approach the door and exert force of equal magnitudes as shown. If the resulting moment about the door pivot axis at O is $25 \text{ N}\cdot\text{m}$, determine the force magnitude F .

Ans. $F = 16.18 \text{ N}$



Problem 2/59

- 2/60** The indicated force-couple system is applied to a small shaft at the center of the plate. Replace this system by a single force and specify the coordinate of the point on the x -axis through which the line of action of this resultant force passes.

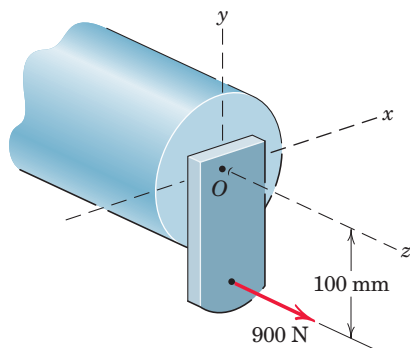


Problem 2/60

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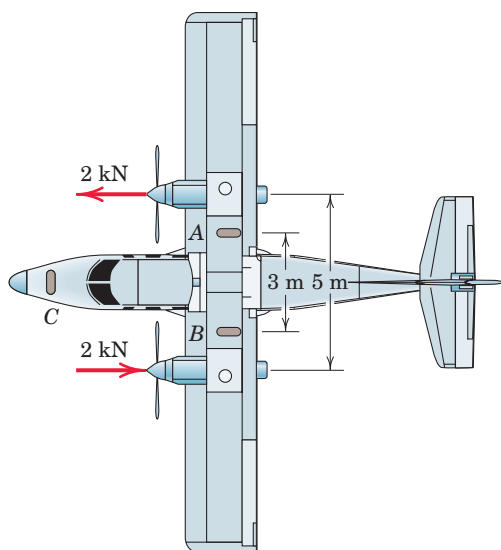
- 2/61** The bracket is spot welded to the end of the shaft at point O . To show the effect of the 900-N force on the weld, replace the force by its equivalent of a force and couple \mathbf{M} at O . Express \mathbf{M} in vector notation.

Ans. $\mathbf{M} = -90\mathbf{i} \text{ N}\cdot\text{m}$



Problem 2/61

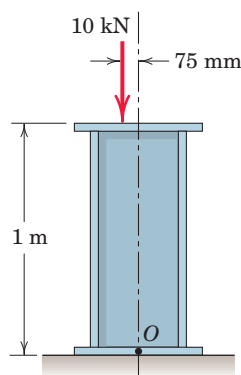
- 2/62** As part of a test, the two aircraft engines are revved up and the propeller pitches are adjusted so as to result in the fore and aft thrusts shown. What force F must be exerted by the ground on each of the main braked wheels at A and B to counteract the turning effect of the two propeller thrusts? Neglect any effects of the nose wheel C , which is turned 90° and unbraked.



Problem 2/62

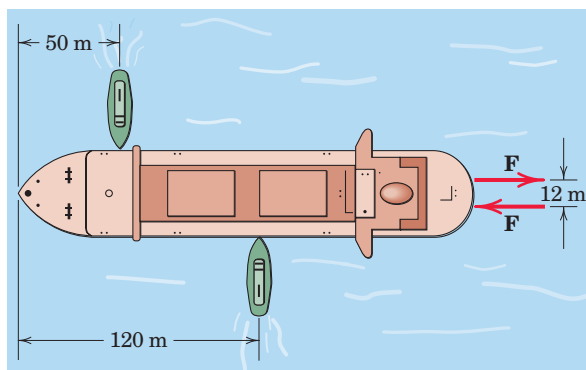
- 2/63** Replace the 10-kN force acting on the steel column by an equivalent force-couple system at point O . This replacement is frequently done in the design of structures.

Ans. $R = 10 \text{ kN}$, $M_O = 0.75 \text{ kN}\cdot\text{m}$ CCW



Problem 2/63

- 2/64** Each propeller of the twin-screw ship develops a full-speed thrust of 300 kN. In maneuvering the ship, one propeller is turning full speed ahead and the other full speed in reverse. What thrust P must each tug exert on the ship to counteract the effect of the ship's propellers?

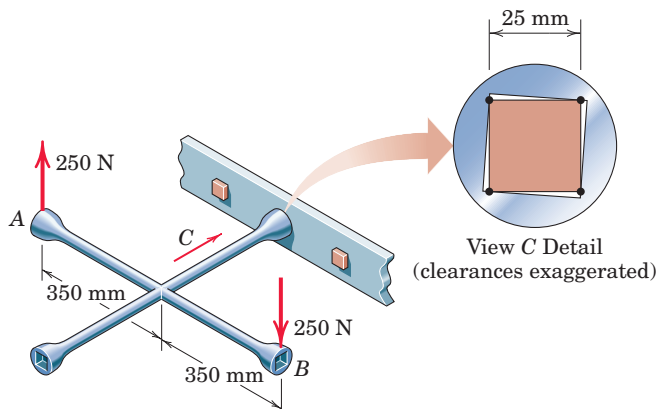


Problem 2/64

Representative Problems

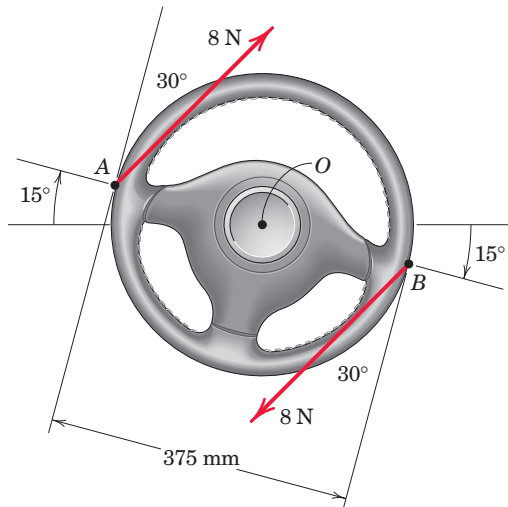
- 2/65** A lug wrench is used to tighten a square-head bolt. If 250-N forces are applied to the wrench as shown, determine the magnitude F of the equal forces exerted on the four contact points on the 25-mm bolt head so that their external effect on the bolt is equivalent to that of the two 250-N forces. Assume that the forces are perpendicular to the flats of the bolt head.

Ans. $F = 3500 \text{ N}$



Problem 2/65

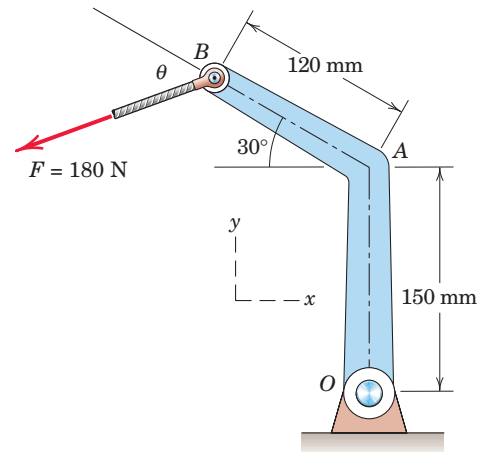
- 2/66** During a steady right turn, a person exerts the forces shown on the steering wheel. Note that each force consists of a tangential component and a radially-inward component. Determine the moment exerted about the steering column at O .



Problem 2/66

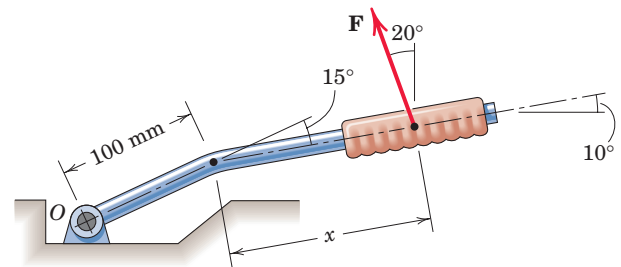
- 2/67** The 180-N force is applied to the end of body OAB . If $\theta = 50^\circ$, determine the equivalent force-couple system at the shaft axis O .

Ans. $\mathbf{F} = -169.1\mathbf{i} - 61.6\mathbf{j} \text{ N}$, $M_O = 41.9 \text{ N}\cdot\text{m CCW}$



Problem 2/67

- 2/68** A force \mathbf{F} of magnitude 50 N is exerted on the automobile parking-brake lever at the position $x = 250 \text{ mm}$. Replace the force by an equivalent force-couple system at the pivot point O .

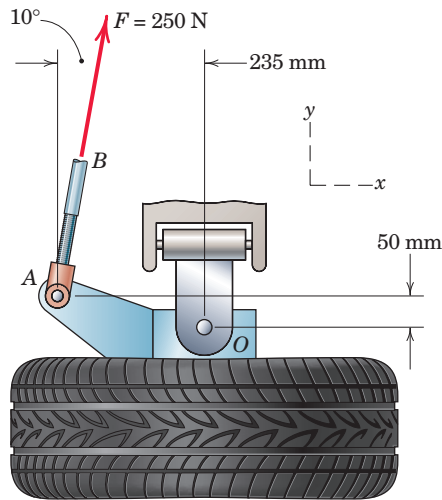


Problem 2/68

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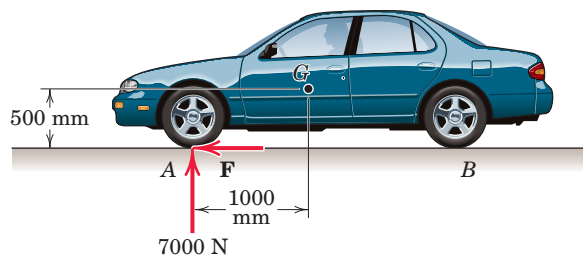
- 2/69** The tie-rod AB exerts the 250-N force on the steering knuckle AO as shown. Replace this force by an equivalent force-couple system at O .

Ans. $\mathbf{F} = 43.4\mathbf{i} + 246\mathbf{j}$ N, $M_O = 60.0$ N·m CW



Problem 2/69

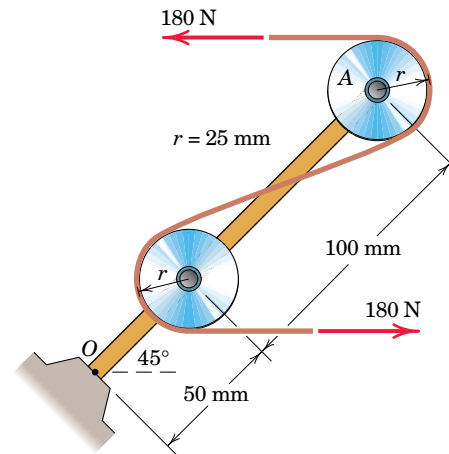
- 2/70** The combined drive wheels of a front-wheel-drive automobile are acted on by a 7000-N normal reaction force and a friction force \mathbf{F} , both of which are exerted by the road surface. If it is known that the resultant of these two forces makes a 15° angle with the vertical, determine the equivalent force-couple system at the car mass center G . Treat this as a two-dimensional problem.



Problem 2/70

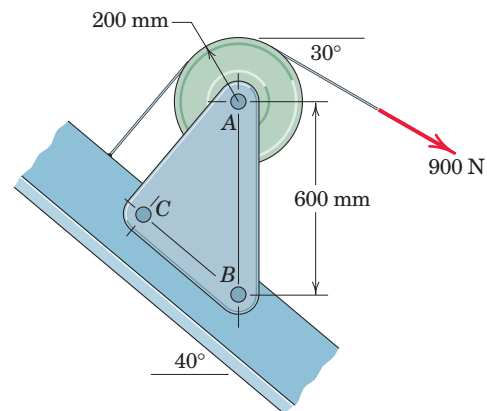
- 2/71** The system consisting of the bar OA , two identical pulleys, and a section of thin tape is subjected to the two 180-N tensile forces shown in the figure. Determine the equivalent force-couple system at point O .

Ans. $M = 21.7$ N·m CCW



Problem 2/71

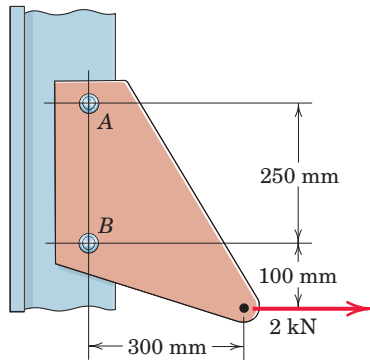
- 2/72** Calculate the moment M_B of the 900-N force about the bolt at B . Simplify your work by first replacing the force by its equivalent force-couple system at A .



Problem 2/72

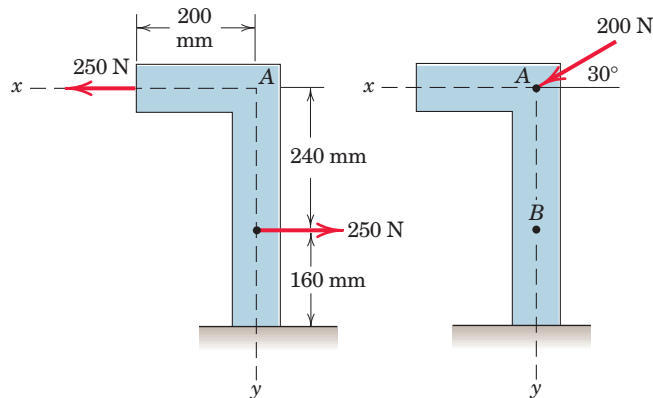
- 2/73** The bracket is fastened to the girder by means of the two rivets A and B and supports the 2-kN force. Replace this force by a force acting along the centerline between the rivets and a couple. Then redistribute this force and couple by replacing it by two forces, one at A and the other at B , and ascertain the forces supported by the rivets.

Ans. $F_A = 0.8 \text{ kN}$
 $F_B = 2.8 \text{ kN}$



Problem 2/73

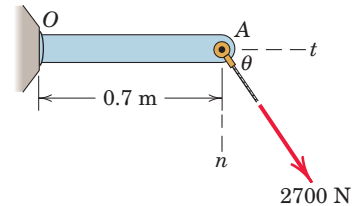
- 2/74** The angle plate is subjected to the two 250-N forces shown. It is desired to replace these forces by an equivalent set consisting of the 200-N force applied at A and a second force applied at B . Determine the y -coordinate of B .



Problem 2/74

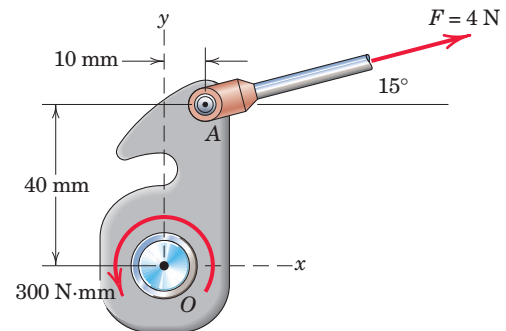
- 2/75** The weld at O can support a maximum of 2500 N of force along each of the n - and t -directions and a maximum of 1400 N·m of moment. Determine the allowable range for the direction θ of the 2700-N force applied at A . The angle θ is restricted to $0 \leq \theta \leq 90^\circ$.

Ans. $22.2^\circ \leq \theta \leq 47.8^\circ$



Problem 2/75

- 2/76** The device shown is a part of an automobile seat-back-release mechanism. The part is subjected to the 4-N force exerted at A and a 300-N·mm restoring moment exerted by a hidden torsional spring. Determine the y -intercept of the line of action of the single equivalent force.



Problem 2/76

2/6 RESULTANTS

The properties of force, moment, and couple were developed in the previous four articles. Now we are ready to describe the resultant action of a group or *system* of forces. Most problems in mechanics deal with a system of forces, and it is usually necessary to reduce the system to its simplest form to describe its action. The *resultant* of a system of forces is the simplest force combination which can replace the original forces without altering the external effect on the rigid body to which the forces are applied.

Equilibrium of a body is the condition in which the resultant of all forces acting on the body is zero. This condition is studied in statics. When the resultant of all forces on a body is not zero, the acceleration of the body is obtained by equating the force resultant to the product of the mass and acceleration of the body. This condition is studied in dynamics. Thus, the determination of resultants is basic to both statics and dynamics.

The most common type of force system occurs when the forces all act in a single plane, say, the x - y plane, as illustrated by the system of three forces \mathbf{F}_1 , \mathbf{F}_2 , and \mathbf{F}_3 in Fig. 2/13a. We obtain the magnitude and direction of the resultant force \mathbf{R} by forming the *force polygon* shown in part b of the figure, where the forces are added head-to-tail in any sequence. Thus, for any system of coplanar forces we may write

$$\begin{aligned}\mathbf{R} &= \mathbf{F}_1 + \mathbf{F}_2 + \mathbf{F}_3 + \cdots = \Sigma \mathbf{F} \\ R_x &= \Sigma F_x \quad R_y = \Sigma F_y \quad R = \sqrt{(\Sigma F_x)^2 + (\Sigma F_y)^2} \\ \theta &= \tan^{-1} \frac{R_y}{R_x} = \tan^{-1} \frac{\Sigma F_y}{\Sigma F_x}\end{aligned}\quad (2/9)$$

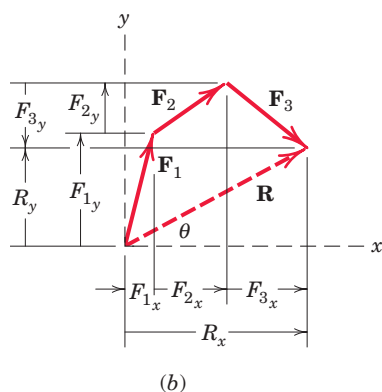
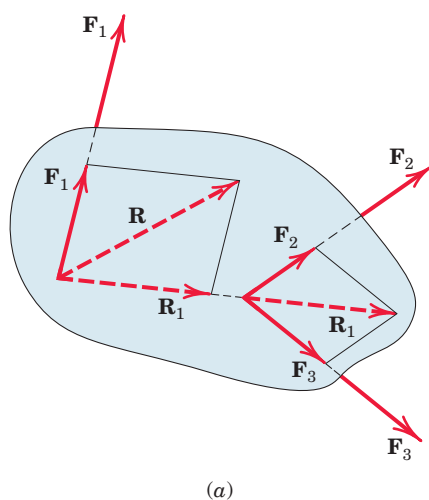


Figure 2/13

Graphically, the correct line of action of \mathbf{R} may be obtained by preserving the correct lines of action of the forces and adding them by the parallelogram law. We see this in part a of the figure for the case of three forces where the sum \mathbf{R}_1 of \mathbf{F}_2 and \mathbf{F}_3 is added to \mathbf{F}_1 to obtain \mathbf{R} . The principle of transmissibility has been used in this process.

Algebraic Method

We can use algebra to obtain the resultant force and its line of action as follows:

1. Choose a convenient reference point and move all forces to that point. This process is depicted for a three-force system in Figs. 2/14a and b , where M_1 , M_2 , and M_3 are the couples resulting from the transfer of forces \mathbf{F}_1 , \mathbf{F}_2 , and \mathbf{F}_3 from their respective original lines of action to lines of action through point O .
2. Add all forces at O to form the resultant force \mathbf{R} , and add all couples to form the resultant couple M_O . We now have the single force-couple system, as shown in Fig. 2/14c.
3. In Fig. 2/14d, find the line of action of \mathbf{R} by requiring \mathbf{R} to have a moment of M_O about point O . Note that the force systems of Figs. 2/14a and 2/14d are equivalent, and that $\Sigma(Fd)$ in Fig. 2/14a is equal to Rd in Fig. 2/14d.

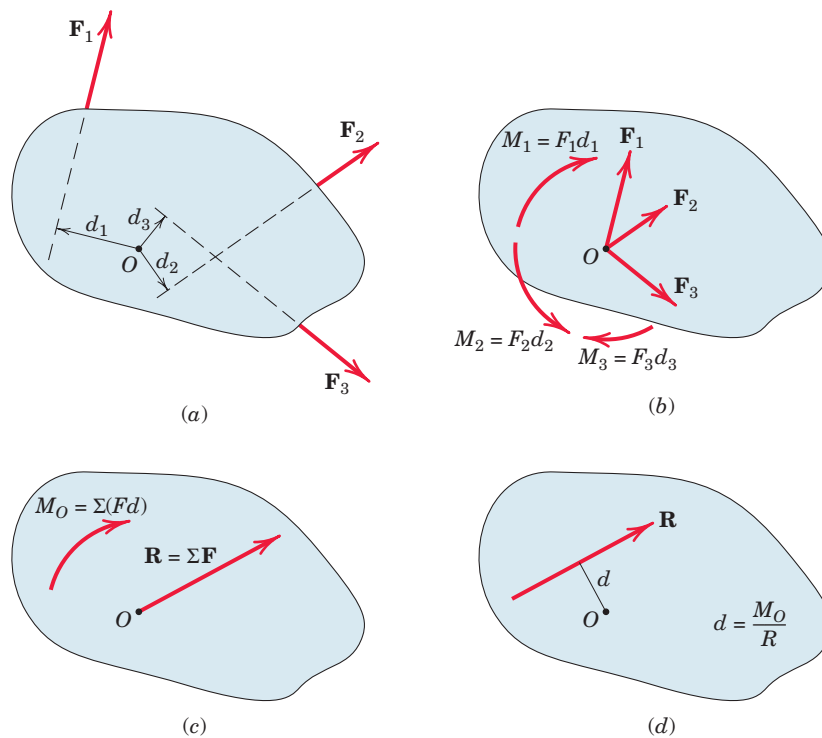


Figure 2/14

Principle of Moments

This process is summarized in equation form by

$$\begin{aligned} \mathbf{R} &= \Sigma \mathbf{F} \\ M_O &= \Sigma M = \Sigma(Fd) \\ Rd &= M_O \end{aligned} \quad (2/10)$$

The first two of Eqs. 2/10 reduce a given system of forces to a force–couple system at an arbitrarily chosen but convenient point O . The last equation specifies the distance d from point O to the line of action of \mathbf{R} , and states that the moment of the resultant force about any point O equals the sum of the moments of the original forces of the system about the same point. This extends Varignon’s theorem to the case of *nonconcurrent* force systems; we call this extension the *principle of moments*.

For a concurrent system of forces where the lines of action of all forces pass through a common point O , the moment sum ΣM_O about that point is zero. Thus, the line of action of the resultant $\mathbf{R} = \Sigma \mathbf{F}$, determined by the first of Eqs. 2/10, passes through point O . For a parallel force system, select a coordinate axis in the direction of the forces. If the resultant force \mathbf{R} for a given force system is zero, the resultant of the system need not be zero because the resultant may be a couple. The three forces in Fig. 2/15, for instance, have a zero resultant force but have a resultant clockwise couple $M = F_3 d$.

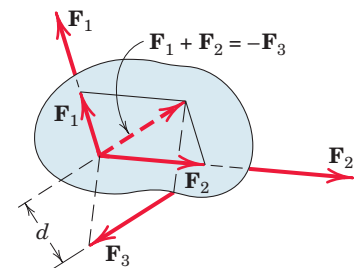


Figure 2/15

60 Chapter 2 Force Systems

Sample Problem 2/9

Determine the resultant of the four forces and one couple which act on the plate shown.

Solution. Point O is selected as a convenient reference point for the force–couple system which is to represent the given system.

$$[R_x = \Sigma F_x] \quad R_x = 40 + 80 \cos 30^\circ - 60 \cos 45^\circ = 66.9 \text{ N}$$

$$[R_y = \Sigma F_y] \quad R_y = 50 + 80 \sin 30^\circ + 60 \cos 45^\circ = 132.4 \text{ N}$$

$$[R = \sqrt{R_x^2 + R_y^2}] \quad R = \sqrt{(66.9)^2 + (132.4)^2} = 148.3 \text{ N}$$

Ans.

$$\left[\theta = \tan^{-1} \frac{R_y}{R_x} \right] \quad \theta = \tan^{-1} \frac{132.4}{66.9} = 63.2^\circ$$

Ans.

$$\textcircled{1} [M_O = \Sigma(Fd)] \quad M_O = 140 - 50(5) + 60 \cos 45^\circ(4) - 60 \sin 45^\circ(7) \\ = -237 \text{ N}\cdot\text{m}$$

The force–couple system consisting of \mathbf{R} and M_O is shown in Fig. a .

We now determine the final line of action of \mathbf{R} such that \mathbf{R} alone represents the original system.

$$[Rd = |M_O|] \quad 148.3d = 237 \quad d = 1.600 \text{ m} \quad \text{Ans.}$$

Hence, the resultant \mathbf{R} may be applied at any point on the line which makes a 63.2° angle with the x -axis and is tangent at point A to a circle of 1.600-m radius with center O , as shown in part b of the figure. We apply the equation $Rd = M_O$ in an absolute-value sense (ignoring any sign of M_O) and let the physics of the situation, as depicted in Fig. a , dictate the final placement of \mathbf{R} . Had M_O been counter-clockwise, the correct line of action of \mathbf{R} would have been the tangent at point B .

The resultant \mathbf{R} may also be located by determining its intercept distance b to point C on the x -axis, Fig. c . With R_x and R_y acting through point C , only R_y exerts a moment about O so that

$$R_y b = |M_O| \quad \text{and} \quad b = \frac{237}{132.4} = 1.792 \text{ m}$$

Alternatively, the y -intercept could have been obtained by noting that the moment about O would be due to R_x only.

A more formal approach in determining the final line of action of \mathbf{R} is to use the vector expression

$$\mathbf{r} \times \mathbf{R} = \mathbf{M}_O$$

where $\mathbf{r} = x\mathbf{i} + y\mathbf{j}$ is a position vector running from point O to any point on the line of action of \mathbf{R} . Substituting the vector expressions for \mathbf{r} , \mathbf{R} , and \mathbf{M}_O and carrying out the cross product result in

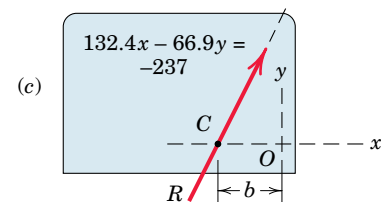
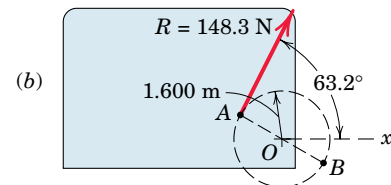
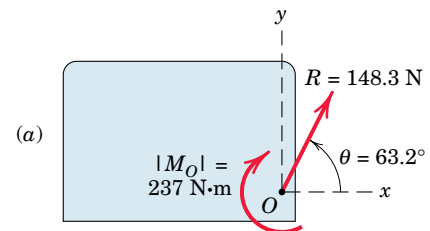
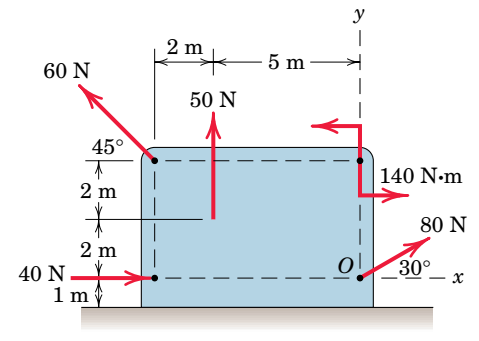
$$(x\mathbf{i} + y\mathbf{j}) \times (66.9\mathbf{i} + 132.4\mathbf{j}) = -237\mathbf{k}$$

$$(132.4x - 66.9y)\mathbf{k} = -237\mathbf{k}$$

Thus, the desired line of action, Fig. c , is given by

$$132.4x - 66.9y = -237$$

$$\textcircled{2} \text{ By setting } y = 0, \text{ we obtain } x = -1.792 \text{ m, which agrees with our earlier calculation of the distance } b.$$



Helpful Hints

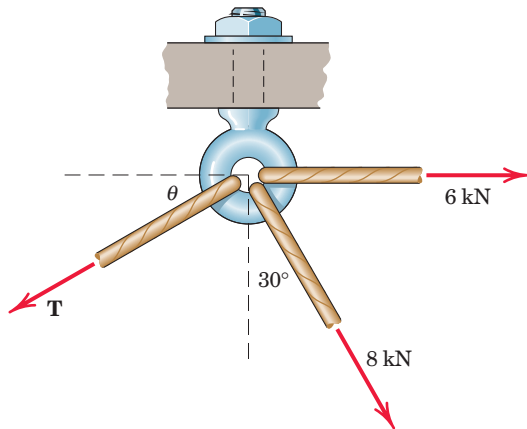
- ① We note that the choice of point O as a moment center eliminates any moments due to the two forces which pass through O . Had the clockwise sign convention been adopted, M_O would have been $+237 \text{ N}\cdot\text{m}$, with the plus sign indicating a sense which agrees with the sign convention. Either sign convention, of course, leads to the conclusion of a clockwise moment M_O .
- ② Note that the vector approach yields sign information automatically, whereas the scalar approach is more physically oriented. You should master both methods.

PROBLEMS

Introductory Problems

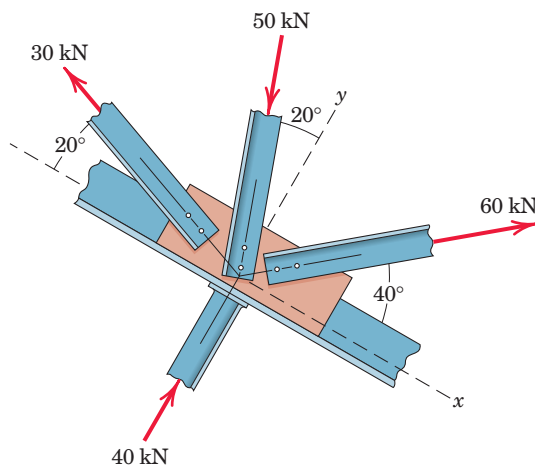
- 2/77** Calculate the magnitude of the tension \mathbf{T} and the angle θ for which the eye bolt will be under a resultant downward force of 15 kN.

Ans. $T = 12.85 \text{ kN}$, $\theta = 38.9^\circ$



Problem 2/77

- 2/78** Determine the resultant \mathbf{R} of the four forces acting on the gusset plate. Also find the magnitude of \mathbf{R} and the angle θ_x which the resultant makes with the x -axis.



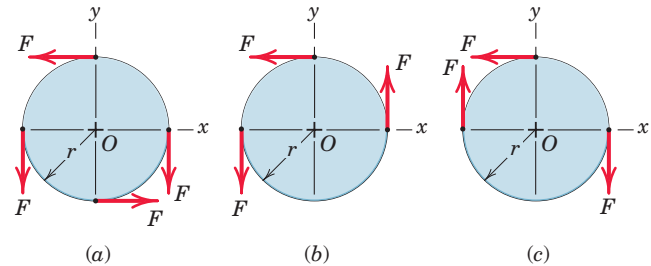
Problem 2/78

- 2/79** Determine the equivalent force-couple system at the origin O for each of the three cases of forces being applied to the edge of a circular disk. If the resultant can be so expressed, replace this force-couple system with a stand-alone force.

Ans. (a) $\mathbf{R} = -2F\mathbf{j}$ along $x = -r$

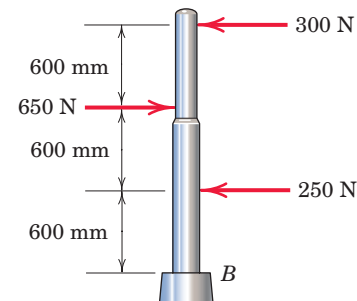
(b) $\mathbf{R} = -F\mathbf{i}$ along $y = 3r$

(c) $\mathbf{R} = -F\mathbf{i}$ along $y = -r$



Problem 2/79

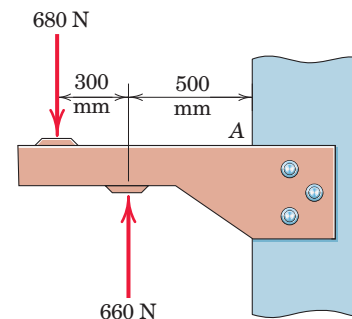
- 2/80** Determine the height h above the base B at which the resultant of the three forces acts.



Problem 2/80

- 2/81** Where does the resultant of the two forces act?

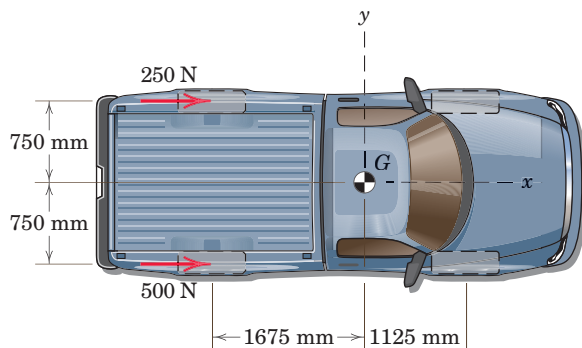
Ans. 10.70 m to the left of A



Problem 2/81

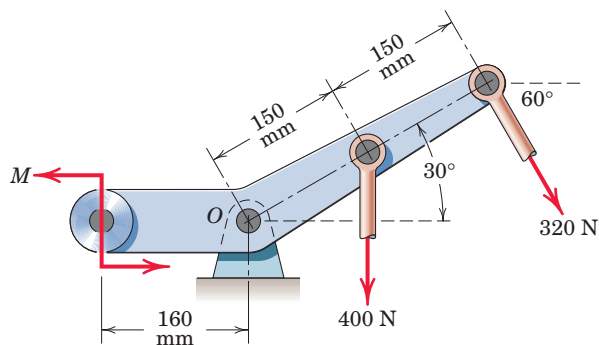
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- 2/82** Under nonuniform and slippery road conditions, the two forces shown are exerted on the two rear-drive wheels of the pickup truck, which has a limited-slip rear differential. Determine the y -intercept of the resultant of this force system.



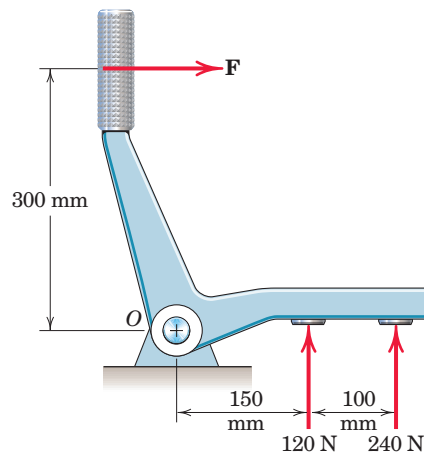
Problem 2/82

- 2/83** If the resultant of the two forces and couple M passes through point O , determine M .
Ans. $M = 148.0 \text{ N} \cdot \text{m}$ CCW



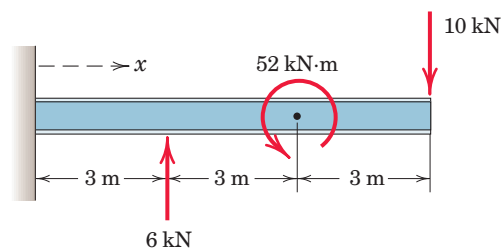
Problem 2/83

- 2/84** Determine the magnitude of the force \mathbf{F} applied to the handle which will make the resultant of the three forces pass through O .



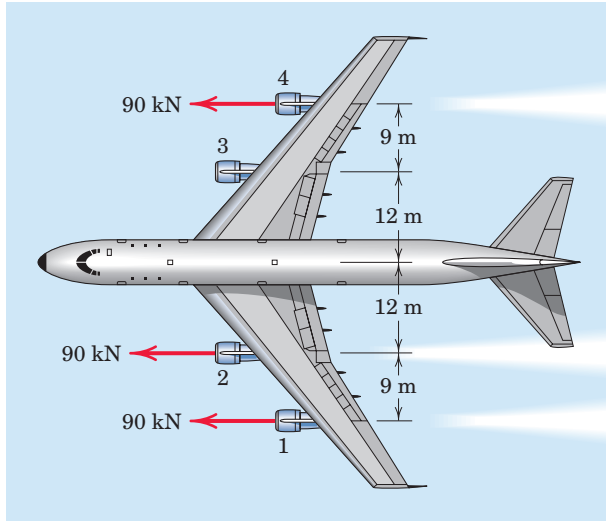
Problem 2/84

- 2/85** Determine and locate the resultant \mathbf{R} of the two forces and one couple acting on the I-beam.
Ans. $R = 4 \text{ kN}$ down at $x = 5 \text{ m}$



Problem 2/85

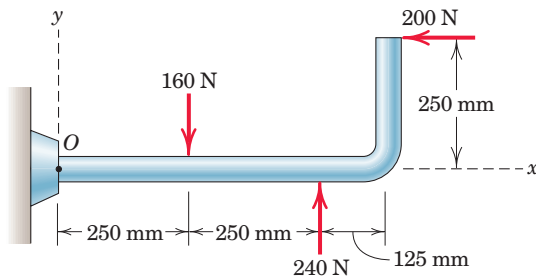
- 2/86** A commercial airliner with four jet engines, each producing 90 kN of forward thrust, is in a steady, level cruise when engine number 3 suddenly fails. Determine and locate the resultant of the three remaining engine thrust vectors. Treat this as a two-dimensional problem.



Problem 2/86

- 2/87** Replace the three forces acting on the bent pipe by a single equivalent force \mathbf{R} . Specify the distance x from point O to the point on the x -axis through which the line of action of \mathbf{R} passes.

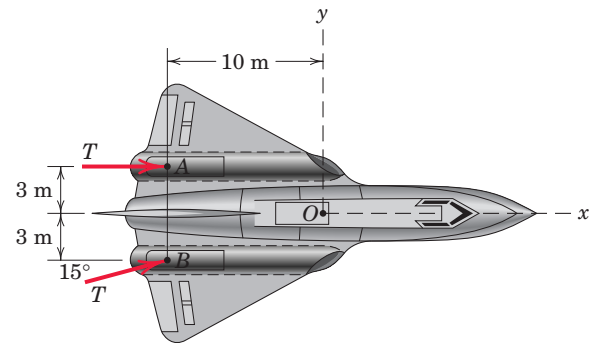
Ans. $\mathbf{R} = -200\mathbf{i} + 80\mathbf{j}$ N, $x = 1.625$ m (off pipe)



Problem 2/87

Representative Problems

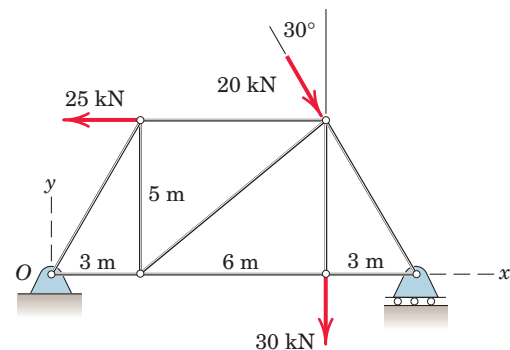
- 2/88** The directions of the two thrust vectors of an experimental aircraft can be independently changed from the conventional forward direction within limits. For the thrust configuration shown, determine the equivalent force-couple system at point O . Then replace this force-couple system by a single force and specify the point on the x -axis through which the line of action of this resultant passes. These results are vital to assessing design performance.



Problem 2/88

- 2/89** Determine the resultant \mathbf{R} of the three forces acting on the simple truss. Specify the points on the x - and y -axes through which \mathbf{R} must pass.

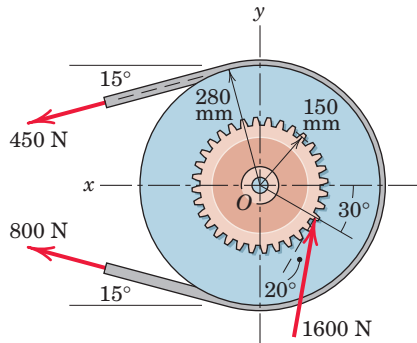
Ans. $\mathbf{R} = -15\mathbf{i} - 47.3\mathbf{j}$ kN
 $x = 7.42$ m, $y = -23.4$ m



Problem 2/89

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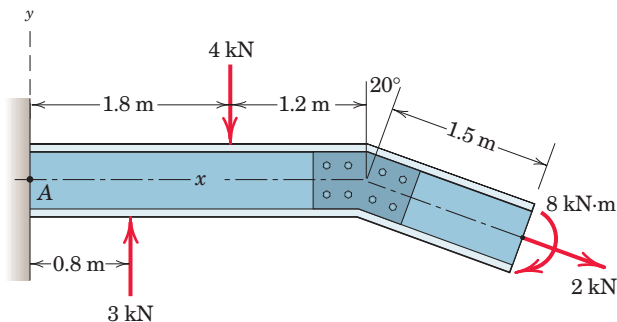
- 2/90** The gear and attached V-belt pulley are turning counterclockwise and are subjected to the tooth load of 1600 N and the 800-N and 450-N tensions in the V-belt. Represent the action of these three forces by a resultant force \mathbf{R} at O and a couple of magnitude M . Is the unit slowing down or speeding up?



Problem 2/90

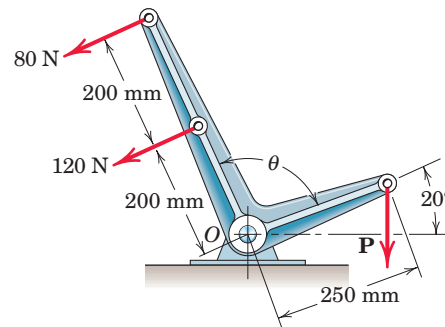
- 2/91** The design specifications for the attachment at A for this beam depend on the magnitude and location of the applied loads. Represent the resultant of the three forces and couple by a single force \mathbf{R} at A and a couple M . Specify the magnitude of \mathbf{R} .

Ans. $\mathbf{R} = 1.879\mathbf{i} - 1.684\mathbf{j}$ kN, $R = 2.52$ kN
 $M = 14.85$ kN·m CW



Problem 2/91

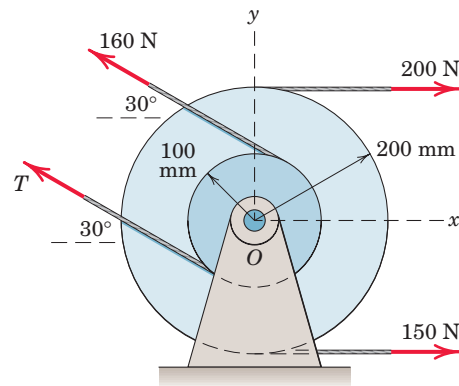
- 2/92** In the equilibrium position shown, the resultant of the three forces acting on the bell crank passes through the bearing O . Determine the vertical force P . Does the result depend on θ ?



Problem 2/92

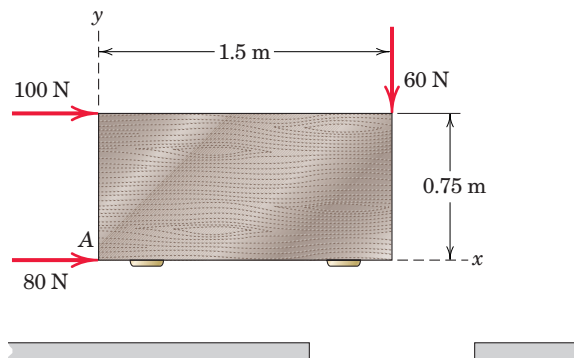
- 2/93** Two integral pulleys are subjected to the belt tensions shown. If the resultant \mathbf{R} of these forces passes through the center O , determine T and the magnitude of \mathbf{R} and the counterclockwise angle θ it makes with the x -axis.

Ans. $T = 60$ N, $R = 193.7$ N, $\theta = 34.6^\circ$



Problem 2/93

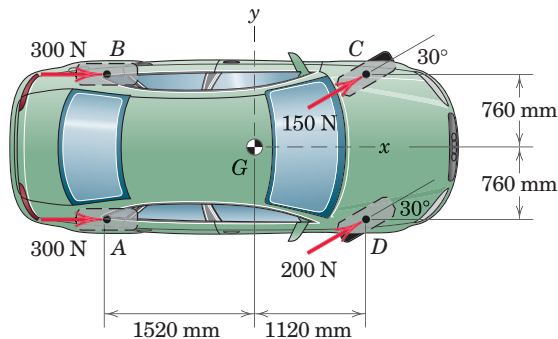
- 2/94** While sliding a desk toward the doorway, three students exert the forces shown in the overhead view. Determine the equivalent force-couple system at point A . Then determine the equation of the line of action of the resultant force.



Problem 2/94

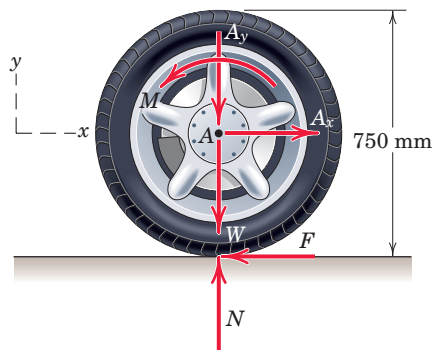
- 2/95** Under nonuniform and slippery road conditions, the four forces shown are exerted on the four drive wheels of the all-wheel-drive vehicle. Determine the resultant of this system and the x - and y -intercepts of its line of action. Note that the front and rear tracks are equal (i.e., $\overline{AB} = \overline{CD}$).

Ans. $\mathbf{R} = 903\mathbf{i} + 175\mathbf{j}$ N, $(0, -0.253)$ m, $(1.308, 0)$ m



Problem 2/95

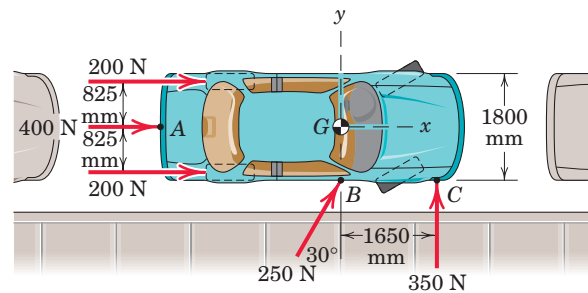
- 2/96** The rolling rear wheel of a front-wheel-drive automobile which is accelerating to the right is subjected to the five forces and one moment shown. The forces $A_x = 240$ N and $A_y = 2000$ N are forces transmitted from the axle to the wheel, $F = 160$ N is the friction force exerted by the road surface on the tire, $N = 2400$ N is the normal reaction force exerted by the road surface, and $W = 400$ N is the weight of the wheel/tire unit. The couple $M = 3$ N·m is the bearing friction moment. Determine and locate the resultant of the system.



Problem 2/96

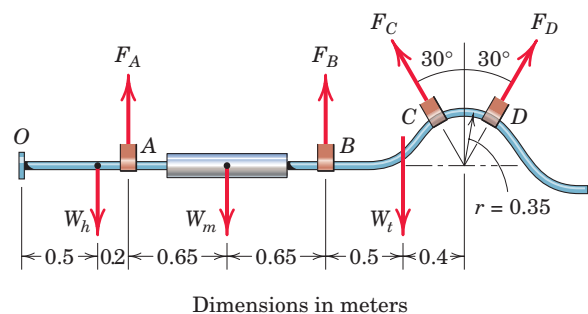
- 2/97** A rear-wheel-drive car is stuck in the snow between other parked cars as shown. In an attempt to free the car, three students exert forces on the car at points A, B, and C while the driver's actions result in a forward thrust of 200 N acting parallel to the plane of rotation of each rear wheel. Treating the problem as two-dimensional, determine the equivalent force-couple system at the car center of mass G and locate the position x of the point on the car center-line through which the resultant passes. Neglect all forces not shown.

Ans. $\mathbf{R} = 925\mathbf{i} + 567\mathbf{j}$ N
 $M_G = 690$ N·m CCW
 $x = 1.218$ m



Problem 2/97

- 2/98** An exhaust system for a pickup truck is shown in the figure. The weights W_h , W_m , and W_t of the headpipe, muffler, and tailpipe are 10, 100, and 50 N, respectively, and act at the indicated points. If the exhaust-pipe hanger at point A is adjusted so that its tension F_A is 50 N, determine the required forces in the hangers at points B, C, and D so that the force-couple system at point O is zero. Why is a zero force-couple system at O desirable?



Problem 2/98

SECTION B THREE-DIMENSIONAL FORCE SYSTEMS

2/7 RECTANGULAR COMPONENTS

Many problems in mechanics require analysis in three dimensions, and for such problems it is often necessary to resolve a force into its three mutually perpendicular components. The force \mathbf{F} acting at point O in Fig. 2/16 has the *rectangular components* F_x , F_y , F_z , where

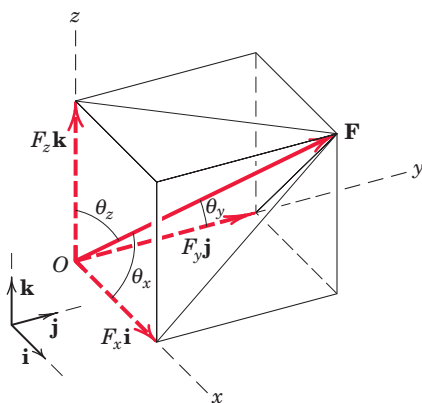


Figure 2/16

$$\begin{aligned} F_x &= F \cos \theta_x & F &= \sqrt{F_x^2 + F_y^2 + F_z^2} \\ F_y &= F \cos \theta_y & \mathbf{F} &= F_x \mathbf{i} + F_y \mathbf{j} + F_z \mathbf{k} \\ F_z &= F \cos \theta_z & \mathbf{F} &= F(\mathbf{i} \cos \theta_x + \mathbf{j} \cos \theta_y + \mathbf{k} \cos \theta_z) \end{aligned} \quad (2/11)$$

The unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} are in the x -, y -, and z -directions, respectively. Using the direction cosines of \mathbf{F} , which are $l = \cos \theta_x$, $m = \cos \theta_y$, and $n = \cos \theta_z$, where $l^2 + m^2 + n^2 = 1$, we may write the force as

$$\mathbf{F} = F(l\mathbf{i} + m\mathbf{j} + n\mathbf{k}) \quad (2/12)$$

We may regard the right-side expression of Eq. 2/12 as the force magnitude F times a unit vector \mathbf{n}_F which characterizes the direction of \mathbf{F} , or

$$\mathbf{F} = F\mathbf{n}_F \quad (2/12a)$$

It is clear from Eqs. 2/12 and 2/12a that $\mathbf{n}_F = l\mathbf{i} + m\mathbf{j} + n\mathbf{k}$, which shows that the scalar components of the unit vector \mathbf{n}_F are the direction cosines of the line of action of \mathbf{F} .

In solving three-dimensional problems, one must usually find the x , y , and z scalar components of a force. In most cases, the direction of a force is described (a) by two points on the line of action of the force or (b) by two angles which orient the line of action.

(a) Specification by two points on the line of action of the force.

If the coordinates of points A and B of Fig. 2/17 are known, the force \mathbf{F} may be written as

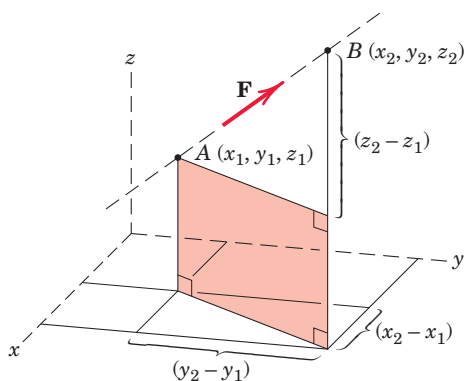


Figure 2/17

$$\mathbf{F} = F\mathbf{n}_F = F \frac{\overrightarrow{AB}}{AB} = F \frac{(x_2 - x_1)\mathbf{i} + (y_2 - y_1)\mathbf{j} + (z_2 - z_1)\mathbf{k}}{\sqrt{(x_2 - x_1)^2 + (y_2 - y_1)^2 + (z_2 - z_1)^2}}$$

Thus the x , y , and z scalar components of \mathbf{F} are the scalar coefficients of the unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} , respectively.

(b) Specification by two angles which orient the line of action of the force. Consider the geometry of Fig. 2/18. We assume that the angles θ and ϕ are known. First resolve \mathbf{F} into horizontal and vertical components.

$$F_{xy} = F \cos \phi$$

$$F_z = F \sin \phi$$

Then resolve the horizontal component F_{xy} into x - and y -components.

$$F_x = F_{xy} \cos \theta = F \cos \phi \cos \theta$$

$$F_y = F_{xy} \sin \theta = F \cos \phi \sin \theta$$

The quantities F_x , F_y , and F_z are the desired scalar components of \mathbf{F} .

The choice of orientation of the coordinate system is arbitrary, with convenience being the primary consideration. However, we must use a right-handed set of axes in our three-dimensional work to be consistent with the right-hand-rule definition of the cross product. When we rotate from the x - to the y -axis through the 90° angle, the positive direction for the z -axis in a right-handed system is that of the advancement of a right-handed screw rotated in the same sense. This is equivalent to the right-hand rule.

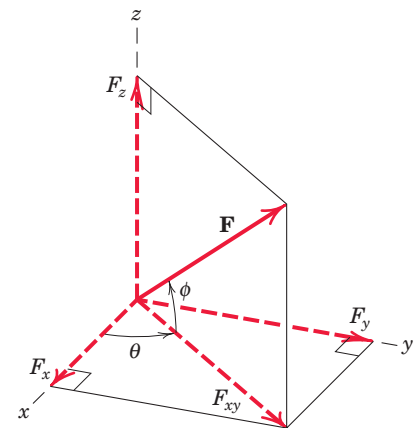


Figure 2/18

Dot Product

We can express the rectangular components of a force \mathbf{F} (or any other vector) with the aid of the vector operation known as the *dot* or *scalar product* (see item 6 in Art. C/7 of Appendix C). The dot product of two vectors \mathbf{P} and \mathbf{Q} , Fig. 2/19a, is defined as the product of their magnitudes times the cosine of the angle α between them. It is written as

$$\mathbf{P} \cdot \mathbf{Q} = PQ \cos \alpha$$

We can view this product either as the orthogonal projection $P \cos \alpha$ of \mathbf{P} in the direction of \mathbf{Q} multiplied by Q , or as the orthogonal projection $Q \cos \alpha$ of \mathbf{Q} in the direction of \mathbf{P} multiplied by P . In either case the dot product of the two vectors is a scalar quantity. Thus, for instance, we can express the scalar component $F_x = F \cos \theta_x$ of the force \mathbf{F} in Fig. 2/16 as $F_x = \mathbf{F} \cdot \mathbf{i}$, where \mathbf{i} is the unit vector in the x -direction.

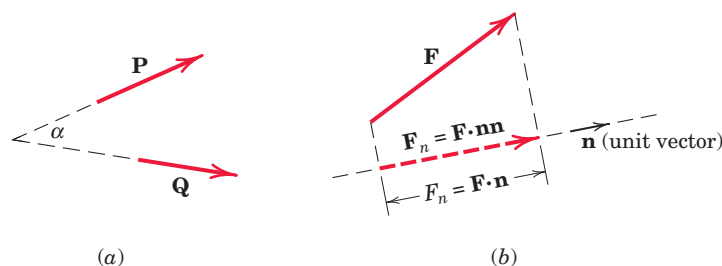


Figure 2/19

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In more general terms, if \mathbf{n} is a unit vector in a specified direction, the projection of \mathbf{F} in the \mathbf{n} -direction, Fig. 2/19b, has the magnitude $F_n = \mathbf{F} \cdot \mathbf{n}$. If we want to express the projection in the \mathbf{n} -direction as a vector quantity, then we multiply its scalar component, expressed by $\mathbf{F} \cdot \mathbf{n}$, by the unit vector \mathbf{n} to give $\mathbf{F}_n = (\mathbf{F} \cdot \mathbf{n})\mathbf{n}$. We may write this as $\mathbf{F}_n = \mathbf{F} \cdot \mathbf{nn}$ without ambiguity because the term \mathbf{nn} is not defined, and so the complete expression cannot be misinterpreted as $\mathbf{F} \cdot (\mathbf{nn})$.

If the direction cosines of \mathbf{n} are α , β , and γ , then we may write \mathbf{n} in vector component form like any other vector as

$$\mathbf{n} = \alpha\mathbf{i} + \beta\mathbf{j} + \gamma\mathbf{k}$$

where in this case its magnitude is unity. If the direction cosines of \mathbf{F} with respect to reference axes x - y - z are l , m , and n , then the projection of \mathbf{F} in the \mathbf{n} -direction becomes

$$\begin{aligned} F_n = \mathbf{F} \cdot \mathbf{n} &= F(l\mathbf{i} + m\mathbf{j} + n\mathbf{k}) \cdot (\alpha\mathbf{i} + \beta\mathbf{j} + \gamma\mathbf{k}) \\ &= F(l\alpha + m\beta + n\gamma) \end{aligned}$$

because

$$\mathbf{i} \cdot \mathbf{i} = \mathbf{j} \cdot \mathbf{j} = \mathbf{k} \cdot \mathbf{k} = 1$$

and

$$\mathbf{i} \cdot \mathbf{j} = \mathbf{j} \cdot \mathbf{i} = \mathbf{i} \cdot \mathbf{k} = \mathbf{k} \cdot \mathbf{i} = \mathbf{j} \cdot \mathbf{k} = \mathbf{k} \cdot \mathbf{j} = 0$$

The latter two sets of equations are true because \mathbf{i} , \mathbf{j} , and \mathbf{k} have unit length and are mutually perpendicular.

Angle between Two Vectors

If the angle between the force \mathbf{F} and the direction specified by the unit vector \mathbf{n} is θ , then from the dot-product definition we have $\mathbf{F} \cdot \mathbf{n} = Fn \cos \theta = F \cos \theta$, where $|\mathbf{n}| = n = 1$. Thus, the angle between \mathbf{F} and \mathbf{n} is given by

$$\theta = \cos^{-1} \frac{\mathbf{F} \cdot \mathbf{n}}{F} \quad (2/13)$$

In general, the angle between any two vectors \mathbf{P} and \mathbf{Q} is

$$\theta = \cos^{-1} \frac{\mathbf{P} \cdot \mathbf{Q}}{PQ} \quad (2/13a)$$

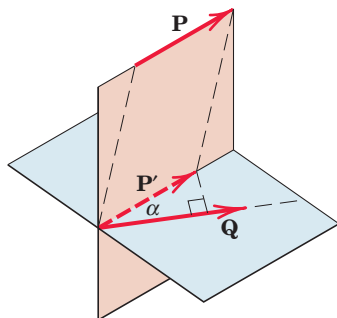


Figure 2/20

If a force \mathbf{F} is perpendicular to a line whose direction is specified by the unit vector \mathbf{n} , then $\cos \theta = 0$, and $\mathbf{F} \cdot \mathbf{n} = 0$. Note that this relationship does not mean that either \mathbf{F} or \mathbf{n} is zero, as would be the case with scalar multiplication where $(A)(B) = 0$ requires that either A or B (or both) be zero.

The dot-product relationship applies to nonintersecting vectors as well as to intersecting vectors. Thus, the dot product of the nonintersecting vectors \mathbf{P} and \mathbf{Q} in Fig. 2/20 is Q times the projection of \mathbf{P}' on \mathbf{Q} , or $P'Q \cos \alpha = PQ \cos \alpha$ because \mathbf{P}' and \mathbf{P} are the same when treated as free vectors.

Sample Problem 2/10

A force \mathbf{F} with a magnitude of 100 N is applied at the origin O of the axes x - y - z as shown. The line of action of \mathbf{F} passes through a point A whose coordinates are 3 m, 4 m, and 5 m. Determine (a) the x , y , and z scalar components of \mathbf{F} , (b) the projection F_{xy} of \mathbf{F} on the x - y plane, and (c) the projection F_{OB} of \mathbf{F} along the line OB .

Solution. Part (a). We begin by writing the force vector \mathbf{F} as its magnitude F times a unit vector \mathbf{n}_{OA} .

$$\begin{aligned}\mathbf{F} &= F\mathbf{n}_{OA} = F \frac{\overrightarrow{OA}}{OA} = 100 \left[\frac{3\mathbf{i} + 4\mathbf{j} + 5\mathbf{k}}{\sqrt{3^2 + 4^2 + 5^2}} \right] \\ &= 100[0.424\mathbf{i} + 0.566\mathbf{j} + 0.707\mathbf{k}] \\ &= 42.4\mathbf{i} + 56.6\mathbf{j} + 70.7\mathbf{k} \text{ N}\end{aligned}$$

The desired scalar components are thus

$$\textcircled{1} \quad F_x = 42.4 \text{ N} \quad F_y = 56.6 \text{ N} \quad F_z = 70.7 \text{ N} \quad \text{Ans.}$$

Part (b). The cosine of the angle θ_{xy} between \mathbf{F} and the x - y plane is

$$\cos \theta_{xy} = \frac{\sqrt{3^2 + 4^2}}{\sqrt{3^2 + 4^2 + 5^2}} = 0.707$$

so that $F_{xy} = F \cos \theta_{xy} = 100(0.707) = 70.7 \text{ N}$

Part (c). The unit vector \mathbf{n}_{OB} along OB is

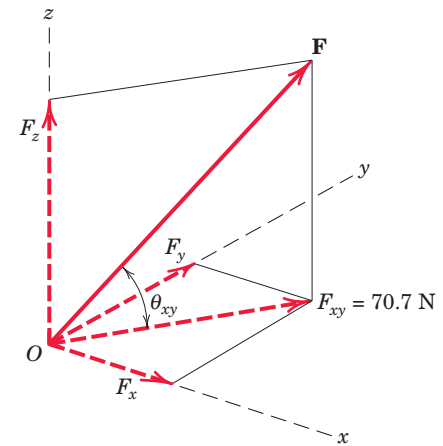
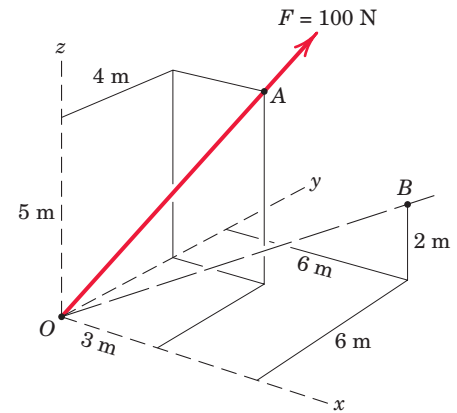
$$\mathbf{n}_{OB} = \frac{\overrightarrow{OB}}{OB} = \frac{6\mathbf{i} + 6\mathbf{j} + 2\mathbf{k}}{\sqrt{6^2 + 6^2 + 2^2}} = 0.688\mathbf{i} + 0.688\mathbf{j} + 0.229\mathbf{k}$$

The scalar projection of \mathbf{F} on OB is

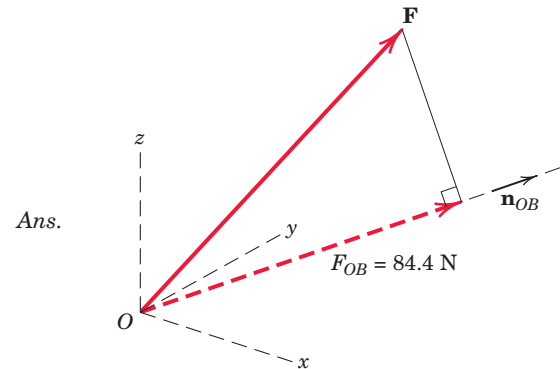
$$\begin{aligned}\textcircled{2} \quad F_{OB} &= \mathbf{F} \cdot \mathbf{n}_{OB} = (42.4\mathbf{i} + 56.6\mathbf{j} + 70.7\mathbf{k}) \cdot (0.688\mathbf{i} + 0.688\mathbf{j} + 0.229\mathbf{k}) \\ &= (42.4)(0.688) + (56.6)(0.688) + (70.7)(0.229) \\ &= 84.4 \text{ N}\end{aligned}$$

If we wish to express the projection as a vector, we write

$$\begin{aligned}\mathbf{F}_{OB} &= \mathbf{F} \cdot \mathbf{n}_{OB} \mathbf{n}_{OB} \\ &= 84.4(0.688\mathbf{i} + 0.688\mathbf{j} + 0.229\mathbf{k}) \\ &= 58.1\mathbf{i} + 58.1\mathbf{j} + 19.35\mathbf{k} \text{ N}\end{aligned}$$



Ans.



Ans.

Helpful Hints

- ① In this example all scalar components are positive. Be prepared for the case where a direction cosine, and hence the scalar component, are negative.
- ② The dot product automatically finds the projection or scalar component of \mathbf{F} along line OB as shown.

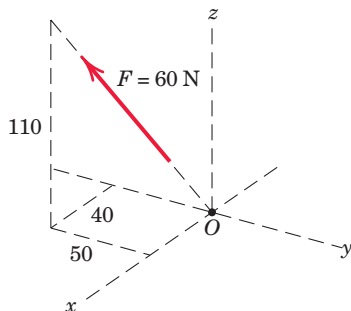
70 Chapter 2 Force Systems

PROBLEMS

Introductory Problems

- 2/99** Express \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} . Determine the angle between \mathbf{F} and the y -axis.

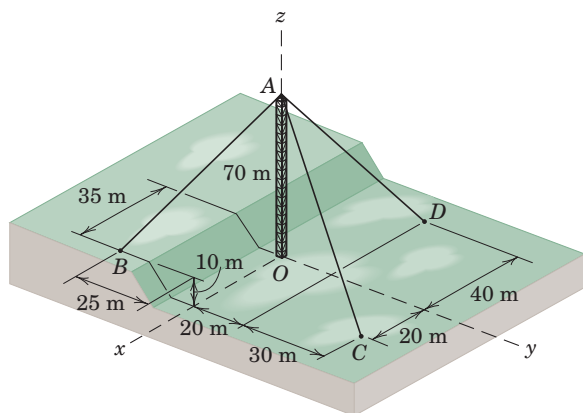
Ans. $\mathbf{F} = 18.86\mathbf{i} - 23.6\mathbf{j} + 51.9\mathbf{k}$ N
 $\theta_y = 113.1^\circ$



Dimensions in millimeters

Problem 2/99

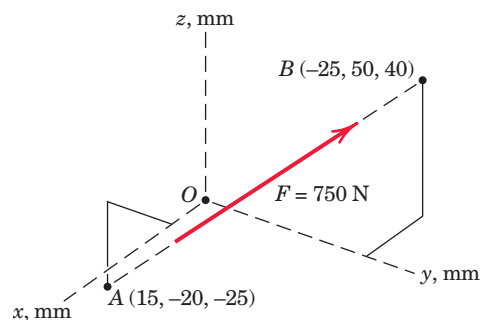
- 2/100** The 70-m microwave transmission tower is steadied by three guy cables as shown. Cable AB carries a tension of 12 kN. Express the corresponding force on point B as a vector.



Problem 2/100

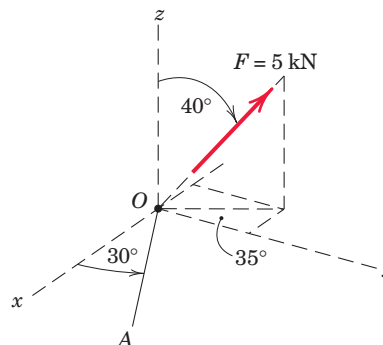
- 2/101** Express the force \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} . Determine the angles θ_x , θ_y , and θ_z which \mathbf{F} makes with the positive x -, y -, and z -axes.

Ans. $\mathbf{F} = -290\mathbf{i} + 507\mathbf{j} + 471\mathbf{k}$ N
 $\theta_x = 112.7^\circ$, $\theta_y = 47.5^\circ$, $\theta_z = 51.1^\circ$



Problem 2/101

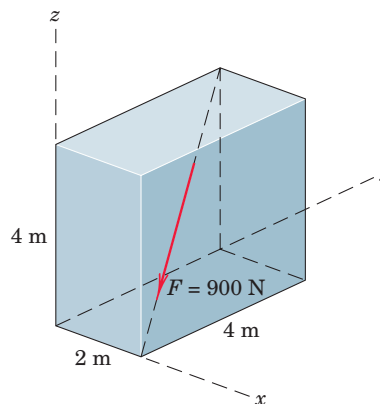
- 2/102** Express \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} . Determine the projection, both as a scalar and as a vector, of \mathbf{F} onto line OA, which lies in the x - y plane.



Problem 2/102

- 2/103** The force \mathbf{F} has a magnitude of 900 N and acts along the diagonal of the parallelepiped as shown. Express \mathbf{F} in terms of its magnitude times the appropriate unit vector and determine its x -, y -, and z -components.

Ans. $\mathbf{F} = 900 \left[\frac{1}{3}\mathbf{i} - \frac{2}{3}\mathbf{j} - \frac{2}{3}\mathbf{k} \right]$ N
 $F_x = 300$ N, $F_y = -600$ N, $F_z = -600$ N

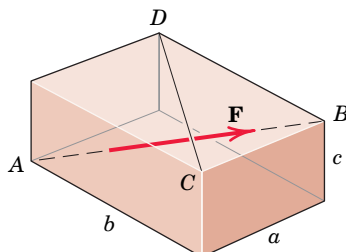


Problem 2/103

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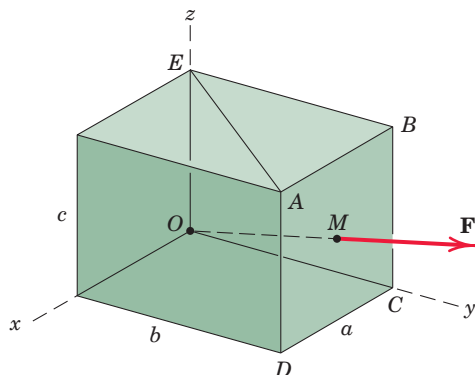
- 2/109** Derive the expression for the projection F_{DC} of the force \mathbf{F} onto the line directed from D to C .

$$\text{Ans. } F_{DC} = \frac{(b^2 - a^2)F}{\sqrt{a^2 + b^2}\sqrt{a^2 + b^2 + c^2}}$$



Problem 2/109

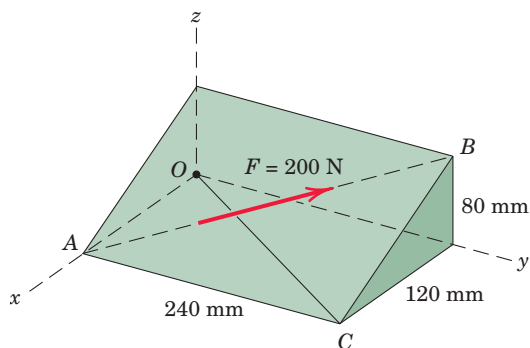
- 2/110** If M is the center of the face $ABCD$ of the rectangular solid, express \mathbf{F} as a vector in terms of the unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} . Determine the scalar projection of \mathbf{F} along line AE . The quantities F , a , b , and c are known.



Problem 2/110

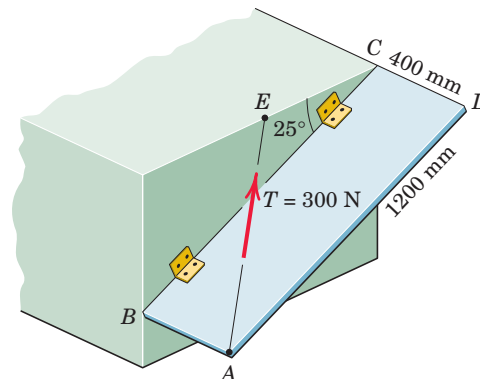
- 2/111** Determine the angle θ between the 200-N force and line OC .

$$\text{Ans. } \theta = 54.9^\circ$$



Problem 2/111

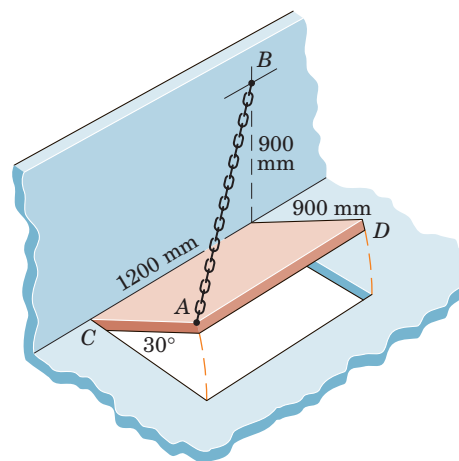
- 2/112** The rectangular plate is supported by hinges along its side BC and by the cable AE . If the cable tension is 300 N, determine the projection onto line BC of the force exerted on the plate by the cable. Note that E is the midpoint of the horizontal upper edge of the structural support.



Problem 2/112

- 2/113** The access door is held in the 30° open position by the chain AB . If the tension in the chain is 100 N, determine the projection of the tension force onto the diagonal axis CD of the door.

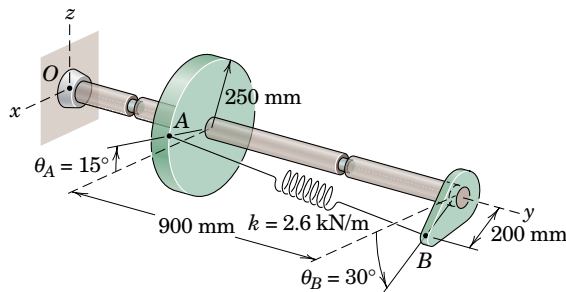
$$\text{Ans. } T_{CD} = 46.0 \text{ N}$$



Problem 2/113

- 2/114 The spring of constant $k = 2.6 \text{ kN/m}$ is attached to the disk at point A and to the end fitting at point B as shown. The spring is unstretched when θ_A and θ_B are both zero. If the disk is rotated 15° clockwise and the end fitting is rotated 30° counterclockwise, determine a vector expression for the force \mathbf{F} which the spring exerts at point A. The magnitude of the spring force is the constant k multiplied by the deflection (lengthening or shortening) of the spring.

Ans. $\mathbf{F} = -3.12\mathbf{i} + 41.1\mathbf{j} - 7.52\mathbf{k} \text{ N}$



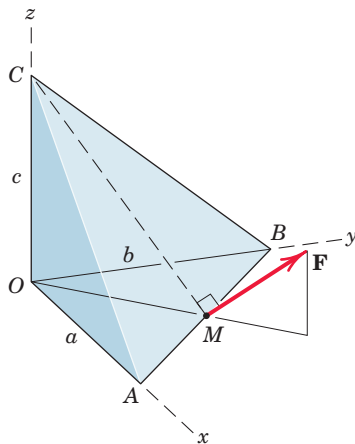
Problem 2/114

- 2/115 Determine the x -, y -, and z -components of force \mathbf{F} which acts on the tetrahedron as shown. The quantities a , b , c , and F are known, and M is the midpoint of edge AB .

$$\text{Ans. } F_x = \frac{2acF}{\sqrt{a^2 + b^2}\sqrt{a^2 + b^2 + 4c^2}}$$

$$F_y = \frac{2bcF}{\sqrt{a^2 + b^2}\sqrt{a^2 + b^2 + 4c^2}}$$

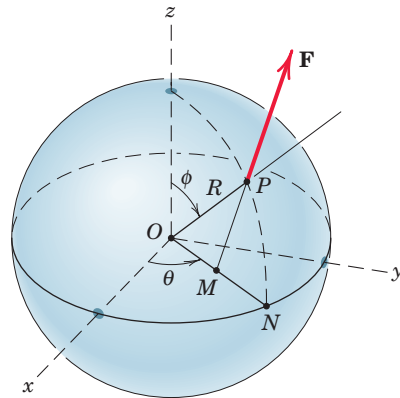
$$F_z = F \sqrt{\frac{a^2 + b^2}{a^2 + b^2 + 4c^2}}$$



Problem 2/115

- 2/116 A force \mathbf{F} is applied to the surface of the sphere as shown. The angles θ and ϕ locate point P , and point M is the midpoint of ON . Express \mathbf{F} in vector form, using the given x -, y -, and z -coordinates.

Ans. $\mathbf{F} = F \left[\frac{(2 \sin \phi - 1)(\mathbf{i} \cos \theta + \mathbf{j} \sin \theta) + \mathbf{k}(2 \cos \phi)}{\sqrt{5 - 4 \sin \phi}} \right]$



Problem 2/116

2/8 MOMENT AND COUPLE

In two-dimensional analyses it is often convenient to determine a moment magnitude by scalar multiplication using the moment-arm rule. In three dimensions, however, the determination of the perpendicular distance between a point or line and the line of action of the force can be a tedious computation. A vector approach with cross-product multiplication then becomes advantageous.

Moments in Three Dimensions

Consider a force \mathbf{F} with a given line of action acting on a body, Fig. 2/21a, and any point O not on this line. Point O and the line of \mathbf{F} establish a plane A . The moment \mathbf{M}_O of \mathbf{F} about an axis through O normal to the plane has the magnitude $M_O = Fd$, where d is the perpendicular distance from O to the line of \mathbf{F} . This moment is also referred to as the moment of \mathbf{F} about the point O .

The vector \mathbf{M}_O is normal to the plane and is directed along the axis through O . We can describe both the magnitude and the direction of \mathbf{M}_O by the vector cross-product relation introduced in Art. 2/4. (Refer to item 7 in Art. C/7 of Appendix C.) The vector \mathbf{r} runs from O to *any* point on the line of action of \mathbf{F} . As described in Art. 2/4, the cross product of \mathbf{r} and \mathbf{F} is written $\mathbf{r} \times \mathbf{F}$ and has the magnitude $(r \sin \alpha)F$, which is the same as Fd , the magnitude of \mathbf{M}_O .

The correct direction and sense of the moment are established by the right-hand rule, described previously in Arts. 2/4 and 2/5. Thus, with \mathbf{r} and \mathbf{F} treated as free vectors emanating from O , Fig. 2/21b, the thumb points in the direction of \mathbf{M}_O if the fingers of the right hand curl in the direction of rotation from \mathbf{r} to \mathbf{F} through the angle α . Therefore, we may write the moment of \mathbf{F} about the axis through O as

$$\mathbf{M}_O = \mathbf{r} \times \mathbf{F} \quad (2/14)$$

The order $\mathbf{r} \times \mathbf{F}$ of the vectors *must* be maintained because $\mathbf{F} \times \mathbf{r}$ would produce a vector with a sense opposite to that of \mathbf{M}_O ; that is, $\mathbf{F} \times \mathbf{r} = -\mathbf{M}_O$.

Evaluating the Cross Product

The cross-product expression for \mathbf{M}_O may be written in the determinant form

$$\mathbf{M}_O = \begin{vmatrix} \mathbf{i} & \mathbf{j} & \mathbf{k} \\ r_x & r_y & r_z \\ F_x & F_y & F_z \end{vmatrix} \quad (2/15)$$

(Refer to item 7 in Art. C/7 of Appendix C if you are not already familiar with the determinant representation of the cross product.) Note the symmetry and order of the terms, and note that a *right-handed* coordinate system must be used. Expansion of the determinant gives

$$\mathbf{M}_O = (r_y F_z - r_z F_y)\mathbf{i} + (r_z F_x - r_x F_z)\mathbf{j} + (r_x F_y - r_y F_x)\mathbf{k}$$

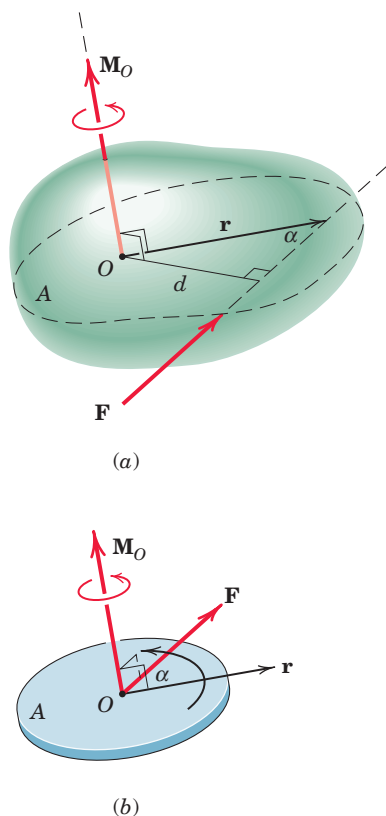


Figure 2/21

To gain more confidence in the cross-product relationship, examine the three components of the moment of a force about a point as obtained from Fig. 2/22. This figure shows the three components of a force \mathbf{F} acting at a point A located relative to O by the vector \mathbf{r} . The scalar magnitudes of the moments of these forces about the positive x -, y -, and z -axes through O can be obtained from the moment-arm rule, and are

$$M_x = r_y F_z - r_z F_y \quad M_y = r_z F_x - r_x F_z \quad M_z = r_x F_y - r_y F_x$$

which agree with the respective terms in the determinant expansion for the cross product $\mathbf{r} \times \mathbf{F}$.

Moment about an Arbitrary Axis

We can now obtain an expression for the moment \mathbf{M}_λ of \mathbf{F} about *any* axis λ through O , as shown in Fig. 2/23. If \mathbf{n} is a unit vector in the λ -direction, then we can use the dot-product expression for the component of a vector as described in Art. 2/7 to obtain $\mathbf{M}_O \cdot \mathbf{n}$, the component of \mathbf{M}_O in the direction of λ . This scalar is the magnitude of the moment \mathbf{M}_λ of \mathbf{F} about λ .

To obtain the vector expression for the moment \mathbf{M}_λ of \mathbf{F} about λ , multiply the magnitude by the directional unit vector \mathbf{n} to obtain

$$\mathbf{M}_\lambda = (\mathbf{r} \times \mathbf{F} \cdot \mathbf{n})\mathbf{n} \quad (2/16)$$

where $\mathbf{r} \times \mathbf{F}$ replaces \mathbf{M}_O . The expression $\mathbf{r} \times \mathbf{F} \cdot \mathbf{n}$ is known as a *triple scalar product* (see item 8 in Art. C/7, Appendix C). It need not be written $(\mathbf{r} \times \mathbf{F}) \cdot \mathbf{n}$ because a cross product cannot be formed by a vector and a scalar. Thus, the association $\mathbf{r} \times (\mathbf{F} \cdot \mathbf{n})$ would have no meaning.

The triple scalar product may be represented by the determinant

$$|\mathbf{M}_\lambda| = M_\lambda = \begin{vmatrix} r_x & r_y & r_z \\ F_x & F_y & F_z \\ \alpha & \beta & \gamma \end{vmatrix} \quad (2/17)$$

where α, β, γ are the direction cosines of the unit vector \mathbf{n} .

Varignon's Theorem in Three Dimensions

In Art. 2/4 we introduced Varignon's theorem in two dimensions. The theorem is easily extended to three dimensions. Figure 2/24 shows a system of concurrent forces $\mathbf{F}_1, \mathbf{F}_2, \mathbf{F}_3, \dots$. The sum of the moments about O of these forces is

$$\begin{aligned} \mathbf{r} \times \mathbf{F}_1 + \mathbf{r} \times \mathbf{F}_2 + \mathbf{r} \times \mathbf{F}_3 + \dots &= \mathbf{r} \times (\mathbf{F}_1 + \mathbf{F}_2 + \mathbf{F}_3 + \dots) \\ &= \mathbf{r} \times \Sigma \mathbf{F} \end{aligned}$$

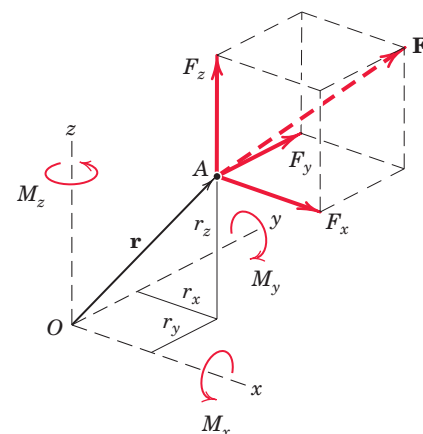


Figure 2/22

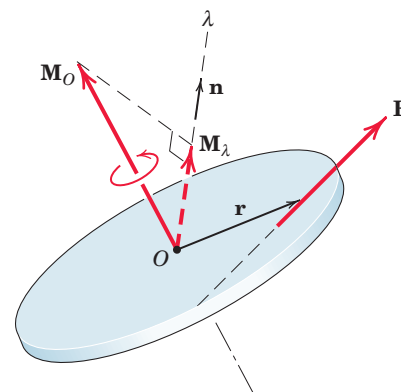


Figure 2/23

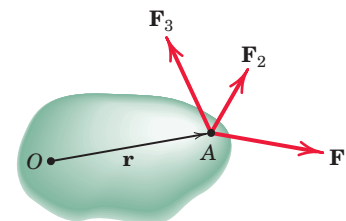


Figure 2/24

where we have used the distributive law for cross products. Using the symbol \mathbf{M}_O to represent the sum of the moments on the left side of the above equation, we have

$$\mathbf{M}_O = \Sigma(\mathbf{r} \times \mathbf{F}) = \mathbf{r} \times \mathbf{R} \quad (2/18)$$

This equation states that the sum of the moments of a system of concurrent forces about a given point equals the moment of their sum about the same point. As mentioned in Art. 2/4, this principle has many applications in mechanics.

Couples in Three Dimensions

The concept of the couple was introduced in Art. 2/5 and is easily extended to three dimensions. Figure 2/25 shows two equal and opposite forces \mathbf{F} and $-\mathbf{F}$ acting on a body. The vector \mathbf{r} runs from *any* point B on the line of action of $-\mathbf{F}$ to *any* point A on the line of action of \mathbf{F} . Points A and B are located by position vectors \mathbf{r}_A and \mathbf{r}_B from *any* point O . The combined moment of the two forces about O is

$$\mathbf{M} = \mathbf{r}_A \times \mathbf{F} + \mathbf{r}_B \times (-\mathbf{F}) = (\mathbf{r}_A - \mathbf{r}_B) \times \mathbf{F}$$

However, $\mathbf{r}_A - \mathbf{r}_B = \mathbf{r}$, so that all reference to the moment center O disappears, and the moment of the couple becomes

$$\mathbf{M} = \mathbf{r} \times \mathbf{F} \quad (2/19)$$

Thus, the moment of a couple is the *same about all points*. The magnitude of \mathbf{M} is $M = Fd$, where d is the perpendicular distance between the lines of action of the two forces, as described in Art. 2/5.

The moment of a couple is a *free vector*, whereas the moment of a force about a point (which is also the moment about a defined axis through the point) is a *sliding vector* whose direction is along the axis through the point. As in the case of two dimensions, a couple tends to produce a pure rotation of the body about an axis normal to the plane of the forces which constitute the couple.

Couple vectors obey all of the rules which govern vector quantities. Thus, in Fig. 2/26 the couple vector \mathbf{M}_1 due to \mathbf{F}_1 and $-\mathbf{F}_1$ may be added

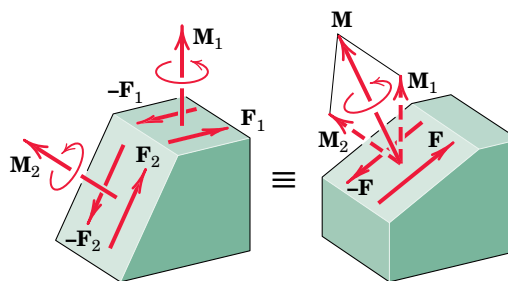


Figure 2/26

as shown to the couple vector \mathbf{M}_2 due to \mathbf{F}_2 and $-\mathbf{F}_2$ to produce the couple \mathbf{M} , which, in turn, can be produced by \mathbf{F} and $-\mathbf{F}$.

In Art. 2/5 we learned how to replace a force by its equivalent force-couple system. You should also be able to carry out this replacement in three dimensions. The procedure is represented in Fig. 2/27, where the force \mathbf{F} acting on a rigid body at point A is replaced by an equal force at point B and the couple $\mathbf{M} = \mathbf{r} \times \mathbf{F}$. By adding the equal and opposite forces \mathbf{F} and $-\mathbf{F}$ at B , we obtain the couple composed of $-\mathbf{F}$ and the original \mathbf{F} . Thus, we see that the couple vector is simply the moment of the original force about the point to which the force is being moved. We emphasize that \mathbf{r} is a vector which runs from B to *any* point on the line of action of the original force passing through A .

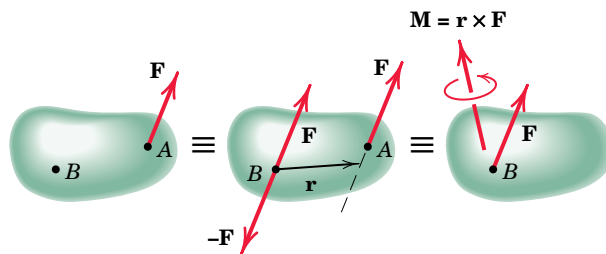


Figure 2/27

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Sample Problem 2/11

Determine the moment of force \mathbf{F} about point O (a) by inspection and (b) by the formal cross-product definition $\mathbf{M}_O = \mathbf{r} \times \mathbf{F}$.

Solution. (a) Because \mathbf{F} is parallel to the y -axis, \mathbf{F} has no moment about that axis. It should be clear that the moment arm from the x -axis to the line of action of \mathbf{F} is c and that the moment of \mathbf{F} about the x -axis is negative. Similarly, the moment arm from the z -axis to the line of action of \mathbf{F} is a and the moment of \mathbf{F} about the z -axis is positive. So we have

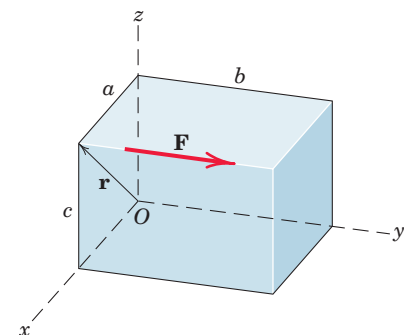
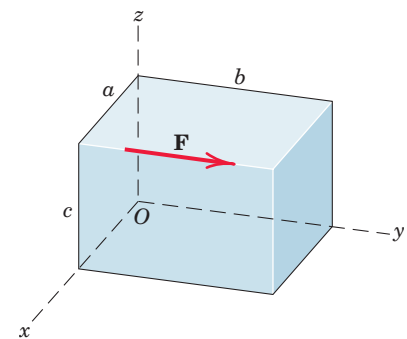
$$\mathbf{M}_O = -cF\mathbf{i} + aF\mathbf{k} = F(-c\mathbf{i} + a\mathbf{k}) \quad \text{Ans.}$$

(b) Formally,

$$\begin{aligned} \textcircled{1} \quad \mathbf{M}_O = \mathbf{r} \times \mathbf{F} &= (a\mathbf{i} + c\mathbf{k}) \times F\mathbf{j} = aF\mathbf{k} - cF\mathbf{i} \\ &= F(-c\mathbf{i} + a\mathbf{k}) \quad \text{Ans.} \end{aligned}$$

Helpful Hint

- ① Again we stress that \mathbf{r} runs from the moment center to the line of action of \mathbf{F} . Another permissible, but less convenient, position vector is $\mathbf{r} = a\mathbf{i} + b\mathbf{j} + c\mathbf{k}$.



Sample Problem 2/12

The turnbuckle is tightened until the tension in cable AB is 2.4 kN. Determine the moment about point O of the cable force acting on point A and the magnitude of this moment.

Solution. We begin by writing the described force as a vector.

$$\begin{aligned} \mathbf{T} = T\mathbf{n}_{AB} &= 2.4 \left[\frac{0.8\mathbf{i} + 1.5\mathbf{j} - 2\mathbf{k}}{\sqrt{0.8^2 + 1.5^2 + 2^2}} \right] \\ &= 0.731\mathbf{i} + 1.371\mathbf{j} - 1.829\mathbf{k} \text{ kN} \end{aligned}$$

The moment of this force about point O is

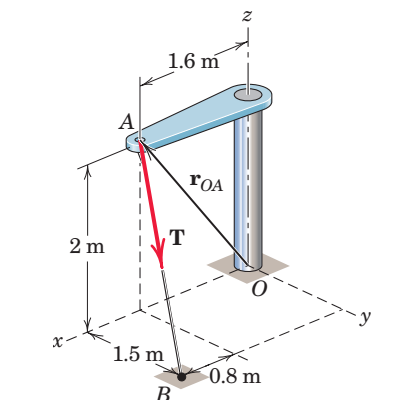
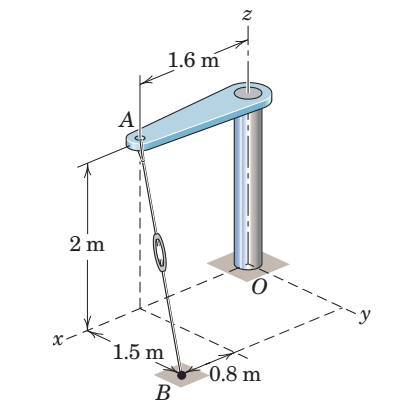
$$\begin{aligned} \textcircled{1} \quad \mathbf{M}_O = \mathbf{r}_{OA} \times \mathbf{T} &= (1.6\mathbf{i} + 2\mathbf{k}) \times (0.731\mathbf{i} + 1.371\mathbf{j} - 1.829\mathbf{k}) \\ &= -2.74\mathbf{i} + 4.39\mathbf{j} + 2.19\mathbf{k} \text{ kN} \cdot \text{m} \quad \text{Ans.} \end{aligned}$$

This vector has a magnitude

$$M_O = \sqrt{2.74^2 + 4.39^2 + 2.19^2} = 5.62 \text{ kN} \cdot \text{m} \quad \text{Ans.}$$

Helpful Hint

- ① The student should verify by inspection the signs of the moment components.



Sample Problem 2/13

A tension \mathbf{T} of magnitude 10 kN is applied to the cable attached to the top A of the rigid mast and secured to the ground at B . Determine the moment M_z of \mathbf{T} about the z -axis passing through the base O .

- Solution (a).** The required moment may be obtained by finding the component along the z -axis of the moment \mathbf{M}_O of \mathbf{T} about point O . The vector \mathbf{M}_O is normal to the plane defined by \mathbf{T} and point O , as shown in the accompanying figure. In the use of Eq. 2/14 to find \mathbf{M}_O , the vector \mathbf{r} is any vector from point O to the line of action of \mathbf{T} . The simplest choice is the vector from O to A , which is written as $\mathbf{r} = 15\mathbf{j}$ m. The vector expression for \mathbf{T} is

$$\begin{aligned}\mathbf{T} &= T\mathbf{n}_{AB} = 10 \left[\frac{12\mathbf{i} - 15\mathbf{j} + 9\mathbf{k}}{\sqrt{(12)^2 + (-15)^2 + (9)^2}} \right] \\ &= 10(0.566\mathbf{i} - 0.707\mathbf{j} + 0.424\mathbf{k}) \text{ kN}\end{aligned}$$

From Eq. 2/14,

$$\begin{aligned}[\mathbf{M}_O = \mathbf{r} \times \mathbf{F}] \quad \mathbf{M}_O &= 15\mathbf{j} \times 10(0.566\mathbf{i} - 0.707\mathbf{j} + 0.424\mathbf{k}) \\ &= 150(-0.566\mathbf{k} + 0.424\mathbf{i}) \text{ kN}\cdot\text{m}\end{aligned}$$

The value M_z of the desired moment is the scalar component of \mathbf{M}_O in the z -direction or $M_z = \mathbf{M}_O \cdot \mathbf{k}$. Therefore,

$$M_z = 150(-0.566\mathbf{k} + 0.424\mathbf{i}) \cdot \mathbf{k} = -84.9 \text{ kN}\cdot\text{m} \quad \text{Ans.}$$

- ② The minus sign indicates that the vector \mathbf{M}_z is in the negative z -direction. Expressed as a vector, the moment is $\mathbf{M}_z = -84.9\mathbf{k}$ kN·m.

- Solution (b).** The force of magnitude T is resolved into components T_z and T_{xy} in the x - y plane. Since T_z is parallel to the z -axis, it can exert no moment about this axis. The moment M_z is, then, due only to T_{xy} and is $M_z = T_{xy}d$, where d is the perpendicular distance from T_{xy} to O . The cosine of the angle between T and T_{xy} is $\sqrt{15^2 + 12^2} / \sqrt{15^2 + 12^2 + 9^2} = 0.906$, and therefore,

$$T_{xy} = 10(0.906) = 9.06 \text{ kN}$$

The moment arm d equals \overline{OA} multiplied by the sine of the angle between T_{xy} and OA , or

$$d = 15 \frac{12}{\sqrt{12^2 + 15^2}} = 9.37 \text{ m}$$

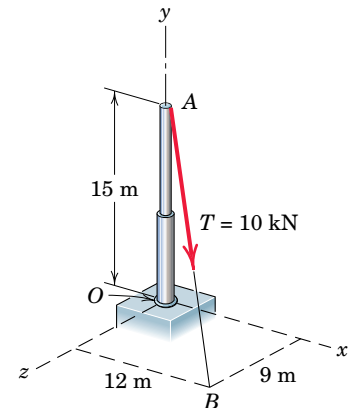
Hence, the moment of \mathbf{T} about the z -axis has the magnitude

$$M_z = 9.06(9.37) = 84.9 \text{ kN}\cdot\text{m} \quad \text{Ans.}$$

and is clockwise when viewed in the x - y plane.

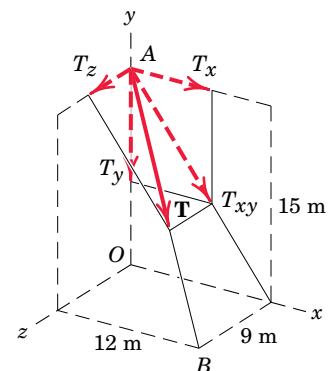
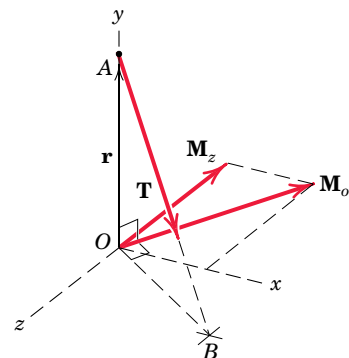
- Solution (c).** The component T_{xy} is further resolved into its components T_x and T_y . It is clear that T_y exerts no moment about the z -axis since it passes through it, so that the required moment is due to T_x alone. The direction cosine of \mathbf{T} with respect to the x -axis is $12/\sqrt{9^2 + 12^2 + 15^2} = 0.566$ so that $T_x = 10(0.566) = 5.66$ kN. Thus,

$$M_z = 5.66(15) = 84.9 \text{ kN}\cdot\text{m} \quad \text{Ans.}$$



Helpful Hints

- ① We could also use the vector from O to B for \mathbf{r} and obtain the same result, but using vector OA is simpler.
- ② It is always helpful to accompany your vector operations with a sketch of the vectors so as to retain a clear picture of the geometry of the problem.
- ③ Sketch the x - y view of the problem and show d .



80 Chapter 2 Force Systems

Sample Problem 2/14

Determine the magnitude and direction of the couple \mathbf{M} which will replace the two given couples and still produce the same external effect on the block. Specify the two forces \mathbf{F} and $-\mathbf{F}$, applied in the two faces of the block parallel to the y - z plane, which may replace the four given forces. The 30-N forces act parallel to the y - z plane.

Solution. The couple due to the 30-N forces has the magnitude $M_1 = 30(0.06) = 1.80 \text{ N}\cdot\text{m}$. The direction of \mathbf{M}_1 is normal to the plane defined by the two forces, and the sense, shown in the figure, is established by the right-hand convention. The couple due to the 25-N forces has the magnitude $M_2 = 25(0.10) = 2.50 \text{ N}\cdot\text{m}$ with the direction and sense shown in the same figure. The two couple vectors combine to give the components

$$M_y = 1.80 \sin 60^\circ = 1.559 \text{ N}\cdot\text{m}$$

$$M_z = -2.50 + 1.80 \cos 60^\circ = -1.600 \text{ N}\cdot\text{m}$$

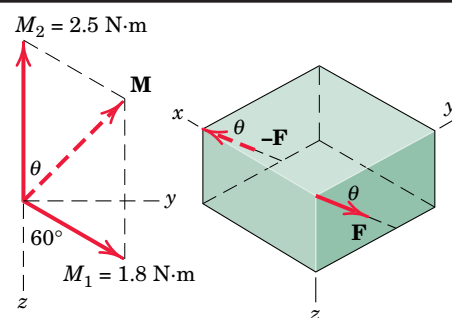
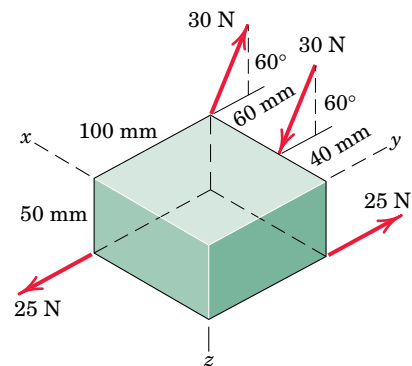
$$\textcircled{1} \text{ Thus, } M = \sqrt{(1.559)^2 + (-1.600)^2} = 2.23 \text{ N}\cdot\text{m} \quad \text{Ans.}$$

$$\text{with } \theta = \tan^{-1} \frac{1.559}{1.600} = \tan^{-1} 0.974 = 44.3^\circ \quad \text{Ans.}$$

The forces \mathbf{F} and $-\mathbf{F}$ lie in a plane normal to the couple \mathbf{M} , and their moment arm as seen from the right-hand figure is 100 mm. Thus, each force has the magnitude

$$[M = Fd] \quad F = \frac{2.23}{0.10} = 22.3 \text{ N} \quad \text{Ans.}$$

and the direction $\theta = 44.3^\circ$.



Helpful Hint

- $\textcircled{1}$ Bear in mind that the couple vectors are *free vectors* and therefore have no unique lines of action.

Sample Problem 2/15

A force of 400 N is applied at A to the handle of the control lever which is attached to the fixed shaft OB. In determining the effect of the force on the shaft at a cross section such as that at O, we may replace the force by an equivalent force at O and a couple. Describe this couple as a vector \mathbf{M} .

Solution. The couple may be expressed in vector notation as $\mathbf{M} = \mathbf{r} \times \mathbf{F}$, where $\mathbf{r} = \overrightarrow{OA} = 0.2\mathbf{j} + 0.125\mathbf{k} \text{ m}$ and $\mathbf{F} = -400\mathbf{i} \text{ N}$. Thus,

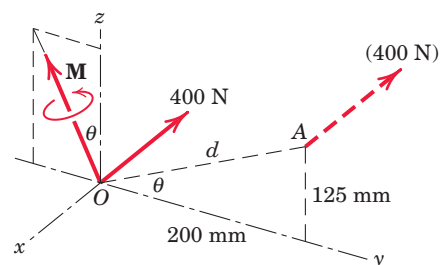
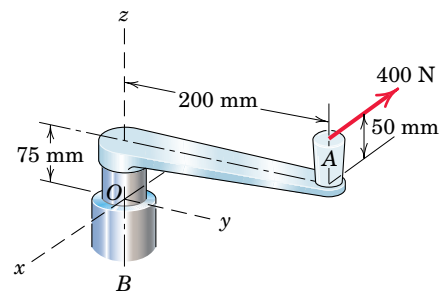
$$\mathbf{M} = (0.2\mathbf{j} + 0.125\mathbf{k}) \times (-400\mathbf{i}) = -50\mathbf{j} + 80\mathbf{k} \text{ N}\cdot\text{m}$$

Alternatively we see that moving the 400-N force through a distance $d = \sqrt{0.125^2 + 0.2^2} = 0.236 \text{ m}$ to a parallel position through O requires the addition of a couple \mathbf{M} whose magnitude is

$$M = Fd = 400(0.236) = 94.3 \text{ N}\cdot\text{m} \quad \text{Ans.}$$

The couple vector is perpendicular to the plane in which the force is shifted, and its sense is that of the moment of the given force about O. The direction of \mathbf{M} in the y - z plane is given by

$$\theta = \tan^{-1} \frac{125}{200} = 32.0^\circ \quad \text{Ans.}$$

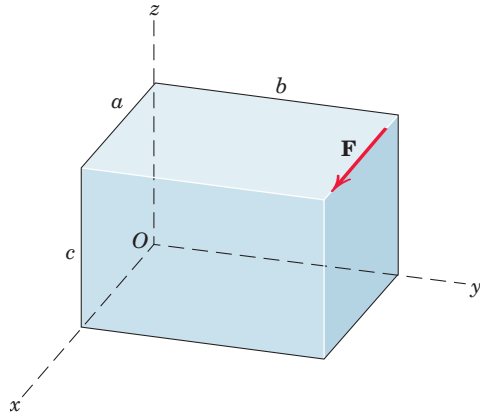


PROBLEMS

Introductory Problems

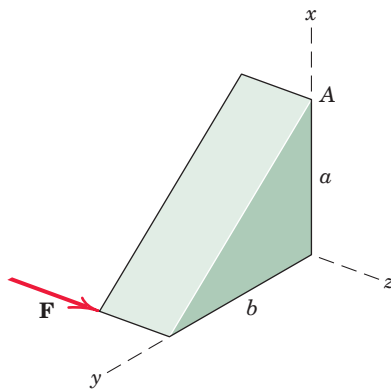
2/117 Determine the moment of force \mathbf{F} about point O .

Ans. $\mathbf{M}_O = F(c\mathbf{j} - b\mathbf{k})$



Problem 2/117

2/118 Determine the moment of force \mathbf{F} about point A .

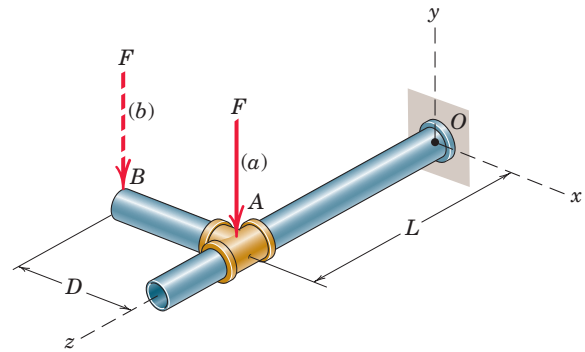


Problem 2/118

2/119 Determine the moment about O of the force of magnitude F for the case (a) when the force \mathbf{F} is applied at A and for the case (b) when \mathbf{F} is applied at B .

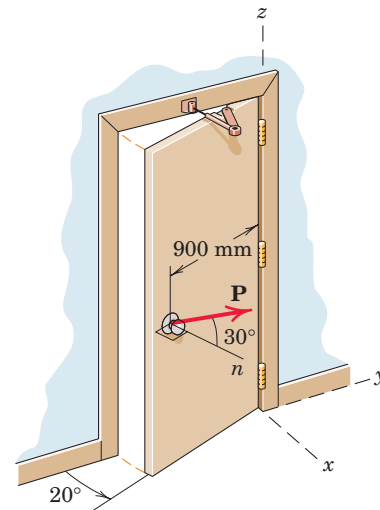
Ans. (a) $\mathbf{M}_O = FL\mathbf{i}$

(b) $\mathbf{M}_O = F(L\mathbf{i} + D\mathbf{k})$



Problem 2/119

2/120 In opening a door which is equipped with a heavy-duty return mechanism, a person exerts a force \mathbf{P} of magnitude 32 N as shown. Force \mathbf{P} and the normal n to the face of the door lie in a vertical plane. Compute the moment of \mathbf{P} about the z -axis.

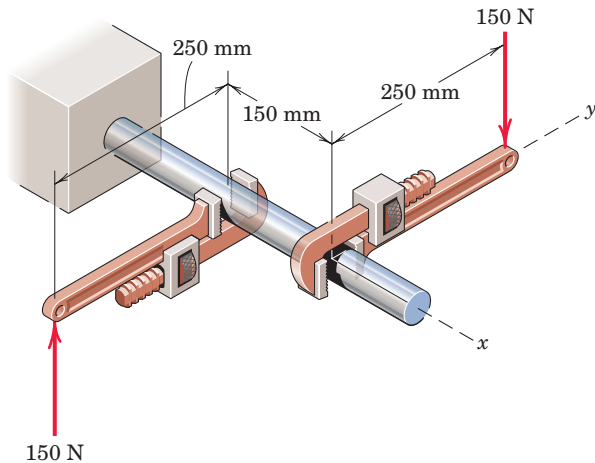


Problem 2/120

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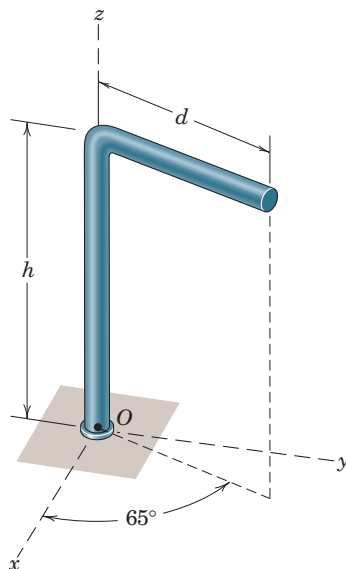
- 2/121** The two forces acting on the handles of the pipe wrenches constitute a couple \mathbf{M} . Express the couple as a vector.

Ans. $\mathbf{M} = -75\mathbf{i} + 22.5\mathbf{j} \text{ N}\cdot\text{m}$



Problem 2/121

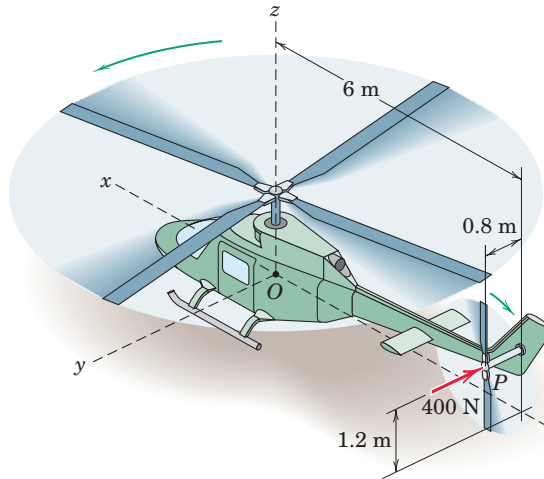
- 2/122** The bent bar has a mass ρ per unit of length. Determine the moment of the weight of the bar about point O .



Problem 2/122

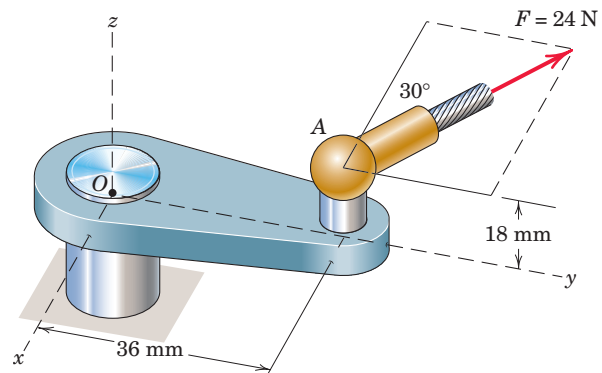
- 2/123** A helicopter is shown here with certain three-dimensional geometry given. During a ground test, a 400-N aerodynamic force is applied to the tail rotor at P as shown. Determine the moment of this force about point O of the airframe.

Ans. $\mathbf{M}_O = 480\mathbf{i} + 2400\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/123

- 2/124** The 24-N force is applied at point A of the crank assembly. Determine the moment of this force about point O .

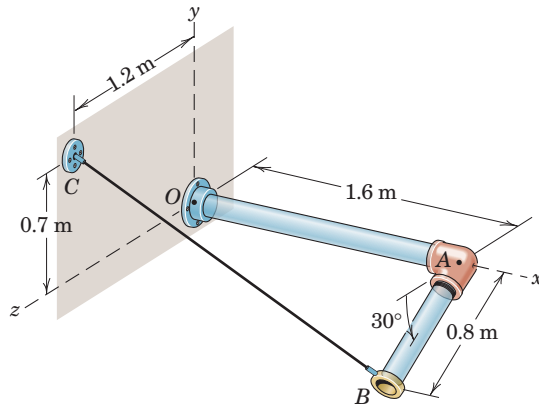


Problem 2/124

- 2/125** The right-angle pipe OAB of Prob. 2/108 is shown again here. Replace the 750-N tensile force which the cable exerts on point B by a force-couple system at point O .

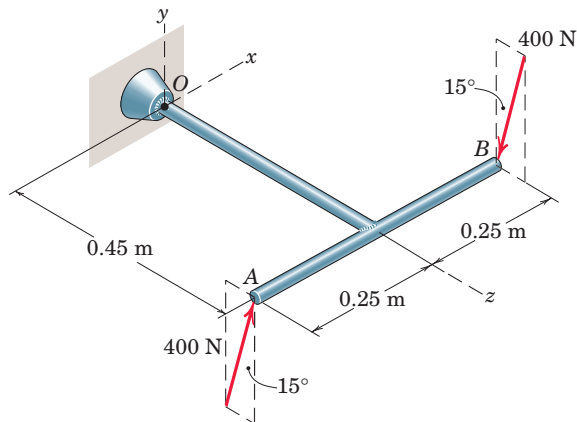
$$\text{Ans. } \mathbf{R} = -598\mathbf{i} + 411\mathbf{j} + 189.5\mathbf{k} \text{ N}$$

$$\mathbf{M}_O = -361\mathbf{i} - 718\mathbf{j} + 419\mathbf{k} \text{ N}\cdot\text{m}$$



Problem 2/125

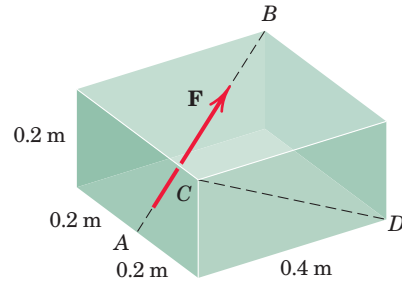
- 2/126** Determine the moment associated with the pair of 400-N forces applied to the T-shaped structure.



Problem 2/126

- 2/127** If the magnitude of the moment of \mathbf{F} about line CD is $50 \text{ N}\cdot\text{m}$, determine the magnitude of \mathbf{F} .

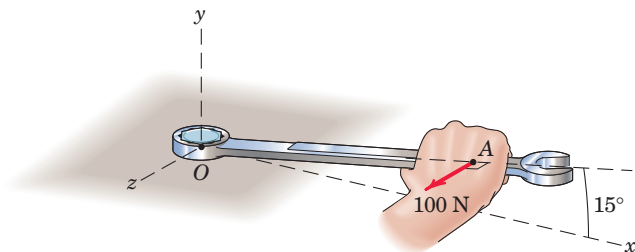
$$\text{Ans. } F = 228 \text{ N}$$



Problem 2/127

Representative Problems

- 2/128** A mechanic applies the horizontal 100-N force perpendicular to the wrench as indicated in the figure. Determine the moment of this force about the bolt center O . The wrench centerline lies in the x - y plane.



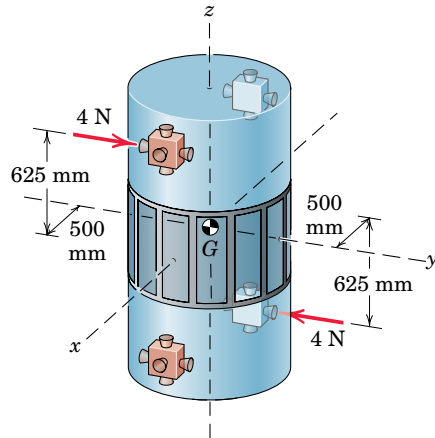
$$\overline{OA} = 185 \text{ mm}$$

Problem 2/128

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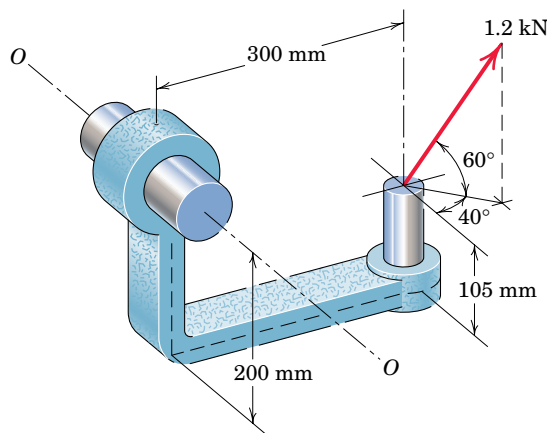
- 2/129** Two 4-N thrusters on the nonrotating satellite are simultaneously fired as shown. Compute the moment associated with this couple and state about which satellite axes rotations will begin to occur.

Ans. $\mathbf{M} = -5\mathbf{i} + 4\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/129

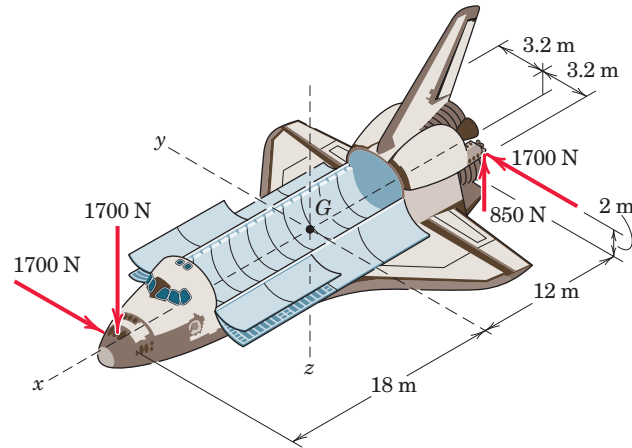
- 2/130** Compute the moment \mathbf{M}_O of the 1.2-kN force about the axis $O-O$.



Problem 2/130

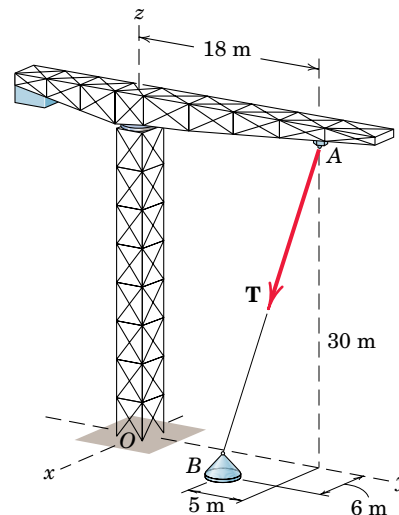
- 2/131** A space shuttle orbiter is subjected to thrusts from five of the engines of its reaction control system. Four of the thrusts are shown in the figure; the fifth is an 850-N upward thrust at the right rear, symmetric to the 850-N thrust shown on the left rear. Compute the moment of these forces about point G and show that the forces have the same moment about all points.

Ans. $\mathbf{M} = 3400\mathbf{i} - 51\,000\mathbf{j} - 51\,000\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/131

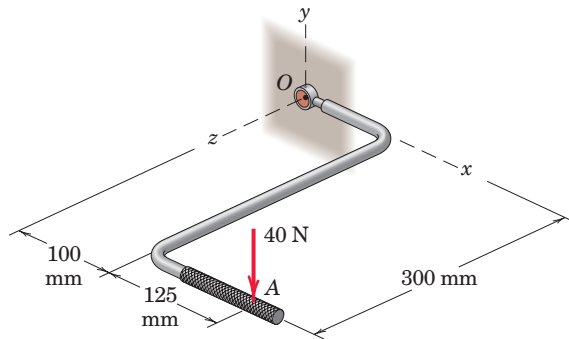
- 2/132** In picking up a load from position B , a cable tension \mathbf{T} of magnitude 24 kN is developed. Calculate the moment which \mathbf{T} produces about the base O of the construction crane.



Problem 2/132

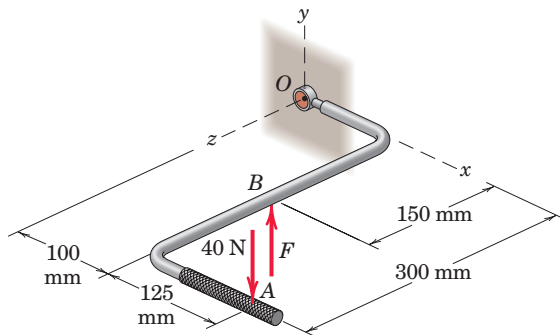
- 2/133** The specialty wrench shown is used for difficult-to-access bolts which have a low torque specification. Determine the moment about O of the 40-N force applied at point A of the wrench handle. Discuss any shortcomings of this wrench.

Ans. $\mathbf{M}_O = 12\mathbf{i} - 9\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/133

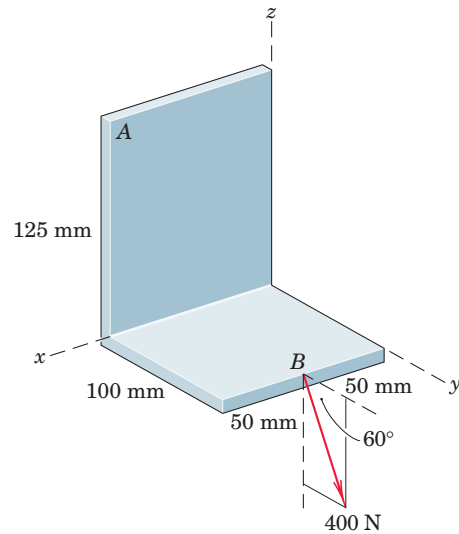
- 2/134** In order to decrease the undesirable moment about the x -axis, a mechanic uses his left hand to support the wrench handle at point B . What upward force F must he exert in order that there be no moment about the x -axis? What then is the net moment about O ?



Problem 2/134

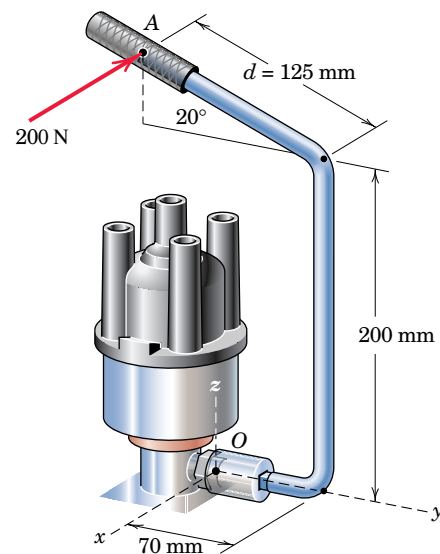
- 2/135** Determine the moment of the 400-N force about point A by (a) using the vector cross-product relation and (b) resolving the force into its components and finding their respective moments.

Ans. $\mathbf{M}_A = -9.64\mathbf{i} - 17.32\mathbf{j} - 10\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/135

- 2/136** The specialty wrench shown in the figure is designed for access to the hold-down bolt on certain automobile distributors. For the configuration shown where the wrench lies in a vertical plane and a horizontal 200-N force is applied at A perpendicular to the handle, calculate the moment \mathbf{M}_O applied to the bolt at O . For what value of the distance d would the z -component of \mathbf{M}_O be zero?

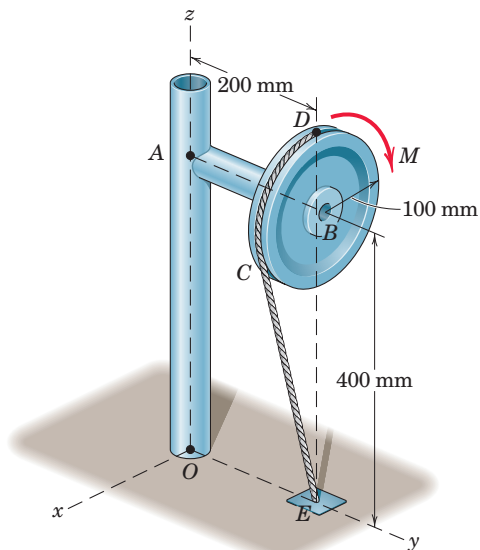


Problem 2/136

86 Chapter 2 Force Systems

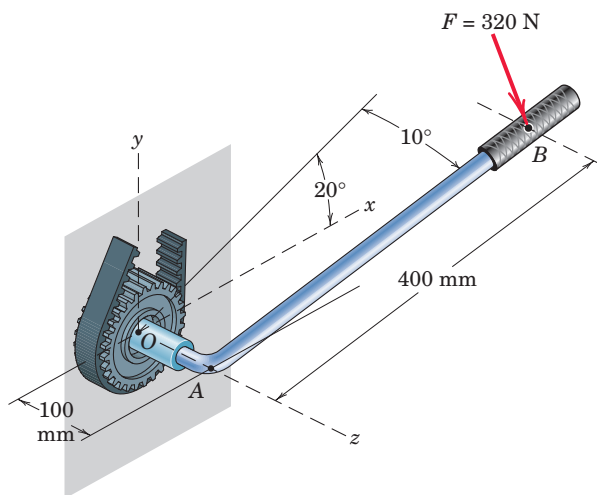
- 2/137** The moment \mathbf{M} applied to the pulley wheel causes a tension $T = 80 \text{ N}$ in the cable which is secured to the wheel at D and to the ground at E . Determine the moment about O of this 80-N force as applied at C .

Ans. $\mathbf{M}_O = -15.49\mathbf{i} + 4\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/137

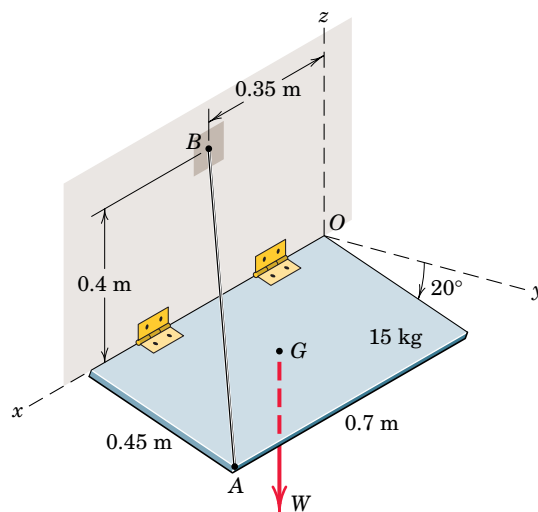
- 2/138** An 320-N force is applied to the end of the wrench as the sprocket wheel is attached to an engine crankshaft. The force is perpendicular to the plane containing points O , A , and B . Determine the moment of this force about point O .



Problem 2/138

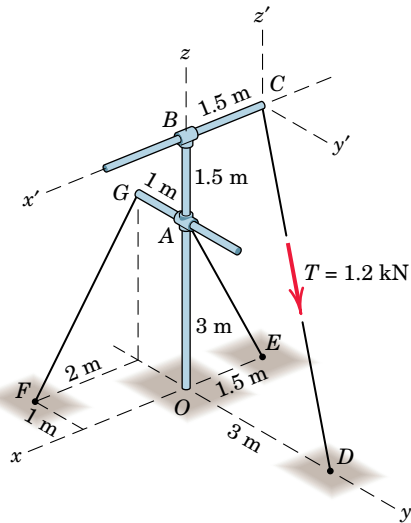
- 2/139** Using the principles to be developed in Chapter 3 on equilibrium, one can determine that the tension in cable AB is 143.4 N. Determine the moment about the x -axis of this tension force acting on point A . Compare your result with the moment of the weight W of the 15-kg uniform plate about the x -axis. What is the moment of the tension force acting at A about line OB ?

Ans. $M_x = 31.1 \text{ N}\cdot\text{m}$, $(M_x)_W = -31.1 \text{ N}\cdot\text{m}$
 $M_{OB} = 0$



Problem 2/139

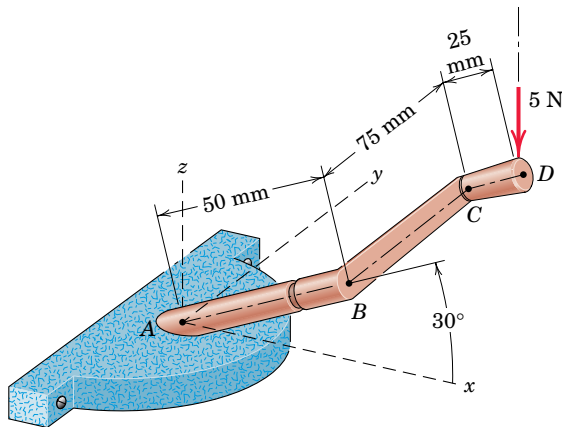
- 2/140** The rigid pole and cross-arm assembly of Prob. 2/105 is shown again here. Determine the vector expression for the moment of the 1.2-kN tension (a) about point O and (b) about the pole z -axis. Find each moment in two different ways.



Problem 2/140

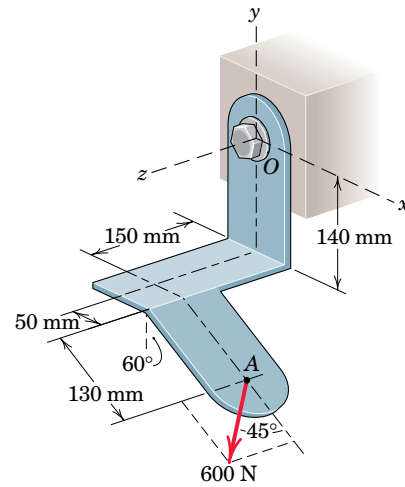
- 2/141** A 5-N vertical force is applied to the knob of the window-opener mechanism when the crank BC is horizontal. Determine the moment of the force about point A and about line AB .

Ans. $\mathbf{M}_A = -375\mathbf{i} + 325\mathbf{j} \text{ N}\cdot\text{mm}$
 $\mathbf{M}_{AB} = -281\mathbf{i} - 162.4\mathbf{k} \text{ N}\cdot\text{mm}$



Problem 2/141

- 2/142** Determine the vector expression for the moment \mathbf{M}_O of the 600-N force about point O . The design specification for the bolt at O would require this result.



Problem 2/142



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The cables of a suspension bridge exert a three-dimensional system of concentrated forces on this bridge tower.

2/9 RESULTANTS

In Art. 2/6 we defined the resultant as the simplest force combination which can replace a given system of forces without altering the external effect on the rigid body on which the forces act. We found the magnitude and direction of the resultant force for the two-dimensional force system by a vector summation of forces, Eq. 2/9, and we located the line of action of the resultant force by applying the principle of moments, Eq. 2/10. These same principles can be extended to three dimensions.

In the previous article we showed that a force could be moved to a parallel position by adding a corresponding couple. Thus, for the system of forces $\mathbf{F}_1, \mathbf{F}_2, \mathbf{F}_3 \dots$ acting on a rigid body in Fig. 2/28a, we may move each of them in turn to the arbitrary point O , provided we also introduce a couple for each force transferred. Thus, for example, we may move force \mathbf{F}_1 to O , provided we introduce the couple $\mathbf{M}_1 = \mathbf{r}_1 \times \mathbf{F}_1$, where \mathbf{r}_1 is a vector from O to any point on the line of action of \mathbf{F}_1 . When all forces are shifted to O in this manner, we have a system of concurrent forces at O and a system of couple vectors, as represented in part *b* of the figure. The concurrent forces may then be added vectorially to produce a resultant force \mathbf{R} , and the couples may also be added to produce a resultant couple \mathbf{M} , Fig. 2/28c. The general force system, then, is reduced to

$$\begin{aligned}\mathbf{R} &= \mathbf{F}_1 + \mathbf{F}_2 + \mathbf{F}_3 + \dots = \Sigma \mathbf{F} \\ \mathbf{M} &= \mathbf{M}_1 + \mathbf{M}_2 + \mathbf{M}_3 + \dots = \Sigma (\mathbf{r} \times \mathbf{F})\end{aligned}\quad (2/20)$$

The couple vectors are shown through point O , but because they are free vectors, they may be represented in any parallel positions. The magnitudes of the resultants and their components are

$$\begin{aligned}R_x &= \Sigma F_x & R_y &= \Sigma F_y & R_z &= \Sigma F_z \\ R &= \sqrt{(\Sigma F_x)^2 + (\Sigma F_y)^2 + (\Sigma F_z)^2} \\ \mathbf{M}_x &= \Sigma (\mathbf{r} \times \mathbf{F})_x & \mathbf{M}_y &= \Sigma (\mathbf{r} \times \mathbf{F})_y & \mathbf{M}_z &= \Sigma (\mathbf{r} \times \mathbf{F})_z \\ M &= \sqrt{M_x^2 + M_y^2 + M_z^2}\end{aligned}\quad (2/21)$$

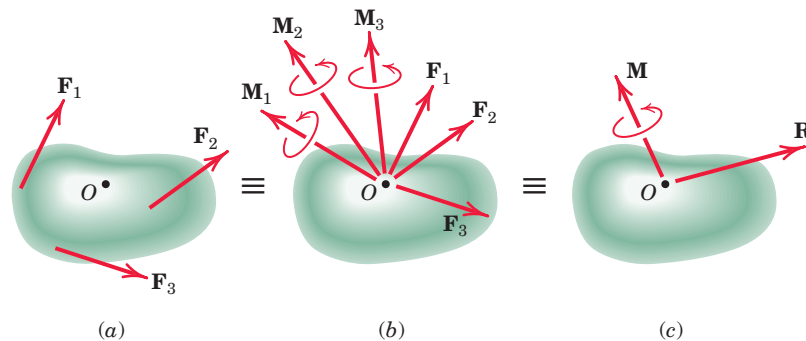


Figure 2/28

The point O selected as the point of concurrency for the forces is arbitrary, and the magnitude and direction of \mathbf{M} depend on the particular point O selected. The magnitude and direction of \mathbf{R} , however, are the same no matter which point is selected.

In general, any system of forces may be replaced by its resultant force \mathbf{R} and the resultant couple \mathbf{M} . In dynamics we usually select the mass center as the reference point. The change in the linear motion of the body is determined by the resultant force, and the change in the angular motion of the body is determined by the resultant couple. In statics, the body is in *complete equilibrium* when the resultant force \mathbf{R} is zero and the resultant couple \mathbf{M} is also zero. Thus, the determination of resultants is essential in both statics and dynamics.

We now examine the resultants for several special force systems.

Concurrent Forces. When forces are concurrent at a point, only the first of Eqs. 2/20 needs to be used because there are no moments about the point of concurrency.

Parallel Forces. For a system of parallel forces not all in the same plane, the magnitude of the parallel resultant force \mathbf{R} is simply the magnitude of the algebraic sum of the given forces. The position of its line of action is obtained from the principle of moments by requiring that $\mathbf{r} \times \mathbf{R} = \mathbf{M}_O$. Here \mathbf{r} is a position vector extending from the force-couple reference point O to the final line of action of \mathbf{R} , and \mathbf{M}_O is the sum of the moments of the individual forces about O . See Sample Problem 2/17 for an example of parallel-force systems.

Coplanar Forces. Article 2/6 was devoted to this force system.

Wrench Resultant. When the resultant couple vector \mathbf{M} is parallel to the resultant force \mathbf{R} , as shown in Fig. 2/29, the resultant is called a *wrench*. By definition a wrench is positive if the couple and force vectors point in the same direction and negative if they point in opposite directions. A common example of a positive wrench is found with the application of a screwdriver, to drive a right-handed screw. Any general force system may be represented by a wrench applied along a unique line of action. This reduction is illustrated in Fig. 2/30, where part *a* of the figure represents, for the general force system, the resultant force \mathbf{R} acting at some point O and the corresponding resultant couple \mathbf{M} . Although \mathbf{M} is a free vector, for convenience we represent it as acting through O .

In part *b* of the figure, \mathbf{M} is resolved into components \mathbf{M}_1 along the direction of \mathbf{R} and \mathbf{M}_2 normal to \mathbf{R} . In part *c* of the figure, the couple \mathbf{M}_2 is replaced by its equivalent of two forces \mathbf{R} and $-\mathbf{R}$ separated by a distance

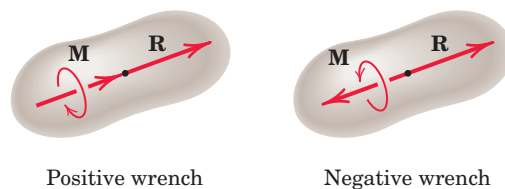


Figure 2/29

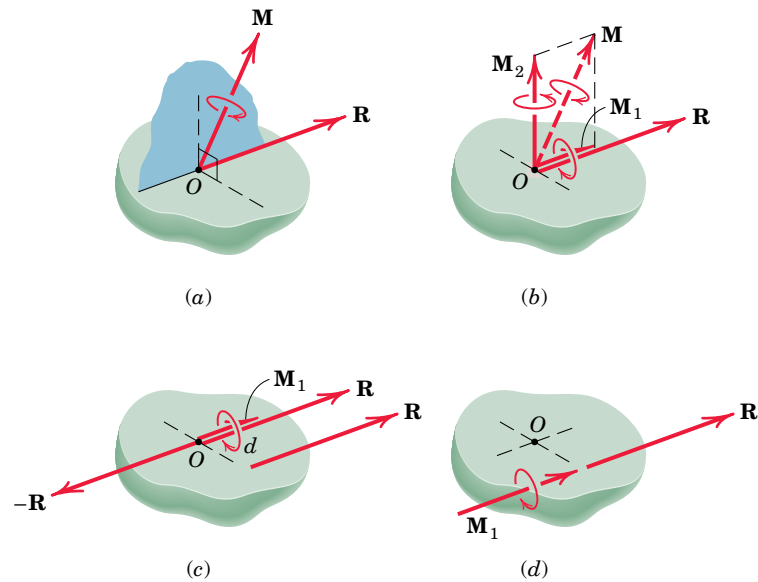


Figure 2/30

$d = M_2/R$ with $-\mathbf{R}$ applied at O to cancel the original \mathbf{R} . This step leaves the resultant \mathbf{R} , which acts along a new and unique line of action, and the parallel couple \mathbf{M}_1 , which is a free vector, as shown in part d of the figure. Thus, the resultants of the original general force system have been transformed into a wrench (positive in this illustration) with its unique axis defined by the new position of \mathbf{R} .

We see from Fig. 2/30 that the axis of the wrench resultant lies in a plane through O normal to the plane defined by \mathbf{R} and \mathbf{M} . The wrench is the simplest form in which the resultant of a general force system may be expressed. This form of the resultant, however, has limited application, because it is usually more convenient to use as the reference point some point O such as the mass center of the body or another convenient origin of coordinates not on the wrench axis.

Sample Problem 2/16

Determine the resultant of the force and couple system which acts on the rectangular solid.

Solution. We choose point O as a convenient reference point for the initial step of reducing the given forces to a force-couple system. The resultant force is

$$\textcircled{1} \quad \mathbf{R} = \Sigma \mathbf{F} = (80 - 80)\mathbf{i} + (100 - 100)\mathbf{j} + (50 - 50)\mathbf{k} = \mathbf{0} \text{ N}$$

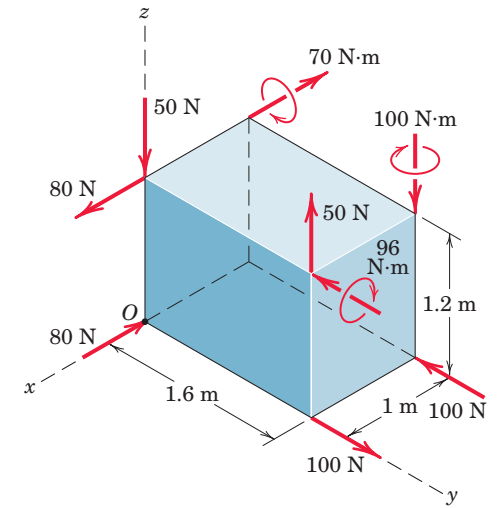
The sum of the moments about O is

$$\textcircled{2} \quad \begin{aligned} \mathbf{M}_O &= [50(1.6) - 70]\mathbf{i} + [80(1.2) - 96]\mathbf{j} + [100(1) - 100]\mathbf{k} \\ &= 10\mathbf{i} \text{ N}\cdot\text{m} \end{aligned}$$

Hence, the resultant consists of a couple, which of course may be applied at any point on the body or the body extended.

Helpful Hints

- ① Since the force summation is zero, we conclude that the resultant, if it exists, must be a couple.
- ② The moments associated with the force pairs are easily obtained by using the $M = Fd$ rule and assigning the unit-vector direction by inspection. In many three-dimensional problems, this may be simpler than the $\mathbf{M} = \mathbf{r} \times \mathbf{F}$ approach.



Sample Problem 2/17

Determine the resultant of the system of parallel forces which act on the plate. Solve with a vector approach.

Solution. Transfer of all forces to point O results in the force-couple system

$$\begin{aligned} \mathbf{R} &= \Sigma \mathbf{F} = (200 + 500 - 300 - 50)\mathbf{j} = 350\mathbf{j} \text{ N} \\ \mathbf{M}_O &= [50(0.35) - 300(0.35)]\mathbf{i} + [-50(0.50) - 200(0.50)]\mathbf{k} \\ &= -87.5\mathbf{i} - 125\mathbf{k} \text{ N}\cdot\text{m} \end{aligned}$$

The placement of \mathbf{R} so that it alone represents the above force-couple system is determined by the principle of moments in vector form

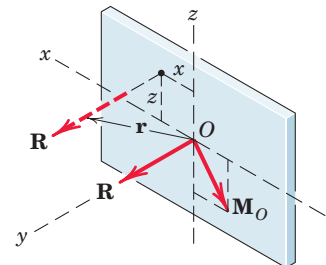
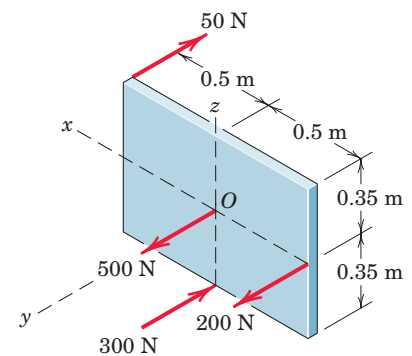
$$\begin{aligned} \mathbf{r} \times \mathbf{R} &= \mathbf{M}_O \\ (x\mathbf{i} + y\mathbf{j} + z\mathbf{k}) \times 350\mathbf{j} &= -87.5\mathbf{i} - 125\mathbf{k} \\ 350x\mathbf{k} - 350z\mathbf{i} &= -87.5\mathbf{i} - 125\mathbf{k} \end{aligned}$$

From the one vector equation we may obtain the two scalar equations

$$350x = -125 \quad \text{and} \quad -350z = -87.5$$

Hence, $x = -0.357 \text{ m}$ and $z = 0.250 \text{ m}$ are the coordinates through which the line of action of \mathbf{R} must pass. The value of y may, of course, be any value, as permitted by the principle of transmissibility. Thus, as expected, the variable y drops out of the above vector analysis.

- ①



Helpful Hint

- ① You should also carry out a scalar solution to this problem.

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Sample Problem 2/18

Replace the two forces and the negative wrench by a single force \mathbf{R} applied at A and the corresponding couple \mathbf{M} .

Solution. The resultant force has the components

$$[R_x = \Sigma F_x] \quad R_x = 500 \sin 40^\circ + 700 \sin 60^\circ = 928 \text{ N}$$

$$[R_y = \Sigma F_y] \quad R_y = 600 + 500 \cos 40^\circ \cos 45^\circ = 871 \text{ N}$$

$$[R_z = \Sigma F_z] \quad R_z = 700 \cos 60^\circ + 500 \cos 40^\circ \sin 45^\circ = 621 \text{ N}$$

$$\text{Thus,} \quad \mathbf{R} = 928\mathbf{i} + 871\mathbf{j} + 621\mathbf{k} \text{ N}$$

$$\text{and} \quad R = \sqrt{(928)^2 + (871)^2 + (621)^2} = 1416 \text{ N}$$

The couple to be added as a result of moving the 500-N force is

$$\textcircled{1} [\mathbf{M} = \mathbf{r} \times \mathbf{F}] \quad \mathbf{M}_{500} = (0.08\mathbf{i} + 0.12\mathbf{j} + 0.05\mathbf{k}) \times 500(\mathbf{i} \sin 40^\circ + \mathbf{j} \cos 40^\circ \cos 45^\circ + \mathbf{k} \cos 40^\circ \sin 45^\circ)$$

where \mathbf{r} is the vector from A to B .

The term-by-term, or determinant, expansion gives

$$\mathbf{M}_{500} = 18.95\mathbf{i} - 5.59\mathbf{j} - 16.90\mathbf{k} \text{ N}\cdot\text{m}$$

- $\textcircled{2}$ The moment of the 600-N force about A is written by inspection of its x - and z -components, which gives

$$\begin{aligned} \mathbf{M}_{600} &= (600)(0.060)\mathbf{i} + (600)(0.040)\mathbf{k} \\ &= 36.0\mathbf{i} + 24.0\mathbf{k} \text{ N}\cdot\text{m} \end{aligned}$$

The moment of the 700-N force about A is easily obtained from the moments of the x - and z -components of the force. The result becomes

$$\begin{aligned} \mathbf{M}_{700} &= (700 \cos 60^\circ)(0.030)\mathbf{i} - [(700 \sin 60^\circ)(0.060) \\ &\quad + (700 \cos 60^\circ)(0.100)]\mathbf{j} - (700 \sin 60^\circ)(0.030)\mathbf{k} \\ &= 10.5\mathbf{i} - 71.4\mathbf{j} - 18.19\mathbf{k} \text{ N}\cdot\text{m} \end{aligned}$$

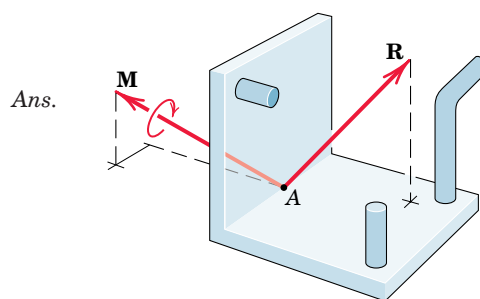
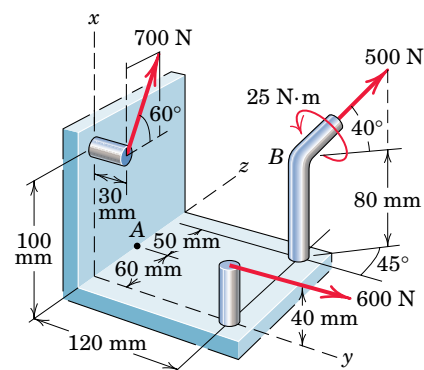
- $\textcircled{3}$ Also, the couple of the given wrench may be written

$$\begin{aligned} \mathbf{M}' &= 25.0(-\mathbf{i} \sin 40^\circ - \mathbf{j} \cos 40^\circ \cos 45^\circ - \mathbf{k} \cos 40^\circ \sin 45^\circ) \\ &= -16.07\mathbf{i} - 13.54\mathbf{j} - 13.54\mathbf{k} \text{ N}\cdot\text{m} \end{aligned}$$

Therefore, the resultant couple on adding together the \mathbf{i} -, \mathbf{j} -, and \mathbf{k} -terms of the four \mathbf{M} 's is

$$\textcircled{4} \quad \mathbf{M} = 49.4\mathbf{i} - 90.5\mathbf{j} - 24.6\mathbf{k} \text{ N}\cdot\text{m}$$

$$\text{and} \quad M = \sqrt{(49.4)^2 + (90.5)^2 + (24.6)^2} = 106.0 \text{ N}\cdot\text{m} \quad \text{Ans.}$$



Helpful Hints

- $\textcircled{1}$ *Suggestion:* Check the cross-product results by evaluating the moments about A of the components of the 500-N force directly from the sketch.
- $\textcircled{2}$ For the 600-N and 700-N forces it is easier to obtain the components of their moments about the coordinate directions through A by inspection of the figure than it is to set up the cross-product relations.
- $\textcircled{3}$ The 25-N·m couple vector of the *wrench* points in the direction opposite to that of the 500-N force, and we must resolve it into its x -, y -, and z -components to be added to the other couple-vector components.
- $\textcircled{4}$ Although the resultant couple vector \mathbf{M} in the sketch of the resultants is shown through A , we recognize that a couple vector is a free vector and therefore has no specified line of action.

Sample Problem 2/19

Determine the wrench resultant of the three forces acting on the bracket. Calculate the coordinates of the point P in the x - y plane through which the resultant force of the wrench acts. Also find the magnitude of the couple \mathbf{M} of the wrench.

Solution. The direction cosines of the couple \mathbf{M} of the wrench must be the same as those of the resultant force \mathbf{R} , assuming that the wrench is positive. The resultant force is

$$\mathbf{R} = 20\mathbf{i} + 40\mathbf{j} + 40\mathbf{k} \quad R = \sqrt{(20)^2 + (40)^2 + (40)^2} = 60 \text{ N}$$

and its direction cosines are

$$\cos \theta_x = 20/60 = 1/3 \quad \cos \theta_y = 40/60 = 2/3 \quad \cos \theta_z = 40/60 = 2/3$$

The moment of the wrench couple must equal the sum of the moments of the given forces about point P through which \mathbf{R} passes. The moments about P of the three forces are

$$(\mathbf{M})_{R_x} = 20y\mathbf{k} \text{ N}\cdot\text{mm}$$

$$(\mathbf{M})_{R_y} = -40(60)\mathbf{i} - 40x\mathbf{k} \text{ N}\cdot\text{mm}$$

$$(\mathbf{M})_{R_z} = 40(80 - y)\mathbf{i} - 40(100 - x)\mathbf{j} \text{ N}\cdot\text{mm}$$

and the total moment is

$$\mathbf{M} = (800 - 40y)\mathbf{i} + (-4000 + 40x)\mathbf{j} + (-40x + 20y)\mathbf{k} \text{ N}\cdot\text{mm}$$

The direction cosines of \mathbf{M} are

$$\cos \theta_x = (800 - 40y)/M$$

$$\cos \theta_y = (-4000 + 40x)/M$$

$$\cos \theta_z = (-40x + 20y)/M$$

where M is the magnitude of \mathbf{M} . Equating the direction cosines of \mathbf{R} and \mathbf{M} gives

$$800 - 40y = \frac{M}{3}$$

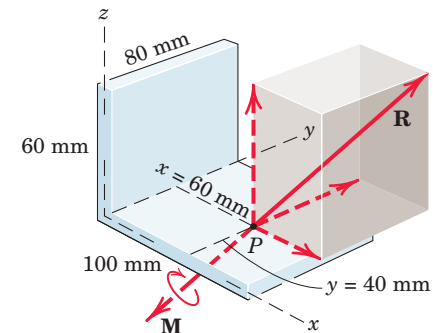
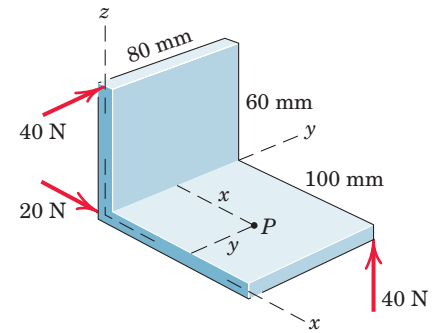
$$-4000 + 40x = \frac{2M}{3}$$

$$-40x + 20y = \frac{2M}{3}$$

Solution of the three equations gives

$$M = -2400 \text{ N}\cdot\text{mm} \quad x = 60 \text{ mm} \quad y = 40 \text{ mm} \quad \text{Ans.}$$

We see that M turned out to be negative, which means that the couple vector is pointing in the direction opposite to \mathbf{R} , which makes the wrench negative.



Helpful Hint

- ① We assume initially that the wrench is positive. If \mathbf{M} turns out to be negative, then the direction of the couple vector is opposite to that of the resultant force.

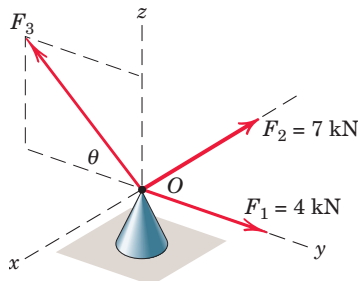
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PROBLEMS

Introductory Problems

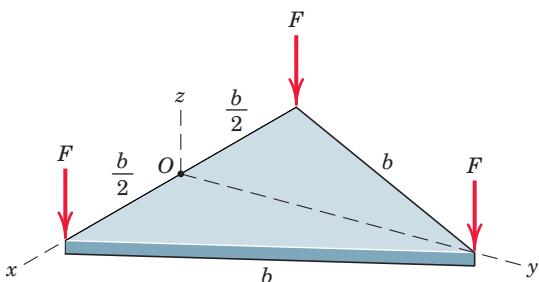
- 2/143** Three forces act at point O . If it is known that the y -component of the resultant \mathbf{R} is -5 kN and that the z -component is 6 kN, determine F_3 , θ , and R .

Ans. $F_3 = 10.82$ kN, $\theta = 33.7^\circ$, $R = 10.49$ kN



Problem 2/143

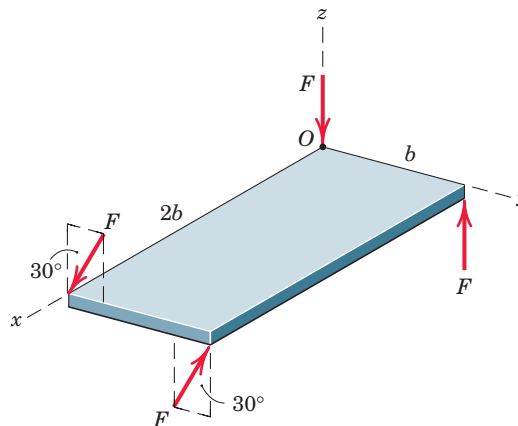
- 2/144** Three equal forces are exerted on the equilateral plate as shown. Reduce the force system to an equivalent force-couple system at point O . Show that \mathbf{R} is perpendicular to \mathbf{M}_O .



Problem 2/144

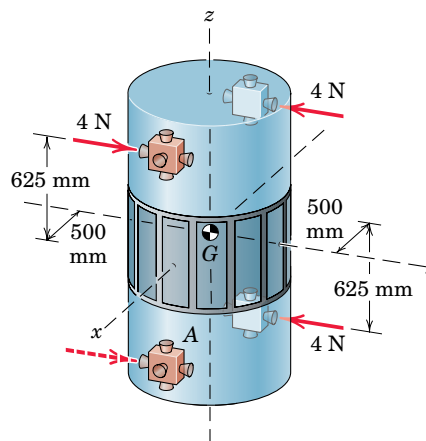
- 2/145** The thin rectangular plate is subjected to the four forces shown. Determine the equivalent force-couple system at O . What is the resultant of the system?

$$\text{Ans. } \mathbf{R} = \mathbf{0}, \mathbf{M}_O = Fb \left(1 + \frac{\sqrt{3}}{2} \right) \mathbf{i}$$



Problem 2/145

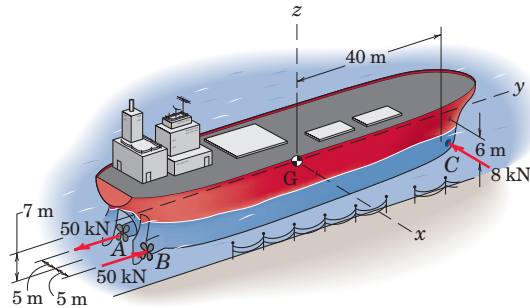
- 2/146** The spacecraft of Prob. 2/129 is repeated here. The plan is to fire four 4-N thrusters as shown in order to spin up the spacecraft about its z -axis, but the thruster at A fails. Determine the equivalent force-couple system at G for the remaining three thrusters.



Problem 2/146

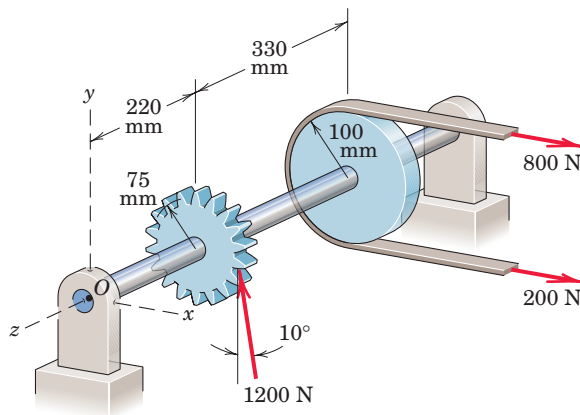
- 2/147** An oil tanker moves away from its docked position under the action of reverse thrust from screw *A*, forward thrust from screw *B*, and side thrust from the bow thruster *C*. Determine the equivalent force-couple system at the mass center *G*.

Ans. $\mathbf{R} = -8\mathbf{i}$ kN, $\mathbf{M}_G = 48\mathbf{j} + 820\mathbf{k}$ kN·m



Problem 2/147

- 2/148** The pulley and gear are subjected to the loads shown. For these forces, determine the equivalent force-couple system at point *O*.

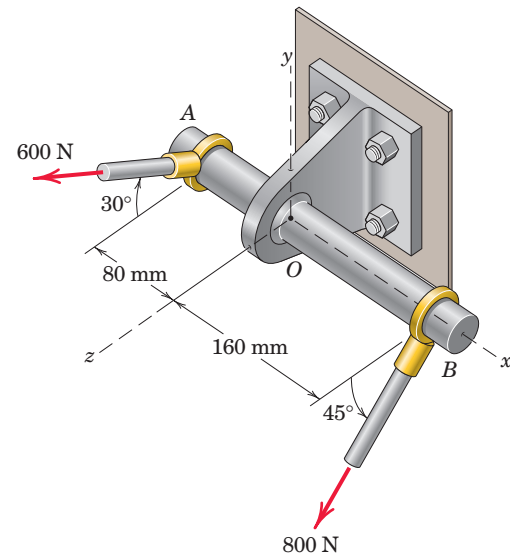


Problem 2/148

- 2/149** Determine the force-couple system at *O* which is equivalent to the two forces applied to the shaft *AOB*. Is \mathbf{R} perpendicular to \mathbf{M}_O ?

Ans. $\mathbf{R} = -266\mathbf{j} + 1085\mathbf{k}$ N

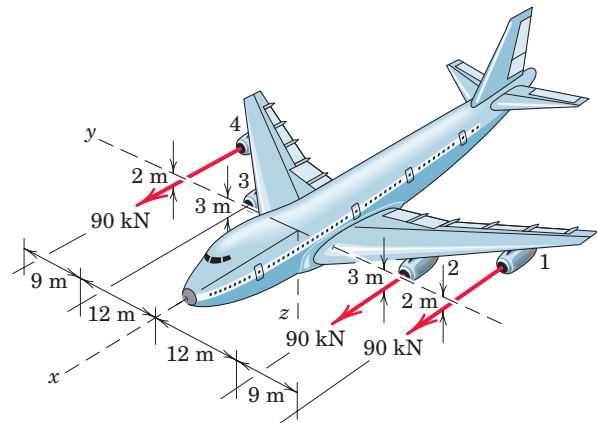
$\mathbf{M}_O = -48.9\mathbf{j} - 114.5\mathbf{k}$ N·m



Problem 2/149

Representative Problems

- 2/150** The commercial airliner of Prob. 2/86 is redrawn here with three-dimensional information supplied. If engine 3 suddenly fails, determine the resultant of the three remaining engine thrust vectors, each of which has a magnitude of 90 kN. Specify the *y*- and *z*-coordinates of the point through which the line of action of the resultant passes. This information would be critical to the design criteria of performance with engine failure.



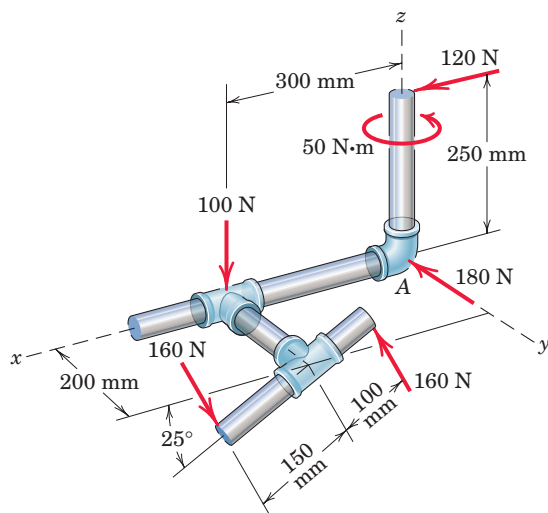
Problem 2/150

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- 2/151** Represent the resultant of the force system acting on the pipe assembly by a single force \mathbf{R} at A and a couple \mathbf{M} .

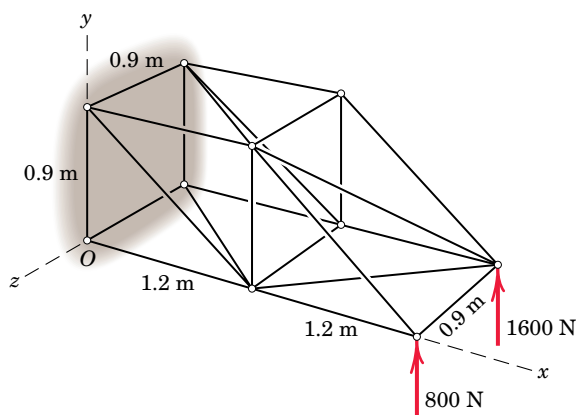
$$\text{Ans. } \mathbf{R} = 120\mathbf{i} - 180\mathbf{j} - 100\mathbf{k} \text{ N}$$

$$\mathbf{M}_A = 100\mathbf{j} + 50\mathbf{k} \text{ N}\cdot\text{m}$$



Problem 2/151

- 2/152** Two upward loads are exerted on the small three-dimensional truss. Reduce these two loads to a single force-couple system at point O. Show that \mathbf{R} is perpendicular to \mathbf{M}_O . Then determine the point in the x - z plane through which the resultant passes.

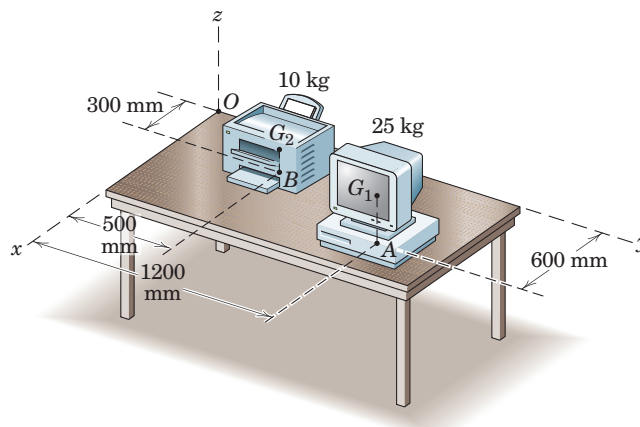


Problem 2/152

- 2/153** A 25-kg computer and a 10-kg laser printer rest on a horizontal tabletop. Determine the resultant of the system of two corresponding weights and specify the coordinates of the point P in the x - y plane through which the resultant passes. The two weights act at the points G_1 and G_2 .

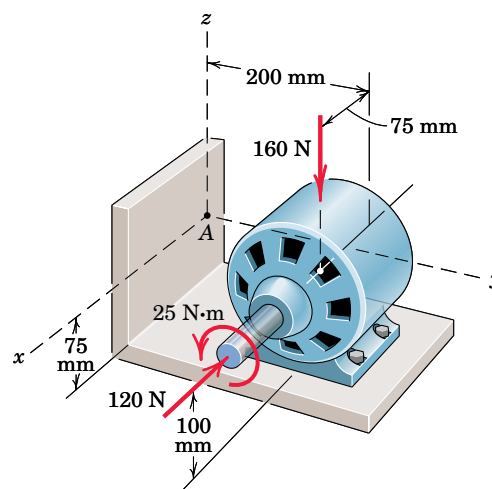
$$\text{Ans. } \mathbf{R} = -343\mathbf{k} \text{ N}, \mathbf{M}_O = -343\mathbf{i} + 176.6\mathbf{j} \text{ N}\cdot\text{m}$$

$$(x, y) = (514, 1000) \text{ mm}$$



Problem 2/153

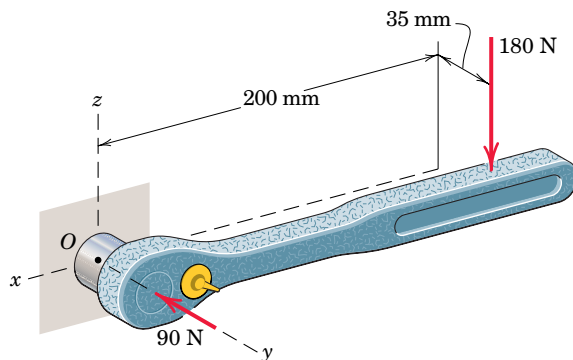
- 2/154** The motor mounted on the bracket is acted on by its 160-N weight, and its shaft resists the 120-N thrust and 25-N·m couple applied to it. Determine the resultant of the force system shown in terms of a force \mathbf{R} at A and a couple \mathbf{M} .



Problem 2/154

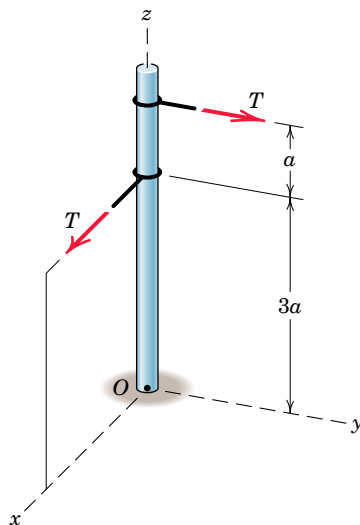
- 2/155** In tightening a bolt whose center is at point O , a person exerts a 180-N force on the ratchet handle with his right hand. In addition, with his left hand he exerts a 90-N force as shown in order to secure the socket onto the bolt head. Determine the equivalent force-couple system at O . Then find the point in the x - y plane through which the line of action of the resultant force of the wrench passes.

Ans. $\mathbf{R} = -90\mathbf{j} - 180\mathbf{k}$ N, $\mathbf{M}_O = -6.3\mathbf{i} - 36\mathbf{j}$ N·m
 $x = -160$ mm, $y = 35$ mm



Problem 2/155

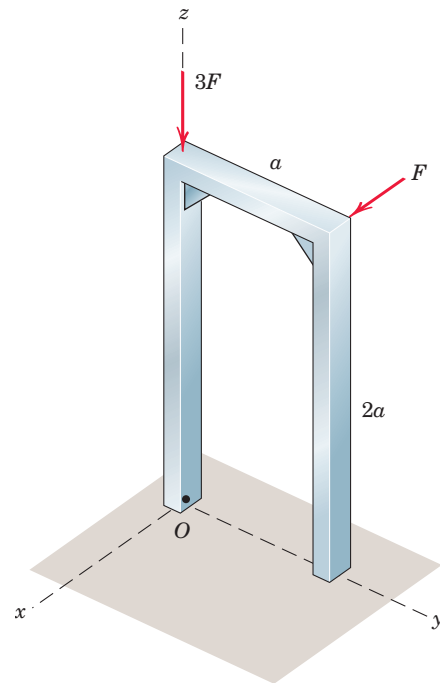
- 2/156** Replace the two forces acting on the pole by a wrench. Write the moment \mathbf{M} associated with the wrench as a vector and specify the coordinates of the point P in the y - z plane through which the line of action of the wrench passes.



Problem 2/156

- 2/157** Replace the two forces acting on the frame by a wrench. Write the moment associated with the wrench as a vector and specify the coordinates of the point P in the y - z plane through which the line of action of the wrench passes. Note that the force of magnitude F is parallel to the x -axis.

Ans. $\mathbf{R} = F(\mathbf{i} - 3\mathbf{k})$, $\mathbf{M} = \frac{3aF}{10}(\mathbf{i} - 3\mathbf{k})$
 $y = \frac{a}{10}$, $z = 2a$



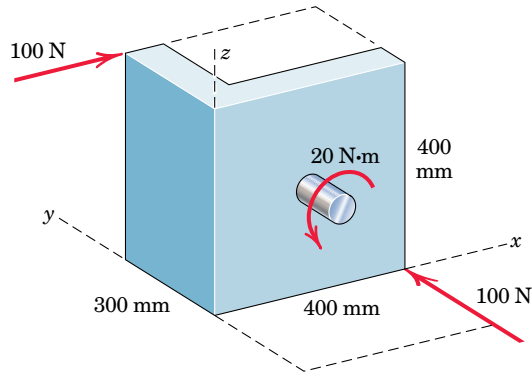
Problem 2/157

- 2/158** For the system of two forces in Prob. 2/149, determine the coordinates of the point in the x - z plane through which the line of action of the resultant of the system passes.

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- **2/159** The resultant of the two forces and couple may be represented by a wrench. Determine the vector expression for the moment \mathbf{M} of the wrench and find the coordinates of the point P in the x - z plane through which the resultant force of the wrench passes.

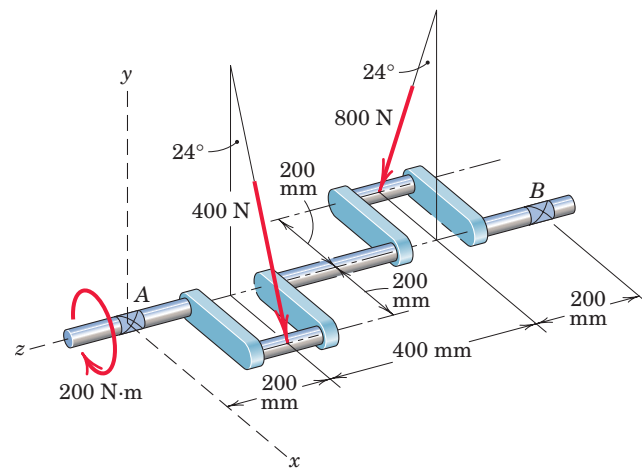
Ans. $\mathbf{M} = 10\mathbf{i} + 10\mathbf{j} \text{ N}\cdot\text{m}$
 $x = z = 0.1 \text{ m}$



Problem 2/159

- **2/160** For the position shown, the crankshaft of a small two-cylinder compressor is subjected to the 400-N and 800-N forces exerted by the connecting rods and the 200-N·m couple. Replace this loading system by a force-couple system at point A. Show that \mathbf{R} is not perpendicular to \mathbf{M}_A . Then replace this force-couple system by a wrench. Determine the magnitude M of the moment of the wrench, the magnitude of the force \mathbf{R} of the wrench, and the coordinates of the point in the x - z plane through which the line of action of the wrench passes.

Ans. $M = 85.8 \text{ N}\cdot\text{m}$, $R = 1108 \text{ N}$
 $x = 0.1158 \text{ m}$, $z = -0.478 \text{ m}$



Problem 2/160

2/10 CHAPTER REVIEW

In Chapter 2 we have established the properties of forces, moments, and couples, and the correct procedures for representing their effects. Mastery of this material is essential for our study of equilibrium in the chapters which follow. Failure to correctly use the procedures of Chapter 2 is a common cause of errors in applying the principles of equilibrium. When difficulties arise, you should refer to this chapter to be sure that the forces, moments, and couples are correctly represented.

Forces

There is frequent need to represent forces as vectors, to resolve a single force into components along desired directions, and to combine two or more concurrent forces into an equivalent resultant force. Specifically, you should be able to:

1. Resolve a given force vector into its components along given directions, and express the vector in terms of the unit vectors along a given set of axes.
2. Express a force as a vector when given its magnitude and information about its line of action. This information may be in the form of two points along the line of action or angles which orient the line of action.
3. Use the dot product to compute the projection of a vector onto a specified line and the angle between two vectors.
4. Compute the resultant of two or more forces concurrent at a point.

Moments

The tendency of a force to rotate a body about an axis is described by a moment (or torque), which is a vector quantity. We have seen that finding the moment of a force is often facilitated by combining the moments of the components of the force. When working with moment vectors you should be able to:

1. Determine a moment by using the moment-arm rule.
2. Use the vector cross product to compute a moment vector in terms of a force vector and a position vector locating the line of action of the force.
3. Utilize Varignon's theorem to simplify the calculation of moments, in both scalar and vector forms.
4. Use the triple scalar product to compute the moment of a force vector about a given axis through a given point.

Couples

A couple is the combined moment of two equal, opposite, and non-collinear forces. The unique effect of a couple is to produce a pure twist or rotation regardless of where the forces are located. The couple is useful in replacing a force acting at a point by a force-couple system at

a different point. To solve problems involving couples you should be able to:

1. Compute the moment of a couple, given the couple forces and either their separation distance or any position vectors locating their lines of action.
2. Replace a given force by an equivalent force–couple system, and vice versa.

Resultants

We can reduce an arbitrary system of forces and couples to a single resultant force applied at an arbitrary point, and a corresponding resultant couple. We can further combine this resultant force and couple into a wrench to give a single resultant force along a unique line of action, together with a parallel couple vector. To solve problems involving resultants you should be able to:

1. Compute the magnitude, direction, and line of action of the resultant of a system of coplanar forces if that resultant is a force; otherwise, compute the moment of the resultant couple.
2. Apply the principle of moments to simplify the calculation of the moment of a system of coplanar forces about a given point.
3. Replace a given general force system by a wrench along a specific line of action.

Equilibrium

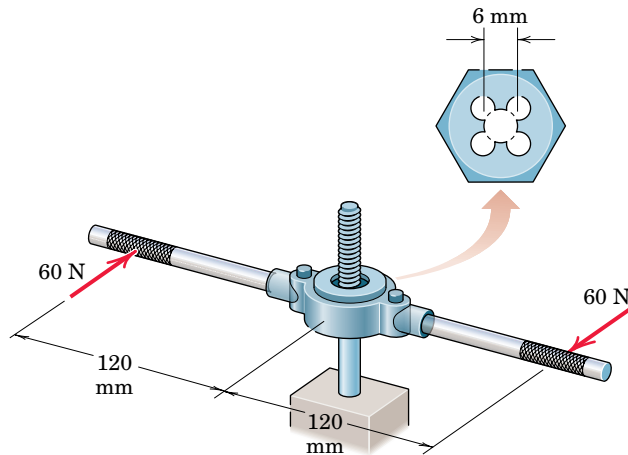
You will use the preceding concepts and methods when you study equilibrium in the following chapters. Let us summarize the concept of equilibrium:

1. When the resultant force on a body is zero ($\Sigma \mathbf{F} = \mathbf{0}$), the body is in *translational* equilibrium. This means that its center of mass is either at rest or moving in a straight line with constant velocity.
2. In addition, if the resultant couple is zero ($\Sigma \mathbf{M} = \mathbf{0}$), the body is in *rotational* equilibrium, either having no rotational motion or rotating with a constant angular velocity.
3. When both resultants are zero, the body is in *complete* equilibrium.

REVIEW PROBLEMS

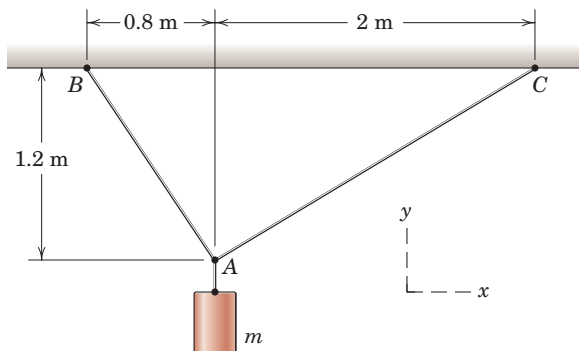
- 2/161** A die is being used to cut threads on a rod. If 60-N forces are applied as shown, determine the magnitude F of the equal forces exerted on the 6-mm rod by each of the four cutting surfaces so that their external effect on the rod is equivalent to that of the two 60-N forces.

Ans. $F = 1200 \text{ N}$



Problem 2/161

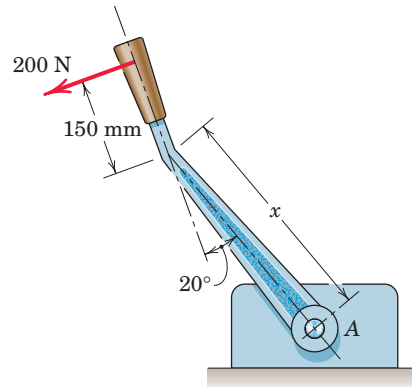
- 2/162** Using the principles of equilibrium to be developed in Chapter 3, you will soon be able to verify that the tension in cable AB is 85.8% of the weight of the cylinder of mass m , while the tension in cable AC is 55.5% of the suspended weight. Write each tension force acting on point A as a vector if the mass m is 60 kg.



Problem 2/162

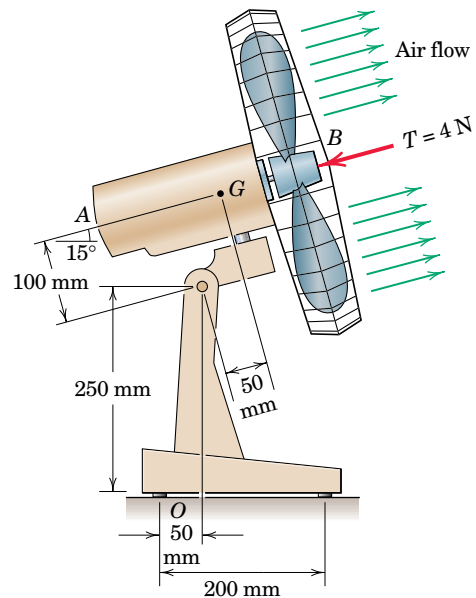
- 2/163** The control lever is subjected to a clockwise couple of $80 \text{ N}\cdot\text{m}$ exerted by its shaft at A and is designed to operate with a 200-N pull as shown. If the resultant of the couple and the force passes through A , determine the proper dimension x of the lever.

Ans. $x = 266 \text{ mm}$



Problem 2/163

- 2/164** The blades of the portable fan generate a 4-N thrust T as shown. Compute the moment M_O of this force about the rear support point O . For comparison, determine the moment about O due to the weight of the motor-fan unit AB , whose weight of 40 N acts at G .

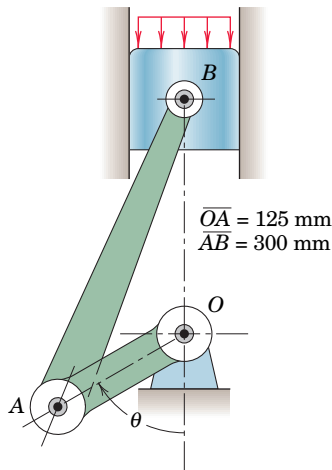


Problem 2/164

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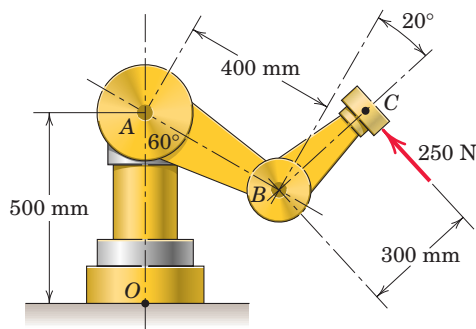
- 2/165** For the angular position $\theta = 60^\circ$ of the crank OA , the gas pressure on the piston induces a compressive force P in the connecting rod along its centerline AB . If this force produces a moment of $720 \text{ N}\cdot\text{m}$ about the crank axis O , calculate P .

Ans. $P = 9.18 \text{ kN}$



Problem 2/165

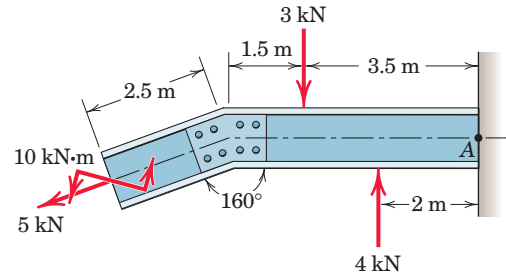
- 2/166** Calculate the moment M_O of the 250-N force about the base point O of the robot.



Problem 2/166

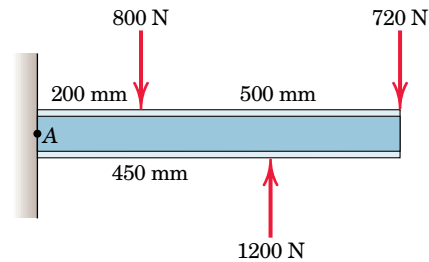
- 2/167** Represent the resultant of the three forces and couple by a force-couple system located at point A .

Ans. $R = 4.75 \text{ kN}$, $M_A = 21.1 \text{ kN}\cdot\text{m}$ CCW



Problem 2/167

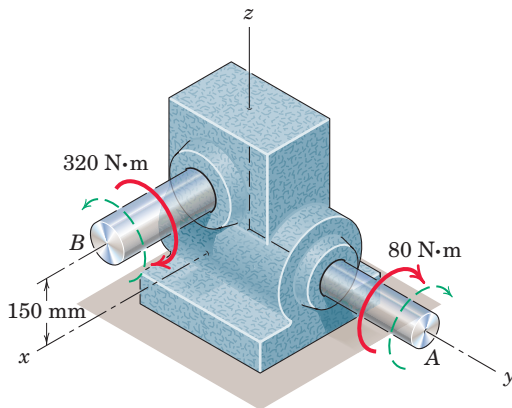
- 2/168** Reduce the given loading system to a force-couple system at point A . Then determine the distance x to the right of point A at which the resultant of the three forces acts.



Problem 2/168

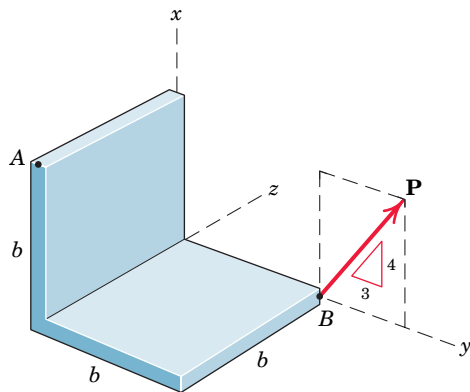
- 2/169** The directions of rotation of the input shaft *A* and output shaft *B* of the worm-gear reducer are indicated by the curved dashed arrows. An input torque (couple) of $80 \text{ N}\cdot\text{m}$ is applied to shaft *A* in the direction of rotation. The output shaft *B* supplies a torque of $320 \text{ N}\cdot\text{m}$ to the machine which it drives (not shown). The shaft of the driven machine exerts an equal and opposite reacting torque on the output shaft of the reducer. Determine the resultant \mathbf{M} of the two couples which act on the reducer unit and calculate the direction cosine of \mathbf{M} with respect to the *x*-axis.

$$\text{Ans. } \mathbf{M} = -320\mathbf{i} - 80\mathbf{j} \text{ N}\cdot\text{m} \\ \cos \theta_x = -0.970$$



Problem 2/169

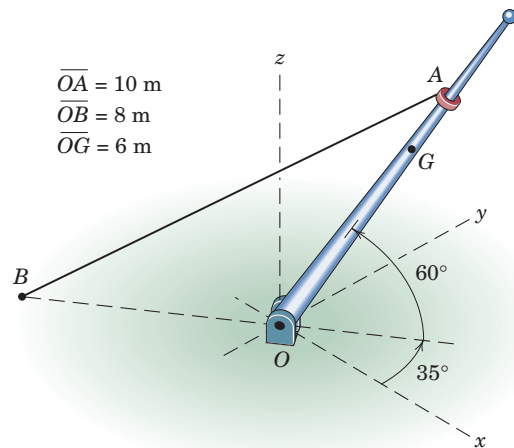
- 2/170** Determine the moment of the force \mathbf{P} about point *A*.



Problem 2/170

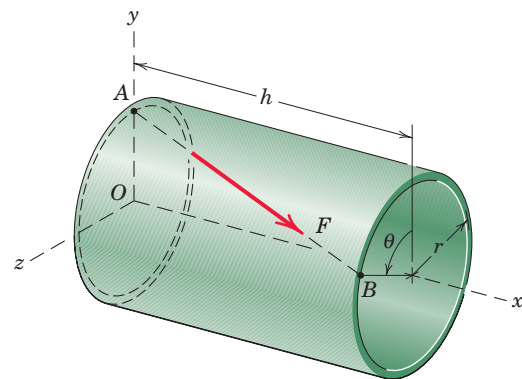
- 2/171** When the pole *OA* is in the position shown, the tension in cable *AB* is 3 kN . (a) Write the tension force exerted on the small collar at point *A* as a vector using the coordinates shown. (b) Determine the moment of this force about point *O* and state the moments about the *x*-, *y*-, and *z*-axes. (c) Determine the projection of this tension force onto line *AO*.

$$\text{Ans. (a) } \mathbf{T}_{AB} = -2.05\mathbf{i} - 1.432\mathbf{j} - 1.663\mathbf{k} \text{ kN} \\ \text{(b) } \mathbf{M}_O = 7.63\mathbf{i} - 10.90\mathbf{j} \text{ kN}\cdot\text{m} \\ \text{(c) } T_{AO} = 2.69 \text{ kN}$$



Problem 2/171

- 2/172** A force \mathbf{F} acts along the line *AB* inside the right circular cylindrical shell as shown. The quantities *r*, *h*, θ , and *F* are known. Using the *x*-, *y*-, and *z*-coordinates shown, express \mathbf{F} as a vector.

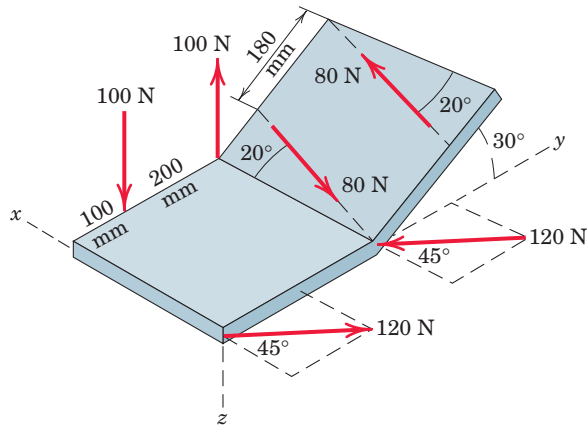


Problem 2/172

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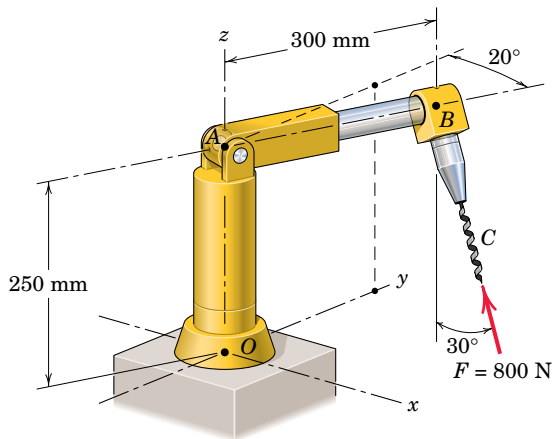
- 2/173** Three couples are formed by the three pairs of equal and opposite forces. Determine the resultant \mathbf{M} of the three couples.

Ans. $\mathbf{M} = -20\mathbf{i} - 6.77\mathbf{j} - 37.2\mathbf{k} \text{ N}\cdot\text{m}$



Problem 2/173

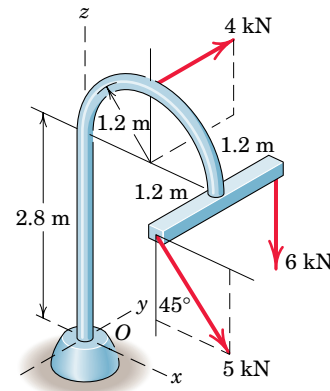
- 2/174** During a drilling operation, the small robotic device is subjected to an 800-N force at point C as shown. Replace this force by an equivalent force-couple system at point O .



Problem 2/174

- 2/175** The combined action of the three forces on the base at O may be obtained by establishing their resultant through O . Determine the magnitudes of \mathbf{R} and the accompanying couple \mathbf{M} .

Ans. $R = 10.93 \text{ kN}$, $M = 38.9 \text{ kN}\cdot\text{m}$

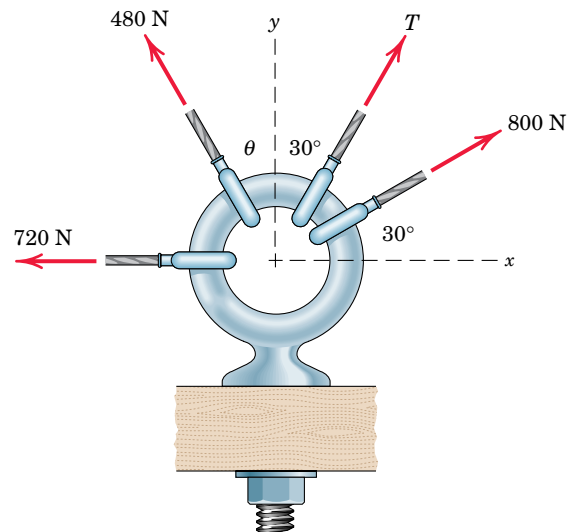


Problem 2/175



*Computer-Oriented Problems

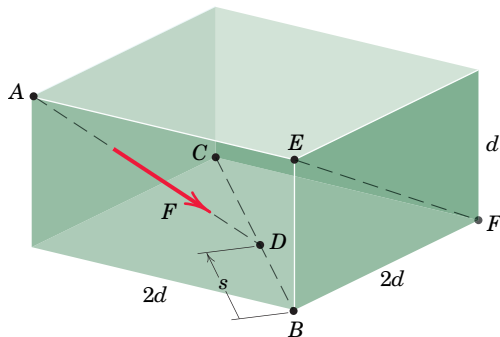
- *2/176** Four forces are exerted on the eyebolt as shown. If the net effect on the bolt is a direct pull of 1200 N in the y -direction, determine the necessary values of T and θ .



Problem 2/176

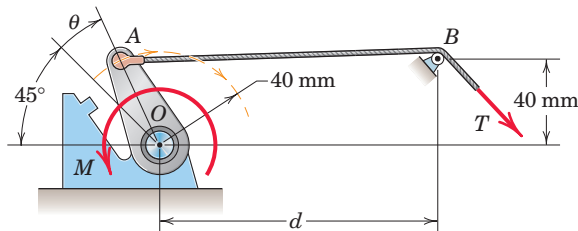
- *2/177** The force \mathbf{F} is directed from A toward D and D is allowed to move from B to C as measured by the variable s . Consider the projection of \mathbf{F} onto line EF as a function of s . In particular, determine and plot the fraction n of the magnitude F which is projected as a function of s/d . Note that s/d varies from 0 to $2\sqrt{2}$.

$$\text{Ans. } n = \frac{\sqrt{2} \frac{s}{d} + 1}{\sqrt{5} \sqrt{\left(\frac{s}{d}\right)^2 + 5} - 2\sqrt{2} \frac{s}{d}}$$



Problem 2/177

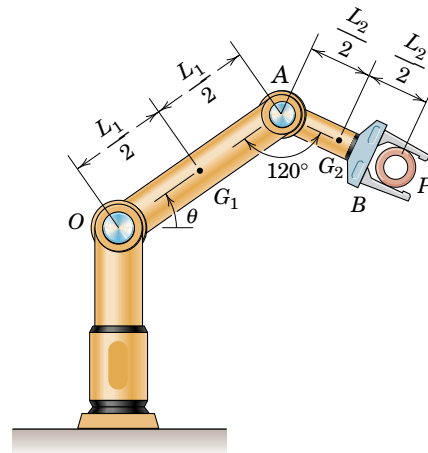
- *2/178** The throttle-control lever OA rotates in the range $0 \leq \theta \leq 90^\circ$. An internal torsional return spring exerts a restoring moment about O given by $M = K(\theta + \pi/4)$, where $K = 500 \text{ N}\cdot\text{mm}/\text{rad}$ and θ is in radians. Determine and plot as a function of θ the tension T required to make the net moment about O zero. Use the two values $d = 60 \text{ mm}$ and $d = 160 \text{ mm}$ and comment on the relative design merits. The effects of the radius of the pulley at B are negligible.



Problem 2/178

- *2/179** With the cylindrical part P of weight 1500 N in its grip, the robotic arm pivots about O through the range $-45^\circ \leq \theta \leq 45^\circ$ with the angle at A locked at 120° . Determine and plot (as a function of θ) the moment at O due to the combined effects of the weight of part P , the 600-N weight of member OA (mass center at G_1), and the 250-N weight of member AB (mass center at G_2). The end grip is included as a part of member AB . The lengths L_1 and L_2 are 900 mm and 600 mm , respectively. What is the maximum value of M_O and at what value of θ does this maximum occur?

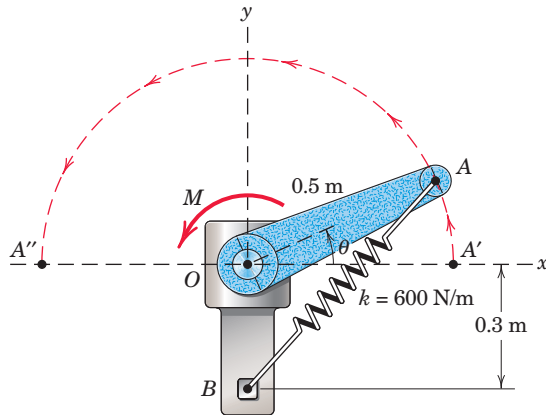
$$\text{Ans. } M_O = 1845 \cos \theta + 975 \cos(60^\circ - \theta) \text{ N}\cdot\text{m} \\ (M_O)_{\max} = 2480 \text{ N}\cdot\text{m at } \theta = 19.90^\circ$$



Problem 2/179

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- *2/180** A motor attached to the shaft at O causes the arm OA to rotate over the range $0 \leq \theta \leq 180^\circ$. The unstretched length of the spring is 0.65 m, and it can support both tension and compression. If the net moment about O must be zero, determine and plot the required motor torque M as a function of θ .

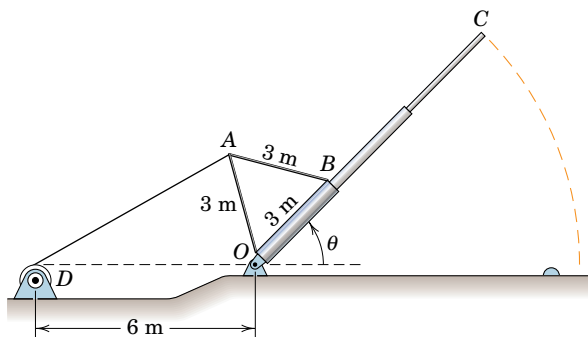


Problem 2/180

- *2/181** A flagpole with attached light triangular frame is shown here for an arbitrary position during its raising. The 75 -N tension in the erecting cable remains constant. Determine and plot the moment about the pivot O of the 75 -N force for the range $0 \leq \theta \leq 90^\circ$. Determine the maximum value of this moment and the elevation angle at which it occurs; comment on the physical significance of the latter. The effects of the diameter of the drum at D may be neglected.

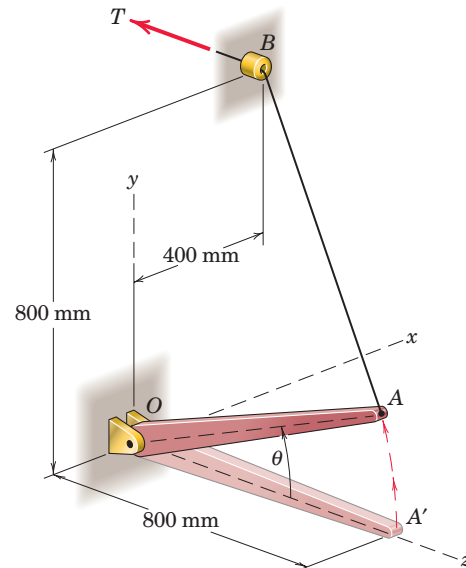
$$\text{Ans. } \mathbf{M}_O = \frac{1350 \sin(\theta + 60^\circ)}{\sqrt{45 + 36 \cos(\theta + 60^\circ)}} \mathbf{k} \text{ N} \cdot \text{m}$$

$$(M_O)_{\max} = 225 \text{ N} \cdot \text{m at } \theta = 60^\circ$$



Problem 2/181

- *2/182** The tension T in cable AB is maintained at a constant value of 120 N. Determine the moment \mathbf{M}_O of this tension about point O over the range $0 \leq \theta \leq 90^\circ$. Plot the x -, y -, and z -components of \mathbf{M}_O as functions of θ .



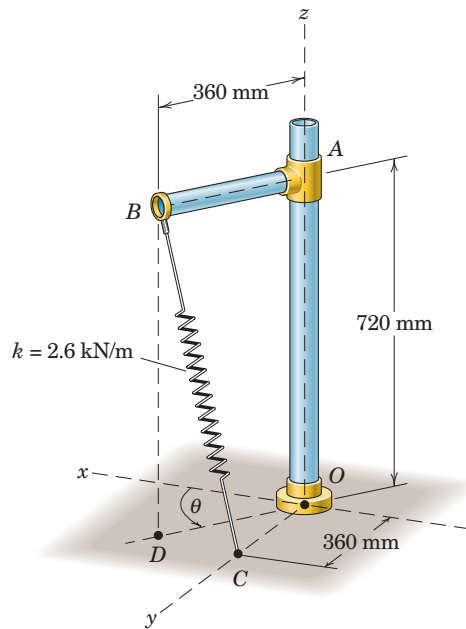
Problem 2/182

***2/183** The arm AB rotates in the range $0 \leq \theta \leq 180^\circ$, and the spring is unstretched when $\theta = 90^\circ$. Determine as a function of θ the moment about O of the spring force as applied at B . Plot the three scalar components of \mathbf{M}_O , and state the maximum absolute value of each component.

$$\text{Ans. } \mathbf{M}_O = \frac{\sqrt{1.5 - 0.5 \sin \theta} - 1}{\sqrt{1.5 - 0.5 \sin \theta}} [-674\mathbf{i} + 337 \cos \theta \mathbf{k}] \text{ N}\cdot\text{m}$$

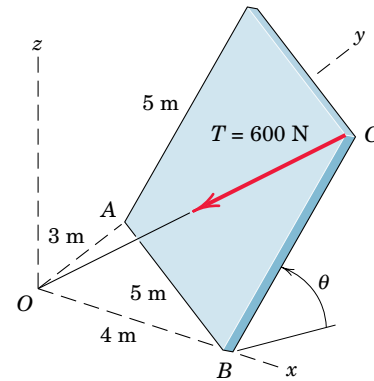
$$(M_{O_x})_{\max} = 123.7 \text{ N}\cdot\text{m at } \theta = 0, 180^\circ$$

$$(M_{O_z})_{\max} = 61.8 \text{ N}\cdot\text{m at } \theta = 0, 180^\circ$$



Problem 2/183

***2/184** The rectangular plate is tilted about its lower edge by a cable tensioned at a constant 600 N. Determine and plot the moment of this tension about the lower edge AB of the plate for the range $0 \leq \theta \leq 90^\circ$.



Problem 2/184



© Peter Bialobrzewski/Laif/Redux

In many applications of mechanics, the sum of the forces acting on a body is zero, and a state of equilibrium exists. This apparatus is designed to hold a car body in equilibrium for a wide range of orientations during vehicle production.

3

EQUILIBRIUM

CHAPTER OUTLINE

3/1 Introduction

SECTION A EQUILIBRIUM IN TWO DIMENSIONS

3/2 System Isolation and the Free-Body Diagram

3/3 Equilibrium Conditions

SECTION B EQUILIBRIUM IN THREE DIMENSIONS

3/4 Equilibrium Conditions

3/5 Chapter Review

3/1 INTRODUCTION

Statics deals primarily with the description of the force conditions necessary and sufficient to maintain the equilibrium of engineering structures. This chapter on equilibrium, therefore, constitutes the most important part of statics, and the procedures developed here form the basis for solving problems in both statics and dynamics. We will make continual use of the concepts developed in Chapter 2 involving forces, moments, couples, and resultants as we apply the principles of equilibrium.

When a body is in equilibrium, the resultant of *all* forces acting on it is zero. Thus, the resultant force \mathbf{R} and the resultant couple \mathbf{M} are both zero, and we have the equilibrium equations

$$\mathbf{R} = \Sigma \mathbf{F} = \mathbf{0} \quad \mathbf{M} = \Sigma \mathbf{M} = \mathbf{0} \quad (3/1)$$

These requirements are both necessary and sufficient conditions for equilibrium.

All physical bodies are three-dimensional, but we can treat many of them as two-dimensional when the forces to which they are subjected act in a single plane or can be projected onto a single plane. When this simplification is not possible, the problem must be treated as three-dimensional. We will follow the arrangement used in Chapter 2, and discuss in Section A the equilibrium of bodies subjected to two-dimensional

force systems and in Section B the equilibrium of bodies subjected to three-dimensional force systems.

SECTION A EQUILIBRIUM IN TWO DIMENSIONS

3/2 SYSTEM ISOLATION AND THE FREE-BODY DIAGRAM

Before we apply Eqs. 3/1, we must define unambiguously the particular body or mechanical system to be analyzed and represent clearly and completely *all* forces acting *on* the body. Omission of a force which acts *on* the body in question, or inclusion of a force which does not act *on* the body, will give erroneous results.

A *mechanical system* is defined as a body or group of bodies which can be conceptually isolated from all other bodies. A system may be a single body or a combination of connected bodies. The bodies may be rigid or nonrigid. The system may also be an identifiable fluid mass, either liquid or gas, or a combination of fluids and solids. In statics we study primarily forces which act on rigid bodies at rest, although we also study forces acting on fluids in equilibrium.

Once we decide which body or combination of bodies to analyze, we then treat this body or combination as a single body *isolated* from all surrounding bodies. This isolation is accomplished by means of the **free-body diagram**, which is a diagrammatic representation of the isolated system treated as a single body. The diagram shows all forces applied to the system by mechanical contact with other bodies, which are imagined to be removed. If appreciable body forces are present, such as gravitational or magnetic attraction, then these forces must also be shown on the free-body diagram of the isolated system. Only after such a diagram has been carefully drawn should the equilibrium equations be written. Because of its critical importance, we emphasize here that

the free-body diagram is the most important single step in the solution of problems in mechanics.

Before attempting to draw a free-body diagram, we must recall the basic characteristics of force. These characteristics were described in Art. 2/2, with primary attention focused on the vector properties of force. Forces can be applied either by direct physical contact or by remote action. Forces can be either internal or external to the system under consideration. Application of force is accompanied by reactive force, and both applied and reactive forces may be either concentrated or distributed. The principle of transmissibility permits the treatment of force as a sliding vector as far as its external effects on a rigid body are concerned.

We will now use these force characteristics to develop conceptual models of isolated mechanical systems. These models enable us to

write the appropriate equations of equilibrium, which can then be analyzed.

Modeling the Action of Forces

Figure 3/1 shows the common types of force application on mechanical systems for analysis in two dimensions. Each example shows the force exerted *on* the body to be *isolated*, *by* the body to be *removed*. Newton's third law, which notes the existence of an equal and opposite reaction to every action, must be carefully observed. The force exerted *on* the body in question *by* a contacting or supporting member is always in the sense to oppose the movement of the isolated body which would occur if the contacting or supporting body were removed.

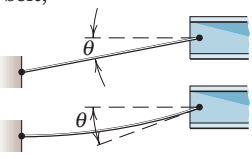
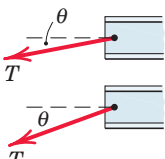
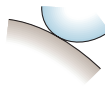
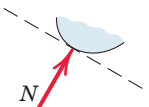

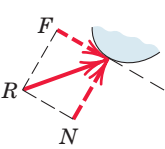
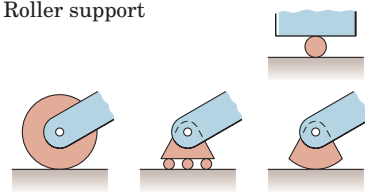
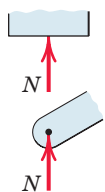
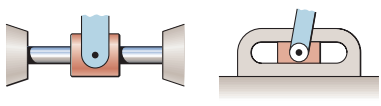
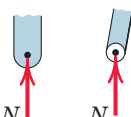
MODELING THE ACTION OF FORCES IN TWO-DIMENSIONAL ANALYSIS	
Type of Contact and Force Origin	Action on Body to Be Isolated
<p>1. Flexible cable, belt, chain, or rope</p> <p>Weight of cable negligible</p> <p>Weight of cable not negligible</p> 	 <p>Force exerted by a flexible cable is always a tension away from the body in the direction of the cable.</p>
<p>2. Smooth surfaces</p> 	 <p>Contact force is compressive and is normal to the surface.</p>
<p>3. Rough surfaces</p> 	 <p>Rough surfaces are capable of supporting a tangential component F (frictional force) as well as a normal component N of the resultant</p>
<p>4. Roller support</p> 	 <p>Roller, rocker, or ball support transmits a compressive force normal to the supporting surface.</p>
<p>5. Freely sliding guide</p> 	 <p>Collar or slider free to move along smooth guides; can support force normal to guide only.</p>

Figure 3/1

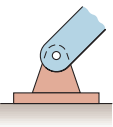
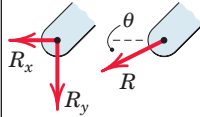
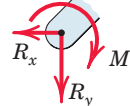
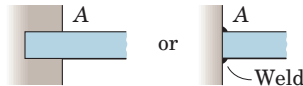
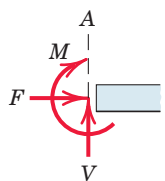
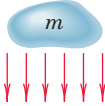
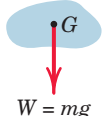
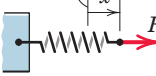
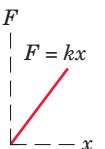
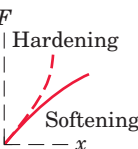
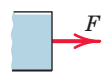
MODELING THE ACTION OF FORCES IN TWO-DIMENSIONAL ANALYSIS (cont.)	
Type of Contact and Force Origin	Action on Body to Be Isolated
<p>6. Pin connection</p> 	<p>Pin free to turn  A freely hinged pin connection is capable of supporting a force in any direction in the plane normal to the pin axis. We may either show two components R_x and R_y or a magnitude R and direction θ. A pin not free to turn also supports a couple M.</p> <p>Pin not free to turn </p>
<p>7. Built-in or fixed support</p> 	 <p>A built-in or fixed support is capable of supporting an axial force F, a transverse force V (shear force), and a couple M (bending moment) to prevent rotation.</p>
<p>8. Gravitational attraction</p> 	 <p>The resultant of gravitational attraction on all elements of a body of mass m is the weight $W = mg$ and acts toward the center of the earth through the center mass G.</p>
<p>9. Spring action</p> <p>Neutral position  Linear  Nonlinear </p>	 <p>Spring force is tensile if spring is stretched and compressive if compressed. For a linearly elastic spring the stiffness k is the force required to deform the spring a unit distance.</p>

Figure 3/1, continued

In Fig. 3/1, Example 1 depicts the action of a flexible cable, belt, rope, or chain on the body to which it is attached. Because of its flexibility, a rope or cable is unable to offer any resistance to bending, shear, or compression and therefore exerts only a tension force in a direction tangent to the cable at its point of attachment. The force exerted by the cable on the body to which it is attached is always *away* from the body. When the tension T is large compared with the weight of the cable, we may assume that the cable forms a straight line. When the cable weight is not negligible compared with its tension, the sag of the cable becomes important, and the tension in the cable changes direction and magnitude along its length.

When the smooth surfaces of two bodies are in contact, as in Example 2, the force exerted by one on the other is *normal* to the tangent to the surfaces and is compressive. Although no actual surfaces are per-

fectly smooth, we can assume this to be so for practical purposes in many instances.

When mating surfaces of contacting bodies are rough, as in Example 3, the force of contact is not necessarily normal to the tangent to the surfaces, but may be resolved into a *tangential* or *frictional component* F and a *normal component* N .

Example 4 illustrates a number of forms of mechanical support which effectively eliminate tangential friction forces. In these cases the net reaction is normal to the supporting surface.

Example 5 shows the action of a smooth guide on the body it supports. There cannot be any resistance parallel to the guide.

Example 6 illustrates the action of a pin connection. Such a connection can support force in any direction normal to the axis of the pin. We usually represent this action in terms of two rectangular components. The correct sense of these components in a specific problem depends on how the member is loaded. When not otherwise initially known, the sense is arbitrarily assigned and the equilibrium equations are then written. If the solution of these equations yields a positive algebraic sign for the force component, the assigned sense is correct. A negative sign indicates the sense is opposite to that initially assigned.

If the joint is free to turn about the pin, the connection can support only the force R . If the joint is not free to turn, the connection can also support a resisting couple M . The sense of M is arbitrarily shown here, but the true sense depends on how the member is loaded.

Example 7 shows the resultants of the rather complex distribution of force over the cross section of a slender bar or beam at a build-in or fixed support. The sense of the reactions F and V and the bending couple M in a given problem depends, of course, on how the member is loaded.

One of the most common forces is that due to gravitational attraction, Example 8. This force affects all elements of mass in a body and is, therefore, distributed throughout it. The resultant of the gravitational forces on all elements is the weight $W = mg$ of the body, which passes through the center of mass G and is directed toward the center of the earth for earthbound structures. The location of G is frequently obvious from the geometry of the body, particularly where there is symmetry. When the location is not readily apparent, it must be determined by experiment or calculations.

Similar remarks apply to the remote action of magnetic and electric forces. These forces of remote action have the same overall effect on a rigid body as forces of equal magnitude and direction applied by direct external contact.

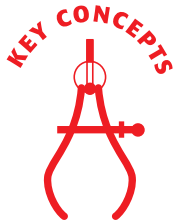
Example 9 illustrates the action of a *linear* elastic spring and of a *nonlinear* spring with either hardening or softening characteristics. The force exerted by a linear spring, in tension or compression, is given by $F = kx$, where k is the *stiffness* of the spring and x is its deformation measured from the neutral or undeformed position.

The representations in Fig. 3/1 are *not* free-body diagrams, but are merely elements used to construct free-body diagrams. Study these nine conditions and identify them in the problem work so that you can draw the correct free-body diagrams.



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Another view of the car-body lifting device shown in the chapter-opening photograph.



Construction of Free-Body Diagrams

The full procedure for drawing a free-body diagram which isolates a body or system consists of the following steps.

Step 1. Decide which system to isolate. The system chosen should usually involve one or more of the desired unknown quantities.

Step 2. Next isolate the chosen system by drawing a diagram which represents its *complete external boundary*. This boundary defines the isolation of the system from *all* other attracting or contacting bodies, which are considered removed. This step is often the most crucial of all. Make certain that you have *completely isolated* the system before proceeding with the next step.

Step 3. Identify all forces which act *on* the isolated system as applied *by* the removed contacting and attracting bodies, and represent them in their proper positions on the diagram of the isolated system. Make a systematic traverse of the entire boundary to identify all contact forces. Include body forces such as weights, where appreciable. Represent all known forces by vector arrows, each with its proper magnitude, direction, and sense indicated. Each unknown force should be represented by a vector arrow with the unknown magnitude or direction indicated by symbol. If the sense of the vector is also unknown, you must arbitrarily assign a sense. The subsequent calculations with the equilibrium equations will yield a positive quantity if the correct sense was assumed and a negative quantity if the incorrect sense was assumed. It is necessary to be *consistent* with the assigned characteristics of unknown forces throughout all of the calculations. If you are consistent, the solution of the equilibrium equations will reveal the correct senses.

Step 4. Show the choice of coordinate axes directly on the diagram. Pertinent dimensions may also be represented for convenience. Note, however, that the free-body diagram serves the purpose of focusing attention on the action of the external forces, and therefore the diagram should not be cluttered with excessive extraneous information. Clearly distinguish force arrows from arrows representing quantities other than forces. For this purpose a colored pencil may be used.

Completion of the foregoing four steps will produce a correct free-body diagram to use in applying the governing equations, both in statics and in dynamics. Be careful not to omit from the free-body diagram certain forces which may not appear at first glance to be needed in the calculations. It is only through *complete* isolation and a systematic representation of *all* external forces that a reliable accounting of the effects of all applied and reactive forces can be made. Very often a force which at first glance may not appear to influence a desired result does indeed have an influence. Thus, the only safe procedure is to include on the free-body diagram all forces whose magnitudes are not obviously negligible.

The free-body method is extremely important in mechanics because it ensures an accurate definition of a mechanical system and focuses attention on the exact meaning and application of the force laws of statics and dynamics. Review the foregoing four steps for constructing a free-body diagram while studying the sample free-body diagrams shown in Fig. 3/2 and the Sample Problems which appear at the end of the next article.

Examples of Free-Body Diagrams

Figure 3/2 gives four examples of mechanisms and structures together with their correct free-body diagrams. Dimensions and magnitudes are omitted for clarity. In each case we treat the entire system as

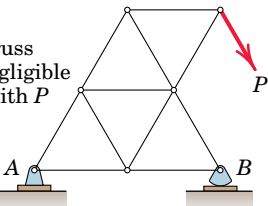
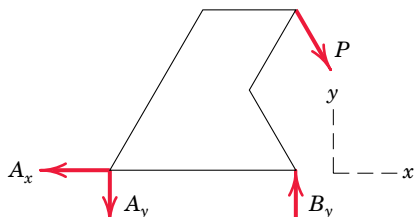
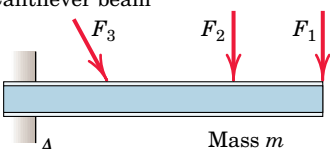
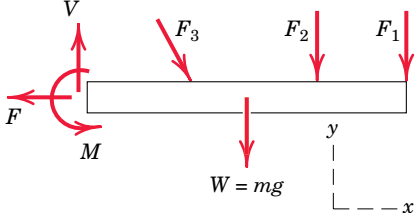
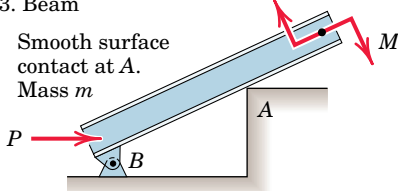
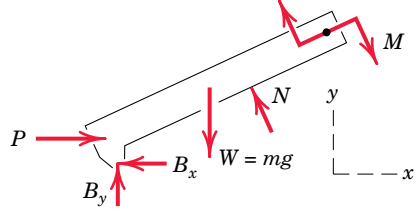
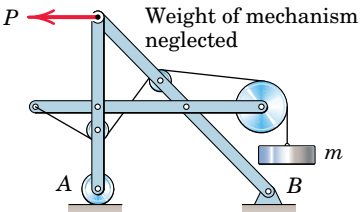
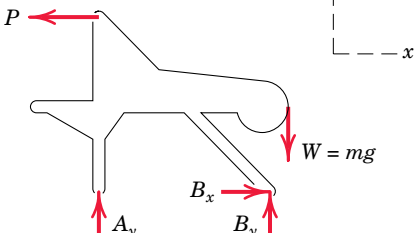
SAMPLE FREE-BODY DIAGRAMS	
Mechanical System	Free-Body Diagram of Isolated Body
<p>1. Plane truss</p> <p>Weight of truss assumed negligible compared with P</p> 	
<p>2. Cantilever beam</p> 	
<p>3. Beam</p> <p>Smooth surface contact at A.</p> <p>Mass m</p> 	
<p>4. Rigid system of interconnected bodies analyzed as a single unit</p> <p>Weight of mechanism neglected</p> 	

Figure 3/2

a single body, so that the internal forces are not shown. The characteristics of the various types of contact forces illustrated in Fig. 3/1 are used in the four examples as they apply.

In Example 1 the truss is composed of structural elements which, taken all together, constitute a rigid framework. Thus, we may remove the entire truss from its supporting foundation and treat it as a single rigid body. In addition to the applied external load P , the free-body diagram must include the reactions on the truss at A and B . The rocker at B can support a vertical force only, and this force is transmitted to the structure at B (Example 4 of Fig. 3/1). The pin connection at A (Example 6 of Fig. 3/1) is capable of supplying both a horizontal and a vertical force component to the truss. If the total weight of the truss members is appreciable compared with P and the forces at A and B , then the weights of the members must be included on the free-body diagram as external forces.

In this relatively simple example it is clear that the vertical component A_y must be directed down to prevent the truss from rotating clockwise about B . Also, the horizontal component A_x will be to the left to keep the truss from moving to the right under the influence of the horizontal component of P . Thus, in constructing the free-body diagram for this simple truss, we can easily perceive the correct sense of each of the components of force exerted on the truss by the foundation at A and can, therefore, represent its correct physical sense on the diagram. When the correct physical sense of a force or its component is not easily recognized by direct observation, it must be assigned arbitrarily, and the correctness of or error in the assignment is determined by the algebraic sign of its calculated value.

In Example 2 the cantilever beam is secured to the wall and subjected to three applied loads. When we isolate that part of the beam to the right of the section at A , we must include the reactive forces applied to the beam by the wall. The resultants of these reactive forces are shown acting on the section of the beam (Example 7 of Fig. 3/1). A vertical force V to counteract the excess of downward applied force is shown, and a tension F to balance the excess of applied force to the right must also be included. Then, to prevent the beam from rotating about A , a counterclockwise couple M is also required. The weight mg of the beam must be represented through the mass center (Example 8 of Fig. 3/1).

In the free-body diagram of Example 2, we have represented the somewhat complex system of forces which actually act on the cut section of the beam by the equivalent force-couple system in which the force is broken down into its vertical component V (shear force) and its horizontal component F (tensile force). The couple M is the bending moment in the beam. The free-body diagram is now complete and shows the beam in equilibrium under the action of six forces and one couple.

In Example 3 the weight $W = mg$ is shown acting through the center of mass of the beam, whose location is assumed known (Example 8 of Fig. 3/1). The force exerted by the corner A on the beam is normal to the smooth surface of the beam (Example 2 of Fig. 3/1). To perceive this action more clearly, visualize an enlargement of the contact point A , which would appear somewhat rounded, and consider the force exerted by this rounded corner on the straight surface of the beam, which is as-

sumed to be smooth. If the contacting surfaces at the corner were not smooth, a tangential frictional component of force could exist. In addition to the applied force P and couple M , there is the pin connection at B , which exerts both an x - and a y -component of force on the beam. The positive senses of these components are assigned arbitrarily.

In Example 4 the free-body diagram of the entire isolated mechanism contains three unknown forces if the loads mg and P are known. Any one of many internal configurations for securing the cable leading from the mass m would be possible without affecting the external response of the mechanism as a whole, and this fact is brought out by the free-body diagram. This hypothetical example is used to show that the forces internal to a rigid assembly of members do not influence the values of the external reactions.

We use the free-body diagram in writing the equilibrium equations, which are discussed in the next article. When these equations are solved, some of the calculated force magnitudes may be zero. This would indicate that the assumed force does not exist. In Example 1 of Fig. 3/2, any of the reactions A_x , A_y , or B_y can be zero for specific values of the truss geometry and of the magnitude, direction, and sense of the applied load P . A zero reaction force is often difficult to identify by inspection, but can be determined by solving the equilibrium equations.

Similar comments apply to calculated force magnitudes which are negative. Such a result indicates that the actual sense is the opposite of the assumed sense. The assumed positive senses of B_x and B_y in Example 3 and B_y in Example 4 are shown on the free-body diagrams. The correctness of these assumptions is proved or disproved according to whether the algebraic signs of the computed forces are plus or minus when the calculations are carried out in an actual problem.

The isolation of the mechanical system under consideration is a crucial step in the formulation of the mathematical model. The most important aspect to the correct construction of the all-important free-body diagram is the clear-cut and unambiguous decision as to what is included and what is excluded. This decision becomes unambiguous only when the boundary of the free-body diagram represents a complete traverse of the body or system of bodies to be isolated, starting at some arbitrary point on the boundary and returning to that same point. The system within this closed boundary is the isolated free body, and all contact forces and all body forces transmitted to the system across the boundary must be accounted for.

The following exercises provide practice with drawing free-body diagrams. This practice is helpful before using such diagrams in the application of the principles of force equilibrium in the next article.



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Even complex pulley systems such as the ones seen here are easily handled with a systematic equilibrium analysis.

FREE-BODY DIAGRAM EXERCISES

3/A In each of the five following examples, the body to be isolated is shown in the left-hand diagram, and an *incomplete* free-body diagram (FBD) of the isolated body is shown on the right. Add whatever forces are

necessary in each case to form a complete free-body diagram. The weights of the bodies are negligible unless otherwise indicated. Dimensions and numerical values are omitted for simplicity.

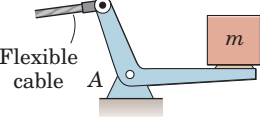
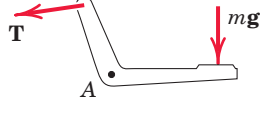
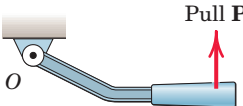
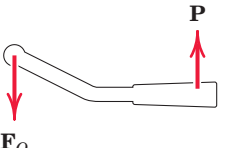
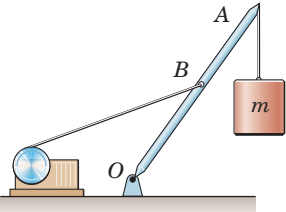
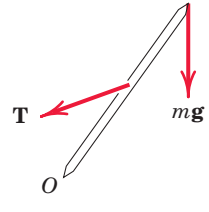
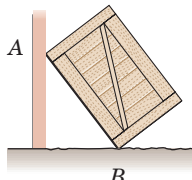
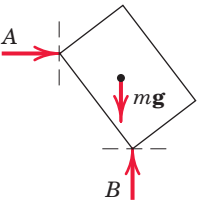
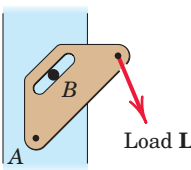
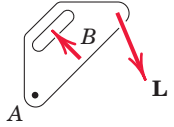
	Body	Incomplete FBD
1. Bell crank supporting mass m with pin support at A .		
2. Control lever applying torque to shaft at O .		
3. Boom OA , of negligible mass compared with mass m . Boom hinged at O and supported by hoisting cable at B .		
4. Uniform crate of mass m leaning against smooth vertical wall and supported on a rough horizontal surface.		
5. Loaded bracket supported by pin connection at A and fixed pin in smooth slot at B .		

Figure 3/A

3/B In each of the five following examples, the body to be isolated is shown in the left-hand diagram, and either a *wrong* or an *incomplete* free-body diagram (FBD) is shown on the right. Make whatever changes or addi-

tions are necessary in each case to form a correct and complete free-body diagram. The weights of the bodies are negligible unless otherwise indicated. Dimensions and numerical values are omitted for simplicity.

	Body	Wrong or Incomplete FBD
1. Lawn roller of mass m being pushed up incline θ .		
2. Prybar lifting body A having smooth horizontal surface. Bar rests on horizontal rough surface.		
3. Uniform pole of mass m being hoisted into position by winch. Horizontal supporting surface notched to prevent slipping of pole.		
4. Supporting angle bracket for frame; pin joints.		
5. Bent rod welded to support at A and subjected to two forces and couple.		

Figure 3/B

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3/C Draw a complete and correct free-body diagram of each of the bodies designated in the statements. The weights of the bodies are significant only if the mass is stated.

All forces, known and unknown, should be labeled. (Note: The sense of some reaction components cannot always be determined without numerical calculation.)

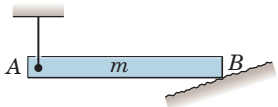
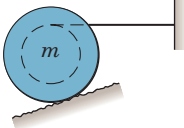
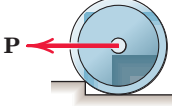
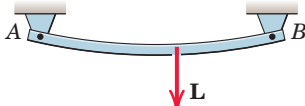
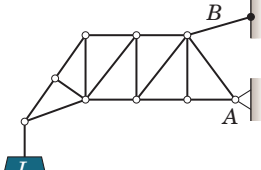
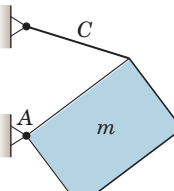
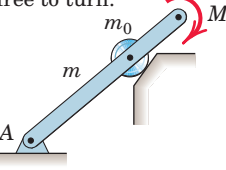
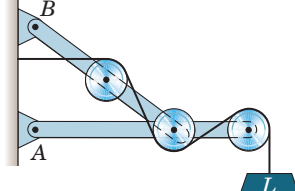
<p>1. Uniform horizontal bar of mass m suspended by vertical cable at A and supported by rough inclined surface at B.</p> 	<p>5. Uniform grooved wheel of mass m supported by a rough surface and by action of horizontal cable.</p> 
<p>2. Wheel of mass m on verge of being rolled over curb by pull P.</p> 	<p>6. Bar, initially horizontal but deflected under load L. Pinned to rigid support at each end.</p> 
<p>3. Loaded truss supported by pin joint at A and by cable at B.</p> 	<p>7. Uniform heavy plate of mass m supported in vertical plane by cable C and hinge A.</p> 
<p>4. Uniform bar of mass m and roller of mass m_0 taken together. Subjected to couple M and supported as shown. Roller is free to turn.</p> 	<p>8. Entire frame, pulleys, and contacting cable to be isolated as a single unit.</p> 

Figure 3/C

3/3 EQUILIBRIUM CONDITIONS

In Art. 3/1 we defined equilibrium as the condition in which the resultant of all forces and moments acting on a body is zero. Stated in another way, a body is in equilibrium if all forces and moments applied to it are in balance. These requirements are contained in the vector equations of equilibrium, Eqs. 3/1, which in two dimensions may be written in scalar form as

$$\Sigma F_x = 0 \quad \Sigma F_y = 0 \quad \Sigma M_O = 0 \quad (3/2)$$

The third equation represents the zero sum of the moments of all forces about any point O on or off the body. Equations 3/2 are the necessary and sufficient conditions for complete equilibrium in two dimensions. They are necessary conditions because, if they are not satisfied, there can be no force or moment balance. They are sufficient because once they are satisfied, there can be no imbalance, and equilibrium is assured.

The equations relating force and acceleration for rigid-body motion are developed in *Vol. 2 Dynamics* from Newton's second law of motion. These equations show that the acceleration of the mass center of a body is proportional to the resultant force $\Sigma \mathbf{F}$ acting on the body. Consequently, if a body moves with constant velocity (zero acceleration), the resultant force on it must be zero, and the body may be treated as in a state of translational equilibrium.

For complete equilibrium in two dimensions, all three of Eqs. 3/2 must hold. However, these conditions are independent requirements, and one may hold without another. Take, for example, a body which slides along a horizontal surface with increasing velocity under the action of applied forces. The force-equilibrium equations will be satisfied in the vertical direction where the acceleration is zero, but not in the horizontal direction. Also, a body, such as a flywheel, which rotates about its fixed mass center with increasing angular speed is not in rotational equilibrium, but the two force-equilibrium equations will be satisfied.

Categories of Equilibrium

Applications of Eqs. 3/2 fall naturally into a number of categories which are easily identified. The categories of force systems acting on bodies in two-dimensional equilibrium are summarized in Fig. 3/3 and are explained further as follows.

Category 1, equilibrium of collinear forces, clearly requires only the one force equation in the direction of the forces (x -direction), since all other equations are automatically satisfied.

Category 2, equilibrium of forces which lie in a plane (x - y plane) and are concurrent at a point O , requires the two force equations only, since the moment sum about O , that is, about a z -axis through O , is necessarily zero. Included in this category is the case of the equilibrium of a particle.

Category 3, equilibrium of parallel forces in a plane, requires the one force equation in the direction of the forces (x -direction) and one moment equation about an axis (z -axis) normal to the plane of the forces.

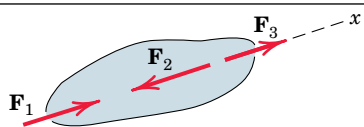
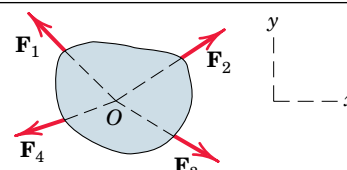
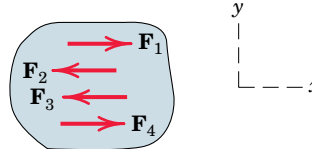
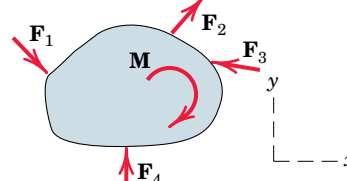
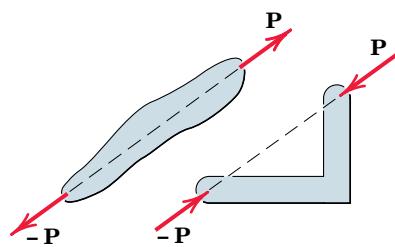
CATEGORIES OF EQUILIBRIUM IN TWO DIMENSIONS		
Force System	Free-Body Diagram	Independent Equations
1. Collinear		$\Sigma F_x = 0$
2. Concurrent at a point		$\Sigma F_x = 0$ $\Sigma F_y = 0$
3. Parallel		$\Sigma F_x = 0$ $\Sigma M_z = 0$
4. General		$\Sigma F_x = 0$ $\Sigma M_z = 0$ $\Sigma F_y = 0$

Figure 3/3

Category 4, equilibrium of a general system of forces in a plane (x - y), requires the two force equations in the plane and one moment equation about an axis (z -axis) normal to the plane.

Two- and Three-Force Members



Two-force members

Figure 3/4

You should be alert to two frequently occurring equilibrium situations. The first situation is the equilibrium of a body under the action of two forces only. Two examples are shown in Fig. 3/4, and we see that for such a *two-force member* to be in equilibrium, the forces must be *equal*, *opposite*, and *collinear*. The shape of the member does not affect this simple requirement. In the illustrations cited, we consider the weights of the members to be negligible compared with the applied forces.

The second situation is a *three-force member*, which is a body under the action of three forces, Fig. 3/5a. We see that equilibrium requires the lines of action of the three forces to be *concurrent*. If they were not concurrent, then one of the forces would exert a resultant moment about the point of intersection of the other two, which would violate the requirement of zero moment about every point. The only exception occurs when the three forces are parallel. In this case we may consider the point of concurrency to be at infinity.

The principle of the concurrency of three forces in equilibrium is of considerable use in carrying out a graphical solution of the force equations. In this case the polygon of forces is drawn and made to close, as shown in Fig. 3/5*b*. Frequently, a body in equilibrium under the action of more than three forces may be reduced to a three-force member by a combination of two or more of the known forces.

Alternative Equilibrium Equations

In addition to Eqs. 3/2, there are two other ways to express the general conditions for the equilibrium of forces in two dimensions. The first way is illustrated in Fig. 3/6, parts (a) and (b). For the body shown in Fig. 3/6*a*, if $\Sigma M_A = 0$, then the resultant, if it still exists, cannot be a couple, but must be a force \mathbf{R} passing through A . If now the equation $\Sigma F_x = 0$ holds, where the x -direction is arbitrary, it follows from Fig. 3/6*b* that the resultant force \mathbf{R} , if it still exists, not only must pass through A , but also must be perpendicular to the x -direction as shown. Now, if $\Sigma M_B = 0$, where B is any point such that the line AB is not perpendicular to the x -direction, we see that \mathbf{R} must be zero, and thus the body is in equilibrium. Therefore, an alternative set of equilibrium equations is

$$\Sigma F_x = 0 \quad \Sigma M_A = 0 \quad \Sigma M_B = 0$$

where the two points A and B must not lie on a line perpendicular to the x -direction.

A third formulation of the equilibrium conditions may be made for a coplanar force system. This is illustrated in Fig. 3/6, parts (c) and (d). Again, if $\Sigma M_A = 0$ for any body such as that shown in Fig. 3/6*c*, the resultant, if any, must be a force \mathbf{R} through A . In addition, if $\Sigma M_B = 0$, the resultant, if one still exists, must pass through B as shown in Fig. 3/6*d*. Such a force cannot exist, however, if $\Sigma M_C = 0$, where C is not

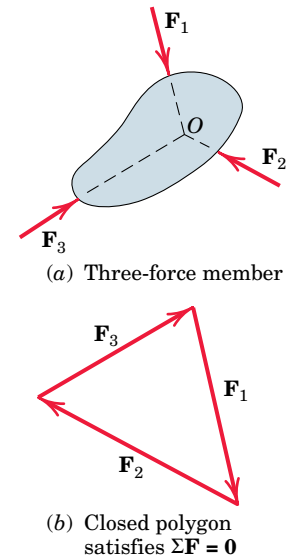


Figure 3/5

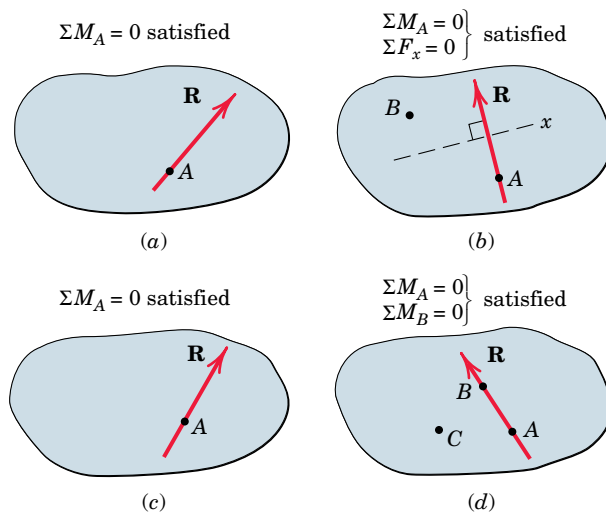


Figure 3/6

collinear with A and B . Thus, we may write the equations of equilibrium as

$$\Sigma M_A = 0 \quad \Sigma M_B = 0 \quad \Sigma M_C = 0$$

where A , B , and C are any three points not on the same straight line.

When equilibrium equations are written which are not independent, redundant information is obtained, and a correct solution of the equations will yield $0 = 0$. For example, for a general problem in two dimensions with three unknowns, three moment equations written about three points which lie on the same straight line are not independent. Such equations will contain duplicated information, and solution of two of them can at best determine two of the unknowns, with the third equation merely verifying the identity $0 = 0$.

Constraints and Statical Determinacy

The equilibrium equations developed in this article are both necessary and sufficient conditions to establish the equilibrium of a body. However, they do not necessarily provide all the information required to calculate all the unknown forces which may act on a body in equilibrium. Whether the equations are adequate to determine all the unknowns depends on the characteristics of the constraints against possible movement of the body provided by its supports. By *constraint* we mean the restriction of movement.

In Example 4 of Fig. 3/1 the roller, ball, and rocker provide constraint normal to the surface of contact, but none tangent to the surface. Thus, a tangential force cannot be supported. For the collar and slider of Example 5, constraint exists only normal to the guide. In Example 6 the fixed-pin connection provides constraint in both directions, but offers no resistance to rotation about the pin unless the pin is not free to turn. The fixed support of Example 7, however, offers constraint against rotation as well as lateral movement.

If the rocker which supports the truss of Example 1 in Fig. 3/2 were replaced by a pin joint, as at A , there would be one additional constraint beyond those required to support an equilibrium configuration with no freedom of movement. The three scalar conditions of equilibrium, Eqs. 3/2, would not provide sufficient information to determine all four unknowns, since A_x and B_x could not be solved for separately; only their sum could be determined. These two components of force would be dependent on the deformation of the members of the truss as influenced by their corresponding stiffness properties. The horizontal reactions A_x and B_x would also depend on any initial deformation required to fit the dimensions of the structure to those of the foundation between A and B . Thus, we cannot determine A_x and B_x by a rigid-body analysis.

Again referring to Fig. 3/2, we see that if the pin B in Example 3 were not free to turn, the support could transmit a couple to the beam through the pin. Therefore, there would be four unknown supporting reactions acting on the beam, namely, the force at A , the two components of force at B , and the couple at B . Consequently the three inde-

pendent scalar equations of equilibrium would not provide enough information to compute all four unknowns.

A rigid body, or rigid combination of elements treated as a single body, which possesses more external supports or constraints than are necessary to maintain an equilibrium position is called *statically indeterminate*. Supports which can be removed without destroying the equilibrium condition of the body are said to be *redundant*. The number of redundant supporting elements present corresponds to the *degree of static indeterminacy* and equals the total number of unknown external forces, minus the number of available independent equations of equilibrium. On the other hand, bodies which are supported by the minimum number of constraints necessary to ensure an equilibrium configuration are called *statically determinate*, and for such bodies the equilibrium equations are sufficient to determine the unknown external forces.

The problems on equilibrium in this article and throughout *Vol. 1 Statics* are generally restricted to statically determinate bodies where the constraints are just sufficient to ensure a stable equilibrium configuration and where the unknown supporting forces can be completely determined by the available independent equations of equilibrium.

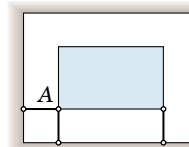
We must be aware of the nature of the constraints before we attempt to solve an equilibrium problem. A body can be recognized as statically indeterminate when there are more unknown external reactions than there are available independent equilibrium equations for the force system involved. It is always well to count the number of unknown variables on a given body and to be certain that an equal number of independent equations can be written; otherwise, effort might be wasted in attempting an impossible solution with the aid of the equilibrium equations only. The unknown variables may be forces, couples, distances, or angles.

Adequacy of Constraints

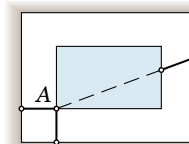
In discussing the relationship between constraints and equilibrium, we should look further at the question of the adequacy of constraints. The existence of three constraints for a two-dimensional problem does not always guarantee a stable equilibrium configuration. Figure 3/7 shows four different types of constraints. In part *a* of the figure, point *A* of the rigid body is fixed by the two links and cannot move, and the third link prevents any rotation about *A*. Thus, this body is *completely fixed* with three *adequate* (proper) constraints.

In part *b* of the figure, the third link is positioned so that the force transmitted by it passes through point *A* where the other two constraint forces act. Thus, this configuration of constraints can offer no initial resistance to rotation about *A*, which would occur when external loads were applied to the body. We conclude, therefore, that this body is *incompletely fixed* under *partial constraints*.

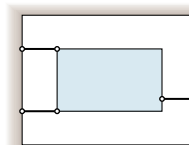
The configuration in part *c* of the figure gives us a similar condition of incomplete fixity because the three parallel links could offer no initial resistance to a small vertical movement of the body as a result of external loads applied to it in this direction. The constraints in these two examples are often termed *improper*.



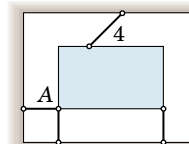
(a) Complete fixity
Adequate constraints



(b) Incomplete fixity
Partial constraints



(c) Incomplete fixity
Partial constraints

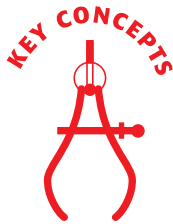


(d) Excessive fixity
Redundant constraint

Figure 3/7

In part *d* of Fig. 3/7 we have a condition of complete fixity, with link 4 acting as a fourth constraint which is unnecessary to maintain a fixed position. Link 4, then, is a *redundant constraint*, and the body is statically indeterminate.

As in the four examples of Fig. 3/7, it is generally possible by direct observation to conclude whether the constraints on a body in two-dimensional equilibrium are adequate (proper), partial (improper), or redundant. As indicated previously, the vast majority of problems in this book are statically determinate with adequate (proper) constraints.



Approach to Solving Problems

The sample problems at the end of this article illustrate the application of free-body diagrams and the equations of equilibrium to typical statics problems. These solutions should be studied thoroughly. In the problem work of this chapter and throughout mechanics, it is important to develop a logical and systematic approach which includes the following steps:

1. Identify clearly the quantities which are known and unknown.
2. Make an unambiguous choice of the body (or system of connected bodies treated as a single body) to be isolated and draw its complete free-body diagram, labeling all external known and unknown but identifiable forces and couples which act on it.
3. Choose a convenient set of reference axes, always using right-handed axes when vector cross products are employed. Choose moment centers with a view to simplifying the calculations. Generally the best choice is one through which as many unknown forces pass as possible. Simultaneous solutions of equilibrium equations are frequently necessary, but can be minimized or avoided by a careful choice of reference axes and moment centers.
4. Identify and state the applicable force and moment principles or equations which govern the equilibrium conditions of the problem. In the following sample problems these relations are shown in brackets and precede each major calculation.
5. Match the number of independent equations with the number of unknowns in each problem.
6. Carry out the solution and check the results. In many problems engineering judgment can be developed by first making a reasonable guess or estimate of the result prior to the calculation and then comparing the estimate with the calculated value.

Sample Problem 3/1

Determine the magnitudes of the forces **C** and **T**, which, along with the other three forces shown, act on the bridge-truss joint.

- Solution.** The given sketch constitutes the free-body diagram of the isolated section of the joint in question and shows the five forces which are in equilibrium.

Solution I (scalar algebra). For the x - y axes as shown we have

$$[\Sigma F_x = 0] \quad 8 + T \cos 40^\circ + C \sin 20^\circ - 16 = 0$$

$$0.766T + 0.342C = 8 \quad (a)$$

$$[\Sigma F_y = 0] \quad T \sin 40^\circ - C \cos 20^\circ - 3 = 0$$

$$0.643T - 0.940C = 3 \quad (b)$$

Simultaneous solution of Eqs. (a) and (b) produces

$$T = 9.09 \text{ kN} \quad C = 3.03 \text{ kN} \quad \text{Ans.}$$

- Solution II (scalar algebra).** To avoid a simultaneous solution, we may use axes x' - y' with the first summation in the y' -direction to eliminate reference to T . Thus,

$$[\Sigma F_{y'} = 0] \quad -C \cos 20^\circ - 3 \cos 40^\circ - 8 \sin 40^\circ + 16 \sin 40^\circ = 0$$

$$C = 3.03 \text{ kN} \quad \text{Ans.}$$

$$[\Sigma F_{x'} = 0] \quad T + 8 \cos 40^\circ - 16 \cos 40^\circ - 3 \sin 40^\circ - 3.03 \sin 20^\circ = 0$$

$$T = 9.09 \text{ kN} \quad \text{Ans.}$$

Solution III (vector algebra). With unit vectors **i** and **j** in the x - and y -directions, the zero summation of forces for equilibrium yields the vector equation

$$[\Sigma \mathbf{F} = \mathbf{0}] \quad 8\mathbf{i} + (T \cos 40^\circ)\mathbf{i} + (T \sin 40^\circ)\mathbf{j} - 3\mathbf{j} + (C \sin 20^\circ)\mathbf{i} - (C \cos 20^\circ)\mathbf{j} - 16\mathbf{i} = \mathbf{0}$$

Equating the coefficients of the **i**- and **j**-terms to zero gives

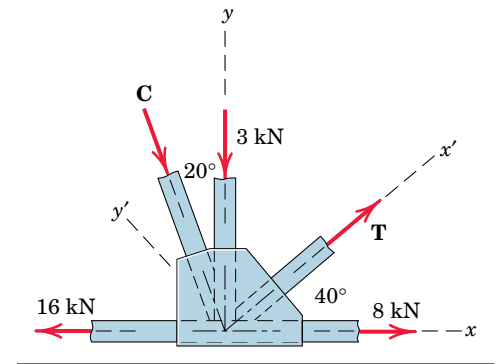
$$8 + T \cos 40^\circ + C \sin 20^\circ - 16 = 0$$

$$T \sin 40^\circ - 3 - C \cos 20^\circ = 0$$

which are the same, of course, as Eqs. (a) and (b), which we solved above.

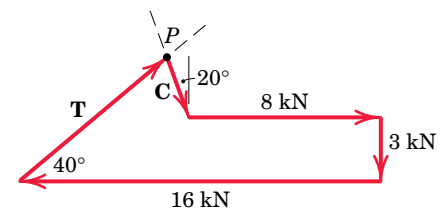
Solution IV (geometric). The polygon representing the zero vector sum of the five forces is shown. Equations (a) and (b) are seen immediately to give the projections of the vectors onto the x - and y -directions. Similarly, projections onto the x' - and y' -directions give the alternative equations in Solution II.

- A graphical solution is easily obtained. The known vectors are laid off head-to-tail to some convenient scale, and the directions of **T** and **C** are then drawn to close the polygon. The resulting intersection at point **P** completes the solution, thus enabling us to measure the magnitudes of **T** and **C** directly from the drawing to whatever degree of accuracy we incorporate in the construction.



Helpful Hints

- ① Since this is a problem of concurrent forces, no moment equation is necessary.
- ② The selection of reference axes to facilitate computation is always an important consideration. Alternatively in this example we could take a set of axes along and normal to the direction of **C** and employ a force summation normal to **C** to eliminate it.



- ③ The known vectors may be added in any order desired, but they must be added before the unknown vectors.

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Sample Problem 3/2

Calculate the tension T in the cable which supports the 500-kg mass with the pulley arrangement shown. Each pulley is free to rotate about its bearing, and the weights of all parts are small compared with the load. Find the magnitude of the total force on the bearing of pulley C .

Solution. The free-body diagram of each pulley is drawn in its relative position to the others. We begin with pulley A , which includes the only known force. With the unspecified pulley radius designated by r , the equilibrium of moments about its center O and the equilibrium of forces in the vertical direction require

$$\begin{aligned} \textcircled{1} [\Sigma M_O = 0] \quad T_1 r - T_2 r = 0 \quad T_1 = T_2 \\ [\Sigma F_y = 0] \quad T_1 + T_2 - 500(9.81) = 0 \quad 2T_1 = 500(9.81) \quad T_1 = T_2 = 2450 \text{ N} \end{aligned}$$

From the example of pulley A we may write the equilibrium of forces on pulley B by inspection as

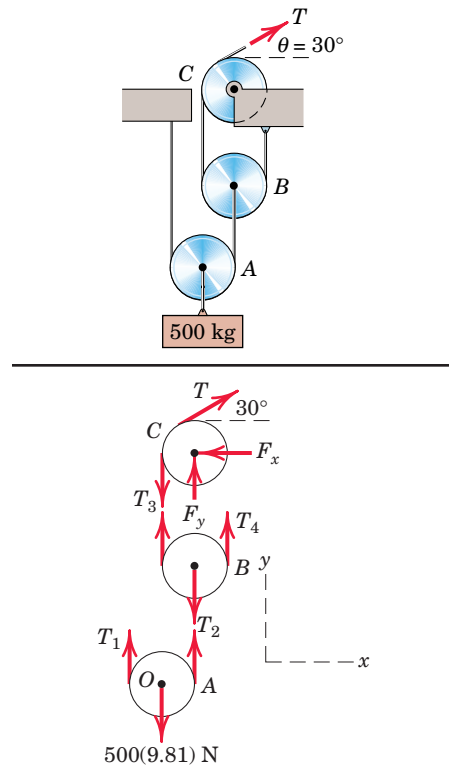
$$T_3 = T_4 = T_2/2 = 1226 \text{ N}$$

For pulley C the angle $\theta = 30^\circ$ in no way affects the moment of T about the center of the pulley, so that moment equilibrium requires

$$T = T_3 \quad \text{or} \quad T = 1226 \text{ N} \quad \text{Ans.}$$

Equilibrium of the pulley in the x - and y -directions requires

$$\begin{aligned} [\Sigma F_x = 0] \quad 1226 \cos 30^\circ - F_x = 0 \quad F_x = 1062 \text{ N} \\ [\Sigma F_y = 0] \quad F_y + 1226 \sin 30^\circ - 1226 = 0 \quad F_y = 613 \text{ N} \\ [F = \sqrt{F_x^2 + F_y^2}] \quad F = \sqrt{(1062)^2 + (613)^2} = 1226 \text{ N} \quad \text{Ans.} \end{aligned}$$



Helpful Hint

- ① Clearly the radius r does not influence the results. Once we have analyzed a simple pulley, the results should be perfectly clear by inspection.

Sample Problem 3/3

The uniform 100-kg I-beam is supported initially by its end rollers on the horizontal surface at A and B . By means of the cable at C it is desired to elevate end B to a position 3 m above end A . Determine the required tension P , the reaction at A , and the angle θ made by the beam with the horizontal in the elevated position.

Solution. In constructing the free-body diagram, we note that the reaction on the roller at A and the weight are vertical forces. Consequently, in the absence of other horizontal forces, P must also be vertical. From Sample Problem 3/2 we see immediately that the tension P in the cable equals the tension P applied to the beam at C .

Moment equilibrium about A eliminates force R and gives

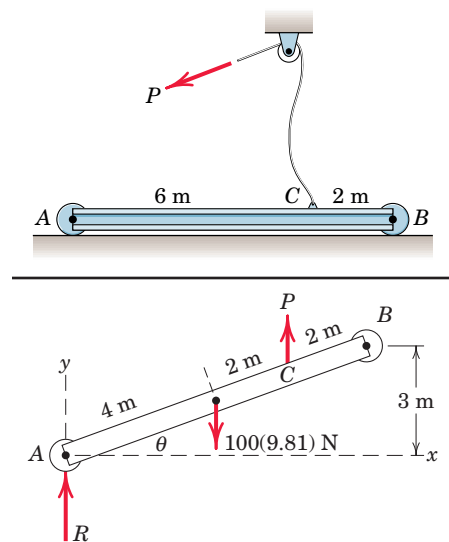
$$\textcircled{1} [\Sigma M_A = 0] \quad P(6 \cos \theta) - 981(4 \cos \theta) = 0 \quad P = 654 \text{ N} \quad \text{Ans.}$$

Equilibrium of vertical forces requires

$$[\Sigma F_y = 0] \quad 654 + R - 981 = 0 \quad R = 327 \text{ N} \quad \text{Ans.}$$

The angle θ depends only on the specified geometry and is

$$\sin \theta = 3/8 \quad \theta = 22.0^\circ \quad \text{Ans.}$$



Helpful Hint

- ① Clearly the equilibrium of this parallel force system is independent of θ .

Sample Problem 3/4

Determine the magnitude T of the tension in the supporting cable and the magnitude of the force on the pin at A for the jib crane shown. The beam AB is a standard 0.5-m I-beam with a mass of 95 kg per meter of length.

Algebraic solution. The system is symmetrical about the vertical x - y plane through the center of the beam, so the problem may be analyzed as the equilibrium of a coplanar force system. The free-body diagram of the beam is shown in the figure with the pin reaction at A represented in terms of its two rectangular components. The weight of the beam is $95(10^{-3})(5)9.81 = 4.66$ kN and acts through its center. Note that there are three unknowns A_x , A_y , and T , which may be found from the three equations of equilibrium. We begin with a moment equation about A , which eliminates two of the three unknowns from the equation.

$$\textcircled{1} \quad [\Sigma M_A = 0] \quad (T \cos 25^\circ)0.25 + (T \sin 25^\circ)(5 - 0.12) - 10(5 - 1.5 - 0.12) - 4.66(2.5 - 0.12) = 0$$

$$\text{from which} \quad T = 19.61 \text{ kN} \quad \text{Ans.}$$

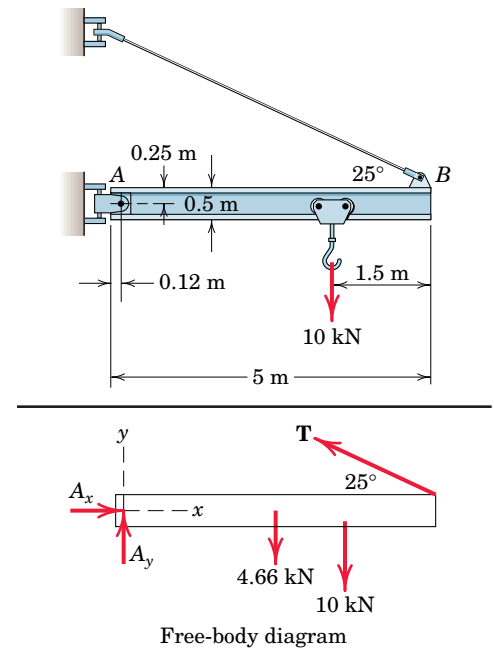
Equating the sums of forces in the x - and y -directions to zero gives

$$[\Sigma F_x = 0] \quad A_x - 19.61 \cos 25^\circ = 0 \quad A_x = 17.77 \text{ kN}$$

$$[\Sigma F_y = 0] \quad A_y + 19.61 \sin 25^\circ - 4.66 - 10 = 0 \quad A_y = 6.37 \text{ kN}$$

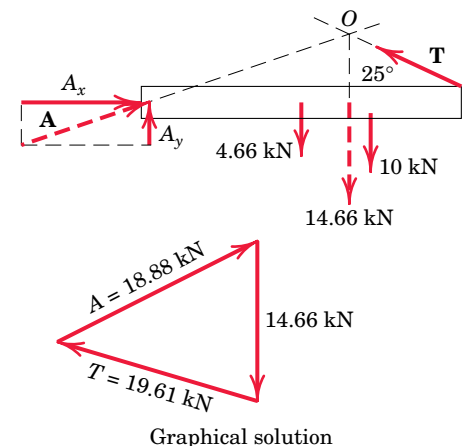
$$\textcircled{2} \quad [A = \sqrt{A_x^2 + A_y^2}] \quad A = \sqrt{(17.77)^2 + (6.37)^2} = 18.88 \text{ kN} \quad \text{Ans.}$$

Graphical solution. The principle that three forces in equilibrium must be concurrent is utilized for a graphical solution by combining the two known vertical forces of 4.66 and 10 kN into a single 14.66-kN force, located as shown on the modified free-body diagram of the beam in the lower figure. The position of this resultant load may easily be determined graphically or algebraically. The intersection of the 14.66-kN force with the line of action of the unknown tension T defines the point of concurrency O through which the pin reaction A must pass. The unknown magnitudes of T and A may now be found by adding the forces head-to-tail to form the closed equilibrium polygon of forces, thus satisfying their zero vector sum. After the known vertical load is laid off to a convenient scale, as shown in the lower part of the figure, a line representing the given direction of the tension T is drawn through the tip of the 14.66-kN vector. Likewise a line representing the direction of the pin reaction A , determined from the concurrency established with the free-body diagram, is drawn through the tail of the 14.66-kN vector. The intersection of the lines representing vectors T and A establishes the magnitudes T and A necessary to make the vector sum of the forces equal to zero. These magnitudes are scaled from the diagram. The x - and y -components of A may be constructed on the force polygon if desired.



Helpful Hints

- ① The justification for this step is Varignon's theorem, explained in Art. 2/4. Be prepared to take full advantage of this principle frequently.
- ② The calculation of moments in two-dimensional problems is generally handled more simply by scalar algebra than by the vector cross product $\mathbf{r} \times \mathbf{F}$. In three dimensions, as we will see later, the reverse is often the case.
- ③ The direction of the force at A could be easily calculated if desired. However, in designing the pin A or in checking its strength, it is only the magnitude of the force that matters.

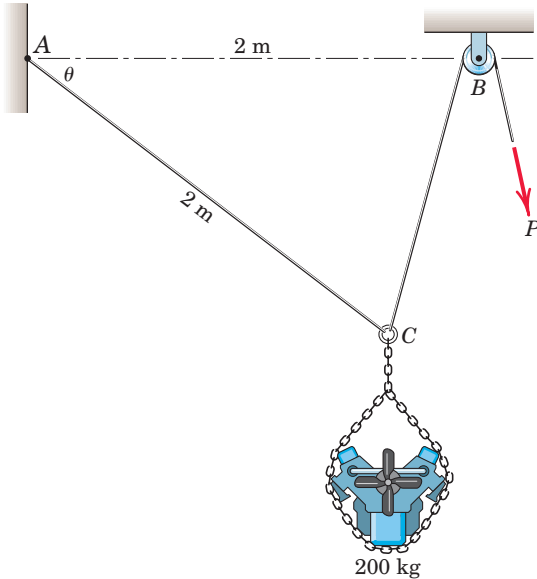


PROBLEMS

Introductory Problems

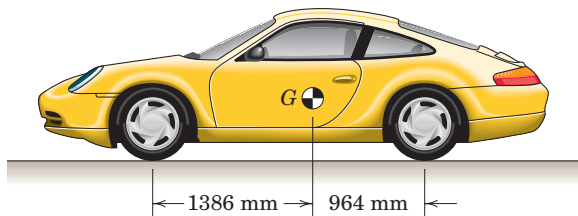
- 3/1** Determine the force P required to maintain the 200-kg engine in the position for which $\theta = 30^\circ$. The diameter of the pulley at B is negligible.

Ans. $P = 1759 \text{ N}$



Problem 3/1

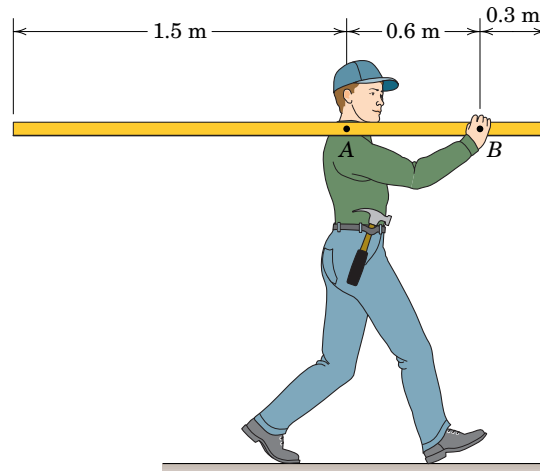
- 3/2** The mass center G of the 1400-kg rear-engine car is located as shown in the figure. Determine the normal force under each tire when the car is in equilibrium. State any assumptions.



Problem 3/2

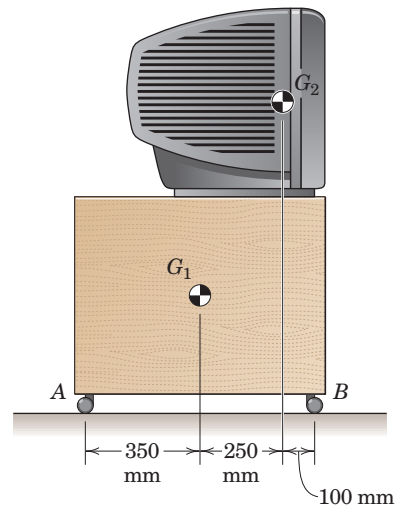
- 3/3** A carpenter carries a 6-kg uniform board as shown. What downward force does he feel on his shoulder at A ?

Ans. $N_A = 88.3 \text{ N}$



Problem 3/3

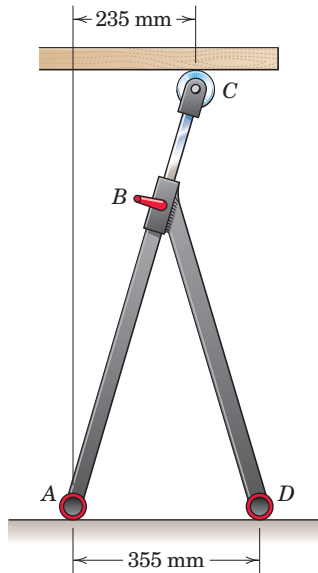
- 3/4** In the side view of a 70-kg television resting on a 24-kg cabinet, the mass centers are labeled G_1 and G_2 . Determine the force reactions at A and B . (Note that the mass center of most televisions is located well forward because of the heavy nature of the front portion of picture tubes.)



Problem 3/4

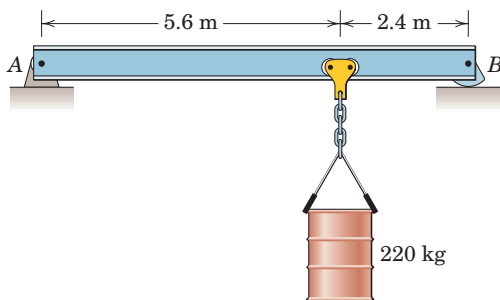
- 3/5** The roller stand is used to support portions of long boards as they are being cut on a table saw. If the board exerts a 25-N downward force on the roller C , determine the vertical reactions at A and D . Note that the connection at B is rigid, and that the feet A and D are fairly lengthy horizontal tubes with a nonslip coating.

Ans. $N_A = 8.45 \text{ N}$, $N_D = 16.55 \text{ N}$



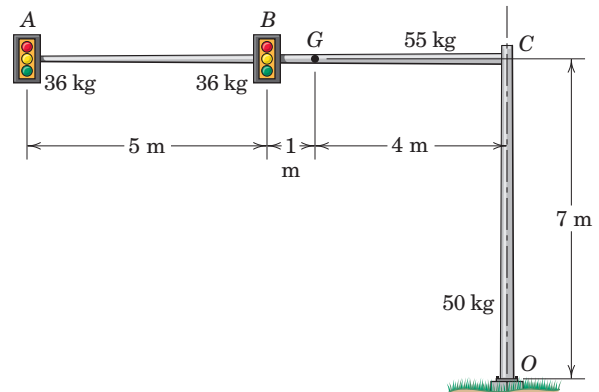
Problem 3/5

- 3/6** The 450-kg uniform I-beam supports the load shown. Determine the reactions at the supports.



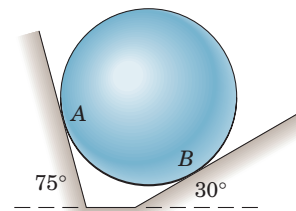
Problem 3/6

- 3/7** Calculate the force and moment reactions at the bolted base O of the overhead traffic-signal assembly. Each traffic signal has a mass of 36 kg, while the masses of members OC and AC are 50 kg and 55 kg, respectively.
- Ans.* $O_x = 0$, $O_y = 1736 \text{ N}$, $M_O = 7460 \text{ N}\cdot\text{m CW}$



Problem 3/7

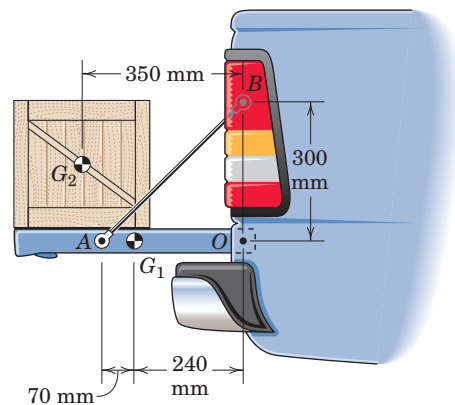
- 3/8** The 20-kg homogeneous smooth sphere rests on the two inclines as shown. Determine the contact forces at A and B .



Problem 3/8

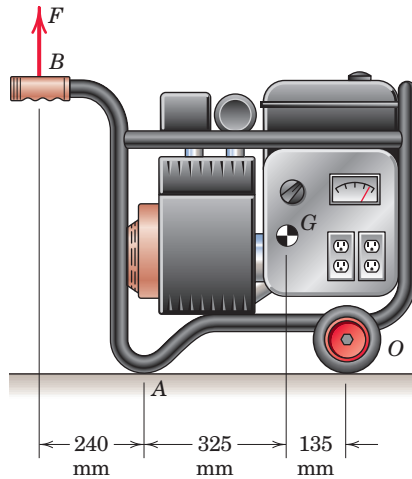
- 3/9** A 54-kg crate rests on the 27-kg pickup tailgate. Calculate the tension T in each of the two restraining cables, one of which is shown. The centers of gravity are at G_1 and G_2 . The crate is located midway between the two cables.

Ans. $T = 577 \text{ N}$



Problem 3/9

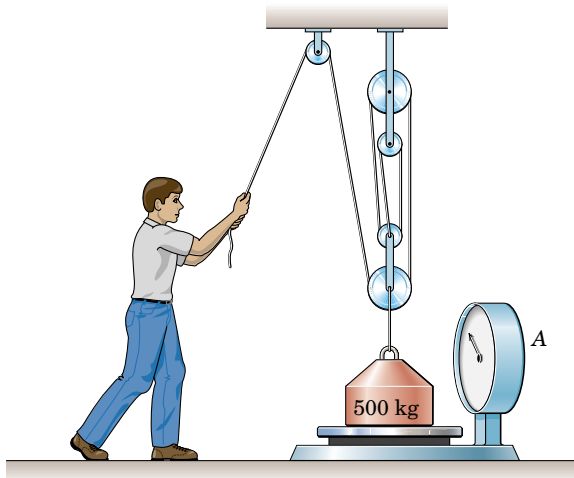
- 3/10** A portable electric generator has a mass of 160 kg with mass center at G . Determine the upward force F necessary to reduce the normal force at A to one-half its nominal ($F = 0$) value.



Problem 3/10

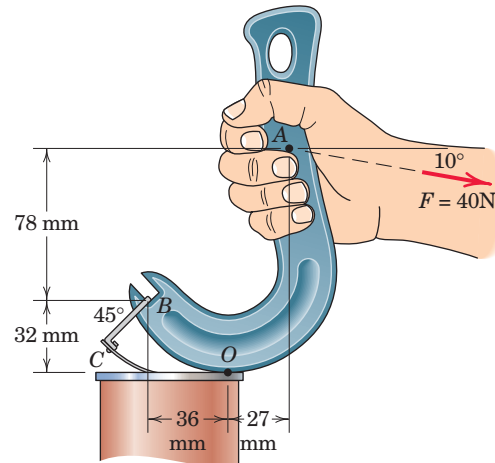
- 3/11** With what force magnitude T must the person pull on the cable in order to cause the scale A to read 2000 N? The weights of the pulleys and cables are negligible. State any assumptions.

Ans. $T = 581$ N



Problem 3/11

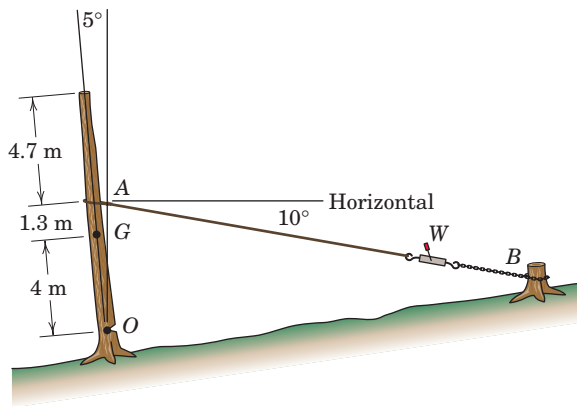
- 3/12** The device shown is designed to aid in the removal of pull-tab tops from cans. If the user exerts a 40-N force at A , determine the tension T in the portion BC of the pull tab.



Problem 3/12

- 3/13** A woodcutter wishes to cause the tree trunk to fall uphill, even though the trunk is leaning downhill. With the aid of the winch W , what tension T in the cable will be required? The 600-kg trunk has a center of gravity at G . The felling notch at O is sufficiently large so that the resisting moment there is negligible.

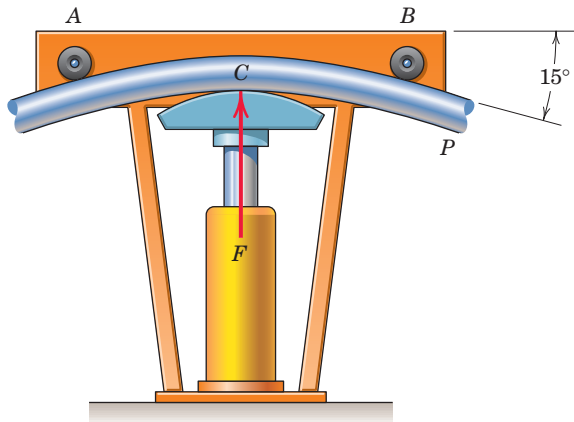
Ans. $T = 401$ N



Problem 3/13

Representative Problems

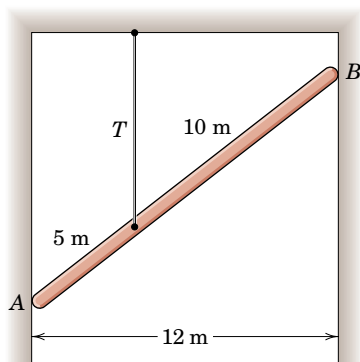
- 3/18** A pipe P is being bent by the pipe bender as shown. If the hydraulic cylinder applies a force of magnitude $F = 24$ kN to the pipe at C , determine the magnitude of the roller reactions at A and B .



Problem 3/18

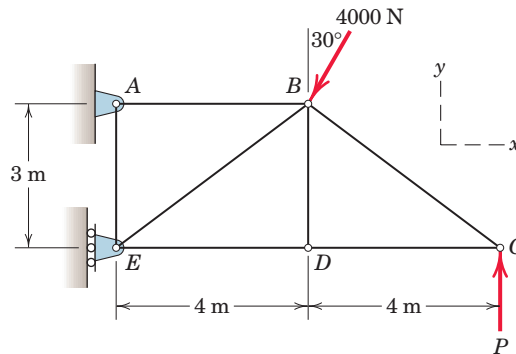
- 3/19** The uniform 15-m pole has a mass of 150 kg and is supported by its smooth ends against the vertical walls and by the tension T in the vertical cable. Compute the reactions at A and B .

Ans. $A = B = 327$ N



Problem 3/19

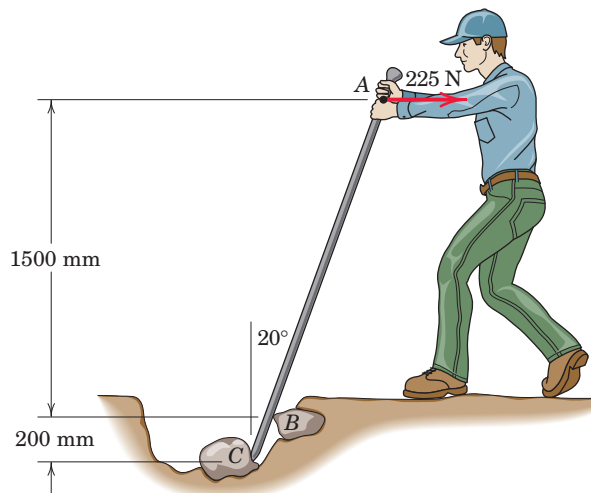
- 3/20** Determine the reactions at A and E if $P = 500$ N. What is the maximum value which P may have for static equilibrium? Neglect the weight of the structure compared with the applied loads.



Problem 3/20

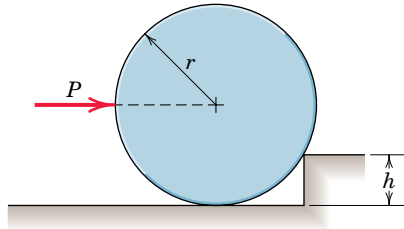
- 3/21** While digging a small hole prior to planting a tree, a homeowner encounters rocks. If he exerts a horizontal 225-N force on the prybar as shown, what is the horizontal force exerted on rock C ? Note that a small ledge on rock C supports a vertical force reaction there. Neglect friction at B . Complete solutions (a) including and (b) excluding the weight of the 18-kg prybar.

Ans. (a) $F_C = 1705$ N, (b) $F_C = 1464$ N



Problem 3/21

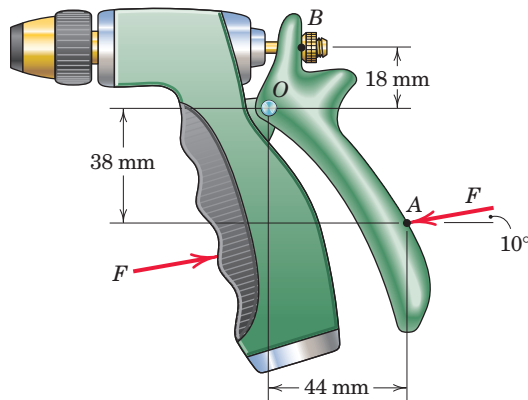
- 3/22** Determine the force P required to begin rolling the uniform cylinder of mass m over the obstruction of height h .



Problem 3/22

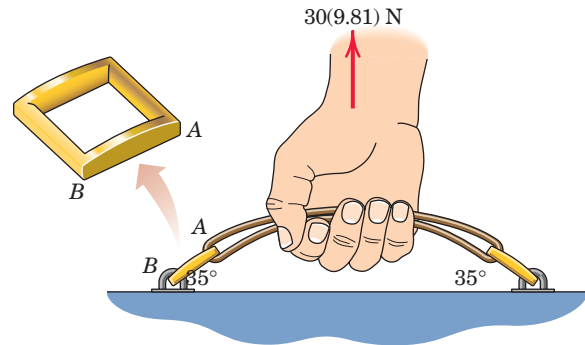
- 3/23** A 35-N axial force at B is required to open the spring-loaded plunger of the water nozzle. Determine the required force F applied to the handle at A and the magnitude of the pin reaction at O . Note that the plunger passes through a vertically-elongated hole in the handle at B , so that negligible vertical force is transmitted there.

Ans. $F = 13.98 \text{ N}$, $O = 48.8 \text{ N}$



Problem 3/23

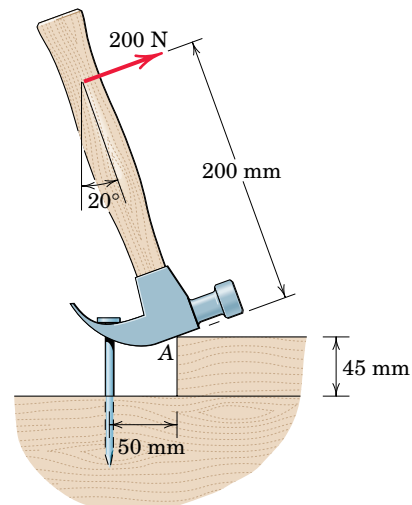
- 3/24** A person holds a 30-kg suitcase by its handle as indicated in the figure. Determine the tension in each of the four identical links AB .



Problem 3/24

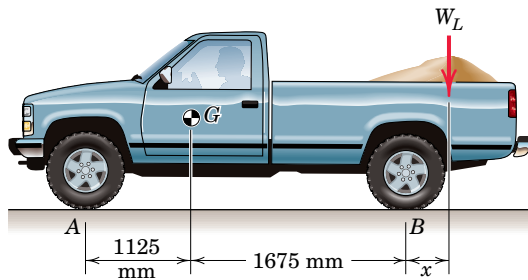
- 3/25** A block placed under the head of the claw hammer as shown greatly facilitates the extraction of the nail. If a 200-N pull on the handle is required to pull the nail, calculate the tension T in the nail and the magnitude A of the force exerted by the hammer head on the block. The contacting surfaces at A are sufficiently rough to prevent slipping.

Ans. $T = 800 \text{ N}$, $A = 755 \text{ N}$



Problem 3/25

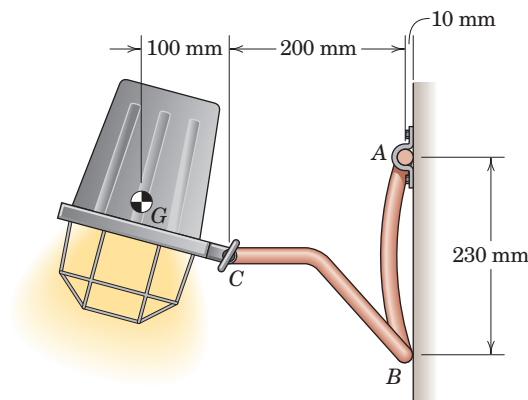
- 3/26** The indicated location of the center of mass of the 1600-kg pickup truck is for the unladen condition. If a load whose center of mass is $x = 400$ mm behind the rear axle is added to the truck, determine the mass m_L for which the normal forces under the front and rear wheels are equal.



Problem 3/26

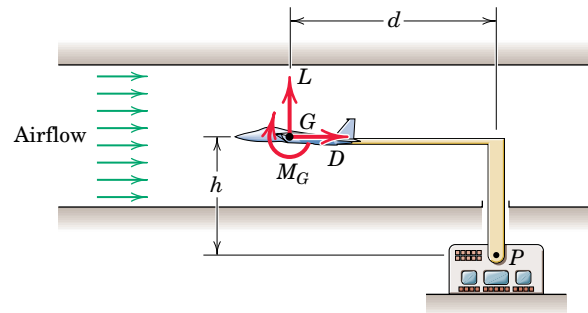
- 3/27** The wall-mounted 2.5-kg light fixture has its mass center at G . Determine the reactions at A and B and also calculate the moment supported by the adjustment thumbscrew at C . (Note that the lightweight frame ABC has about 250 mm of horizontal tubing, directed into and out of the paper, at both A and B .)

Ans. $B_x = 32.0$ N
 $A_x = 32.0$ N, $A_y = 24.5$ N
 $M_C = 2.45$ N·m CW



Problem 3/27

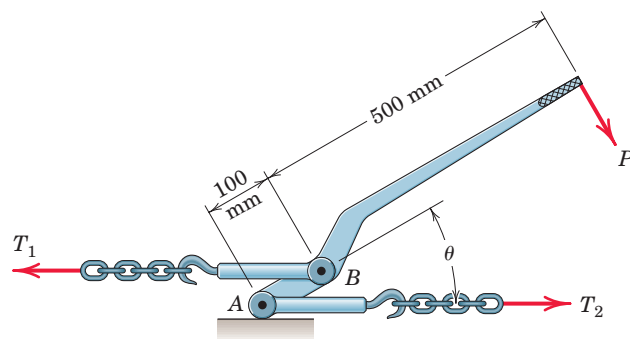
- 3/28** To test the validity of aerodynamic assumptions made in the design of the aircraft, its model is being tested in a wind tunnel. The support bracket is connected to a force and moment balance, which is zeroed when there is no airflow. Under test conditions, the lift L , drag D , and pitching moment M_G act as shown. The force balance records the lift, drag, and a moment M_P . Determine M_G in terms of L , D , and M_P .



Problem 3/28

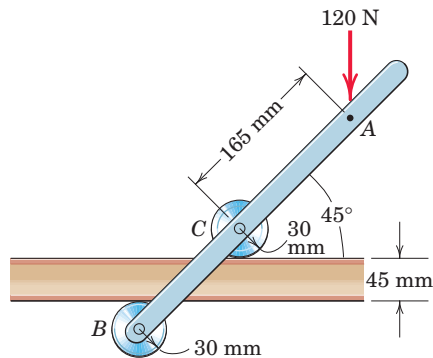
- 3/29** The chain binder is used to secure loads of logs, lumber, pipe, and the like. If the tension T_1 is 2 kN when $\theta = 30^\circ$, determine the force P required on the lever and the corresponding tension T_2 for this position. Assume that the surface under A is perfectly smooth.

Ans. $P = 166.7$ N, $T_2 = 1.917$ kN



Problem 3/29

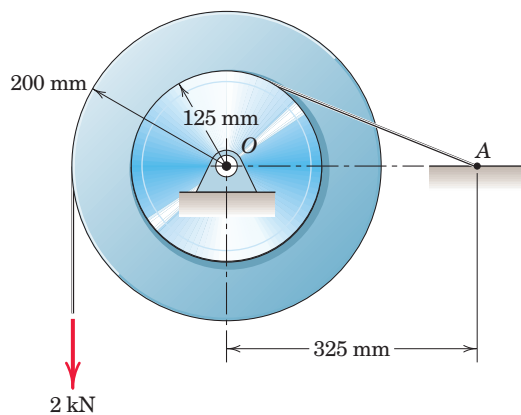
- 3/30** The device shown is designed to apply pressure when bonding laminate to each side of a countertop near an edge. If a 120-N force is applied to the handle, determine the force which each roller exerts on its corresponding surface.



Problem 3/30

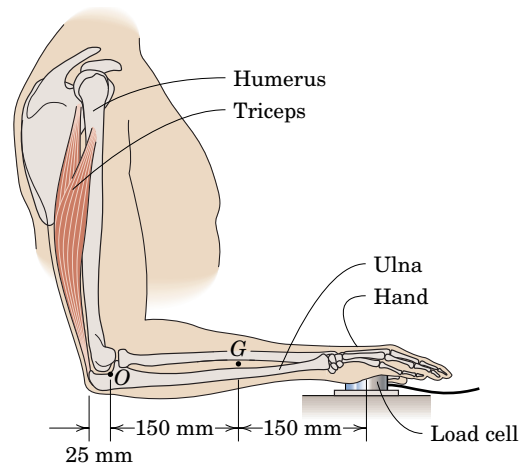
- 3/31** The two light pulleys are fastened together and form an integral unit. They are prevented from turning about their bearing at O by a cable wound securely around the smaller pulley and fastened to point A . Calculate the magnitude R of the force supported by the bearing O for the applied 2-kN load.

Ans. $R = 4.38$ kN



Problem 3/31

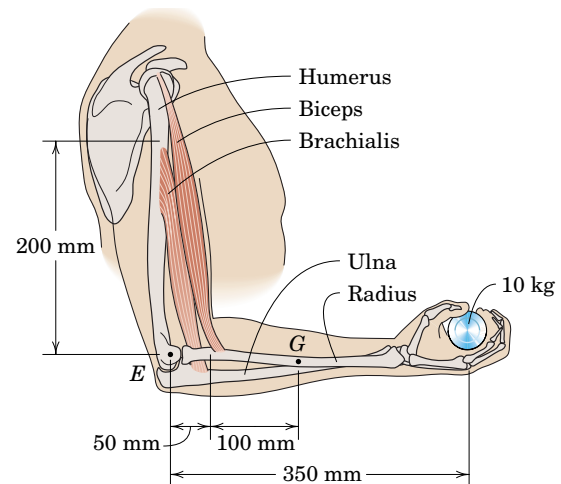
- 3/32** In a procedure to evaluate the strength of the triceps muscle, a person pushes down on a load cell with the palm of his hand as indicated in the figure. If the load-cell reading is 160 N, determine the vertical tensile force F generated by the triceps muscle. The mass of the lower arm is 1.5 kg with mass center at G . State any assumptions.



Problem 3/32

- 3/33** A person is performing slow arm curls with a 10-kg weight as indicated in the figure. The brachialis muscle group (consisting of the biceps and brachialis muscles) is the major factor in this exercise. Determine the magnitude F of the brachialis-muscle-group force and the magnitude E of the elbow joint reaction at point E for the forearm position shown in the figure. Take the dimensions shown to locate the effective points of application of the two muscle groups; these points are 200 mm directly above E and 50 mm directly to the right of E . Include the effect of the 1.5-kg forearm mass with mass center at point G . State any assumptions.

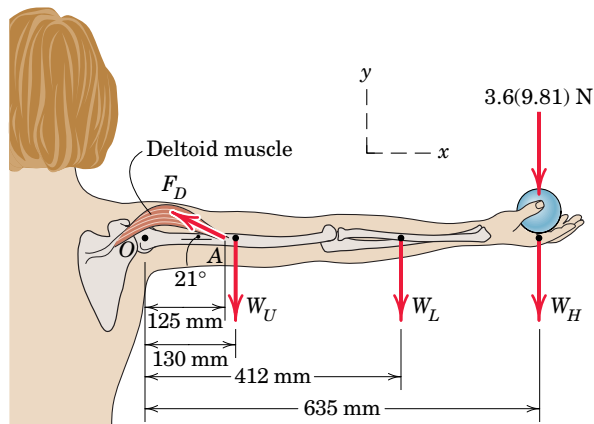
Ans. $F = 753$ N, $E = 644$ N



Problem 3/33

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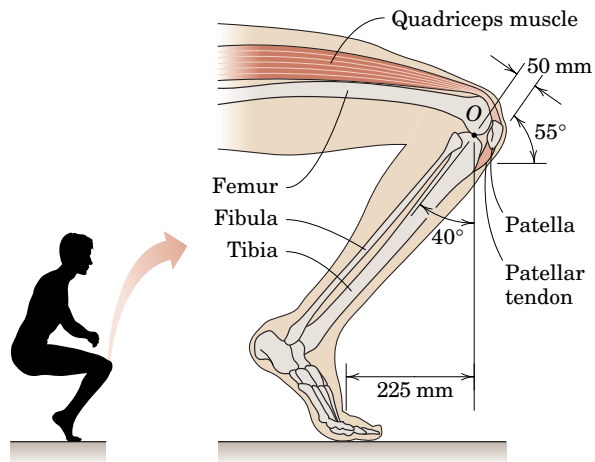
- 3/34** A woman is holding a 3.6-kg sphere in her hand with the entire arm held horizontally as shown in the figure. A tensile force in the deltoid muscle prevents the arm from rotating about the shoulder joint O ; this force acts at the 21° angle shown. Determine the force exerted by the deltoid muscle on the upper arm at A and the x - and y -components of the force reaction at the shoulder joint O . The mass of the upper arm is $m_U = 1.9$ kg, the mass of the lower arm is $m_L = 1.1$ kg, and the mass of the hand is $m_H = 0.4$ kg; all the corresponding weights act at the locations shown in the figure.



Problem 3/34

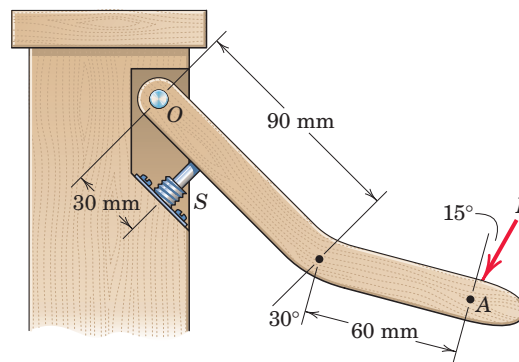
- 3/35** With his weight W equally distributed on both feet, a man begins to slowly rise from a squatting position as indicated in the figure. Determine the tensile force F in the patellar tendon and the magnitude of the force reaction at point O , which is the contact area between the tibia and the femur. Note that the line of action of the patellar tendon force is along its midline. Neglect the weight of the lower leg.

Ans. $F = 2.25W$, $O = 2.67W$



Problem 3/35

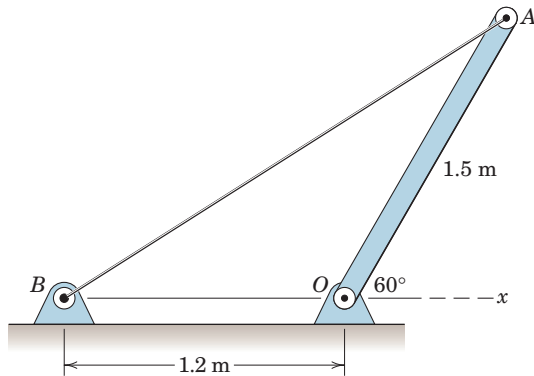
- 3/36** The elements of an on-off mechanism for a table lamp are shown in the figure. The electrical switch S requires a 4-N force in order to depress it. What corresponding force F must be exerted on the handle at A ?



Problem 3/36

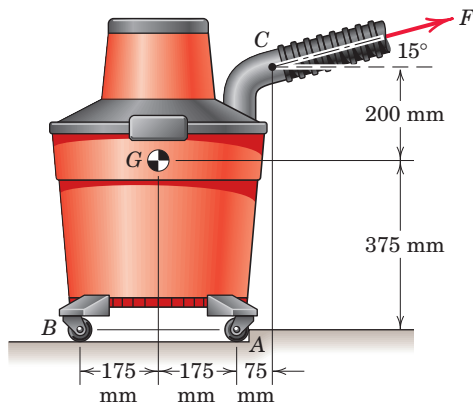
- 3/37** The uniform 18-kg bar OA is held in the position shown by the smooth pin at O and the cable AB . Determine the tension T in the cable and the magnitude and direction of the external pin reaction at O .

Ans. $T = 99.5 \text{ N}$, $O = 246 \text{ N}$, 70.3° CCW from x -axis



Problem 3/37

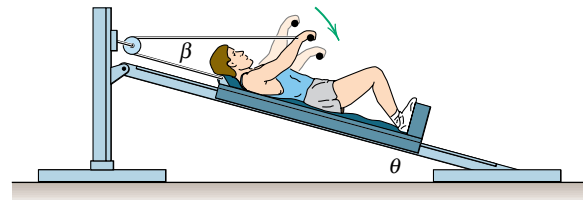
- 3/38** A person attempts to move a 20-kg shop vacuum by pulling on the hose as indicated. What force F will cause the unit to tip clockwise if wheel A is against an obstruction?



Problem 3/38

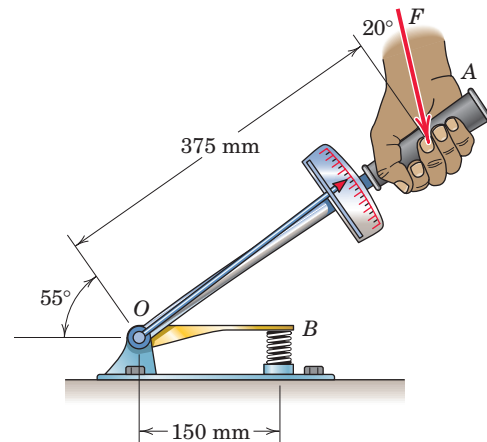
- 3/39** The exercise machine is designed with a lightweight cart which is mounted on small rollers so that it is free to move along the inclined ramp. Two cables are attached to the cart—one for each hand. If the hands are together so that the cables are parallel and if each cable lies essentially in a vertical plane, determine the force P which each hand must exert on its cable in order to maintain an equilibrium position. The mass of the person is 70 kg, the ramp angle θ is 15° , and the angle β is 18° . In addition, calculate the force R which the ramp exerts on the cart.

Ans. $P = 45.5 \text{ N}$, $R = 691 \text{ N}$



Problem 3/39

- 3/40** The device shown is used to test automobile-engine valve springs. The torque wrench is directly connected to arm OB . The specification for the automotive intake-valve spring is that 370 N of force should reduce its length from 50 mm (unstressed length) to 42 mm. What is the corresponding reading M on the torque wrench, and what force F exerted on the torque-wrench handle is required to produce this reading? Neglect the small effects of changes in the angular position of arm OB .

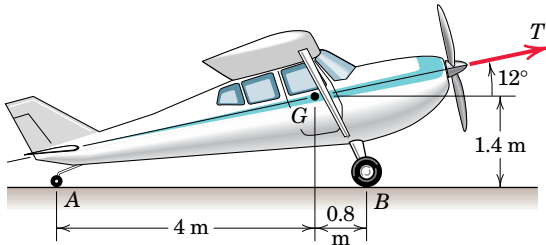


Problem 3/40

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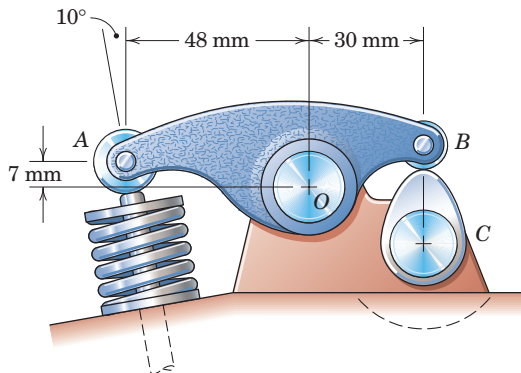
- 3/41** During an engine test on the ground, a propeller thrust $T = 3000$ N is generated on the 1800-kg airplane with mass center at G . The main wheels at B are locked and do not skid; the small tail wheel at A has no brake. Compute the percent change n in the normal forces at A and B as compared with their “engine-off” values.

Ans. $n_A = -32.6\%$, $n_B = 2.28\%$



Problem 3/41

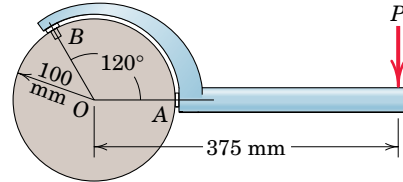
- 3/42** A rocker arm with rollers at A and B is shown in the position when the valve is open and the valve spring is fully compressed. In this position, the spring force is 900 N. Determine the force which the rocker arm exerts on the camshaft C . Also calculate the magnitude of the force supported by the rocker-arm shaft O .



Problem 3/42

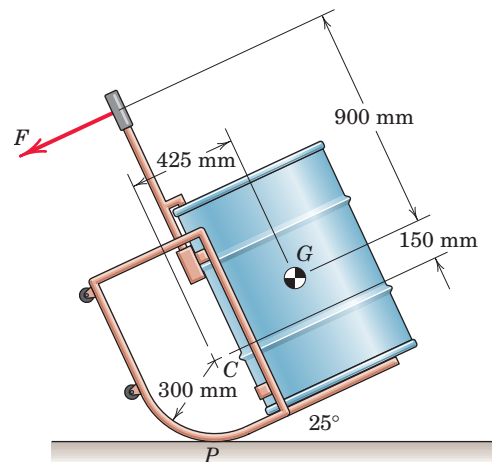
- 3/43** The hook wrench or pin spanner is used to turn shafts and collars. If a moment of 80 N·m is required to turn the 200-mm-diameter collar about its center O under the action of the applied force P , determine the contact force R on the smooth surface at A . Engagement of the pin at B may be considered to occur at the periphery of the collar.

Ans. $R = 1047$ N



Problem 3/43

- 3/44** The dolly shown is useful in the handling of large drums. Determine the force F necessary to hold a drum in the position shown. You may neglect the weight of the dolly in comparison with that of the 250-kg drum, whose center of mass is at G . There is sufficient friction to prevent slipping at the contact point P .

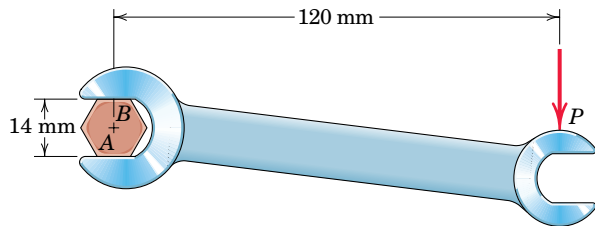


Problem 3/44

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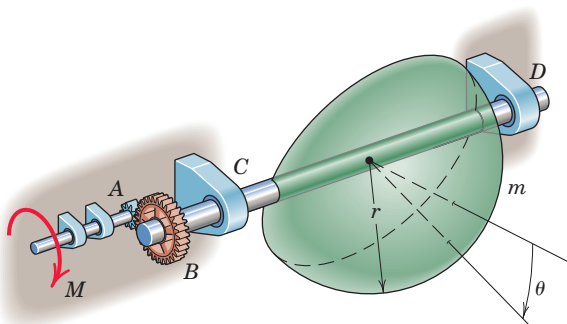
- 3/49** A torque (moment) of $24 \text{ N}\cdot\text{m}$ is required to turn the bolt about its axis. Determine P and the forces between the smooth hardened jaws of the wrench and the corners A and B of the hexagonal head. Assume that the wrench fits easily on the bolt so that contact is made at corners A and B only.

Ans. $P = 200 \text{ N}$, $A = 2870 \text{ N}$, $B = 3070 \text{ N}$



Problem 3/49

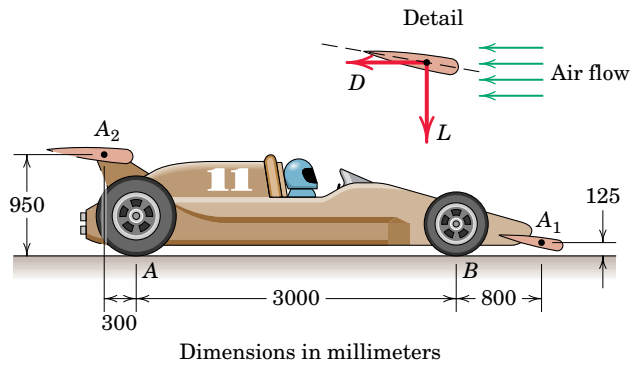
- 3/50** Determine the moment M which must be applied to the shaft in order to hold the homogeneous hemisphere in an arbitrary angular position as measured by the angle θ . The radii of gear A , gear B , and the hemisphere are r_A , r_B , and r , respectively. Assume the friction in all bearings to be negligible.



Problem 3/50

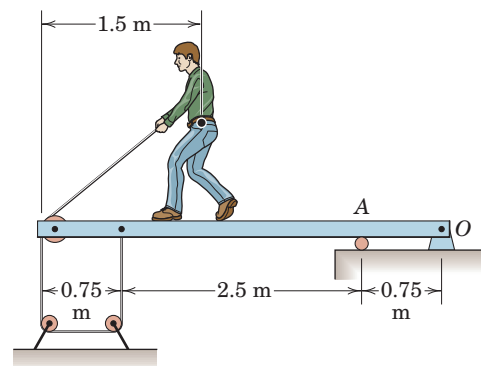
- 3/51** The car complete with driver has a mass of 815 kg and without the two airfoils has a 50%–50% front–rear weight distribution at a certain speed at which there is no lift on the car. It is estimated that at this speed each of the airfoils A_1 and A_2 will generate 2 kN of downward force L and 250 N of drag force D on the car. Specify the vertical reactions N_A and N_B under the two pairs of wheels at that speed when the airfoils are added. Assume that the addition of the airfoils does not affect the drag and zero-lift conditions of the car body itself and that the engine has sufficient power for equilibrium at that speed. The weight of the airfoils may be neglected.

Ans. $N_A = 5750 \text{ N}$ (48.0%), $N_B = 6240 \text{ N}$ (52.0%)



Problem 3/51

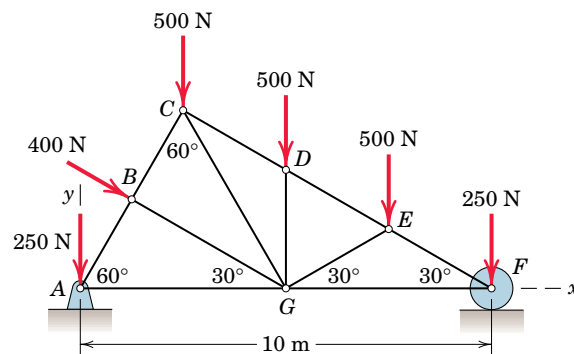
- 3/52** To test the deflection of the uniform 100-kg beam the 50-kg boy exerts a pull of 150 N on the rope rigged as shown. Compute the force supported by the pin at the hinge O .



Problem 3/52

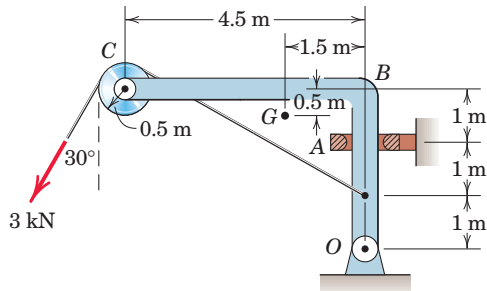
- 3/53** Determine the external reactions at A and F for the roof truss loaded as shown. The vertical loads represent the effect of the supported roofing materials, while the 400-N force represents a wind load.

Ans. $A_x = 346 \text{ N}$, $A_y = 1100 \text{ N}$, $F_y = 1100 \text{ N}$



Problem 3/53

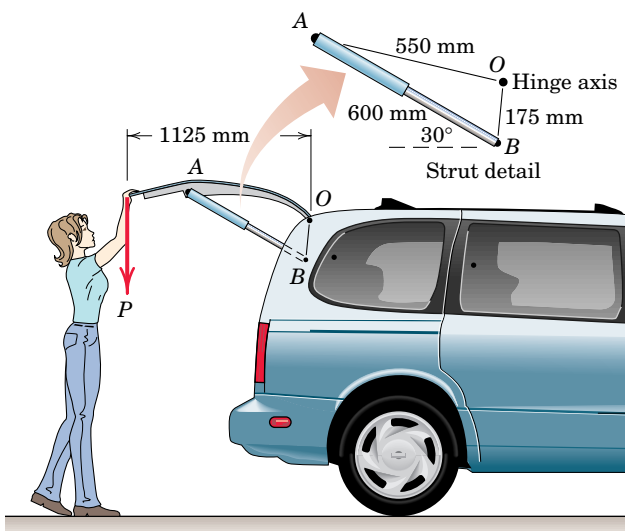
- 3/54** The member OBC and sheave at C together have a mass of 500 kg, with a combined center of mass at G . Calculate the magnitude of the force supported by the pin connection at O when the 3-kN load is applied. The collar at A can provide support in the horizontal direction only.



Problem 3/54

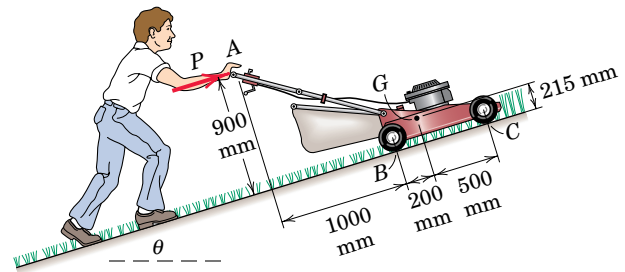
- 3/55** It is desired that a person be able to begin closing the van hatch from the open position shown with a 40-N vertical force P . As a design exercise, determine the necessary force in each of the two hydraulic struts AB . The mass center of the 40-kg door is 37.5 mm directly below point A . Treat the problem as two-dimensional.

Ans. $F = 803 \text{ N}$



Problem 3/55

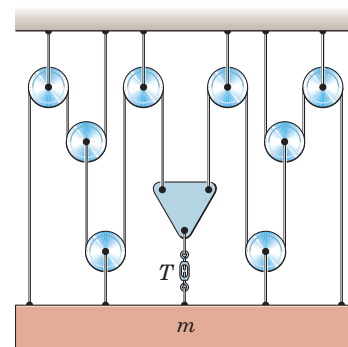
- 3/56** The man pushes the lawn mower at a steady speed with a force P which is parallel to the incline. The mass of the mower with attached grass bag is 50 kg with mass center at G . If $\theta = 15^\circ$, determine the normal forces N_B and N_C under each pair of wheels B and C . Neglect friction. Compare with the normal forces for the conditions of $\theta = 0$ and $P = 0$.



Problem 3/56

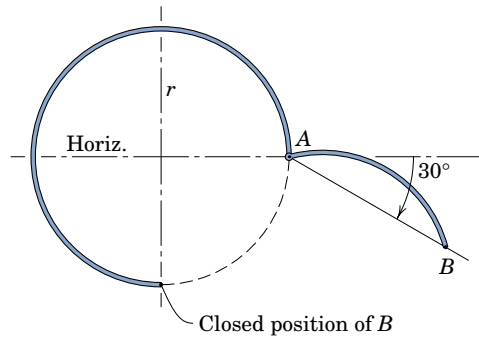
- 3/57** Determine the tension T in the turnbuckle for the pulley-cable system in terms of the mass m of the body which it supports. Neglect the mass of the pulleys and cable.

Ans. $T = \frac{2}{7}mg$



Problem 3/57

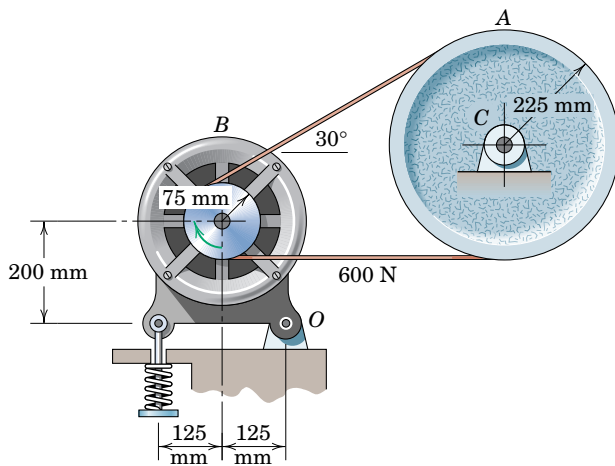
- 3/58** The cargo door for an airplane of circular fuselage section consists of the uniform quarter-circular segment AB of mass m . A detent in the hinge at A holds the door open in the position shown. Determine the moment exerted by the hinge on the door.



Problem 3/58

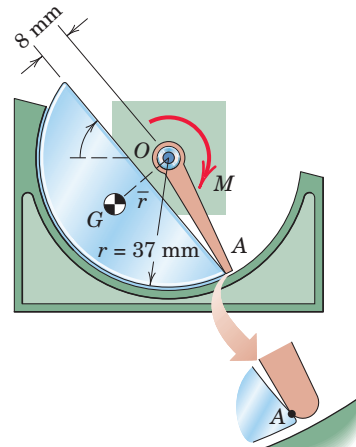
- 3/59** Pulley A delivers a steady torque (moment) of $100 \text{ N}\cdot\text{m}$ to a pump through its shaft at C . The tension in the lower side of the belt is 600 N . The driving motor B has a mass of 100 kg and rotates clockwise. As a design consideration, determine the magnitude R of the force on the supporting pin at O .

Ans. $R = 1.167 \text{ kN}$



Problem 3/59

- 3/60** Certain elements of an in-refrigerator ice-cube maker are shown in the figure. (A “cube” has the form of a cylindrical segment!) Once the cube freezes and a small heater (not shown) forms a thin film of water between the cube and supporting surface, a motor rotates the ejector arm OA to remove the cube. If there are eight cubes and eight arms, determine the required torque M as a function of θ . The mass of eight cubes is 0.25 kg , and the center-of-mass distance $\bar{r} = 0.55r$. Neglect friction, and assume that the resultant of the distributed normal force acting on the cube passes through point O .



Problem 3/60

SECTION B EQUILIBRIUM IN THREE DIMENSIONS

3/4 EQUILIBRIUM CONDITIONS

We now extend our principles and methods developed for two-dimensional equilibrium to the case of three-dimensional equilibrium. In Art. 3/1 the general conditions for the equilibrium of a body were stated in Eqs. 3/1, which require that the resultant force and resultant couple on a body in equilibrium be zero. These two vector equations of equilibrium and their scalar components may be written as

$$\begin{array}{ll} \Sigma \mathbf{F} = \mathbf{0} & \text{or} \quad \begin{cases} \Sigma F_x = 0 \\ \Sigma F_y = 0 \\ \Sigma F_z = 0 \end{cases} \\ \Sigma \mathbf{M} = \mathbf{0} & \text{or} \quad \begin{cases} \Sigma M_x = 0 \\ \Sigma M_y = 0 \\ \Sigma M_z = 0 \end{cases} \end{array} \quad (3/3)$$

The first three scalar equations state that there is no resultant force acting on a body in equilibrium in any of the three coordinate directions. The second three scalar equations express the further equilibrium requirement that there be no resultant moment acting on the body about any of the coordinate axes or about axes parallel to the coordinate axes. These six equations are both necessary and sufficient conditions for complete equilibrium. The reference axes may be chosen arbitrarily as a matter of convenience, the only restriction being that a right-handed coordinate system should be chosen when vector notation is used.

The six scalar relationships of Eqs. 3/3 are independent conditions because any of them can be valid without the others. For example, for a car which accelerates on a straight and level road in the x -direction, Newton's second law tells us that the resultant force on the car equals its mass times its acceleration. Thus $\Sigma F_x \neq 0$, but the remaining two force-equilibrium equations are satisfied because all other acceleration components are zero. Similarly, if the flywheel of the engine of the accelerating car is rotating with increasing angular speed about the x -axis, it is not in rotational equilibrium about this axis. Thus, for the flywheel alone, $\Sigma M_x \neq 0$ along with $\Sigma F_x \neq 0$, but the remaining four equilibrium equations for the flywheel would be satisfied for its mass-center axes.

In applying the vector form of Eqs. 3/3, we first express each of the forces in terms of the coordinate unit vectors \mathbf{i} , \mathbf{j} , and \mathbf{k} . For the first equation, $\Sigma \mathbf{F} = \mathbf{0}$, the vector sum will be zero only if the coefficients of \mathbf{i} , \mathbf{j} , and \mathbf{k} in the expression are, respectively, zero. These three sums, when each is set equal to zero, yield precisely the three scalar equations of equilibrium, $\Sigma F_x = 0$, $\Sigma F_y = 0$, and $\Sigma F_z = 0$.

For the second equation, $\Sigma \mathbf{M} = \mathbf{0}$, where the moment sum may be taken about any convenient point O , we express the moment of each force as the cross product $\mathbf{r} \times \mathbf{F}$, where \mathbf{r} is the position vector from O to any point on the line of action of the force \mathbf{F} . Thus $\Sigma \mathbf{M} = \Sigma(\mathbf{r} \times \mathbf{F}) = \mathbf{0}$. When the coefficients of \mathbf{i} , \mathbf{j} , and \mathbf{k} in the resulting moment equation are

set equal to zero, respectively, we obtain the three scalar moment equations $\Sigma M_x = 0$, $\Sigma M_y = 0$, and $\Sigma M_z = 0$.

Free-Body Diagrams

The summations in Eqs. 3/3 include the effects of *all* forces on the body under consideration. We learned in the previous article that the free-body diagram is the only reliable method for disclosing all forces and moments which should be included in our equilibrium equations. In three dimensions the free-body diagram serves the same essential purpose as it does in two dimensions and should *always* be drawn. We have our choice either of drawing a pictorial view of the isolated body with all external forces represented or of drawing the orthogonal projections of the free-body diagram. Both representations are illustrated in the sample problems at the end of this article.

The correct representation of forces on the free-body diagram requires a knowledge of the characteristics of contacting surfaces. These characteristics were described in Fig. 3/1 for two-dimensional problems, and their extension to three-dimensional problems is represented in Fig. 3/8 for the most common situations of force transmission. The representations in both Figs. 3/1 and 3/8 will be used in three-dimensional analysis.

The essential purpose of the free-body diagram is to develop a reliable picture of the physical action of all forces (and couples if any) acting on a body. So it is helpful to represent the forces in their correct physical sense whenever possible. In this way, the free-body diagram becomes a closer model to the actual physical problem than it would be if the forces were arbitrarily assigned or always assigned in the same mathematical sense as that of the assigned coordinate axis.

For example, in part 4 of Fig. 3/8, the correct sense of the unknowns R_x and R_y may be known or perceived to be in the sense opposite to those of the assigned coordinate axes. Similar conditions apply to the sense of couple vectors, parts 5 and 6, where their sense by the right-hand rule may be assigned opposite to that of the respective coordinate direction. By this time, you should recognize that a negative answer for an unknown force or couple vector merely indicates that its physical action is in the sense opposite to that assigned on the free-body diagram. Frequently, of course, the correct physical sense is not known initially, so that an arbitrary assignment on the free-body diagram become necessary.

Categories of Equilibrium

Application of Eqs. 3/3 falls into four categories which we identify with the aid of Fig. 3/9. These categories differ in the number and type (force or moment) of independent equilibrium equations required to solve the problem.

Category 1, equilibrium of forces all concurrent at point O , requires all three force equations, but no moment equations because the moment of the forces about any axis through O is zero.

Category 2, equilibrium of forces which are concurrent with a line, requires all equations except the moment equation about that line, which is automatically satisfied.

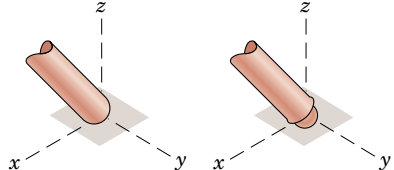
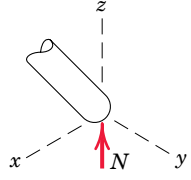
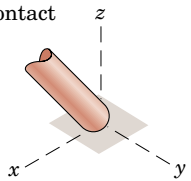
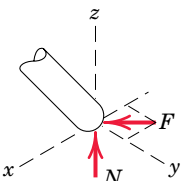
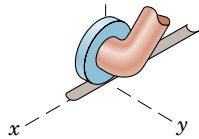
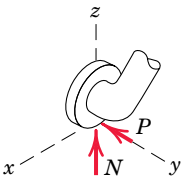
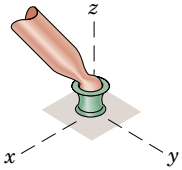
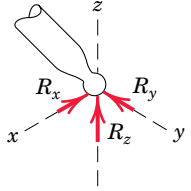
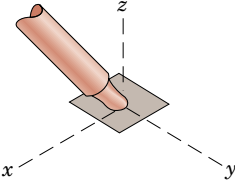
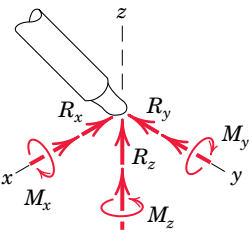
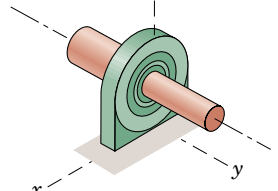
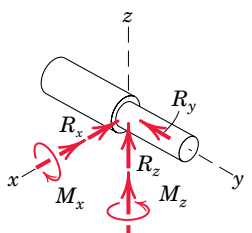
MODELING THE ACTION OF FORCES IN THREE-DIMENSIONAL ANALYSIS	
Type of Contact and Force Origin	Action on Body to Be Isolated
<p>1. Member in contact with smooth surface, or ball-supported member</p> 	 <p>Force must be normal to the surface and directed toward the member.</p>
<p>2. Member in contact with rough surface</p> 	 <p>The possibility exists for a force F tangent to the surface (friction force) to act on the member, as well as a normal force N.</p>
<p>3. Roller or wheel support with lateral constraint</p> 	 <p>A lateral force P exerted by the guide on the wheel can exist, in addition to the normal force N.</p>
<p>4. Ball-and-socket joint</p> 	 <p>A ball-and-socket joint free to pivot about the center of the ball can support a force \mathbf{R} with all three components.</p>
<p>5. Fixed connection (embedded or welded)</p> 	 <p>In addition to three components of force, a fixed connection can support a couple \mathbf{M} represented by its three components.</p>
<p>6. Thrust-bearing support</p> 	 <p>Thrust bearing is capable of supporting axial force R_y as well as radial forces R_x and R_z. Couples M_x and M_z must, in some cases, be assumed zero in order to provide statical determinacy.</p>

Figure 3/8

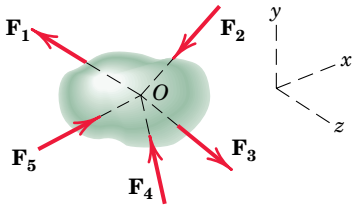
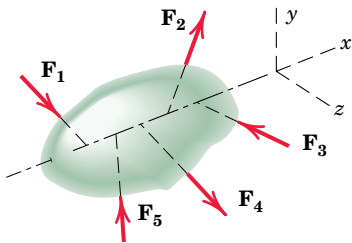
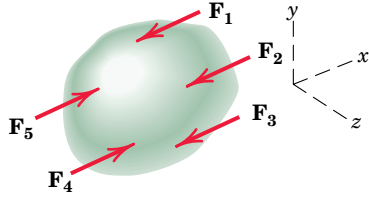
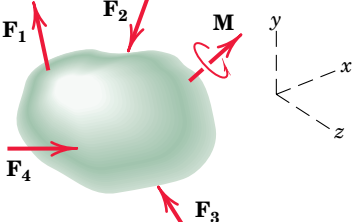
CATEGORIES OF EQUILIBRIUM IN THREE DIMENSIONS		
Force System	Free-Body Diagram	Independent Equations
1. Concurrent at a point		$\Sigma F_x = 0$ $\Sigma F_y = 0$ $\Sigma F_z = 0$
2. Concurrent with a line		$\Sigma F_x = 0$ $\Sigma M_y = 0$ $\Sigma F_y = 0$ $\Sigma M_z = 0$ $\Sigma F_z = 0$
3. Parallel		$\Sigma F_x = 0$ $\Sigma M_y = 0$ $\Sigma M_z = 0$
4. General		$\Sigma F_x = 0$ $\Sigma M_x = 0$ $\Sigma F_y = 0$ $\Sigma M_y = 0$ $\Sigma F_z = 0$ $\Sigma M_z = 0$

Figure 3/9

Category 3, equilibrium of parallel forces, requires only one force equation, the one in the direction of the forces (x -direction as shown), and two moment equations about the axes (y and z) which are normal to the direction of the forces.

Category 4, equilibrium of a general system of forces, requires all three force equations and all three moment equations.

The observations contained in these statements are generally quite evident when a given problem is being solved.

Constraints and Statical Determinacy

The six scalar relations of Eqs. 3/3, although necessary and sufficient conditions to establish equilibrium, do not necessarily provide all of the information required to calculate the unknown forces acting in a three-dimensional equilibrium situation. Again, as we found with two dimensions, the question of adequacy of information is decided by the

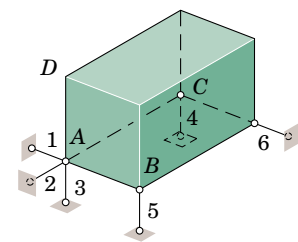
characteristics of the constraints provided by the supports. An analytical criterion for determining the adequacy of constraints is available, but it is beyond the scope of this treatment.* In Fig. 3/10, however, we cite four examples of constraint conditions to alert the reader to the problem.

Part *a* of Fig. 3/10 shows a rigid body whose corner point *A* is completely fixed by the links 1, 2, and 3. Links 4, 5, and 6 prevent rotations about the axes of links 1, 2, and 3, respectively, so that the body is *completely fixed* and the constraints are said to be *adequate*. Part *b* of the figure shows the same number of constraints, but we see that they provide no resistance to a moment which might be applied about axis *AE*. Here the body is *incompletely fixed* and only *partially constrained*.

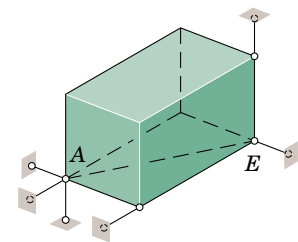
Similarly, in Fig. 3/10*c* the constraints provide no resistance to an unbalanced force in the *y*-direction, so here also is a case of incomplete fixity with partial constraints. In Fig. 3/10*d*, if a seventh constraining link were imposed on a system of six constraints placed properly for complete fixity, more supports would be provided than would be necessary to establish the equilibrium position, and link 7 would be *redundant*. The body would then be *statically indeterminate* with such a seventh link in place. With only a few exceptions, the supporting constraints for rigid bodies in equilibrium in this book are adequate, and the bodies are statically determinate.



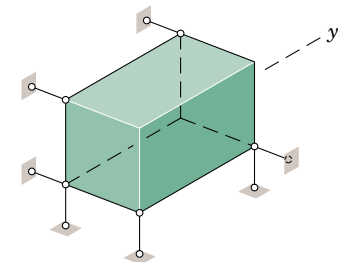
The equilibrium of structural components such as these shell-like panels is an issue both during and after construction. This structure will be used to house airships.



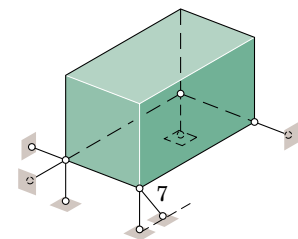
(a) Complete fixity
Adequate constraints



(b) Incomplete fixity
Partial constraints



(c) Incomplete fixity
Partial constraints



(d) Excessive fixity
Redundant constraints

Figure 3/10

*See the first author's *Statics, 2nd Edition SI Version*, 1975, Art. 16.

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Sample Problem 3/5

The uniform 7-m steel shaft has a mass of 200 kg and is supported by a ball-and-socket joint at A in the horizontal floor. The ball end B rests against the smooth vertical walls as shown. Compute the forces exerted by the walls and the floor on the ends of the shaft.

- Solution.** The free-body diagram of the shaft is first drawn where the contact forces acting on the shaft at B are shown normal to the wall surfaces. In addition to the weight $W = mg = 200(9.81) = 1962$ N, the force exerted by the floor on the ball joint at A is represented by its x -, y -, and z -components. These components are shown in their correct physical sense, as should be evident from the requirement that A be held in place. The vertical position of B is found from $7 = \sqrt{2^2 + 6^2 + h^2}$, $h = 3$ m. Right-handed coordinate axes are assigned as shown.

Vector solution. We will use A as a moment center to eliminate reference to the forces at A. The position vectors needed to compute the moments about A are

$$\mathbf{r}_{AG} = -\mathbf{i} - 3\mathbf{j} + 1.5\mathbf{k} \text{ m} \quad \text{and} \quad \mathbf{r}_{AB} = -2\mathbf{i} - 6\mathbf{j} + 3\mathbf{k} \text{ m}$$

where the mass center G is located halfway between A and B.

The vector moment equation gives

$$\begin{aligned} [\Sigma \mathbf{M}_A = \mathbf{0}] \quad & \mathbf{r}_{AB} \times (\mathbf{B}_x + \mathbf{B}_y) + \mathbf{r}_{AG} \times \mathbf{W} = \mathbf{0} \\ & (-2\mathbf{i} - 6\mathbf{j} + 3\mathbf{k}) \times (B_x\mathbf{i} + B_y\mathbf{j}) + (-\mathbf{i} - 3\mathbf{j} + 1.5\mathbf{k}) \times (-1962\mathbf{k}) = \mathbf{0} \\ & \begin{vmatrix} \mathbf{i} & \mathbf{j} & \mathbf{k} \\ -2 & -6 & 3 \\ B_x & B_y & 0 \end{vmatrix} + \begin{vmatrix} \mathbf{i} & \mathbf{j} & \mathbf{k} \\ -1 & -3 & 1.5 \\ 0 & 0 & -1962 \end{vmatrix} = \mathbf{0} \\ & (-3B_y + 5890)\mathbf{i} + (3B_x - 1962)\mathbf{j} + (-2B_y + 6B_x)\mathbf{k} = \mathbf{0} \end{aligned}$$

Equating the coefficients of \mathbf{i} , \mathbf{j} , and \mathbf{k} to zero and solving give

$$\textcircled{2} \quad B_x = 654 \text{ N} \quad \text{and} \quad B_y = 1962 \text{ N} \quad \text{Ans.}$$

The forces at A are easily determined by

$$[\Sigma \mathbf{F} = \mathbf{0}] \quad (654 - A_x)\mathbf{i} + (1962 - A_y)\mathbf{j} + (-1962 + A_z)\mathbf{k} = \mathbf{0}$$

$$\text{and} \quad A_x = 654 \text{ N} \quad A_y = 1962 \text{ N} \quad A_z = 1962 \text{ N}$$

$$\begin{aligned} \text{Finally} \quad A &= \sqrt{A_x^2 + A_y^2 + A_z^2} \\ &= \sqrt{(654)^2 + (1962)^2 + (1962)^2} = 2850 \text{ N} \quad \text{Ans.} \end{aligned}$$

Scalar solution. Evaluating the scalar moment equations about axes through A parallel, respectively, to the x - and y -axes, gives

$$[\Sigma M_{A_x} = 0] \quad 1962(3) - 3B_y = 0 \quad B_y = 1962 \text{ N}$$

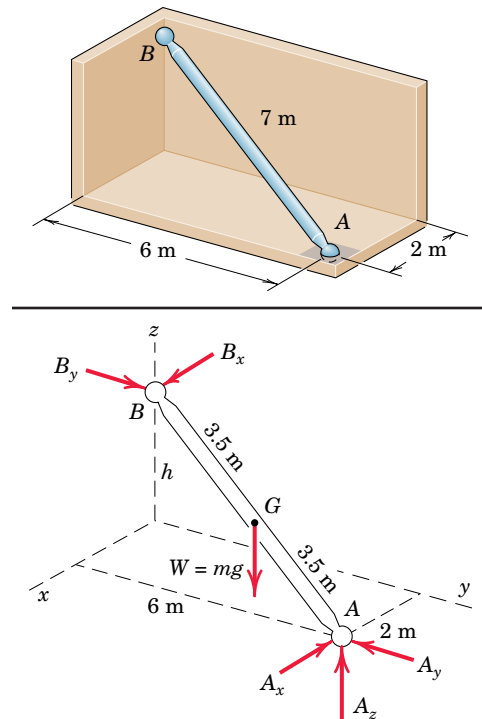
$$\textcircled{3} \quad [\Sigma M_{A_y} = 0] \quad -1962(1) + 3B_x = 0 \quad B_x = 654 \text{ N}$$

The force equations give, simply,

$$[\Sigma F_x = 0] \quad -A_x + 654 = 0 \quad A_x = 654 \text{ N}$$

$$[\Sigma F_y = 0] \quad -A_y + 1962 = 0 \quad A_y = 1962 \text{ N}$$

$$[\Sigma F_z = 0] \quad A_z - 1962 = 0 \quad A_z = 1962 \text{ N}$$

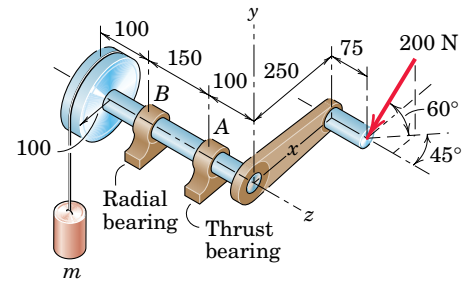


Helpful Hints

- ① We could, of course, assign all of the unknown components of force in the positive mathematical sense, in which case A_x and A_y would turn out to be negative upon computation. The free-body diagram describes the physical situation, so it is generally preferable to show the forces in their correct physical senses wherever possible.
- ② Note that the third equation $-2B_y + 6B_x = 0$ merely checks the results of the first two equations. This result could be anticipated from the fact that an equilibrium system of forces concurrent with a line requires only two moment equations (Category 2 under *Categories of Equilibrium*).
- ③ We observe that a moment sum about an axis through A parallel to the z -axis merely gives us $6B_x - 2B_y = 0$, which serves only as a check as noted previously. Alternatively we could have first obtained A_z from $\Sigma F_z = 0$ and then taken our moment equations about axes through B to obtain A_x and A_y .

Sample Problem 3/6

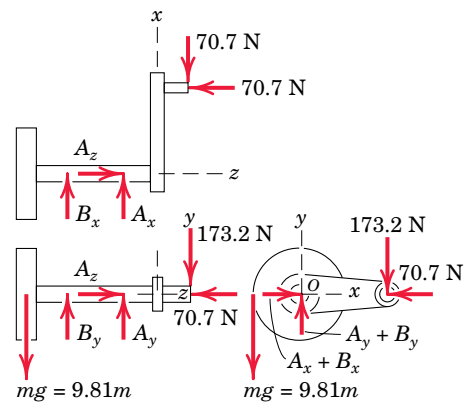
A 200-N force is applied to the handle of the hoist in the direction shown. The bearing A supports the thrust (force in the direction of the shaft axis), while bearing B supports only radial load (load normal to the shaft axis). Determine the mass m which can be supported and the total radial force exerted on the shaft by each bearing. Assume neither bearing to be capable of supporting a moment about a line normal to the shaft axis.



Dimensions in millimeters

- Solution.** The system is clearly three-dimensional with no lines or planes of symmetry, and therefore the problem must be analyzed as a general space system of forces. A scalar solution is used here to illustrate this approach, although a solution using vector notation would also be satisfactory. The free-body diagram of the shaft, lever, and drum considered a single body could be shown by a space view if desired, but is represented here by its three orthogonal projections.

The 200-N force is resolved into its three components, and each of the three views shows two of these components. The correct directions of A_x and B_x may be seen by inspection by observing that the line of action of the resultant of the two 70.7-N forces passes between A and B . The correct sense of the forces A_y and B_y cannot be determined until the magnitudes of the moments are obtained, so they are arbitrarily assigned. The x - y projection of the bearing forces is shown in terms of the sums of the unknown x - and y -components. The addition of A_z and the weight $W = mg$ completes the free-body diagrams. It should be noted that the three views represent three two-dimensional problems related by the corresponding components of the forces.



- ② From the x - y projection

$$[\Sigma M_O = 0] \quad 100(9.81m) - 250(173.2) = 0 \quad m = 44.1 \text{ kg} \quad \text{Ans.}$$

From the x - z projection

$$[\Sigma M_A = 0] \quad 150B_x + 175(70.7) - 250(70.7) = 0 \quad B_x = 35.4 \text{ N}$$

$$[\Sigma F_x = 0] \quad A_x + 35.4 - 70.7 = 0 \quad A_x = 35.4 \text{ N}$$

- ③ The y - z view gives

$$[\Sigma M_A = 0] \quad 150B_y + 175(173.2) - 250(44.1)(9.81) = 0 \quad B_y = 520 \text{ N}$$

$$[\Sigma F_y = 0] \quad A_y + 520 - 173.2 - (44.1)(9.81) = 0 \quad A_y = 86.8 \text{ N}$$

$$[\Sigma F_z = 0] \quad A_z = 70.7 \text{ N}$$

The total radial forces on the bearings become

$$[A_r = \sqrt{A_x^2 + A_y^2}] \quad A_r = \sqrt{(35.4)^2 + (86.8)^2} = 93.5 \text{ N} \quad \text{Ans.}$$

$$④ [B = \sqrt{B_x^2 + B_y^2}] \quad B = \sqrt{(35.4)^2 + (520)^2} = 521 \text{ N} \quad \text{Ans.}$$

Helpful Hints

- ① If the standard three views of orthographic projection are not entirely familiar, then review and practice them. Visualize the three views as the images of the body projected onto the front, top, and end surfaces of a clear plastic box placed over and aligned with the body.
- ② We could have started with the x - z projection rather than with the x - y projection.
- ③ The y - z view could have followed immediately after the x - y view since the determination of A_y and B_y may be made after m is found.
- ④ Without the assumption of zero moment supported by each bearing about a line normal to the shaft axis, the problem would be statically indeterminate.

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Sample Problem 3/7

The welded tubular frame is secured to the horizontal x - y plane by a ball-and-socket joint at A and receives support from the loose-fitting ring at B . Under the action of the 2-kN load, rotation about a line from A to B is prevented by the cable CD , and the frame is stable in the position shown. Neglect the weight of the frame compared with the applied load and determine the tension T in the cable, the reaction at the ring, and the reaction components at A .

Solution. The system is clearly three-dimensional with no lines or planes of symmetry, and therefore the problem must be analyzed as a general space system of forces. The free-body diagram is drawn, where the ring reaction is shown in terms of its two components. All unknowns except \mathbf{T} may be eliminated by a moment sum about the line AB . The direction of AB is specified by the unit

- ① vector $\mathbf{n} = \frac{1}{\sqrt{6^2 + 4.5^2}} (4.5\mathbf{j} + 6\mathbf{k}) = \frac{1}{5}(3\mathbf{j} + 4\mathbf{k})$. The moment of \mathbf{T} about AB

is the component in the direction of AB of the vector moment about the point A and equals $\mathbf{r}_1 \times \mathbf{T} \cdot \mathbf{n}$. Similarly the moment of the applied load F about AB is $\mathbf{r}_2 \times \mathbf{F} \cdot \mathbf{n}$. With $CD = \sqrt{46.2}$ m, the vector expressions for \mathbf{T} , \mathbf{F} , \mathbf{r}_1 , and \mathbf{r}_2 are

$$\mathbf{T} = \frac{T}{\sqrt{46.2}} (2\mathbf{i} + 2.5\mathbf{j} - 6\mathbf{k}) \quad \mathbf{F} = 2\mathbf{j} \text{ kN}$$

- ② $\mathbf{r}_1 = -\mathbf{i} + 2.5\mathbf{j} \text{ m} \quad \mathbf{r}_2 = 2.5\mathbf{i} + 6\mathbf{k} \text{ m}$

The moment equation now becomes

$$[\Sigma M_{AB} = 0] \quad (-\mathbf{i} + 2.5\mathbf{j}) \times \frac{T}{\sqrt{46.2}} (2\mathbf{i} + 2.5\mathbf{j} - 6\mathbf{k}) \cdot \frac{1}{5}(3\mathbf{j} + 4\mathbf{k}) \\ + (2.5\mathbf{i} + 6\mathbf{k}) \times (2\mathbf{j}) \cdot \frac{1}{5}(3\mathbf{j} + 4\mathbf{k}) = 0$$

Completion of the vector operations gives

$$-\frac{48T}{\sqrt{46.2}} + 20 = 0 \quad T = 2.83 \text{ kN} \quad \text{Ans.}$$

and the components of T become

$$T_x = 0.833 \text{ kN} \quad T_y = 1.042 \text{ kN} \quad T_z = -2.50 \text{ kN}$$

We may find the remaining unknowns by moment and force summations as follows:

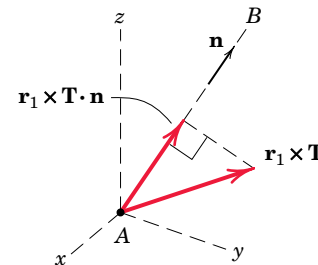
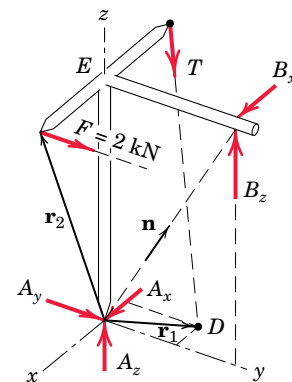
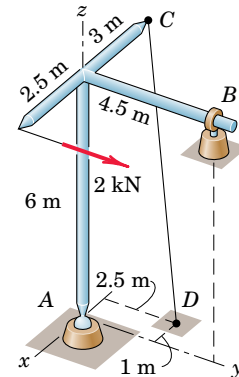
$$[\Sigma M_z = 0] \quad 2(2.5) - 4.5B_x - 1.042(3) = 0 \quad B_x = 0.417 \text{ kN} \quad \text{Ans.}$$

$$[\Sigma M_x = 0] \quad 4.5B_z - 2(6) - 1.042(6) = 0 \quad B_z = 4.06 \text{ kN} \quad \text{Ans.}$$

$$[\Sigma F_x = 0] \quad A_x + 0.417 + 0.833 = 0 \quad A_x = -1.250 \text{ kN} \quad \text{Ans.}$$

③ $[\Sigma F_y = 0] \quad A_y + 2 + 1.042 = 0 \quad A_y = -3.04 \text{ kN} \quad \text{Ans.}$

$$[\Sigma F_z = 0] \quad A_z + 4.06 - 2.50 = 0 \quad A_z = -1.556 \text{ kN} \quad \text{Ans.}$$



Helpful Hints

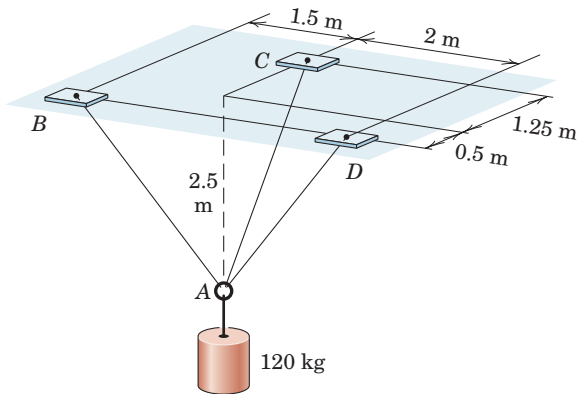
- ① The advantage of using vector notation in this problem is the freedom to take moments directly about any axis. In this problem this freedom permits the choice of an axis that eliminates five of the unknowns.
- ② Recall that the vector \mathbf{r} in the expression $\mathbf{r} \times \mathbf{F}$ for the moment of a force is a vector from the moment center to any point on the line of action of the force. Instead of \mathbf{r}_1 , an equally simple choice would be the vector \mathbf{AC} .
- ③ The negative signs associated with the A -components indicate that they are in the opposite direction to those shown on the free-body diagram.

PROBLEMS

Introductory Problems

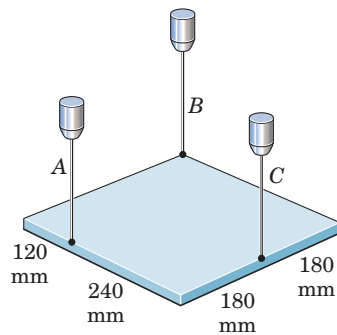
3/61 Determine the tensions in cables AB , AC , and AD .

Ans. $T_{AB} = 569 \text{ N}$, $T_{AC} = 376 \text{ N}$, $T_{AD} = 467 \text{ N}$



Problem 3/61

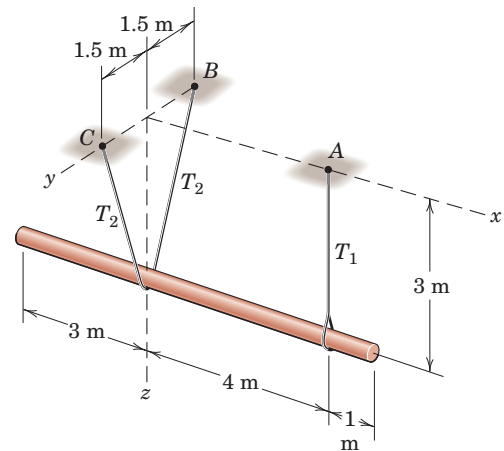
3/62 A uniform steel plate 360 mm square with a mass of 15 kg is suspended in the horizontal plane by the three vertical wires as shown. Calculate the tension in each wire.



Problem 3/62

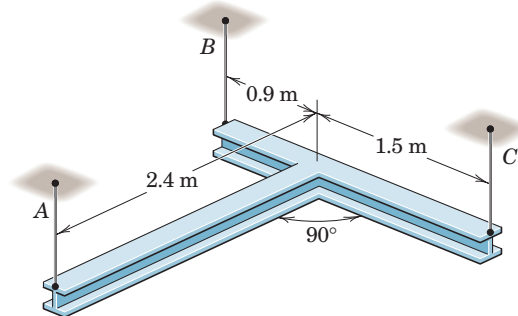
3/63 The horizontal steel shaft has a mass of 480 kg and is suspended by a vertical cable from A and by a second cable BC which lies in a vertical transverse plane and loops underneath the shaft. Calculate the tensions T_1 and T_2 in the cables.

Ans. $T_1 = 1177 \text{ N}$, $T_2 = 1974 \text{ N}$



Problem 3/63

3/64 Two steel I-beams, each with a mass of 100 kg, are welded together at right angles and lifted by vertical cables so that the beams remain in a horizontal plane. Compute the tension in each of the cables A , B , and C .

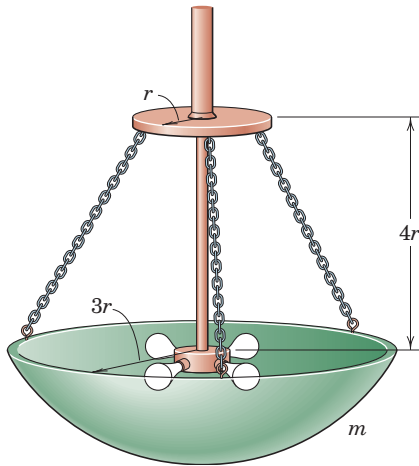


Problem 3/64

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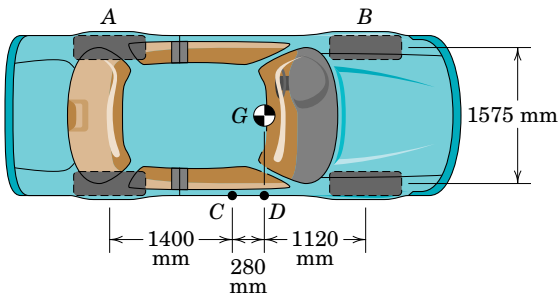
- 3/65** The chain-supported portion of a light fixture is in the shape of part of a spherical shell. If the mass of the glass unit is m , determine the tension T in each of the three chains.

Ans. $T = 0.373mg$



Problem 3/65

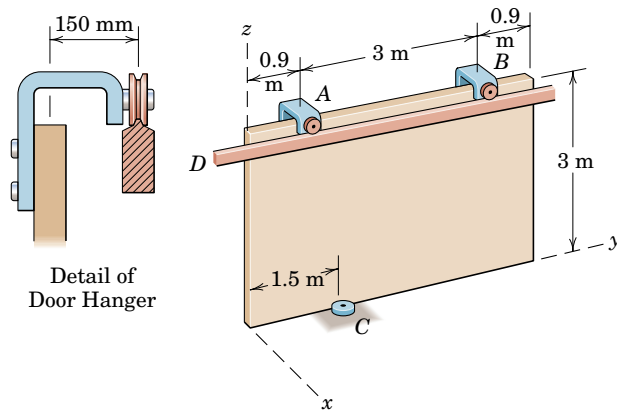
- 3/66** An overhead view of a car is shown in the figure. Two different locations C and D are considered for a single jack. In each case, the entire right side of the car is lifted just off the ground. Determine the normal reaction forces at A and B and the vertical jacking force required for the case of each jacking location. Consider the 1600-kg car to be rigid. The mass center G is on the midline of the car.



Problem 3/66

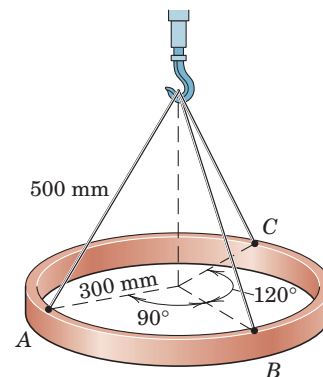
- 3/67** The 600-kg industrial door is a uniform rectangular panel which rolls along the fixed rail D on its hanger-mounted wheels A and B . The door is maintained in a vertical plane by the floor-mounted guide roller C , which bears against the bottom edge. For the position shown compute the horizontal side thrust on each of the wheels A and B , which must be accounted for in the design of the brackets.

Ans. $A_x = 235 \text{ N}$, $B_x = 58.9 \text{ N}$



Problem 3/67

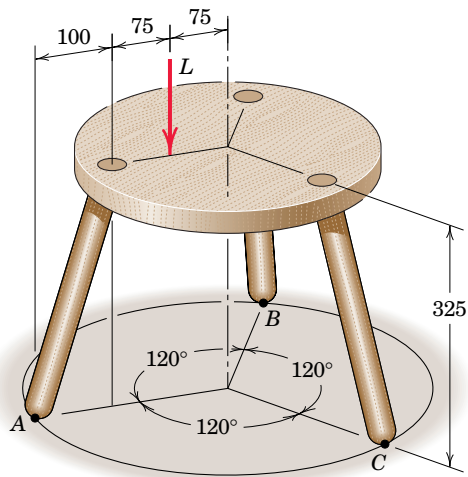
- 3/68** A uniform steel ring 600 mm in diameter has a mass of 50 kg and is lifted by the three cables, each 500 mm long, attached at points A , B , and C as shown. Compute the tension in each cable.



Problem 3/68

- 3/69** A three-legged stool is subjected to the load L as shown. determine the vertical force reaction under each leg. Neglect the weight of the stool.

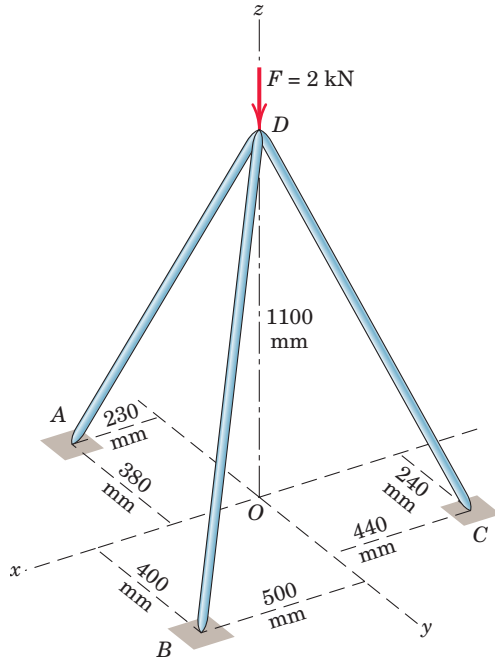
Ans. $N_A = 0.533L$, $N_B = N_C = 0.233L$



Dimensions in millimeters

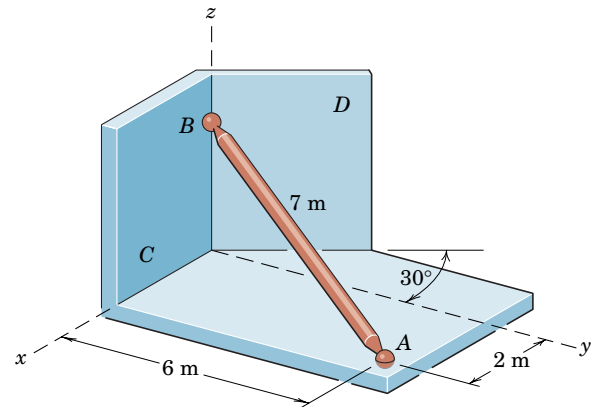
Problem 3/69

- 3/70** Determine the compression in each of the three legs of the tripod subjected to the vertical 2-kN force. The weight of the legs is negligible compared with the applied load. Solve by using the force-equilibrium equation $\Sigma \mathbf{F} = \mathbf{0}$.

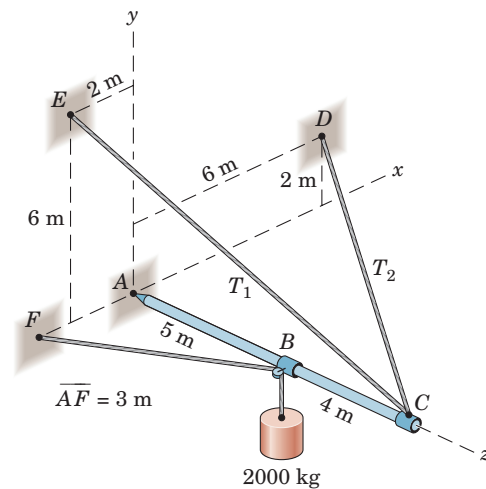
**Problem 3/70****Representative Problems**

- 3/71** One of the vertical walls supporting end B of the 200-kg uniform shaft of Sample Problem 3/5 is turned through a 30° angle as shown here. End A is still supported by the ball-and-socket connection in the horizontal x - y plane. Calculate the magnitudes of the forces \mathbf{P} and \mathbf{R} exerted on the ball end B of the shaft by the vertical walls C and D , respectively.

Ans. $P = 1584 \text{ N}$, $R = 755 \text{ N}$

**Problem 3/71**

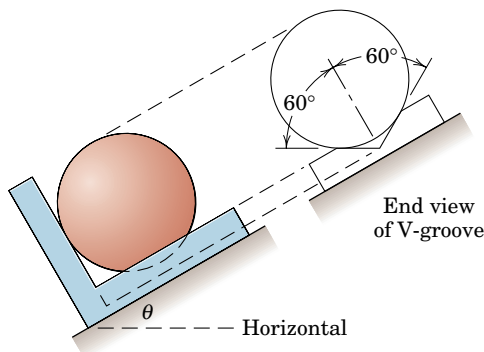
- 3/72** The 9-m steel boom has a mass of 600 kg with center of mass at midlength. It is supported by a ball-and-socket joint at A and the two cables under tensions T_1 and T_2 . The cable which supports the 2000-kg load leads through a sheave (pulley) at B and is secured to the vertical x - y plane at F . Calculate the magnitude of the tension T_1 . (Hint: Write a moment equation which eliminates all unknowns except T_1 .)

**Problem 3/72**

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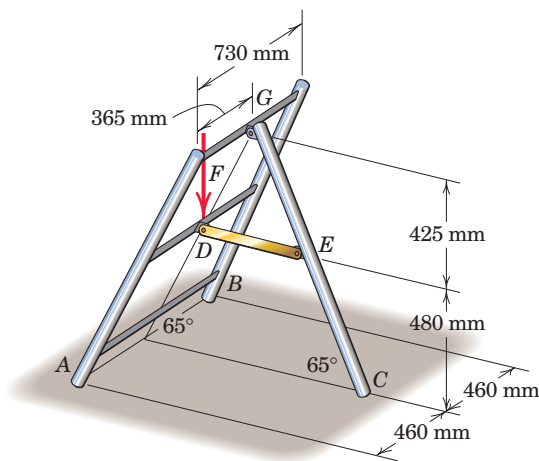
- 3/73** The smooth homogeneous sphere rests in the 120° groove and bears against the end plate, which is normal to the direction of the groove. Determine the angle θ , measured from the horizontal, for which the reaction on each side of the groove equals the force supported by the end plate.

Ans. $\theta = 30^\circ$



Problem 3/73

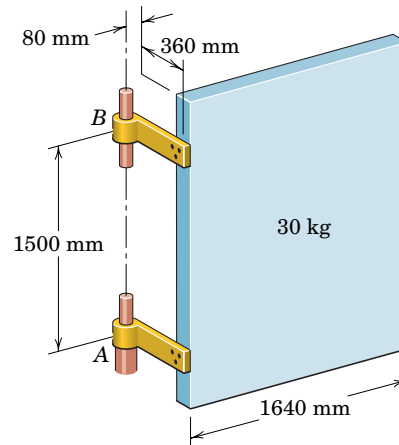
- 3/74** The small tripod-like stepladder is useful for supporting one end of a walking board. If F denotes the magnitude of the downward load from such a board (not shown), determine the reaction at each of the three feet A , B , and C . Neglect friction.



Problem 3/74

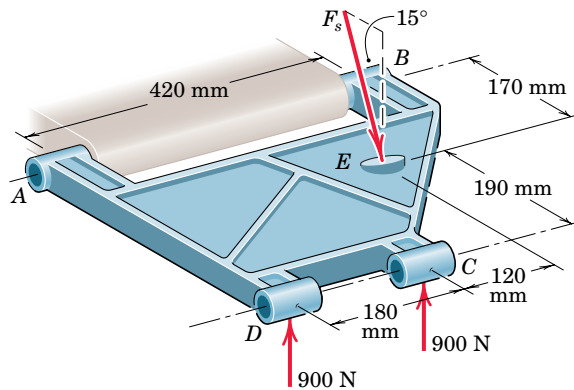
- 3/75** The mass center of the 30-kg door is in the center of the panel. If the weight of the door is supported entirely by the lower hinge A , calculate the magnitude of the total force supported by the hinge at B .

Ans. $B = 190.2 \text{ N}$



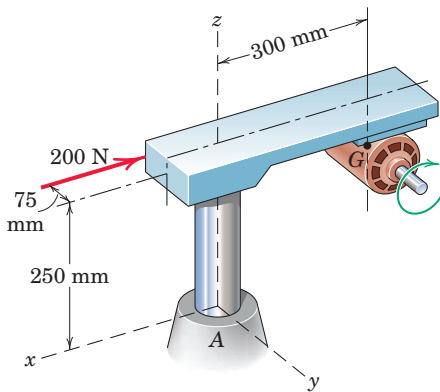
Problem 3/75

- 3/76** As part of a check on its design, a lower A-arm (part of an automobile suspension) is supported by bearings at A and B and subjected to the pair of 900-N forces at C and D . The suspension spring, not shown for clarity, exerts a force F_s at E as shown, where E is in plane $ABCD$. Determine the magnitude F_s of the spring force and the magnitudes F_A and F_B of the bearing forces at A and B which are perpendicular to the hinge axis AB .



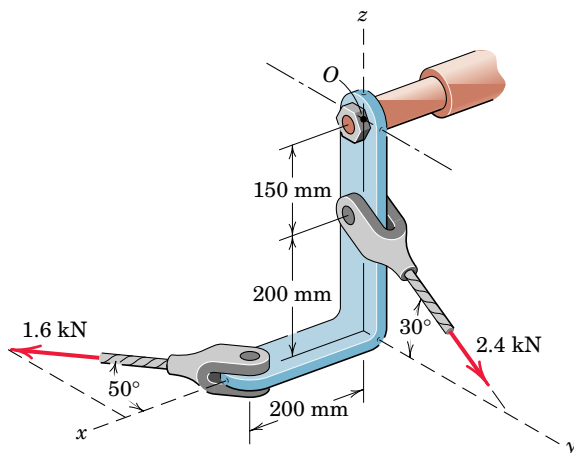
Problem 3/76

- 3/77** The rigid unit of post, bracket, and motor has a mass of 30 kg with its mass center at G located 300 mm from the vertical centerline of the post. The post is welded to the fixed base at A . The motor, which drives a machine through a flexible shaft, turns in the direction indicated and delivers a torque of 200 N·m. In addition, a 200-N force is applied to the bracket as shown. Determine the vector expressions for the total force \mathbf{R} and moment \mathbf{M} applied to the post at A by the supporting base. (*Caution:* Be careful to assign the torque (couple) which acts on the motor shaft in its correct sense consistent with Newton's third law.)
- Ans.* $\mathbf{R} = 200\mathbf{i} + 294\mathbf{k}$ N, $\mathbf{M} = -61.7\mathbf{j} + 15\mathbf{k}$ N·m



Problem 3/77

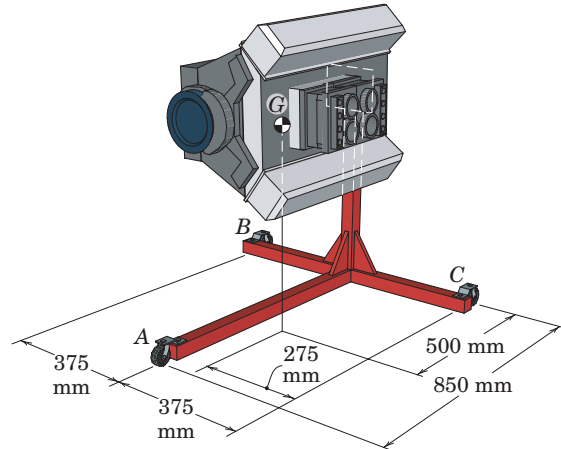
- 3/78** Determine the magnitudes of the force \mathbf{R} and couple \mathbf{M} exerted by the nut and bolt on the loaded bracket at O to maintain equilibrium.



Problem 3/78

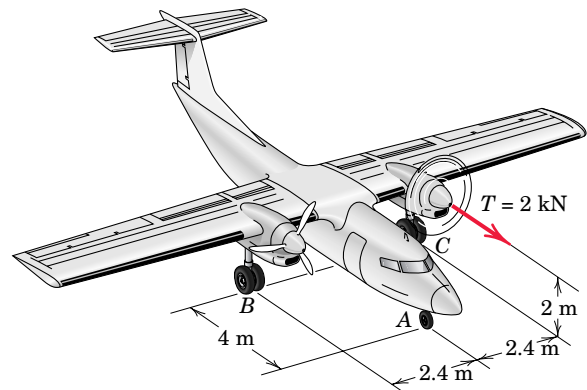
- 3/79** The 220-kg V-8 engine is supported on an engine stand and rotated 90° from its upright position so that its center of gravity G is in the position shown. Determine the vertical reaction at each roller of the stand. Neglect the weight of the stand itself.

Ans. $N_A = 1270$ N, $N_B = 156.6$ N, $N_C = 732$ N



Problem 3/79

- 3/80** During a test, the left engine of the twin-engine airplane is revved up and a 2-kN thrust is generated. The main wheels at B and C are braked in order to prevent motion. Determine the change (compared with the nominal values with both engines off) in the normal reaction forces at A , B , and C .

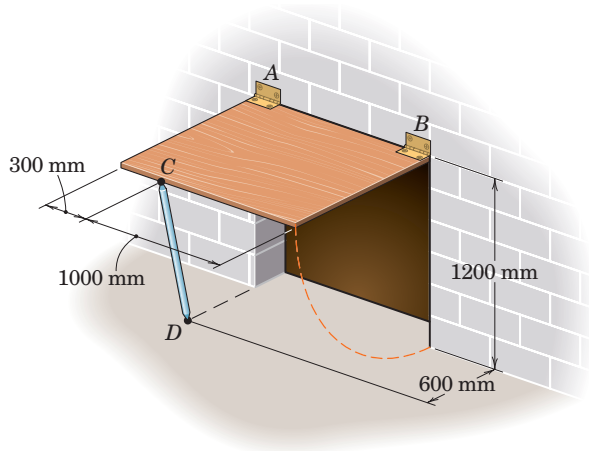


Problem 3/80

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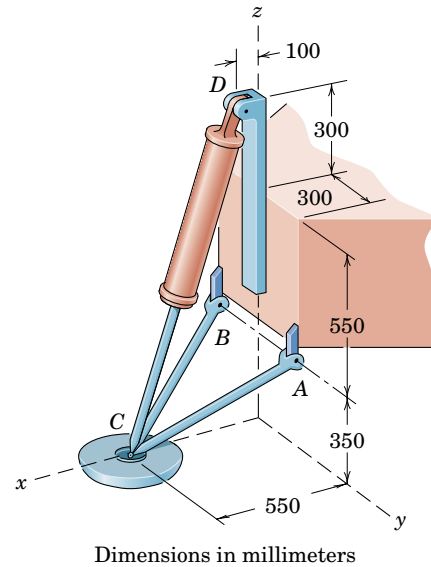
- 3/81** The 25-kg rectangular access door is held in the 90° open position by the single prop CD . Determine the force F in the prop and the magnitude of the force normal to the hinge axis AB in each of the small hinges A and B .

Ans. $F = 140.5 \text{ N}$, $A_n = 80.6 \text{ N}$, $B_n = 95.4 \text{ N}$



Problem 3/81

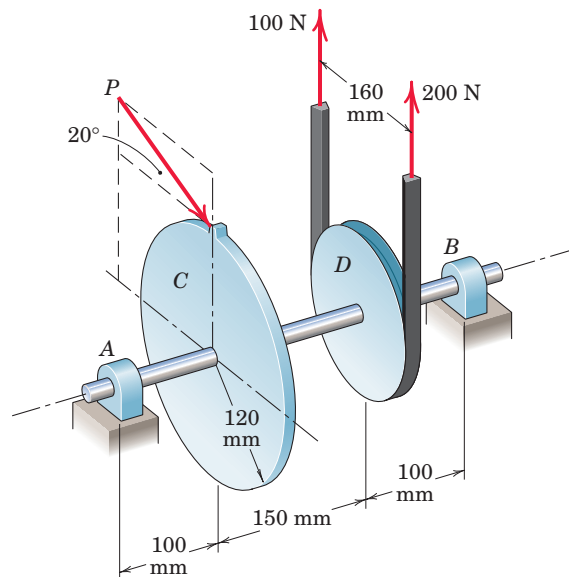
- 3/82** One of the three landing pads for a proposed Mars lander is shown in the figure. As part of a design check on the distribution of force in the landing struts, compute the force in each of the struts AC , BC , and CD when the lander is resting on a horizontal surface on Mars. The arrangement is symmetrical with respect to the x - z plane. The mass of the lander is 600 kg. (Assume equal support by the pads and consult Table D/2 in Appendix D as needed.)



Problem 3/82

- 3/83** Gear C drives the V-belt pulley D at a constant speed. For the belt tensions shown calculate the gear-tooth force P and the magnitudes of the total forces supported by the bearings at A and B .

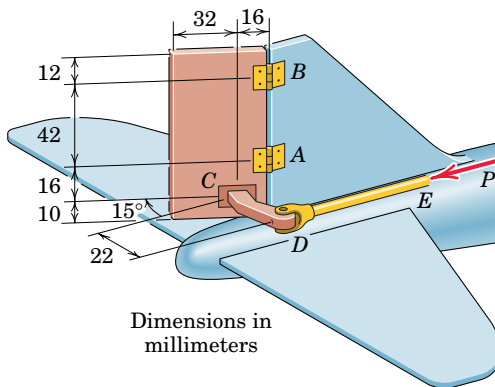
Ans. $P = 70.9 \text{ N}$, $A = 83.3 \text{ N}$, $B = 208 \text{ N}$



Problem 3/83

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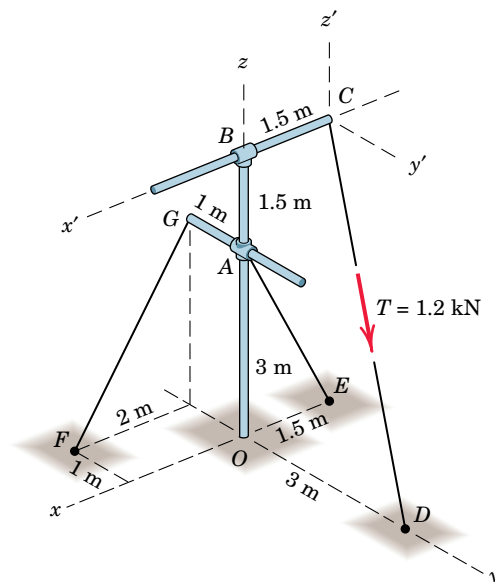
- 3/88** Consider the rudder assembly of a radio-controlled model airplane. For the 15° position shown in the figure, the net pressure acting on the left side of the rectangular rudder area is $p = 4(10^{-5}) \text{ N/mm}^2$. Determine the required force P in the control rod DE and the horizontal components of the reactions at hinges A and B which are parallel to the rudder surface. Assume the aerodynamic pressure to be uniform.



Problem 3/88

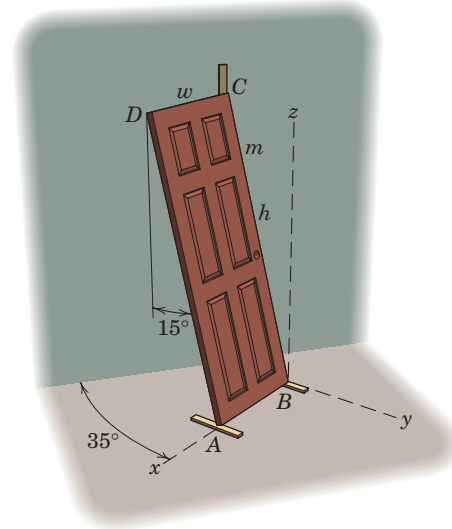
- 3/89** The rigid pole and cross-arms of Prob. 2/105 are shown again here. Determine the tensions T_{AE} and T_{GF} in the two supporting cables resulting from the 1.2-kN tension in cable CD . Assume the absence of any resisting moments on the base of the pole at O about the x - and y -axes, but not about the z -axis.

Ans. $T_{AE} = 4.30 \text{ kN}$, $T_{GF} = 3.47 \text{ kN}$



Problem 3/89

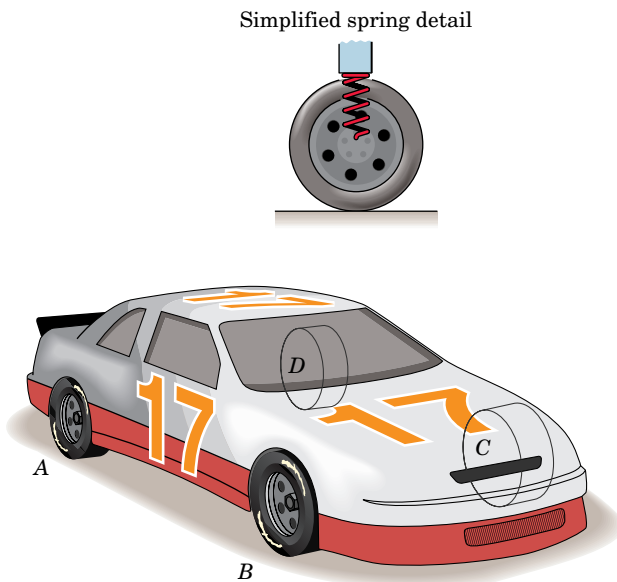
- 3/90** A homogeneous door of mass m , height h , and width w is leaned against a wall for painting. Small wooden strips are placed beneath corners A , B , and C . There is negligible friction at C , but friction at A and B is sufficient to prevent slipping. Determine the y - and z -components of the force reactions at A and B and the force normal to the wall at C .



Problem 3/90

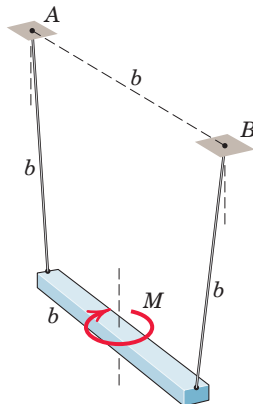
- 3/91** The upper ends of the vertical coil springs in the stock racecar can be moved up and down by means of a screw mechanism not shown. This adjustment permits a change in the downward force at each wheel as an optimum handling setup is sought. Initially, scales indicate the normal forces to be 3600 N, 3600 N, 4500 N, and 4500 N at A , B , C , and D , respectively. If the top of the right rear spring at A is lowered so that the scale at A reads an additional 450 N, determine the corresponding changes in the normal forces at B , C , and D . Neglect the effects of the small attitude changes (pitch and roll angles) caused by the spring adjustment. The front wheels are the same distance apart as the rear wheels.

Ans. $\Delta N_B = -450 \text{ N}$, $\Delta N_C = 450 \text{ N}$
 $\Delta N_D = -450 \text{ N}$



Problem 3/91

- 3/92** A uniform bar of length b and mass m is suspended at its ends by two wires, each of length b , from points A and B in the horizontal plane a distance b apart. A couple M is applied to the bar causing it to rotate about a vertical axis to the equilibrium position shown. Derive an expression for the height h which it rises from its original equilibrium position where it hangs freely with no applied moment. What value of M is required to raise the bar the maximum amount b ?

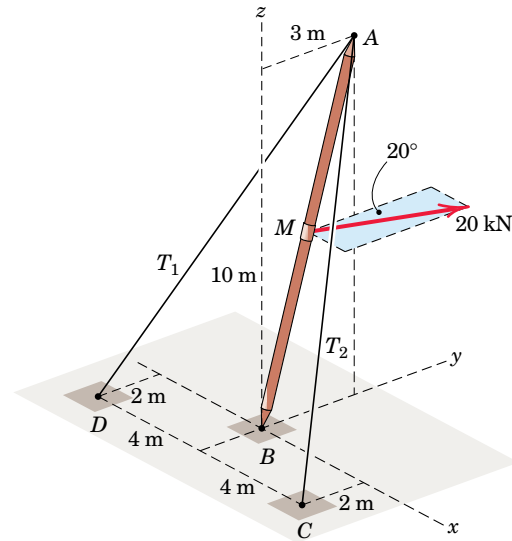


Problem 3/92

- 3/93** The boom AB lies in the vertical y - z plane and is supported by a ball-and-socket joint at B and by the two cables at A . Calculate the tension in each cable resulting from the 20-kN force acting in the horizontal

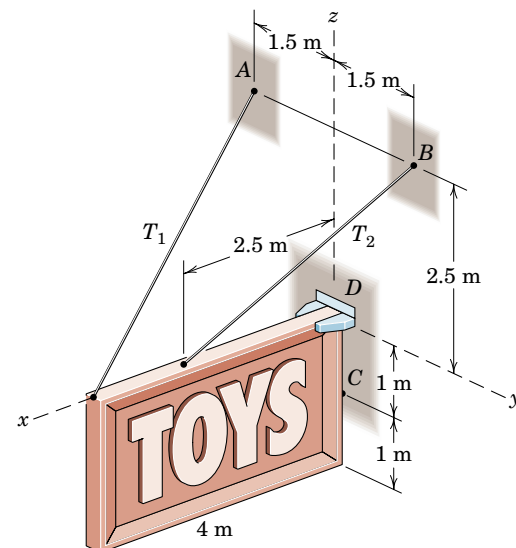
plane and applied at the midpoint M of the boom. Neglect the weight of the boom.

Ans. $T_1 = 33.0$ kN, $T_2 = 22.8$ kN



Problem 3/93

- 3/94** A rectangular sign over a store has a mass of 100 kg, with the center of mass in the center of the rectangle. The support against the wall at point C may be treated as a ball-and-socket joint. At corner D support is provided in the y -direction only. Calculate the tensions T_1 and T_2 in the supporting wires, the total force supported at C , and the lateral force R supported at D .

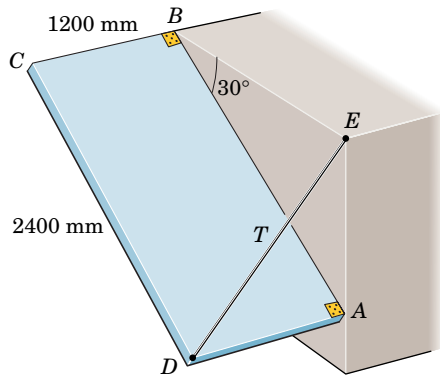


Problem 3/94

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- 3/95 The uniform rectangular panel $ABCD$ has a mass of 40 kg and is hinged at its corners A and B to the fixed vertical surface. A wire from E to D keeps edges BC and AD horizontal. Hinge A can support thrust along the hinge axis AB , whereas hinge B supports force normal to the hinge axis only. Find the tension T in the wire and the magnitude B of the force supported by hinge B .

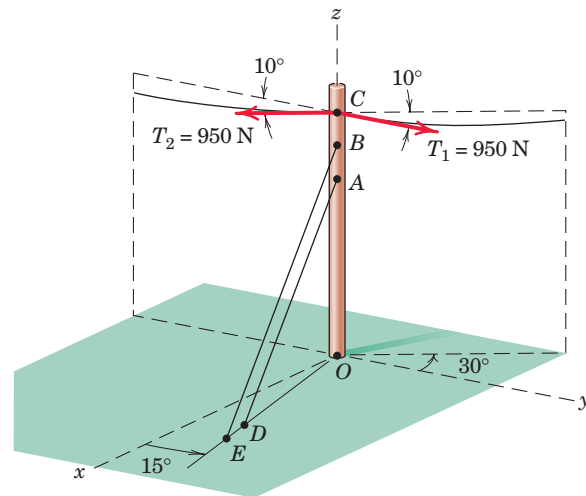
Ans. $T = 277 \text{ N}$, $B = 169.9 \text{ N}$



Problem 3/95

- 3/96 The vertical plane containing the utility cable turns 30° at the vertical pole OC . The tensions T_1 and T_2 are both 950 N. In order to prevent long-term leaning of the pole, guy wires AD and BE are utilized. If the two guy wires are adjusted so as to carry equal tensions T which together reduce the moment at O to zero, determine the net horizontal reaction at O . Determine the required value of T . Neglect the weight of the pole.

Ans. $O = 144.9 \text{ N}$, $T = 471 \text{ N}$



$$\begin{aligned}\overline{OA} &= 9 \text{ m} & \overline{OD} &= 8 \text{ m} \\ \overline{OB} &= 11 \text{ m} & \overline{OE} &= 10 \text{ m} \\ \overline{OC} &= 13 \text{ m}\end{aligned}$$

Problem 3/96

3/5 CHAPTER REVIEW

In Chapter 3 we have applied our knowledge of the properties of forces, moments, and couples studied in Chapter 2 to solve problems involving rigid bodies in equilibrium. Complete equilibrium of a body requires that the vector resultant of all forces acting on it be zero ($\Sigma \mathbf{F} = \mathbf{0}$) and the vector resultant of all moments on the body about a point (or axis) also be zero ($\Sigma \mathbf{M} = \mathbf{0}$). We are guided in all of our solutions by these two requirements, which are easily comprehended physically.

Frequently, it is not the theory but its application which presents difficulty. The crucial steps in applying our principles of equilibrium should be quite familiar by now. They are:

1. Make an unequivocal decision as to which system (a body or collection of bodies) in equilibrium is to be analyzed.
2. Isolate the system in question from all contacting bodies by drawing its *free-body diagram* showing *all* forces and couples acting *on* the isolated system from external sources.
3. Observe the principle of action and reaction (Newton's third law) when assigning the sense of each force.
4. Label all forces and couples, known and unknown.
5. Choose and label reference axes, always choosing a right-handed set when vector notation is used (which is usually the case for three-dimensional analysis).
6. Check the adequacy of the constraints (supports) and match the number of unknowns with the number of available independent equations of equilibrium.

When solving an equilibrium problem, we should first check to see that the body is statically determinate. If there are more supports than are necessary to hold the body in place, the body is statically indeterminate, and the equations of equilibrium by themselves will not enable us to solve for all of the external reactions. In applying the equations of equilibrium, we choose scalar algebra, vector algebra, or graphical analysis according to both preference and experience; vector algebra is particularly useful for many three-dimensional problems.

The algebra of a solution can be simplified by the choice of a moment axis which eliminates as many unknowns as possible or by the choice of a direction for a force summation which avoids reference to certain unknowns. A few moments of thought to take advantage of these simplifications can save appreciable time and effort.

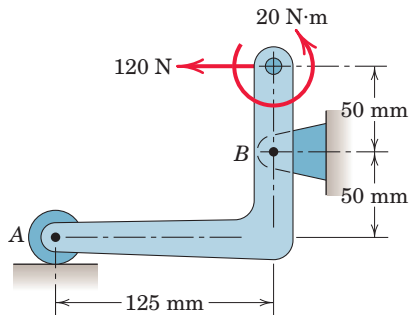
The principles and methods covered in Chapter 2 and 3 constitute the most basic part of statics. They lay the foundation for what follows not only in statics but in dynamics as well.

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REVIEW PROBLEMS

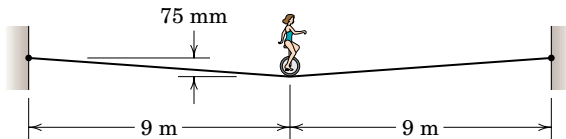
- 3/97** Calculate the magnitude of the force supported by the pin at B for the bell crank loaded and supported as shown.

Ans. $B = 240 \text{ N}$



Problem 3/97

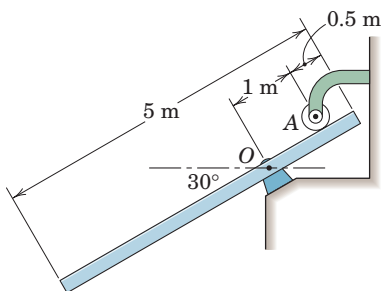
- 3/98** A 50-kg acrobat pedals her unicycle across the taut but slightly elastic cable. If the deflection at the center of the 18-m span is 75 mm, determine the tension in the cable. Neglect the effects of the weights of the cable and unicycle.



Problem 3/98

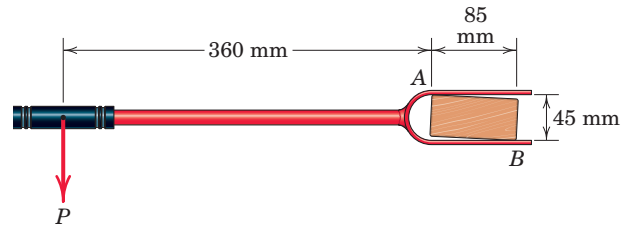
- 3/99** The uniform 5-m bar with a mass of 100 kg is hinged at O and prevented from rotating in the vertical plane beyond the 30° position by the fixed roller at A . Calculate the magnitude of the total force supported by the pin at O .

Ans. $O = 1769 \text{ N}$



Problem 3/99

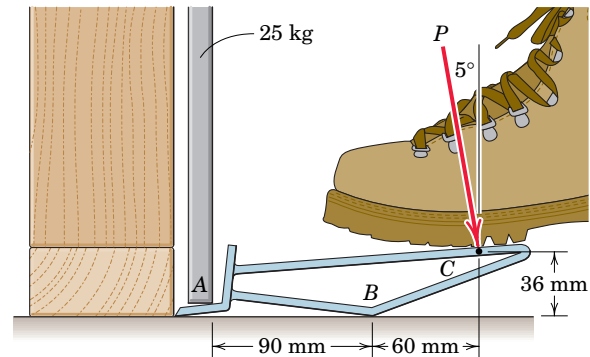
- 3/100** The tool shown is used for straightening twisted members as wooden framing is completed. If the force $P = 150 \text{ N}$ is applied to the handle as shown, determine the normal forces applied to the installed stud at points A and B . Ignore friction.



Problem 3/100

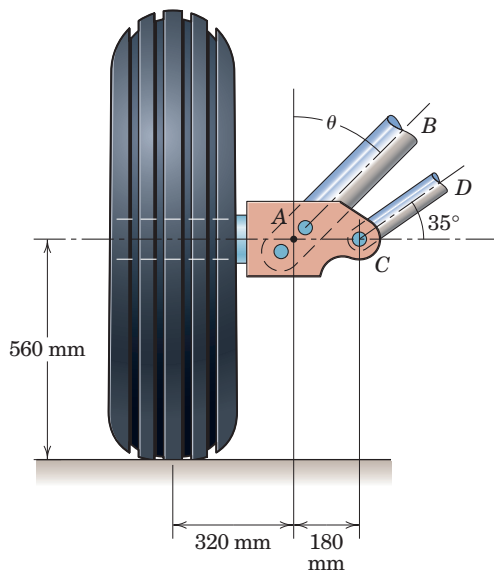
- 3/101** The device shown in the figure is useful for lifting drywall panels into position prior to fastening to the stud wall. Estimate the magnitude P of the force required to lift the 25-kg panel. State any assumptions.

Ans. $P = 351 \text{ N}$



Problem 3/101

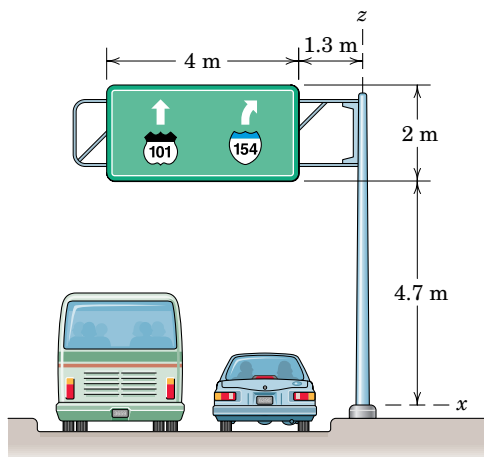
- 3/102** The designers of an aircraft landing-gear system wish to cause the forces in both struts AB and CD to act along their respective lengths. What angle θ should they specify for strut AB ? The weights of all members are small compared with the forces which act on the system shown. Treat as two-dimensional.



Problem 3/102

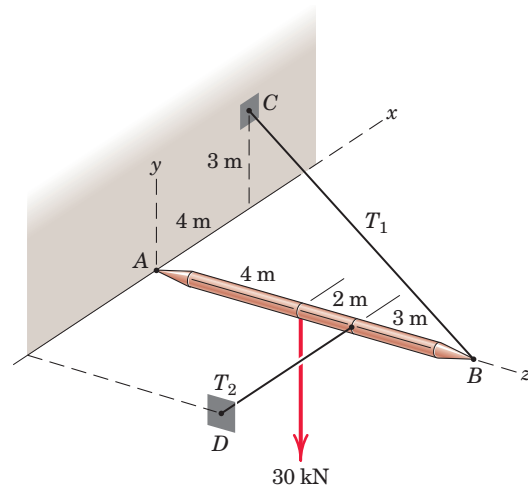
3/103 A freeway sign measuring 4 m by 2 m is supported by the single mast as shown. The sign, supporting framework, and mast together have a mass of 300 kg, with center of mass 3.3 m away from the vertical centerline of the mast. When the sign is subjected to the direct blast of a 125-km/h wind, an average pressure difference of 700 Pa is developed between the front and back sides of the sign, with the resultant of the wind-pressure forces acting at the center of the sign. Determine the magnitudes of the force and moment reactions at the base of the mast. Such results would be instrumental in the design of the base.

Ans. $R = 6330 \text{ N}$, $M = 38.1 \text{ kN}\cdot\text{m}$



Problem 3/103

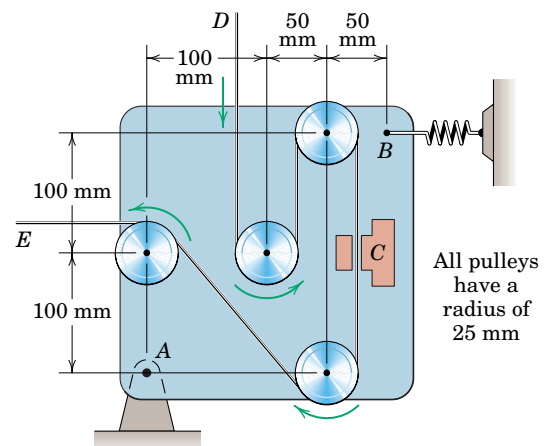
3/104 If the weight of the boom is negligible compared with the applied 30-kN load, determine the cable tensions T_1 and T_2 and the force acting at the ball joint at A.



Problem 3/104

3/105 Magnetic tape under a tension of 10 N at D passes around the guide pulleys and through the erasing head at C at constant speed. As a result of a small amount of friction in the bearings of the pulleys, the tape at E is under a tension of 11 N. Determine the tension T in the supporting spring at B. The plate lies in a horizontal plane and is mounted on a precision needle bearing at A.

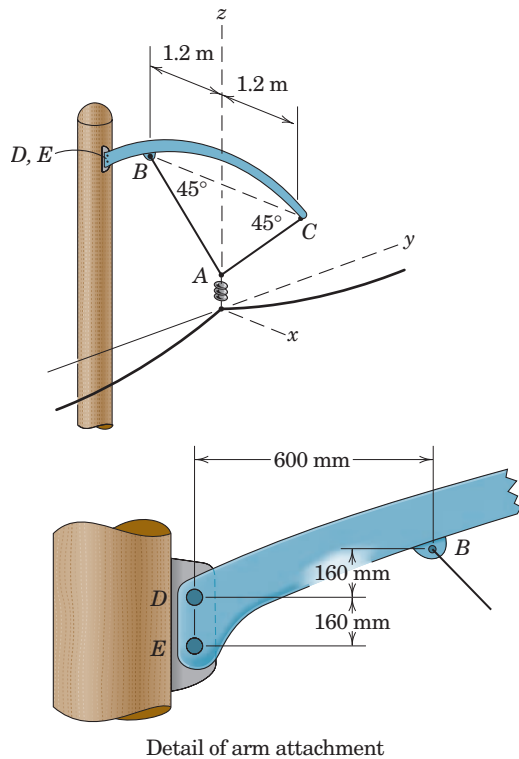
Ans. $T = 10.62 \text{ N}$



Problem 3/105

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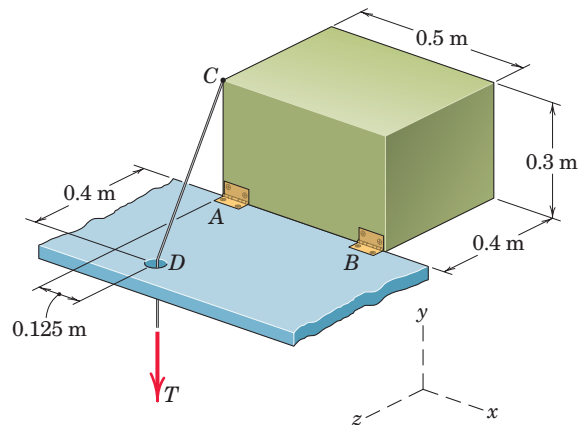
3/106 The curved arm BC and attached cables AB and AC support a power line which lies in the vertical y - z plane. The tangents to the power line at the insulator below A make 15° angles with the horizontal y -axis. If the tension in the power line at the insulator is 1.3 kN, calculate the total force supported by the bolt at D on the pole bracket. The weight of the arm BC can be neglected compared with the other forces, and it can be assumed that the bolt at E supports horizontal force only.



Problem 3/106

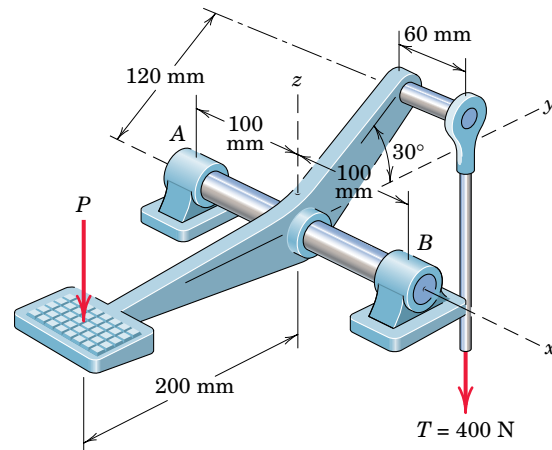
3/107 Determine the tension T required to hold the rectangular solid in the position shown. The 125 -kg rectangular solid is homogeneous. Friction at D is negligible.

Ans. $T = 1053$ N



Problem 3/107

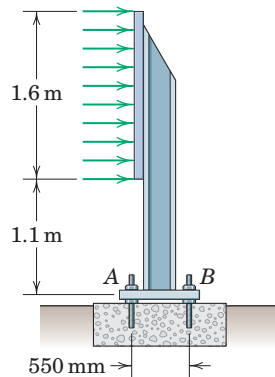
3/108 A vertical force P on the foot pedal of the bell crank is required to produce a tension T of 400 N in the vertical control rod. Determine the corresponding bearing reactions at A and B .



Problem 3/108

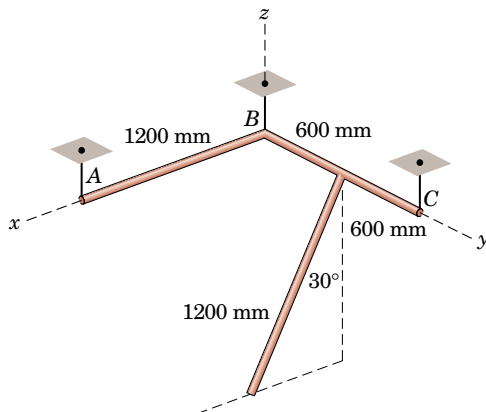
- 3/109** The wind blowing normal to the plane of the rectangular sign exerts a uniform pressure of 175 N/m^2 as indicated in the figure. Determine the changes in the forces exerted at A and B by each support. There are two symmetrically placed I-beam up-rights, and the width of the sign is 3 m .

Ans. $\Delta A = 1451 \text{ N down}$, $\Delta B = 1451 \text{ N up}$



Problem 3/109

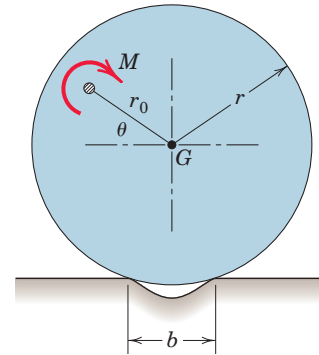
- 3/110** Each of the three uniform 1200-mm bars has a mass of 20 kg . The bars are welded together into the configuration shown and suspended by three vertical wires. Bars AB and BC lie in the horizontal x - y plane, and the third bar lies in a plane parallel to the x - z plane. Compute the tension in each wire.



Problem 3/110

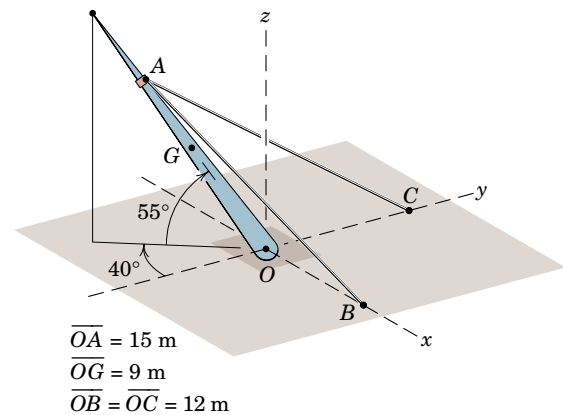
- 3/111** A wheel of mass m and radius r with its mass center G at the geometric center rests in a small depression of width b . Determine the minimum couple M applied to the wheel at the location shown in order to roll the wheel out of the depression. Assume no slipping occurs. What is the influence of r_0 , r , and θ ?

Ans. $M = \frac{1}{2}mgb$



Problem 3/111

- 3/112** The 400-kg boom with center of mass at G is held in the position shown by a ball-and-socket joint at O and the two cables AB and AC . Determine the two cable tensions and the x -, y -, and z -components of the force reaction at O .



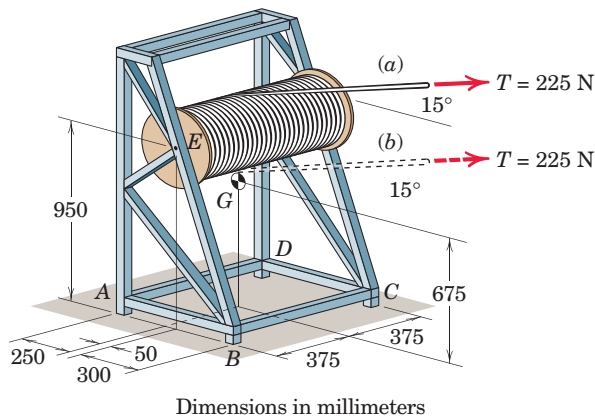
Problem 3/112

168 Chapter 3 Equilibrium

3/113 The stand is used for storing and dispensing cable. The combined mass of the stand, reel, and cable is 150 kg with center of mass at G . Determine the normal forces at all four feet (A , B , C , and D) if (a) the cable comes off at the top of the reel and (b) the cable comes off at the bottom of the reel. Note that the cable departs the reel at its horizontal midpoint in both cases. The effective radius of the reel is 150 mm.

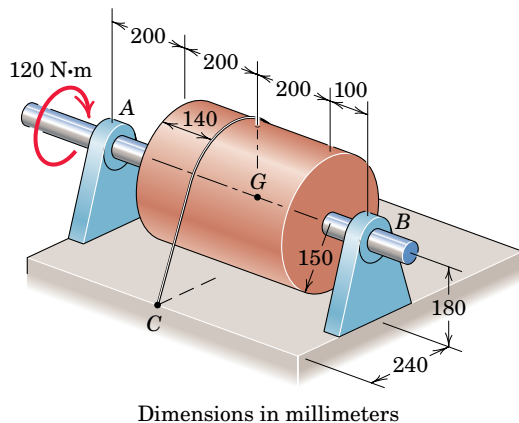
Ans. (a) $N_A = N_D = 214 \text{ N}$, $N_B = N_C = 492 \text{ N}$

(b) $N_A = N_D = 271 \text{ N}$, $N_B = N_C = 436 \text{ N}$



Problem 3/113

3/114 The drum and shaft are welded together and have a mass of 50 kg with mass center at G . The shaft is subjected to a torque (couple) of $120 \text{ N}\cdot\text{m}$, and the drum is prevented from rotating by the cord wrapped securely around it and attached to point C . Calculate the magnitudes of the forces supported by bearings A and B .



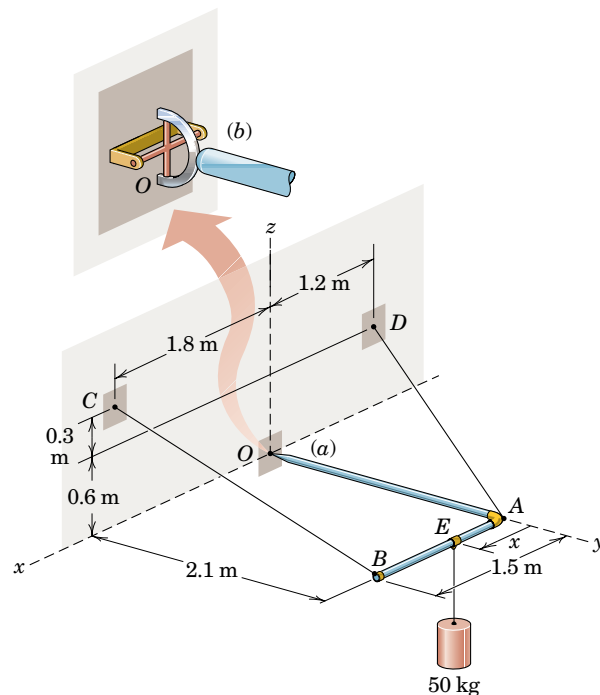
Problem 3/114

3/115 The L-shaped bar is supported by a ball-and-socket joint at O [case (a)] and the two cables as shown. Explain why this configuration is improperly constrained. (b) The ball-and-socket joint is now replaced by the universal joint which can support, in addition to three force reactions, a moment about the y -axis but no moments about the x - and z -axes. Plot the two cable tensions, the magnitude of the force reaction at O , and the moment reaction at O as functions of the position x of the 50-kg cylinder over the range $0.15 \leq x \leq 1.35 \text{ m}$. Explain any unusual characteristics of these plots. Neglect the weight of the bar throughout.

Ans. $T_{AD} = 1019 \text{ N}$, $T_{BC} = 628 \text{ N}$

$O = 1488 \text{ N}$, all constant

$M_{O_y} = -490x + 368 \text{ N}\cdot\text{m}$

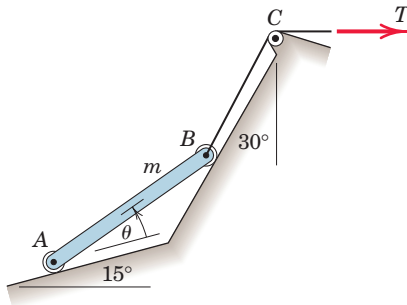


Problem 3/115



*Computer-Oriented Problems

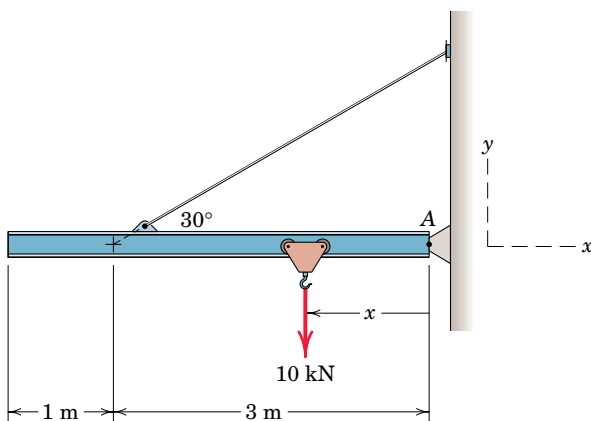
- *3/116** Determine and plot the tension ratio T/mg required to hold the uniform slender bar in equilibrium for any angle θ from just above zero to just under 45° . The bar AB of mass m is uniform.



Problem 3/116

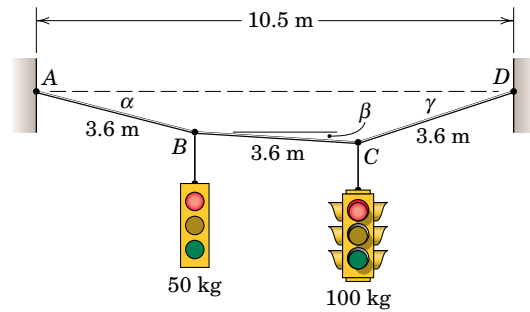
- *3/117** The jib crane is designed for a maximum capacity of 10 kN, and its uniform I-beam has a mass of 200 kg. (a) Plot the magnitude R of the force on the pin at A as a function of x through its operating range of $x = 0.2$ m to $x = 3.8$ m. On the same set of axes, plot the x - and y -components of the pin reaction at A. (b) Determine the minimum value of R and the corresponding value of x . (c) For what value of R should the pin at A be designed? (Use $g = 10 \text{ m/s}^2$.)

Ans. (a) $R = \frac{1}{3} \{400x^2 - 400x + 1072\}^{1/2}$
 (b) $R_{\min} = 10.39 \text{ kN}$ at $x = 0.5 \text{ m}$
 (c) $R_{\max} = 24.3 \text{ kN}$ at $x = 3.8 \text{ m}$



Problem 3/117

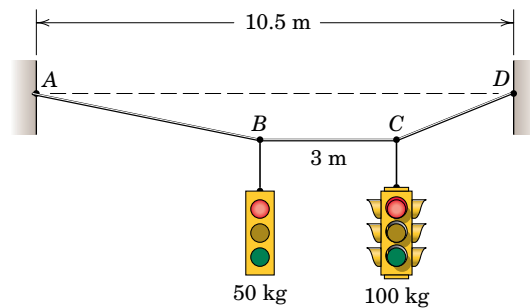
- *3/118** Two traffic signals are attached to the 10.8-m support cable at equal intervals as shown. Determine the equilibrium configuration angles α , β , and γ , as well as the tension in each cable segment.



Problem 3/118

- *3/119** The two traffic signals of Prob. 3/118 are now repositioned so that segment BC of the 10.8-m support cable is 3 m in length and is horizontal. Specify the necessary lengths AB and CD and the tensions in all three cable segments.

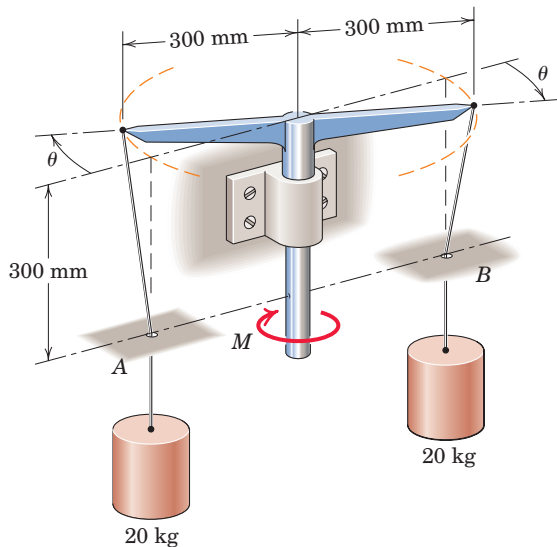
Ans. $\overline{AB} = 5.10 \text{ m}$, $\overline{CD} = 2.70 \text{ m}$
 $T_{AB} = 2470 \text{ N}$, $T_{BC} = 2420 \text{ N}$, $T_{CD} = 2610 \text{ N}$



Problem 3/119

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- *3/120** The vertical position of the two 20-kg cylinders is controlled by the torque M applied to the central shaft. The cords attached to the ends of the arms pass through smooth holes in a fixed surface at A and B . Plot M as a function of θ from $\theta = 0^\circ$ to $\theta = 180^\circ$. Determine the maximum value of M and the corresponding value of θ . Take $g = 10 \text{ m/s}^2$ for your calculations.

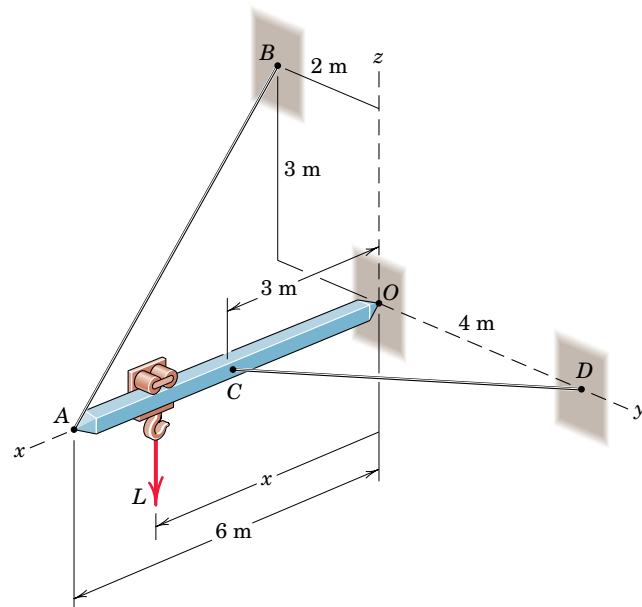


Problem 3/120

- *3/121** The horizontal boom is supported by the cables AB and CD and by a ball-and-socket joint at O . To determine the influence on the reaction at O of the position of the vertical load L along the boom, we may neglect the weight of the boom. If R represents the magnitude of the total force at O , determine by calculus the minimum ratio R/L and the corresponding value of x . Then write a computer program for R/L and plot the results for $0 < x < 6 \text{ m}$ as a check on your calculations.

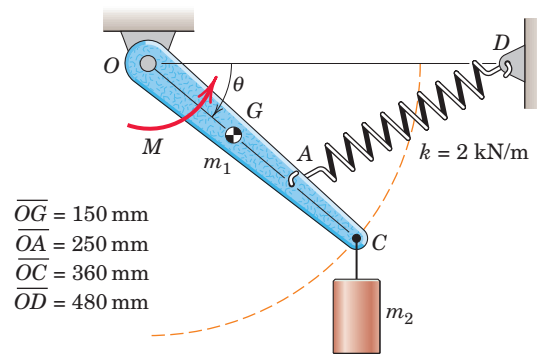
$$\text{Ans. } R/L = \sqrt{47x^2/162 - x/3 + 1}$$

$$(R/L)_{\min} = 0.951 \text{ at } x = 0.574 \text{ m}$$



Problem 3/121

- *3/122** The mass center of the 10-kg arm OC is located at G , and the spring of constant $k = 1.2 \text{ kN/m}$ is unstretched when $\theta = 0$. Plot the applied moment M required for static equilibrium over the range $0 \leq \theta \leq 180^\circ$. Determine the value of θ for which $M = 0$ (if any) and the minimum and maximum values of M along with the corresponding values of θ at which these extremes occur. The motion of this mechanism occurs in a vertical plane. Take M to be positive when counterclockwise. The value of m_2 is 3 kg.



Problem 3/122



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This view from Lisbon, Portugal, shows a variety of structures. In all cases, however, the engineers had to calculate the force supported by each major component of the overall structure.

4

STRUCTURES

CHAPTER OUTLINE

- 4/1 Introduction
- 4/2 Plane Trusses
- 4/3 Method of Joints
- 4/4 Method of Sections
- 4/5 Space Trusses
- 4/6 Frames and Machines
- 4/7 Chapter Review

4/1 INTRODUCTION

In Chapter 3 we studied the equilibrium of a single rigid body or a system of connected members treated as a single rigid body. We first drew a free-body diagram of the body showing all forces external to the isolated body and then we applied the force and moment equations of equilibrium. In Chapter 4 we focus on the determination of the forces internal to a structure, that is, forces of action and reaction between the connected members. An engineering structure is any connected system of members built to support or transfer forces and to safely withstand the loads applied to it. To determine the forces internal to an engineering structure, we must dismember the structure and analyze separate free-body diagrams of individual members or combinations of members. This analysis requires careful application of Newton's third law, which states that each action is accompanied by an equal and opposite reaction.

In Chapter 4 we analyze the internal forces acting in several types of structures, namely, trusses, frames, and machines. In this treatment we consider only *statically determinate* structures, which do not have more supporting constraints than are necessary to maintain an equilibrium configuration. Thus, as we have already seen, the equations of equilibrium are adequate to determine all unknown reactions.

The analysis of trusses, frames and machines, and beams under concentrated loads constitutes a straightforward application of the material developed in the previous two chapters. The basic procedure developed in Chapter 3 for isolating a body by constructing a correct free-body diagram is essential for the analysis of statically determinate structures.

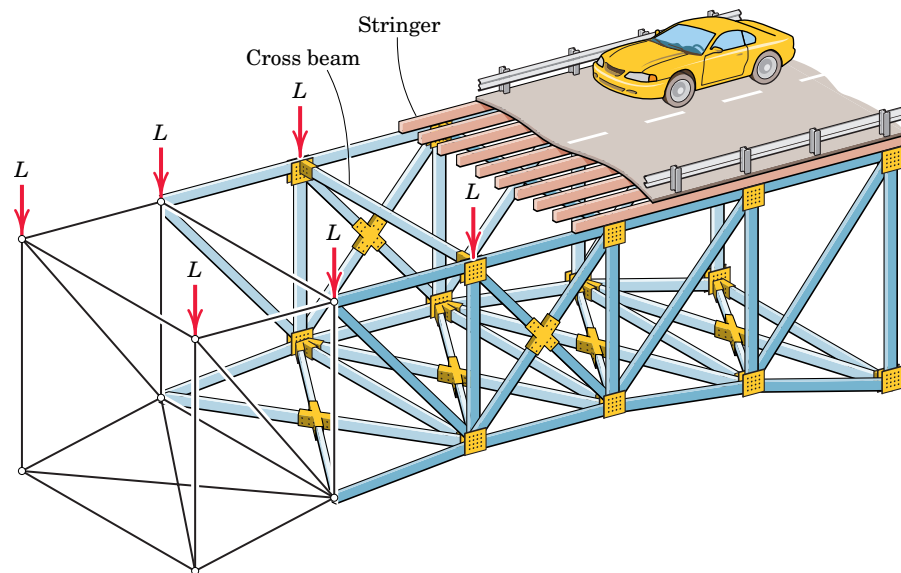


Figure 4/1

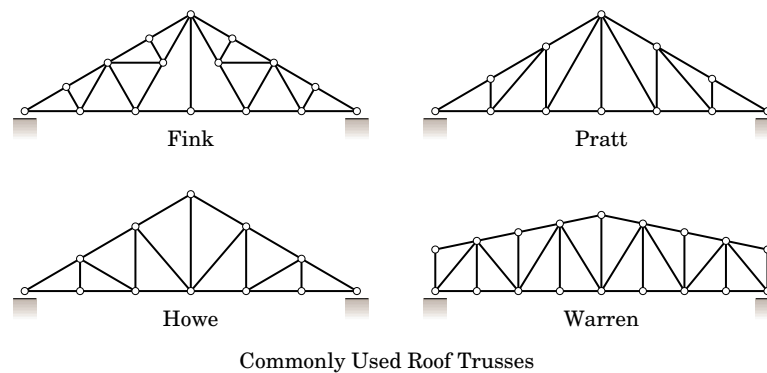
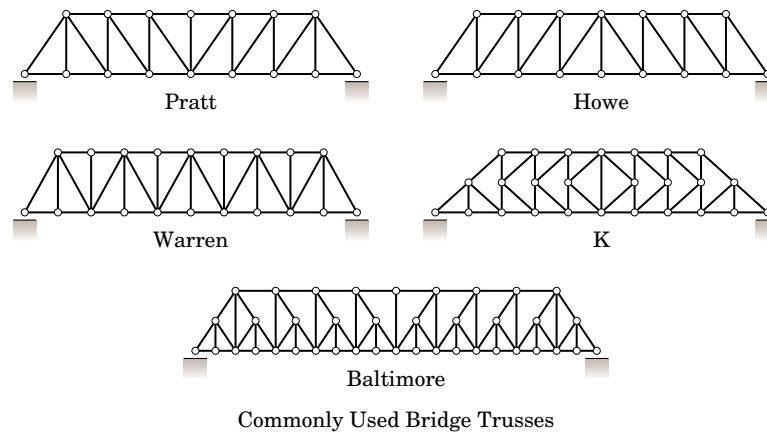


Figure 4/2

4/2 PLANE TRUSSES

A framework composed of members joined at their ends to form a rigid structure is called a *truss*. Bridges, roof supports, derricks, and other such structures are common examples of trusses. Structural members commonly used are I-beams, channels, angles, bars, and special shapes which are fastened together at their ends by welding, riveted connections, or large bolts or pins. When the members of the truss lie essentially in a single plane, the truss is called a *plane truss*.

For bridges and similar structures, plane trusses are commonly utilized in pairs with one truss assembly placed on each side of the structure. A section of a typical bridge structure is shown in Fig. 4/1. The combined weight of the roadway and vehicles is transferred to the longitudinal stringers, then to the cross beams, and finally, with the weights of the stringers and cross beams accounted for, to the upper joints of the two plane trusses which form the vertical sides of the structure. A simplified model of the truss structure is indicated at the left side of the illustration; the forces L represent the joint loadings.

Several examples of commonly used trusses which can be analyzed as plane trusses are shown in Fig. 4/2.

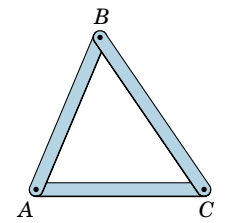
Simple Trusses

The basic element of a plane truss is the triangle. Three bars joined by pins at their ends, Fig. 4/3a, constitute a rigid frame. The term *rigid* is used to mean noncollapsible and also to mean that deformation of the members due to induced internal strains is negligible. On the other hand, four or more bars pin-jointed to form a polygon of as many sides constitute a nonrigid frame. We can make the nonrigid frame in Fig. 4/3b rigid, or stable, by adding a diagonal bar joining A and D or B and C and thereby forming two triangles. We can extend the structure by adding additional units of two end-connected bars, such as DE and CE or AF and DF , Fig. 4/3c, which are pinned to two fixed joints. In this way the entire structure will remain rigid.

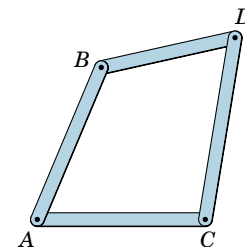
Structures built from a basic triangle in the manner described are known as *simple trusses*. When more members are present than are needed to prevent collapse, the truss is statically indeterminate. A statically indeterminate truss cannot be analyzed by the equations of equilibrium alone. Additional members or supports which are not necessary for maintaining the equilibrium configuration are called *redundant*.

To design a truss we must first determine the forces in the various members and then select appropriate sizes and structural shapes to withstand the forces. Several assumptions are made in the force analysis of simple trusses. First, we assume all members to be *two-force members*. A two-force member is one in equilibrium under the action of two forces only, as defined in general terms with Fig. 3/4 in Art. 3/3. Each member of a truss is normally a straight link joining the two points of application of force. The two forces are applied at the ends of the member and are necessarily equal, opposite, and *collinear* for equilibrium.

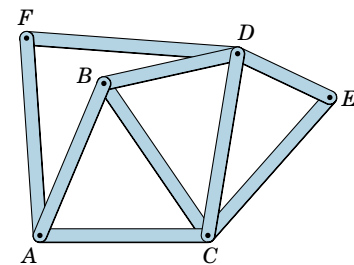
The member may be in tension or compression, as shown in Fig. 4/4. When we represent the equilibrium of a portion of a two-force member, the tension T or compression C acting on the cut section is the same



(a)

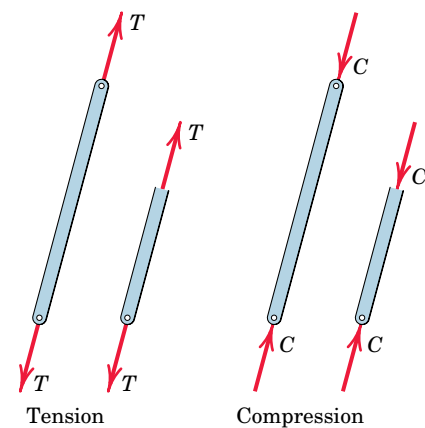


(b)



(c)

Figure 4/3



Two-Force Members

Figure 4/4

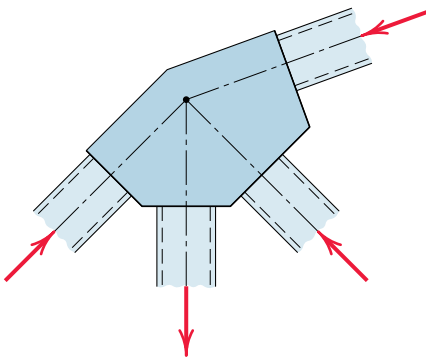
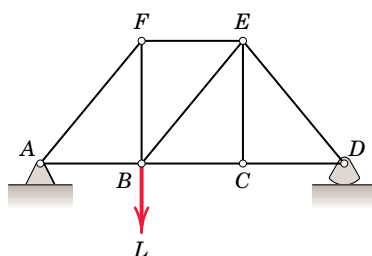
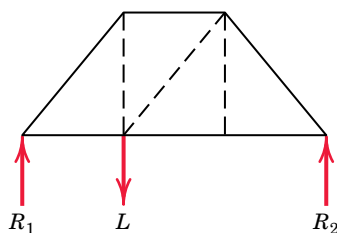


Figure 4/5



(a)



(b)

Figure 4/6

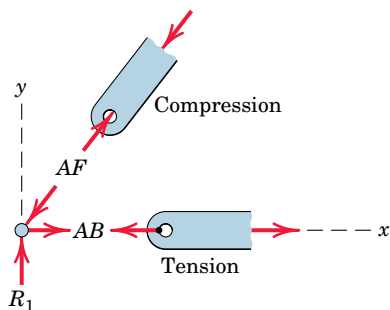


Figure 4/7

for all sections. We assume here that the weight of the member is small compared with the force it supports. If it is not, or if we must account for the small effect of the weight, we can replace the weight W of the member by two forces, each $W/2$ if the member is uniform, with one force acting at each end of the member. These forces, in effect, are treated as loads externally applied to the pin connections. Accounting for the weight of a member in this way gives the correct result for the average tension or compression along the member but will not account for the effect of bending of the member.

Truss Connections and Supports

When welded or riveted connections are used to join structural members, we may usually assume that the connection is a pin joint if the centerlines of the members are concurrent at the joint as in Fig. 4/5.

We also assume in the analysis of simple trusses that all external forces are applied at the pin connections. This condition is satisfied in most trusses. In bridge trusses the deck is usually laid on cross beams which are supported at the joints, as shown in Fig. 4/1.

For large trusses, a roller, rocker, or some kind of slip joint is used at one of the supports to provide for expansion and contraction due to temperature changes and for deformation from applied loads. Trusses and frames in which no such provision is made are statically indeterminate, as explained in Art. 3/3. Figure 3/1 shows examples of such joints.

Two methods for the force analysis of simple trusses will be given. Each method will be explained for the simple truss shown in Fig. 4/6a. The free-body diagram of the truss as a whole is shown in Fig. 4/6b. The external reactions are usually determined first, by applying the equilibrium equations to the truss as a whole. Then the force analysis of the remainder of the truss is performed.

4/3 METHOD OF JOINTS

This method for finding the forces in the members of a truss consists of satisfying the conditions of equilibrium for the forces acting on the connecting pin of each joint. The method therefore deals with the equilibrium of concurrent forces, and only two independent equilibrium equations are involved.

We begin the analysis with any joint where at least one known load exists and where not more than two unknown forces are present. The solution may be started with the pin at the left end. Its free-body diagram is shown in Fig. 4/7. With the joints indicated by letters, we usually designate the force in each member by the two letters defining the ends of the member. The proper directions of the forces should be evident by inspection for this simple case. The free-body diagrams of portions of members AF and AB are also shown to clearly indicate the mechanism of the action and reaction. The member AB actually makes contact on the left side of the pin, although the force AB is drawn from the right side and is shown acting away from the pin. Thus, if we consistently draw the force arrows on the *same* side of the pin as the member, then tension (such as AB) will always be indicated by an arrow *away*

from the pin, and compression (such as AF) will always be indicated by an arrow *toward* the pin. The magnitude of AF is obtained from the equation $\Sigma F_y = 0$ and AB is then found from $\Sigma F_x = 0$.

Joint F may be analyzed next, since it now contains only two unknowns, EF and BF . Proceeding to the next joint having no more than two unknowns, we subsequently analyze joints B , C , E , and D in that order. Figure 4/8 shows the free-body diagram of each joint and its corresponding force polygon, which represents graphically the two equilibrium conditions $\Sigma F_x = 0$ and $\Sigma F_y = 0$. The numbers indicate the order in which the joints are analyzed. We note that, when joint D is finally reached, the computed reaction R_2 must be in equilibrium with the forces in members CD and ED , which were determined previously from the two neighboring joints. This requirement provides a check on the correctness of our work. Note that isolation of joint C shows that the force in CE is zero when the equation $\Sigma F_y = 0$ is applied. The force in



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This New York City bridge structure suggests that members of a simple truss need not be straight.

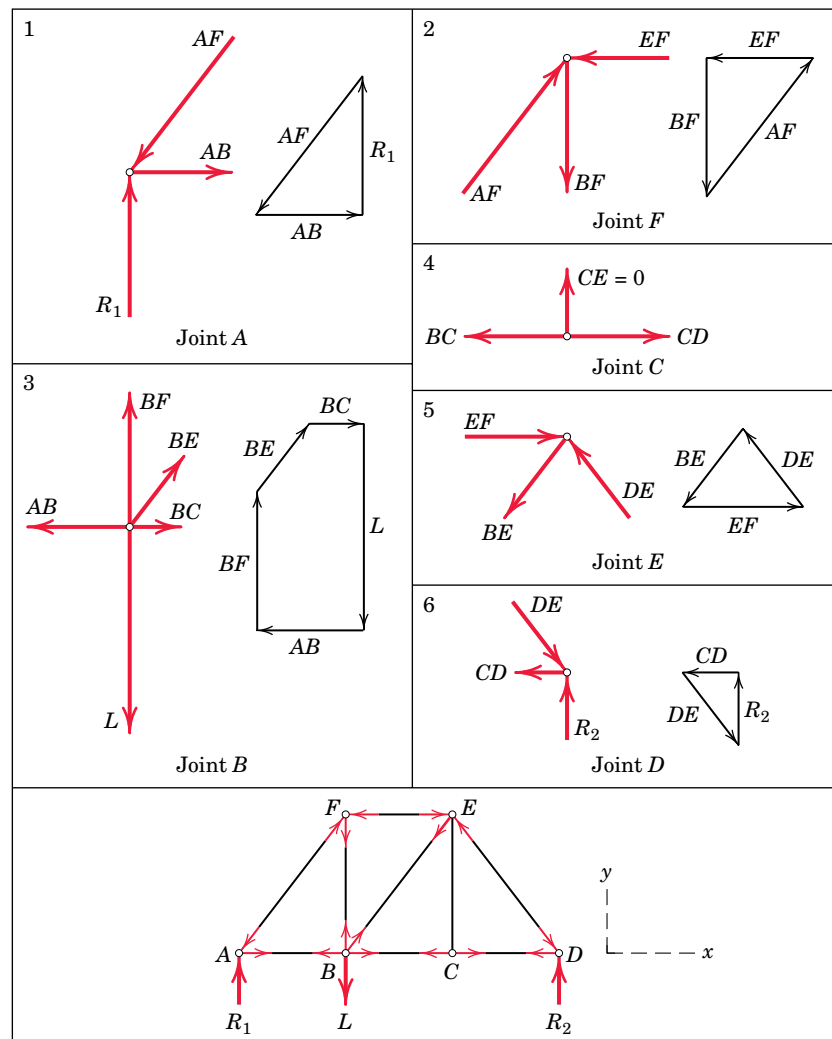


Figure 4/8

this member would not be zero, of course, if an external vertical load were applied at C .

It is often convenient to indicate the tension T and compression C of the various members directly on the original truss diagram by drawing arrows away from the pins for tension and toward the pins for compression. This designation is illustrated at the bottom of Fig. 4/8.

Sometimes we cannot initially assign the correct direction of one or both of the unknown forces acting on a given pin. If so, we may make an arbitrary assignment. A negative computed force value indicates that the initially assumed direction is incorrect.

Internal and External Redundancy

If a plane truss has more external supports than are necessary to ensure a stable equilibrium configuration, the truss as a whole is statically indeterminate, and the extra supports constitute *external* redundancy. If a truss has more internal members than are necessary to prevent collapse when the truss is removed from its supports, then the extra members constitute *internal* redundancy and the truss is again statically indeterminate.

For a truss which is statically determinate externally, there is a definite relation between the number of its members and the number of its joints necessary for internal stability without redundancy. Because we can specify the equilibrium of each joint by two scalar force equations, there are in all $2j$ such equations for a truss with j joints. For the entire truss composed of m two-force members and having the maximum of three unknown support reactions, there are in all $m + 3$ unknowns (m tension or compression forces and three reactions). Thus, for any plane truss, the equation $m + 3 = 2j$ will be satisfied if the truss is statically determinate internally.

A *simple* plane truss, formed by starting with a triangle and adding two new members to locate each new joint with respect to the existing structure, satisfies the relation automatically. The condition holds for the initial triangle, where $m = j = 3$, and m increases by 2 for each added joint while j increases by 1. Some other (nonsimple) statically determinate trusses, such as the K-truss in Fig. 4/2, are arranged differently, but can be seen to satisfy the same relation.

This equation is a necessary condition for stability but it is not a sufficient condition, since one or more of the m members can be arranged in such a way as not to contribute to a stable configuration of the entire truss. If $m + 3 > 2j$, there are more members than independent equations, and the truss is statically indeterminate internally with redundant members present. If $m + 3 < 2j$, there is a deficiency of internal members, and the truss is unstable and will collapse under load.

Special Conditions

We often encounter several special conditions in the analysis of trusses. When two collinear members are under compression, as indicated in Fig. 4/9a, it is necessary to add a third member to maintain



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Harbour Bridge in Sydney, Australia

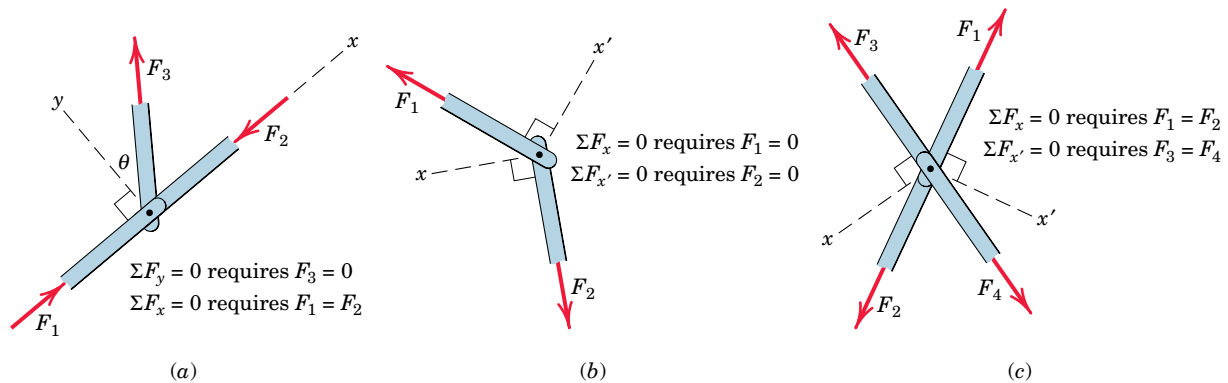


Figure 4/9

alignment of the two members and prevent buckling. We see from a force summation in the y -direction that the force F_3 in the third member must be zero and from the x -direction that $F_1 = F_2$. This conclusion holds regardless of the angle θ and holds also if the collinear members are in tension. If an external force with a component in the y -direction were applied to the joint, then F_3 would no longer be zero.

When two noncollinear members are joined as shown in Fig. 4/9b, then in the absence of an externally applied load at this joint, the forces in both members must be zero, as we can see from the two force summations.

When two pairs of collinear members are joined as shown in Fig. 4/9c, the forces in each pair must be equal and opposite. This conclusion follows from the force summations indicated in the figure.

Truss panels are frequently cross-braced as shown in Fig. 4/10a. Such a panel is statically indeterminate if each brace can support either tension or compression. However, when the braces are flexible members incapable of supporting compression, as are cables, then only the tension member acts and we can disregard the other member. It is usually evident from the asymmetry of the loading how the panel will deflect. If the deflection is as indicated in Fig. 4/10b, then member AB should be retained and CD disregarded. When this choice cannot be made by inspection, we may arbitrarily select the member to be retained. If the assumed tension turns out to be positive upon calculation, then the choice was correct. If the assumed tension force turns out to be negative, then the opposite member must be retained and the calculation redone.

We can avoid simultaneous solution of the equilibrium equations for two unknown forces at a joint by a careful choice of reference axes. Thus, for the joint indicated schematically in Fig. 4/11 where L is known and F_1 and F_2 are unknown, a force summation in the x -direction eliminates reference to F_1 and a force summation in the x' -direction eliminates reference to F_2 . When the angles involved are not easily found, then a simultaneous solution of the equations using one set of reference directions for both unknowns may be preferable.

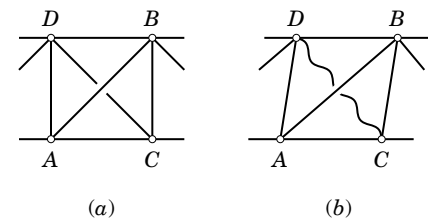


Figure 4/10

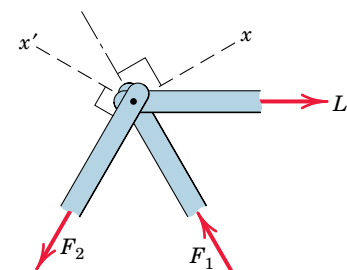


Figure 4/11

Sample Problem 4/1

Compute the force in each member of the loaded cantilever truss by the method of joints.

Solution. If it were not desired to calculate the external reactions at D and E , the analysis for a cantilever truss could begin with the joint at the loaded end. However, this truss will be analyzed completely, so the first step will be to compute the external forces at D and E from the free-body diagram of the truss as a whole. The equations of equilibrium give

$$\begin{aligned} [\Sigma M_E = 0] \quad & 5T - 20(5) - 30(10) = 0 & T &= 80 \text{ kN} \\ [\Sigma F_x = 0] \quad & 80 \cos 30^\circ - E_x = 0 & E_x &= 69.3 \text{ kN} \\ [\Sigma F_y = 0] \quad & 80 \sin 30^\circ + E_y - 20 - 30 = 0 & E_y &= 10 \text{ kN} \end{aligned}$$

Next we draw free-body diagrams showing the forces acting on each of the connecting pins. The correctness of the assigned directions of the forces is verified when each joint is considered in sequence. There should be no question about the correct direction of the forces on joint A . Equilibrium requires

$$\begin{aligned} [\Sigma F_y = 0] \quad & 0.866AB - 30 = 0 & AB &= 34.6 \text{ kN } T & \text{Ans.} \\ [\Sigma F_x = 0] \quad & AC - 0.5(34.6) = 0 & AC &= 17.32 \text{ kN } C & \text{Ans.} \end{aligned}$$

① where T stands for tension and C stands for compression.

Joint B must be analyzed next, since there are more than two unknown forces on joint C . The force BC must provide an upward component, in which case BD must balance the force to the left. Again the forces are obtained from

$$\begin{aligned} [\Sigma F_y = 0] \quad & 0.866BC - 0.866(34.6) = 0 & BC &= 34.6 \text{ kN } C & \text{Ans.} \\ [\Sigma F_x = 0] \quad & BD - 2(0.5)(34.6) = 0 & BD &= 34.6 \text{ kN } T & \text{Ans.} \end{aligned}$$

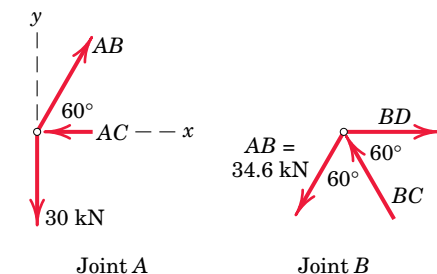
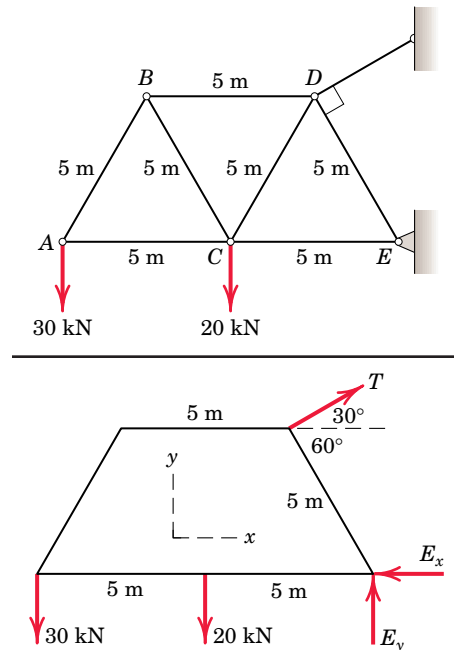
Joint C now contains only two unknowns, and these are found in the same way as before:

$$\begin{aligned} [\Sigma F_y = 0] \quad & 0.866CD - 0.866(34.6) - 20 = 0 & & & \\ & CD = 57.7 \text{ kN } T & \text{Ans.} & & \\ [\Sigma F_x = 0] \quad & CE - 17.32 - 0.5(34.6) - 0.5(57.7) = 0 & & & \\ & CE = 63.5 \text{ kN } C & \text{Ans.} & & \end{aligned}$$

Finally, from joint E there results

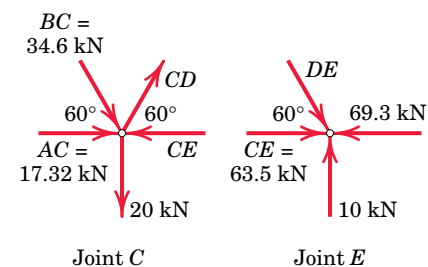
$$[\Sigma F_y = 0] \quad 0.866DE = 10 \quad DE = 11.55 \text{ kN } C \quad \text{Ans.}$$

and the equation $\Sigma F_x = 0$ checks.



Helpful Hint

① It should be stressed that the tension/compression designation refers to the member, not the joint. Note that we draw the force arrow on the same side of the joint as the member which exerts the force. In this way tension (arrow away from the joint) is distinguished from compression (arrow toward the joint).

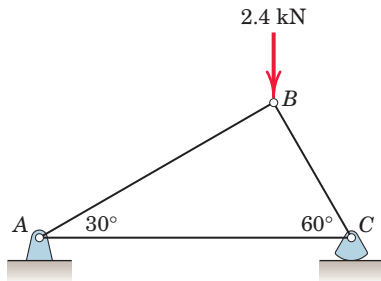


PROBLEMS

Introductory Problems

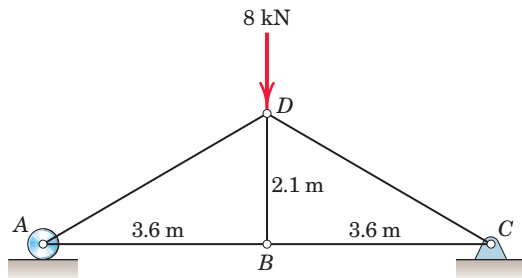
- 4/1** Determine the force in each member of the loaded truss. Explain why knowledge of the lengths of the members is unnecessary.

Ans. $AB = 1.2 \text{ kN C}$, $AC = 1.039 \text{ kN T}$
 $BC = 2.08 \text{ kN C}$



Problem 4/1

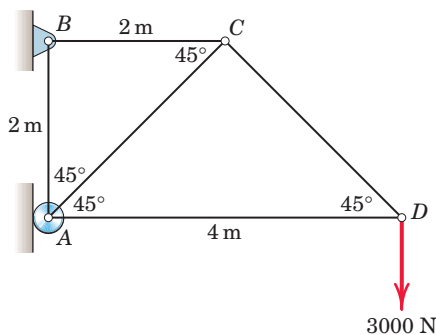
- 4/2** Determine the force in each member of the loaded truss. Identify any zero-force members by inspection.



Problem 4/2

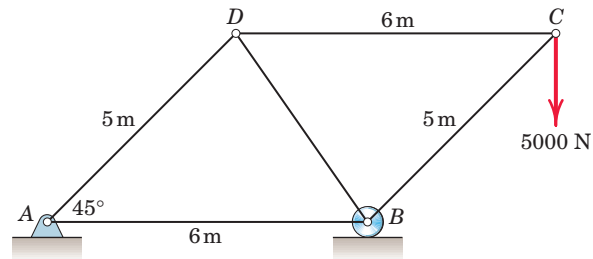
- 4/3** Determine the force in each member of the loaded truss.

Ans. $AB = 3000 \text{ N T}$, $AC = 4240 \text{ N C}$, $CD = 4240 \text{ N T}$
 $AD = 3000 \text{ N C}$, $BC = 6000 \text{ N T}$



Problem 4/3

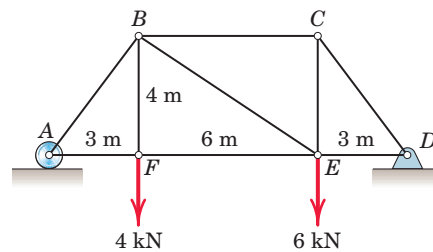
- 4/4** Determine the force in each member of the loaded truss.



Problem 4/4

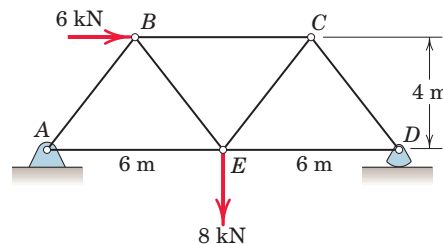
- 4/5** Determine the force in each member of the loaded truss.

Ans. $AB = 5.63 \text{ kN C}$, $AF = 3.38 \text{ kN T}$
 $BC = 4.13 \text{ kN C}$, $BE = 0.901 \text{ kN T}$
 $BF = 4 \text{ kN T}$, $CD = 6.88 \text{ kN C}$
 $CE = 5.50 \text{ kN T}$, $DE = 4.13 \text{ kN T}$
 $EF = 3.38 \text{ kN T}$



Problem 4/5

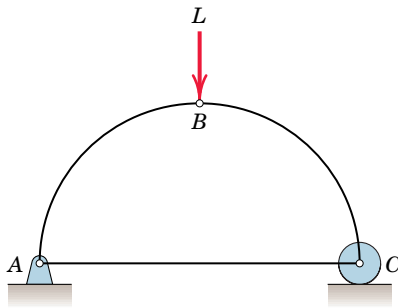
- 4/6** Calculate the force in each member of the loaded truss. All triangles are isosceles.



Problem 4/6

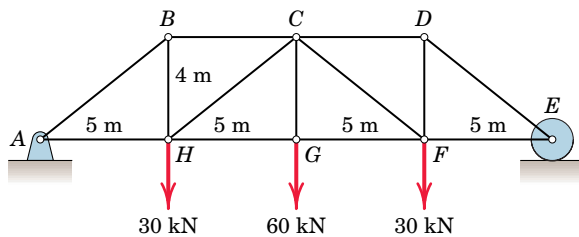
- 4/7** Determine the force in member AC of the loaded truss. The two quarter-circular members act as two-force members.

$$\text{Ans. } AC = \frac{L}{2} T$$



Problem 4/7

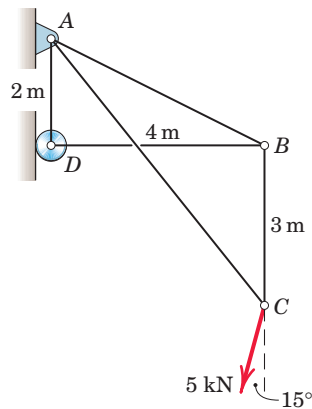
- 4/8** Determine the force in each member of the loaded truss. Make use of the symmetry of the truss and of the loading.



Problem 4/8

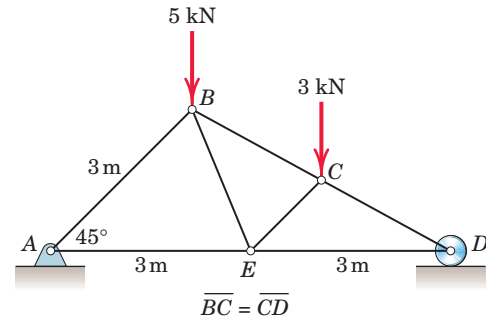
- 4/9** Determine the force in each member of the loaded truss.

$$\text{Ans. } AB = 14.42 \text{ kN } T, AC = 2.07 \text{ kN } C, AD = 0 \\ BC = 6.45 \text{ kN } T, BD = 12.89 \text{ kN } C$$



Problem 4/9

- 4/10** Determine the forces in members BE and CE of the loaded truss.

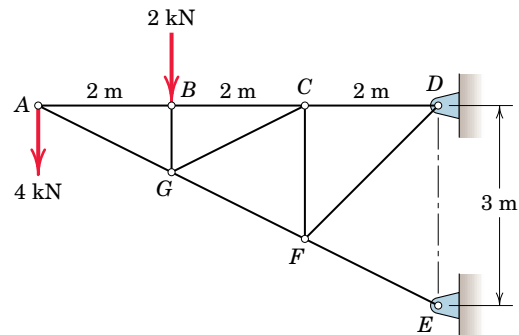


Problem 4/10

Representative Problems

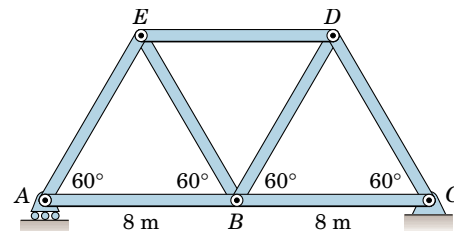
- 4/11** Calculate the forces in members CG and CF for the truss shown.

$$\text{Ans. } CG = 2.24 \text{ kN } T, CF = 1 \text{ kN } C$$



Problem 4/11

- 4/12** Each member of the truss is a uniform 8-m bar with a mass of 400 kg. Calculate the average tension or compression in each member due to the weights of the members.

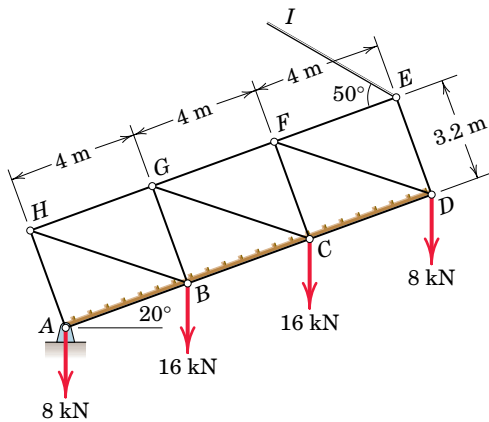


Problem 4/12

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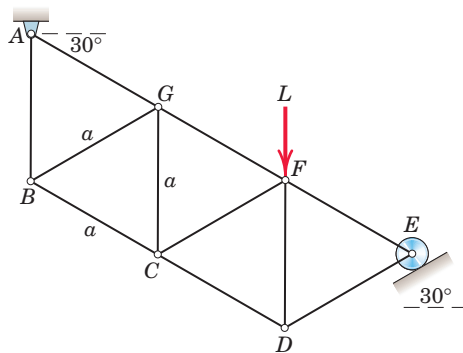
- 4/13** A drawbridge is being raised by a cable EI . The four joint loadings shown result from the weight of the roadway. Determine the forces in members EF , DE , DF , CD , and FG .

Ans. $EF = 15.46 \text{ kN C}$, $DE = 18.43 \text{ kN T}$
 $DF = 17.47 \text{ kN C}$, $CD = 10.90 \text{ kN T}$
 $FG = 29.1 \text{ kN C}$



Problem 4/13

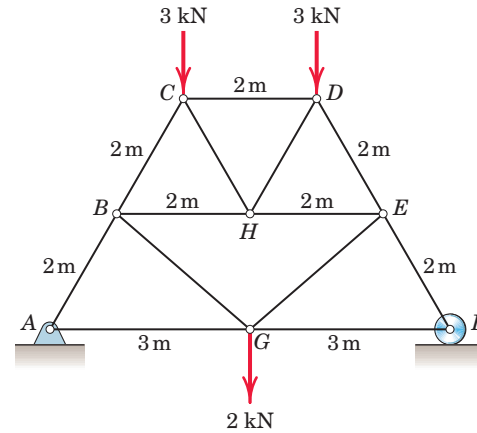
- 4/14** The truss is composed of equilateral triangles of sides a and is loaded and supported as shown. Determine the forces in members EF , DE , and DF .



Problem 4/14

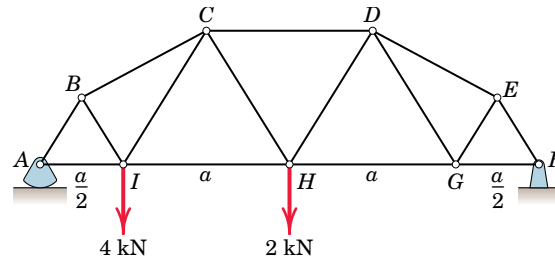
- 4/15** Determine the forces in members BC and BG of the loaded truss.

Ans. $BC = 3.46 \text{ kN C}$, $BG = 1.528 \text{ kN T}$



Problem 4/15

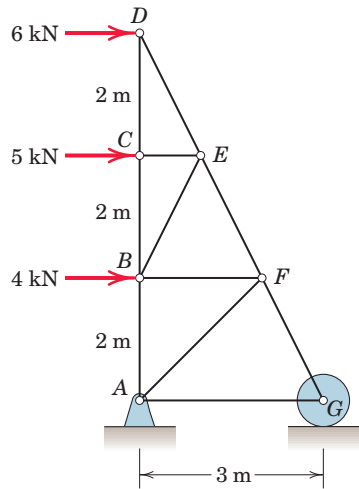
- 4/16** Determine the forces in members BI , CI , and HI for the loaded truss. All angles are 30° , 60° , or 90° .



Problem 4/16

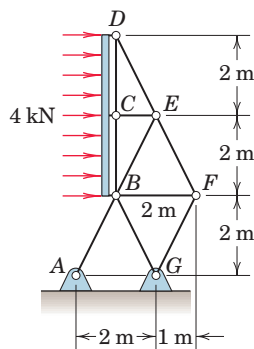
- 4/17** Determine the forces in members AF , BE , BF , and CE of the loaded truss.

Ans. $AF = 6.13 \text{ kN T}$, $BE = 5.59 \text{ kN T}$
 $BF = 6.50 \text{ kN C}$, $CE = 5 \text{ kN C}$



Problem 4/17

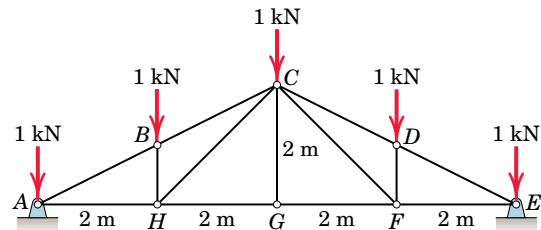
- 4/18** The signboard truss is designed to support a horizontal wind load of 4 kN. A separate analysis shows that $\frac{5}{8}$ of this force is transmitted to the center connection at C and the rest is equally divided between D and B . Calculate the forces in members BE and BC .



Problem 4/18

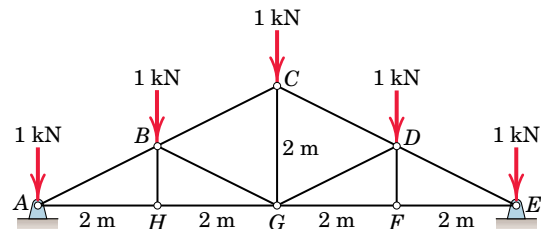
- 4/19** A snow load transfers the forces shown to the upper joints of a Pratt roof truss. Neglect any horizontal reactions at the supports and solve for the forces in all members.

Ans. $AB = DE = BC = CD = 3.35 \text{ kN C}$
 $AH = EF = 3 \text{ kN T}$, $BH = DF = 1 \text{ kN C}$
 $CF = CH = 1.414 \text{ kN T}$, $FG = GH = 2 \text{ kN T}$



Problem 4/19

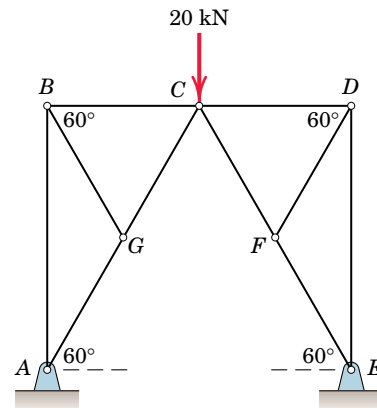
- 4/20** The loading of Prob. 4/19 is shown applied to a Howe roof truss. Neglect any horizontal reactions at the supports and solve for the forces in all members. Compare with the results of Prob. 4/19.



Problem 4/20

- 4/21** Determine the force in each member of the pair of trusses which support the 20-kN load at their common joint C .

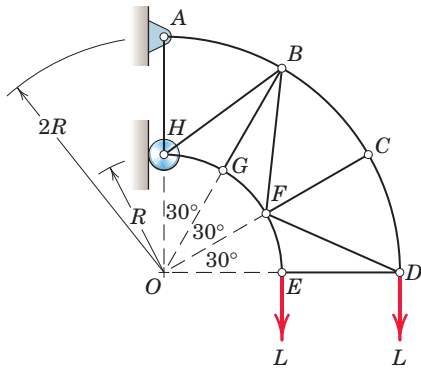
Ans. $AB = BC = BG = 0$, $AG = CG = 11.55 \text{ kN C}$



Problem 4/21

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- 4/22** Determine the force in member BF of the loaded truss.



Problem 4/22

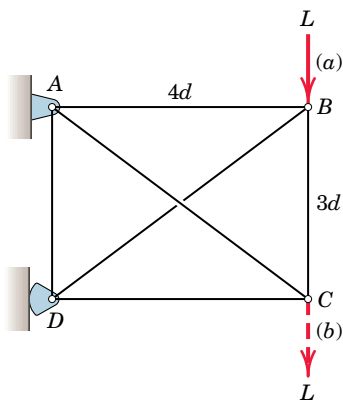
- 4/23** The rectangular frame is composed of four perimeter two-force members and two cables AC and BD which are incapable of supporting compression. Determine the forces in all members due to the load L in position (a) and then in position (b).

Ans. (a) $AB = AD = BD = 0$, $BC = L$ C

$$AC = \frac{5L}{3} \text{ T}, CD = \frac{4L}{3} \text{ C}$$

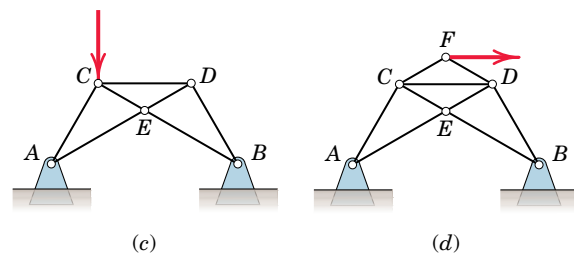
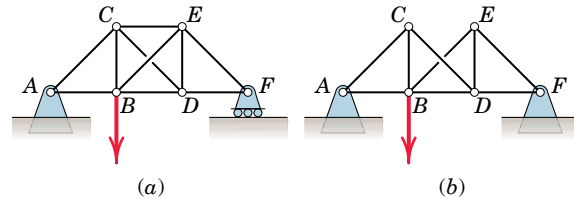
(b) $AB = AD = BC = BD = 0$

$$AC = \frac{5L}{3} \text{ T}, CD = \frac{4L}{3} \text{ C}$$



Problem 4/23

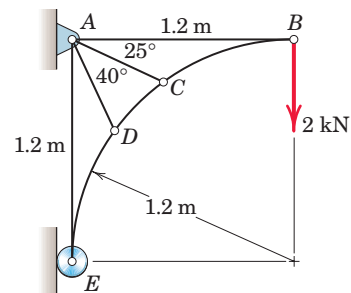
- 4/24** Verify the fact that each of the trusses contains one or more elements of redundancy and propose two separate changes, either one of which would remove the redundancy and produce complete statical determinacy. All members can support compression as well as tension.



Problem 4/24

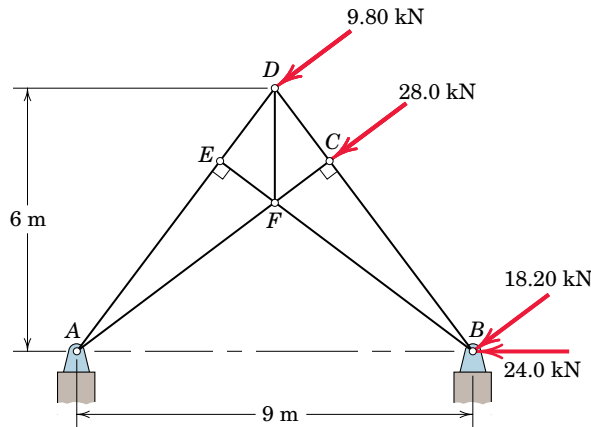
- 4/25** Determine the forces in members AC and AD of the loaded truss.

Ans. $AC = AD = 3.12 \text{ kN C}$



Problem 4/25

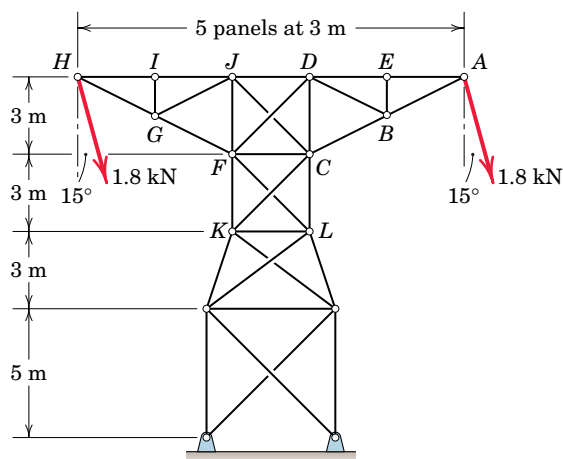
- 4/26** Analysis of the wind acting on a small Hawaiian church, which withstood the 280-km/h winds of Hurricane Iniki in 1992, showed the forces transmitted to each roof truss panel to be as shown. Treat the structure as a symmetrical simple truss and neglect any horizontal component of the support reaction at A. Identify the truss member which supports the largest force, tension or compression, and calculate this force.



Problem 4/26

- 4/27** The tower for a transmission line is modeled by the truss shown. The crossed members in the center sections of the truss may be assumed to be capable of supporting tension only. For the loads of 1.8 kN applied in the vertical plane, compute the forces induced in members AB, DB, and CD.

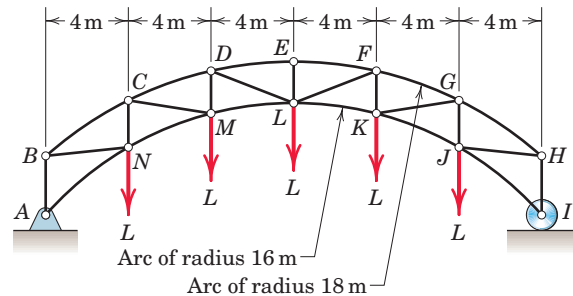
Ans. $AB = 3.89 \text{ kN C}$, $DB = 0$, $CD = 0.932 \text{ kN C}$



Problem 4/27

- 4/28** Determine the force in member CM of the loaded truss.

Ans. $CM = 3.41L \text{ T}$



Problem 4/28

4/4 METHOD OF SECTIONS

When analyzing plane trusses by the method of joints, we need only two of the three equilibrium equations because the procedures involve concurrent forces at each joint. We can take advantage of the third or moment equation of equilibrium by selecting an entire section of the truss for the free body in equilibrium under the action of a nonconcurrent system of forces. This *method of sections* has the basic advantage that the force in almost any desired member may be found directly from an analysis of a section which has cut that member. Thus, it is not necessary to proceed with the calculation from joint to joint until the member in question has been reached. In choosing a section of the truss, we note that, in general, not more than three members whose forces are unknown should be cut, since there are only three available independent equilibrium relations.

Illustration of the Method

The method of sections will now be illustrated for the truss in Fig. 4/6, which was used in the explanation of the method of joints. The truss is shown again in Fig. 4/12a for ready reference. The external reactions are first computed as with the method of joints, by considering the truss as a whole.

Let us determine the force in the member BE , for example. An imaginary section, indicated by the dashed line, is passed through the truss, cutting it into two parts, Fig. 4/12b. This section has cut three members whose forces are initially unknown. In order for the portion of the truss on each side of the section to remain in equilibrium, it is necessary to apply to each cut member the force which was exerted on it by the member cut away. For simple trusses composed of straight two-force members, these forces, either tensile or compressive, will always be in the directions of the respective members. The left-hand section is in equilibrium under the action of the applied load L , the end reaction R_1 , and the three forces exerted on the cut members by the right-hand section which has been removed.

We can usually draw the forces with their proper senses by a visual approximation of the equilibrium requirements. Thus, in balancing the moments about point B for the left-hand section, the force EF is clearly to the left, which makes it compressive, because it acts toward the cut section of member EF . The load L is greater than the reaction R_1 , so that the force BE must be up and to the right to supply the needed upward component for vertical equilibrium. Force BE is therefore tensile, since it acts away from the cut section.

With the approximate magnitudes of R_1 and L in mind we see that the balance of moments about point E requires that BC be to the right. A casual glance at the truss should lead to the same conclusion when it is realized that the lower horizontal member will stretch under the tension caused by bending. The equation of moments about joint B eliminates three forces from the relation, and EF can be determined directly. The force BE is calculated from the equilibrium equation for the y -direction. Finally, we determine BC by balancing moments about point E . In this

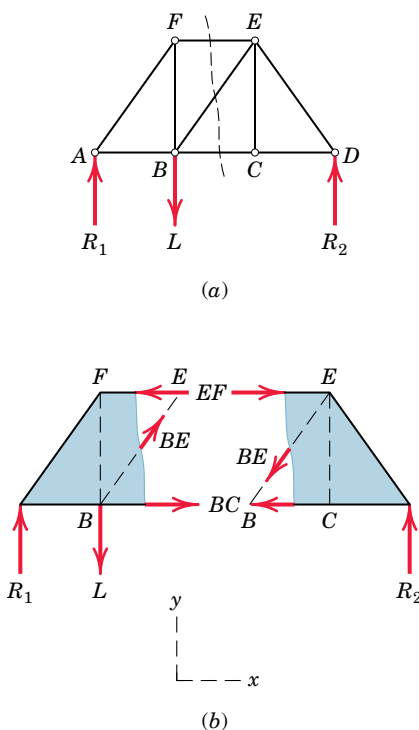


Figure 4/12

way each of the three unknowns has been determined independently of the other two.

The right-hand section of the truss, Fig. 4/12*b*, is in equilibrium under the action of R_2 and the same three forces in the cut members applied in the directions opposite to those for the left section. The proper sense for the horizontal forces can easily be seen from the balance of moments about points B and E .

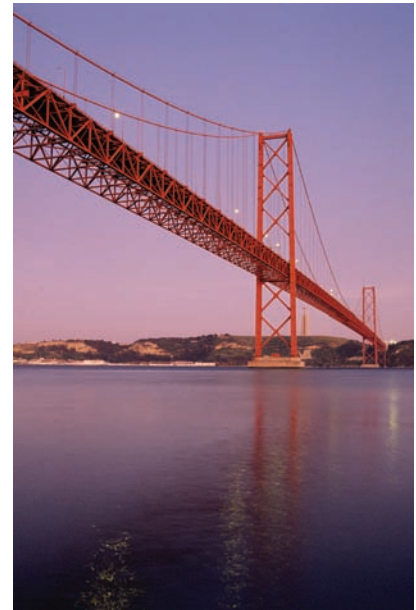
Additional Considerations

It is essential to understand that in the method of sections an entire portion of the truss is considered a single body in equilibrium. Thus, the forces in members internal to the section are not involved in the analysis of the section as a whole. To clarify the free body and the forces acting externally on it, the cutting section is preferably passed through the members and not the joints. We may use either portion of a truss for the calculations, but the one involving the smaller number of forces will usually yield the simpler solution.

In some cases the methods of sections and joints can be combined for an efficient solution. For example, suppose we wish to find the force in a central member of a large truss. Furthermore, suppose that it is not possible to pass a section through this member without passing through at least four unknown members. It may be possible to determine the forces in nearby members by the method of sections and then progress to the unknown member by the method of joints. Such a combination of the two methods may be more expedient than exclusive use of either method.

The moment equations are used to great advantage in the method of sections. One should choose a moment center, either on or off the section, through which as many unknown forces as possible pass.

It is not always possible to assign the proper sense of an unknown force when the free-body diagram of a section is initially drawn. Once an arbitrary assignment is made, a positive answer will verify the assumed sense and a negative result will indicate that the force is in the sense opposite to that assumed. An alternative notation preferred by some is to assign all unknown forces arbitrarily as positive in the tension direction (away from the section) and let the algebraic sign of the answer distinguish between tension and compression. Thus, a plus sign would signify tension and a minus sign compression. On the other hand, the advantage of assigning forces in their correct sense on the free-body diagram of a section wherever possible is that doing so emphasizes the physical action of the forces more directly, and this practice is the one which is preferred here.



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Many simple trusses are periodic in that there are repeated and identical structural sections.

Sample Problem 4/3

Calculate the forces induced in members KL , CL , and CB by the 200-kN load on the cantilever truss.

Solution. Although the vertical components of the reactions at A and M are statically indeterminate with the two fixed supports, all members other than AM are statically determinate. We may pass a section directly through members KL , CL , and CB and analyze the portion of the truss to the left of this section as a statically determinate rigid body.

The free-body diagram of the portion of the truss to the left of the section is shown. A moment sum about L quickly verifies the assignment of CB as compression, and a moment sum about C quickly discloses that KL is in tension. The direction of CL is not quite so obvious until we observe that KL and CB intersect at a point P to the right of G . A moment sum about P eliminates reference to KL and CB and shows that CL must be compressive to balance the moment of the 200-kN force about P . With these considerations in mind the solution becomes straightforward, as we now see how to solve for each of the three unknowns independently of the other two.

- ② Summing moments about L requires finding the moment arm $\overline{BL} = 4 + (6.5 - 4)/2 = 5.25$ m. Thus,

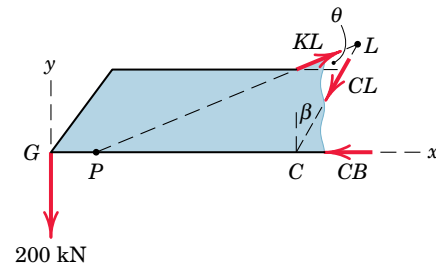
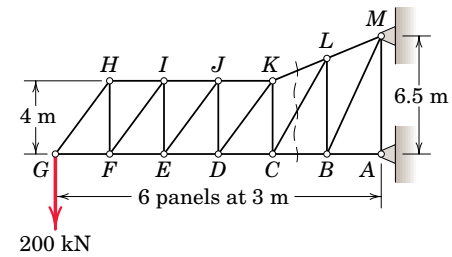
$$[\Sigma M_L = 0] \quad 200(5)(3) - CB(5.25) = 0 \quad CB = 571 \text{ kN } C \quad \text{Ans.}$$

Next we take moments about C , which requires a calculation of $\cos \theta$. From the given dimensions we see $\theta = \tan^{-1}(5/12)$ so that $\cos \theta = 12/13$. Therefore,

$$[\Sigma M_C = 0] \quad 200(4)(3) - \frac{12}{13}KL(4) = 0 \quad KL = 650 \text{ kN } T \quad \text{Ans.}$$

Finally, we may find CL by a moment sum about P , whose distance from C is given by $\overline{PC}/4 = 6/(6.5 - 4)$ or $\overline{PC} = 9.60$ m. We also need β , which is given by $\beta = \tan^{-1}(\overline{CB}/\overline{BL}) = \tan^{-1}(3/5.25) = 29.7^\circ$ and $\cos \beta = 0.868$. We now have

- ③ $[\Sigma M_P = 0] \quad 200(12 - 9.60) - CL(0.868)(9.60) = 0$
 $CL = 57.6 \text{ kN } C \quad \text{Ans.}$



Helpful Hints

- ① We note that analysis by the method of joints would necessitate working with eight joints in order to calculate the three forces in question. Thus, the method of sections offers a considerable advantage in this case.
- ② We could have started with moments about C or P just as well.
- ③ We could also have determined CL by a force summation in either the x - or y -direction.

Sample Problem 4/4

Calculate the force in member DJ of the Howe roof truss illustrated. Neglect any horizontal components of force at the supports.

Solution. It is not possible to pass a section through DJ without cutting four members whose forces are unknown. Although three of these cut by section 2 are concurrent at J and therefore the moment equation about J could be used to obtain DE , the force in DJ cannot be obtained from the remaining two equilibrium principles. It is necessary to consider first the adjacent section 1 before analyzing section 2.

The free-body diagram for section 1 is drawn and includes the reaction of 18.33 kN at A , which is previously calculated from the equilibrium of the truss as a whole. In assigning the proper directions for the forces acting on the three cut members, we see that a balance of moments about A eliminates the effects of CD and JK and clearly requires that CJ be up and to the left. A balance of moments about C eliminates the effect of the three forces concurrent at C and indicates that JK must be to the right to supply sufficient counterclockwise moment. Again it should be fairly obvious that the lower chord is under tension because of the bending tendency of the truss. Although it should also be apparent that the top chord is under compression, for purposes of illustration the force in CD will be arbitrarily assigned as tension.

- ① By the analysis of section 1, CJ is obtained from

$$[\Sigma M_A = 0] \quad 0.707CJ(12) - 10(4) - 10(8) = 0 \quad CJ = 14.14 \text{ kN C}$$

In this equation the moment of CJ is calculated by considering its horizontal and vertical components acting at point J . Equilibrium of moments about J requires

$$[\Sigma M_J = 0] \quad 0.894CD(6) + 18.33(12) - 10(4) - 10(8) = 0$$

$$CD = -18.63 \text{ kN}$$

The moment of CD about J is calculated here by considering its two components as acting through D . The minus sign indicates that CD was assigned in the wrong direction.

- ② Hence, $CD = 18.63 \text{ kN C}$

- ③ From the free-body diagram of section 2, which now includes the known value of CJ , a balance of moments about G is seen to eliminate DE and JK . Thus,

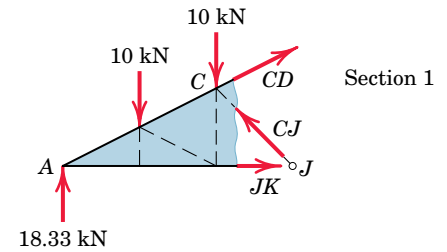
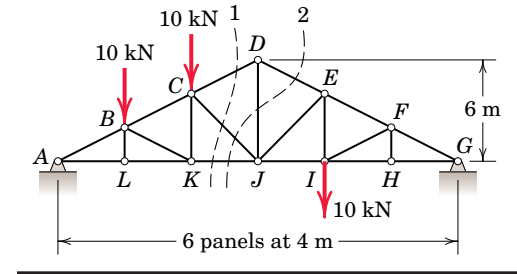
$$[\Sigma M_G = 0] \quad 12DJ + 10(16) + 10(20) - 18.33(24) - 14.14(0.707)(12) = 0$$

$$DJ = 16.67 \text{ kN T}$$

Ans.

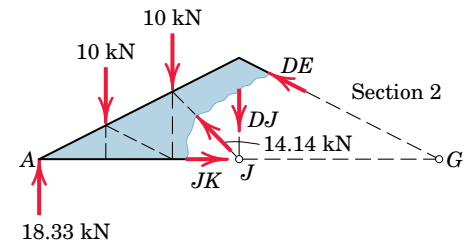
Again the moment of CJ is determined from its components considered to be acting at J . The answer for DJ is positive, so that the assumed tensile direction is correct.

An alternative approach to the entire problem is to utilize section 1 to determine CD and then use the method of joints applied at D to determine DJ .



Helpful Hints

- ① There is no harm in assigning one or more of the forces in the wrong direction, as long as the calculations are consistent with the assumption. A negative answer will show the need for reversing the direction of the force.
- ② If desired, the direction of CD may be changed on the free-body diagram and the algebraic sign of CD reversed in the calculations, or else the work may be left as it stands with a note stating the proper direction.



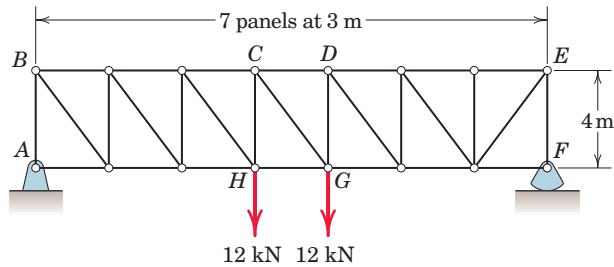
- ③ Observe that a section through members CD , DJ , and DE could be taken which would cut only three unknown members. However, since the forces in these three members are all concurrent at D , a moment equation about D would yield no information about them. The remaining two force equations would not be sufficient to solve for the three unknowns.

PROBLEMS

Introductory Problems

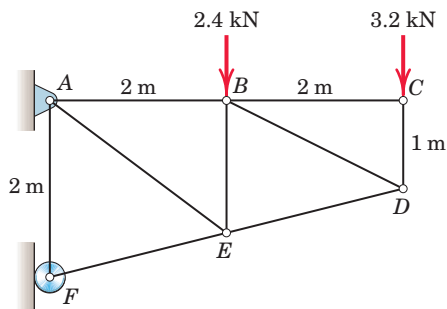
4/29 Determine the forces in members CG and GH .

Ans. $CG = 0$, $GH = 27 \text{ kN T}$



Problem 4/29

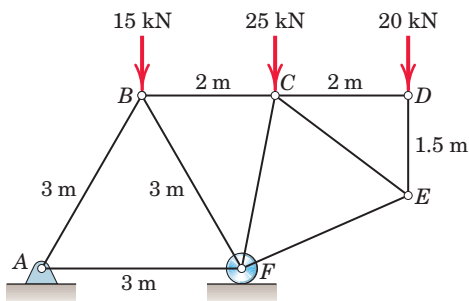
4/30 Determine the force in member AE of the loaded truss.



Problem 4/30

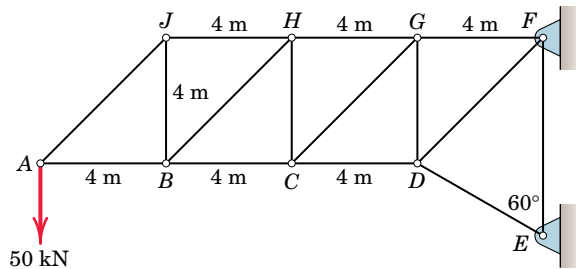
4/31 Determine the force in member BC of the loaded truss.

Ans. $BC = 24.1 \text{ kN T}$



Problem 4/31

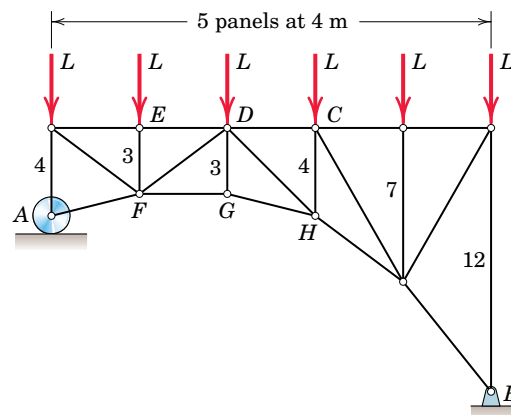
4/32 Determine the forces in members GH and CG for the truss loaded and supported as shown. Does the static indeterminacy of the supports affect your calculation?



Problem 4/32

4/33 Determine the force in member DG of the loaded truss.

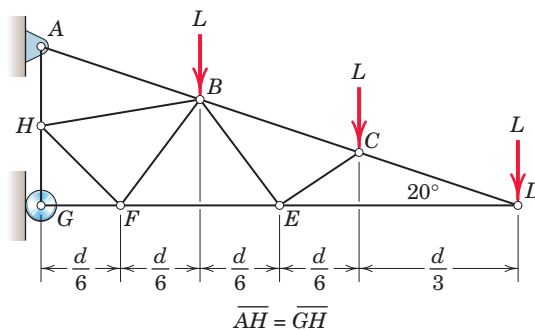
Ans. $DG = L \text{ T}$



Dimensions in meters

Problem 4/33

4/34 Determine the force in member BE of the loaded truss.

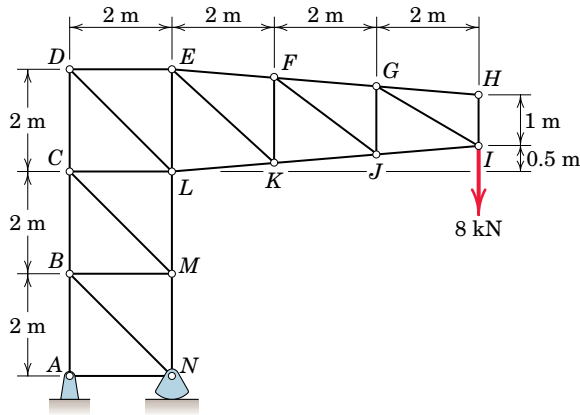


Problem 4/34

Representative Problems

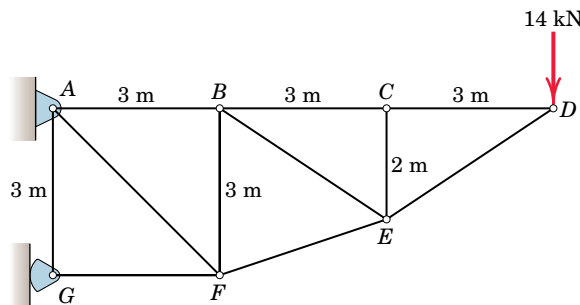
4/35 Determine the forces in members DE and DL .

Ans. $DE = 24 \text{ kN T}$, $DL = 33.9 \text{ kN C}$



Problem 4/35

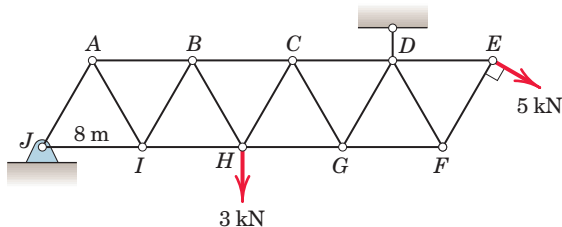
4/36 Calculate the forces in members BC , BE , and EF . Solve for each force from an equilibrium equation which contains that force as the only unknown.



Problem 4/36

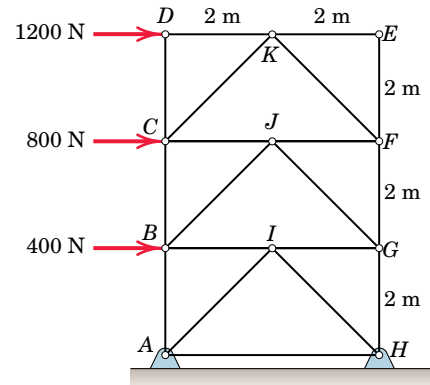
4/37 Calculate the forces in members BC , CD , and CG of the loaded truss composed of equilateral triangles, each of side length 8 m.

Ans. $BC = 1.155 \text{ kN T}$, $CD = 5.20 \text{ kN T}$
 $CG = 4.04 \text{ kN C}$



Problem 4/37

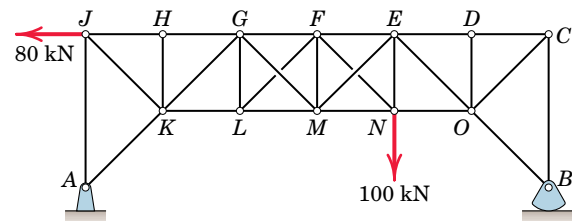
4/38 Determine the forces in members BC and FG of the loaded symmetrical truss. Show that this calculation can be accomplished by using one section and two equations, each of which contains only one of the two unknowns. Are the results affected by the static indeterminacy of the supports at the base?



Problem 4/38

4/39 The truss shown is composed of 45° right triangles. The crossed members in the center two panels are slender tie rods incapable of supporting compression. Retain the two rods which are under tension and compute the magnitudes of their tensions. Also find the force in member MN .

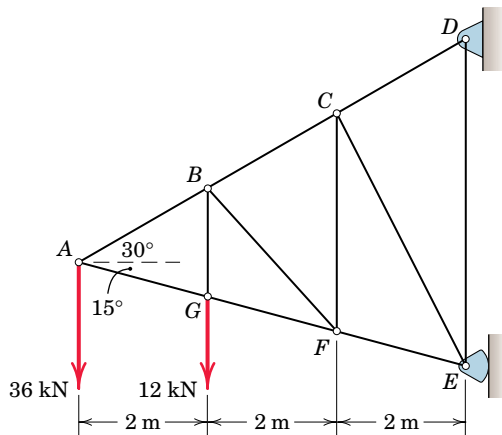
Ans. $FN = GM = 84.8 \text{ kN T}$, $MN = 20 \text{ kN T}$



Problem 4/39

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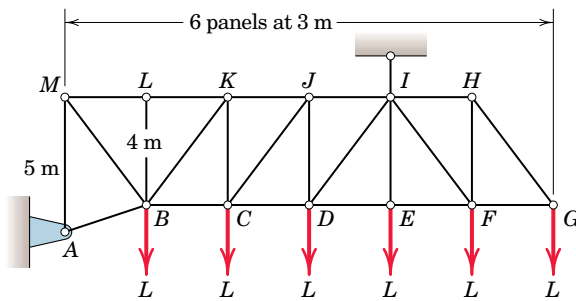
4/40 Determine the force in member BF .



Problem 4/40

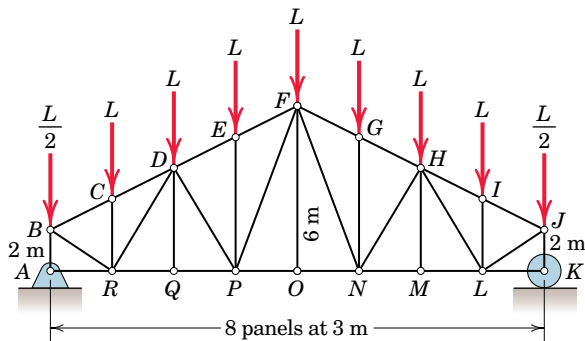
4/41 Determine the forces in members CD , CJ , and DJ .

Ans. $CD = 0.562L$ C, $CJ = 1.562L$ T
 $DJ = 1.250L$ C



Problem 4/41

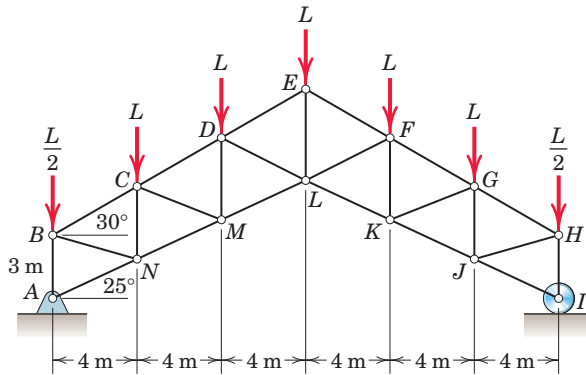
4/42 Compute the force in member HN of the loaded truss.



Problem 4/42

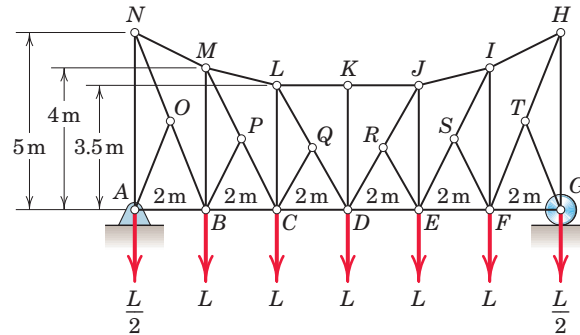
4/43 Determine the forces in members DE , DL , LM , and EL of the loaded symmetrical truss.

Ans. $DE = 4.80L$ C, $DL = 0.0446L$ T, $LM = 4.54L$ T
 $EL = 3.80L$ T



Problem 4/43

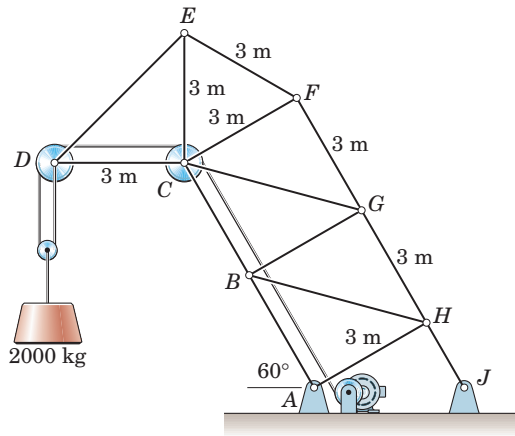
4/44 Determine the forces in members DQ and CQ of the loaded symmetrical truss.



Problem 4/44

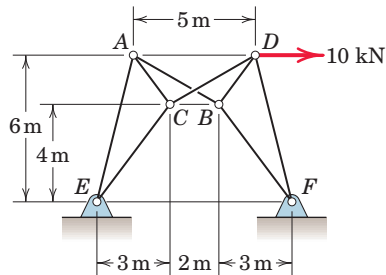
4/45 Calculate the forces in members CB , CG , and FG for the loaded truss without first calculating the force in any other member.

Ans. $CB = 56.2$ kN C, $CG = 13.87$ kN T
 $FG = 19.62$ kN T



Problem 4/45

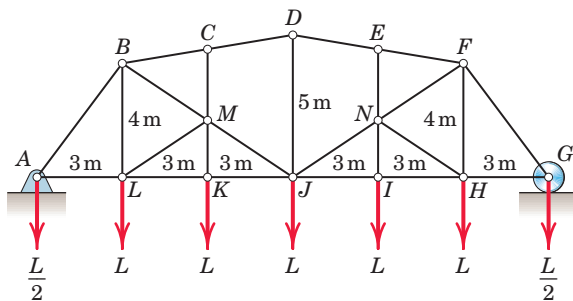
- 4/46** The hinged frames ACE and DFB are connected by two hinged bars, AB and CD , which cross without being connected. Compute the force in AB .



Problem 4/46

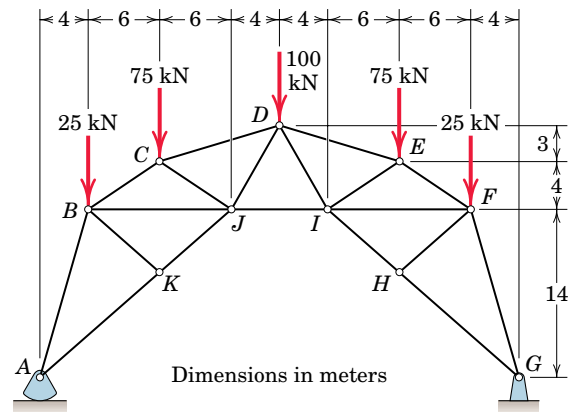
- 4/47** Determine the force in member JM of the loaded truss.

Ans. $JM = 0.0901L$ T



Problem 4/47

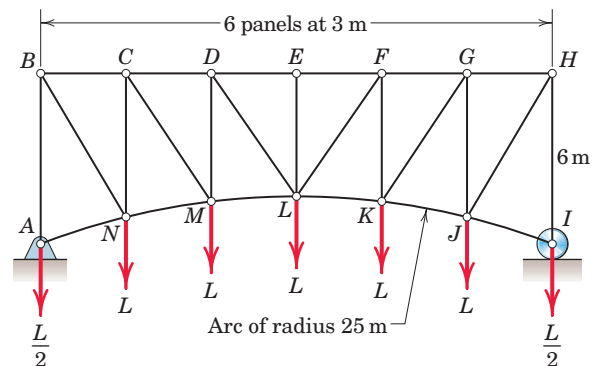
- 4/48** Determine the forces in members DE , EI , FI , and HI of the arched roof truss.



Problem 4/48

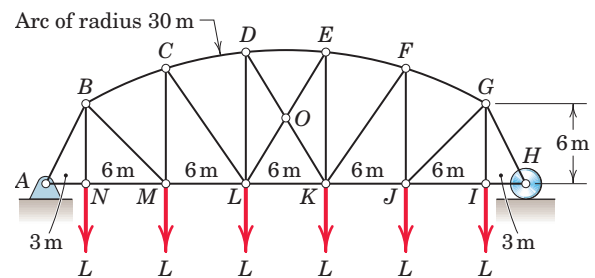
- 4/49** Determine the force in member GK of the loaded symmetrical truss.

Ans. $GK = 2.13L$ T



Problem 4/49

- 4/50** Determine the force in member CL of the loaded truss. The radius of curvature of the upper chord $BCDEFG$ is 30 m.

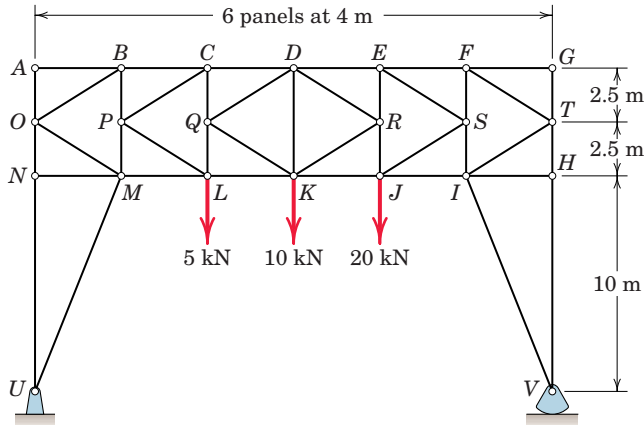


Problem 4/50

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- 4/51 Determine the force in member DK of the loaded overhead sign truss.

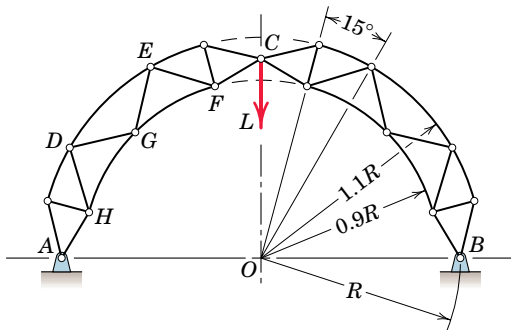
Ans. $DK = 5 \text{ kN T}$



Problem 4/51

- 4/52 Determine the force in member DG of the compound truss. The joints all lie on radial lines subtending angles of 15° as indicated, and the curved members act as two-force members. Distance $\overline{OC} = \overline{OA} = \overline{OB} = R$.

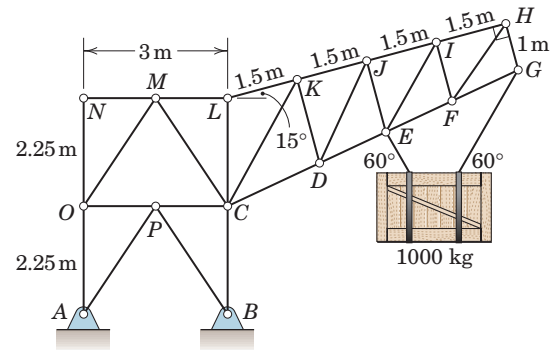
Ans. $DG = 0.569L C$



Problem 4/52

- 4/53 Determine the force in member CK of the loaded truss.

Ans. $CK = 9290 \text{ N C}$

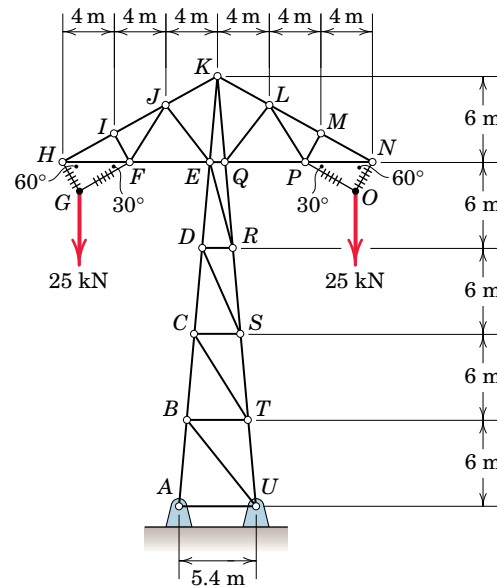


Problem 4/53

- 4/54 A design model for a transmission-line tower is shown in the figure. Members GH , FG , OP , and NO are insulated cables; all other members are steel bars. For the loading shown, compute the forces in members FI , FJ , EJ , EK , and ER . Use a combination of methods if desired.

Ans. $FI = ER = 0$, $FJ = 7.81 \text{ kN T}$

$EJ = 3.61 \text{ kN C}$, $EK = 22.4 \text{ kN C}$



Problem 4/54

4/5 SPACE TRUSSES

A *space truss* is the three-dimensional counterpart of the plane truss described in the three previous articles. The idealized space truss consists of rigid links connected at their ends by ball-and-socket joints (such a joint is illustrated in Fig. 3/8 in Art. 3/4). Whereas a triangle of pin-connected bars forms the basic noncollapsible unit for the plane truss, a space truss, on the other hand, requires six bars joined at their ends to form the edges of a tetrahedron as the basic noncollapsible unit. In Fig. 4/13a the two bars AD and BD joined at D require a third support CD to keep the triangle ADB from rotating about AB . In Fig. 4/13b the supporting base is replaced by three more bars AB , BC , and AC to form a tetrahedron not dependent on the foundation for its own rigidity.

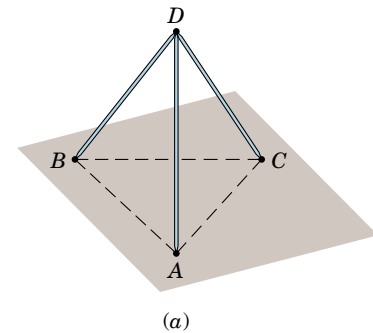
We may form a new rigid unit to extend the structure with three additional concurrent bars whose ends are attached to three fixed joints on the existing structure. Thus, in Fig. 4/13c the bars AF , BF , and CF are attached to the foundation and therefore fix point F in space. Likewise point H is fixed in space by the bars AH , DH , and CH . The three additional bars CG , FG , and HG are attached to the three fixed points C , F , and H and therefore fix G in space. The fixed point E is similarly created. We see now that the structure is entirely rigid. The two applied loads shown will result in forces in all of the members. A space truss formed in this way is called a *simple space truss*.

Ideally there must be point support, such as that given by a ball-and-socket joint, at the connections of a space truss to prevent bending in the members. As in riveted and welded connections for plane trusses, if the centerlines of joined members intersect at a point, we can justify the assumption of two-force members under simple tension and compression.

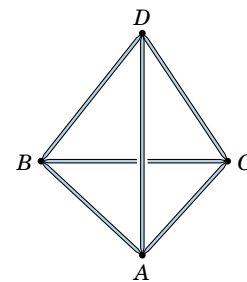
Statically Determinate Space Trusses

When a space truss is supported externally so that it is statically determinate as an entire unit, a relationship exists between the number of its joints and the number of its members necessary for internal stability without redundancy. Because the equilibrium of each joint is specified by three scalar force equations, there are in all $3j$ such equations for a space truss with j joints. For the entire truss composed of m members there are m unknowns (the tensile or compressive forces in the members) plus six unknown support reactions in the general case of a statically determinate space structure. Thus, for any space truss, the equation $m + 6 = 3j$ will be satisfied if the truss is statically determinate internally. A *simple space truss* satisfies this relation automatically. Starting with the initial tetrahedron, for which the equation holds, the structure is extended by adding three members and one joint at a time, thus preserving the equality.

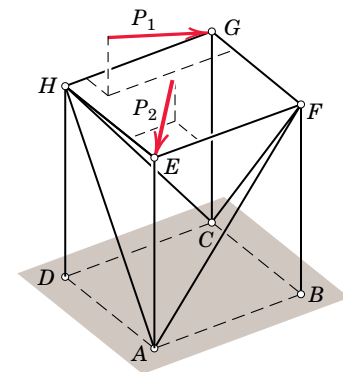
As in the case of the plane truss, this relation is a necessary condition for stability, but it is not a sufficient condition, since one or more of the m members can be arranged in such a way as not to contribute to a stable configuration of the entire truss. If $m + 6 > 3j$, there are more members than there are independent equations, and the truss is statically indeterminate internally with redundant members present.



(a)



(b)



(c)

Figure 4/13

If $m + 6 < 3j$, there is a deficiency of internal members, and the truss is unstable and subject to collapse under load. This relationship between the number of joints and the number of members is very helpful in the preliminary design of a stable space truss, since the configuration is not as obvious as with a plane truss, where the geometry for statical determinacy is generally quite apparent.

Method of Joints for Space Trusses

The method of joints developed in Art. 4/3 for plane trusses may be extended directly to space trusses by satisfying the complete vector equation

$$\Sigma \mathbf{F} = \mathbf{0} \quad (4/1)$$

for each joint. We normally begin the analysis at a joint where at least one known force acts and not more than three unknown forces are present. Adjacent joints on which not more than three unknown forces act may then be analyzed in turn.

This step-by-step joint technique tends to minimize the number of simultaneous equations to be solved when we must determine the forces in all members of the space truss. For this reason, although it is not readily reduced to a routine, such an approach is recommended. As an alternative procedure, however, we may simply write $3j$ joint equations by applying Eq. 4/1 to all joints of the space frame. The number of unknowns will be $m + 6$ if the structure is noncollapsible when removed from its supports and those supports provide six external reactions. If, in addition, there are no redundant members, then the number of equations ($3j$) equals the number of unknowns ($m + 6$), and the entire system of equations can be solved simultaneously for the unknowns. Because of the large number of coupled equations, a computer solution is usually required. With this latter approach, it is not necessary to begin at a joint where at least one known and no more than three unknown forces act.

Method of Sections for Space Trusses

The method of sections developed in the previous article may also be applied to space trusses. The two vector equations

$$\Sigma \mathbf{F} = \mathbf{0} \quad \text{and} \quad \Sigma \mathbf{M} = \mathbf{0}$$

must be satisfied for any section of the truss, where the zero moment sum will hold for all moment axes. Because the two vector equations are equivalent to six scalar equations, we conclude that, in general, a section should not be passed through more than six members whose forces are unknown. The method of sections for space trusses is not widely used, however, because a moment axis can seldom be found which eliminates all but one unknown, as in the case of plane trusses.

Vector notation for expressing the terms in the force and moment equations for space trusses is of considerable advantage and is used in the sample problem which follows.



© Achim Gaasterland/laif/Redux

This space truss is used to support observation equipment in Bottrop, Germany.

Sample Problem 4/5

The space truss consists of the rigid tetrahedron $ABCD$ anchored by a ball-and-socket connection at A and prevented from any rotation about the x -, y -, or z -axes by the respective links 1, 2, and 3. The load L is applied to joint E , which is rigidly fixed to the tetrahedron by the three additional links. Solve for the forces in the members at joint E and indicate the procedure for the determination of the forces in the remaining members of the truss.

Solution. We note first that the truss is supported with six properly placed constraints, which are the three at A and the links 1, 2, and 3. Also, with $m = 9$ members and $j = 5$ joints, the condition $m + 6 = 3j$ for a sufficiency of members to provide a noncollapsible structure is satisfied.

The external reactions at A , B , and D can be calculated easily as a first step, although their values will be determined from the solution of all forces on each of the joints in succession.

- ① We start with a joint on which at least one known force and not more than three unknown forces act, which in this case is joint E . The free-body diagram of joint E is shown with all force vectors arbitrarily assumed in their positive tension directions (away from the joint).
- ② The vector expressions for the three unknown forces are

$$\mathbf{F}_{EB} = \frac{F_{EB}}{\sqrt{2}}(-\mathbf{i} - \mathbf{j}), \mathbf{F}_{EC} = \frac{F_{EC}}{5}(-3\mathbf{i} - 4\mathbf{k}), \mathbf{F}_{ED} = \frac{F_{ED}}{5}(-3\mathbf{j} - 4\mathbf{k})$$

Equilibrium of joint E requires

$$[\Sigma \mathbf{F} = \mathbf{0}] \quad \mathbf{L} + \mathbf{F}_{EB} + \mathbf{F}_{EC} + \mathbf{F}_{ED} = \mathbf{0} \quad \text{or}$$

$$-L\mathbf{i} + \frac{F_{EB}}{\sqrt{2}}(-\mathbf{i} - \mathbf{j}) + \frac{F_{EC}}{5}(-3\mathbf{i} + 4\mathbf{k}) + \frac{F_{ED}}{5}(-3\mathbf{j} - 4\mathbf{k}) = \mathbf{0}$$

Rearranging terms gives

$$\left(-L - \frac{F_{EB}}{\sqrt{2}} - \frac{3F_{EC}}{5}\right)\mathbf{i} + \left(-\frac{F_{EB}}{\sqrt{2}} - \frac{3F_{ED}}{5}\right)\mathbf{j} + \left(-\frac{4F_{EC}}{5} - \frac{4F_{ED}}{5}\right)\mathbf{k} = \mathbf{0}$$

Equating the coefficients of the \mathbf{i} -, \mathbf{j} -, and \mathbf{k} -unit vectors to zero gives the three equations

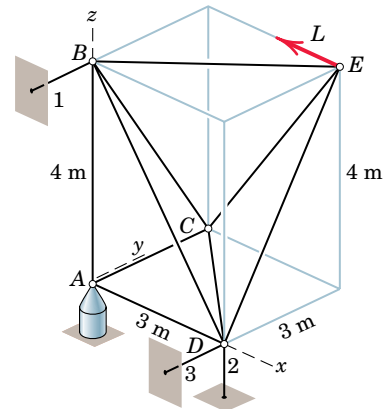
$$\frac{F_{EB}}{\sqrt{2}} + \frac{3F_{EC}}{5} = -L \quad \frac{F_{EB}}{\sqrt{2}} + \frac{3F_{ED}}{5} = 0 \quad F_{EC} + F_{ED} = 0$$

Solving the equations gives us

$$F_{EB} = -L/\sqrt{2} \quad F_{EC} = -5L/6 \quad F_{ED} = 5L/6 \quad \text{Ans.}$$

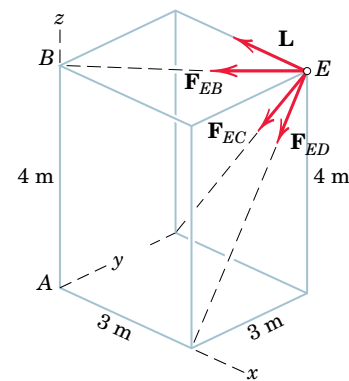
Thus, we conclude that F_{EB} and F_{EC} are compressive forces and F_{ED} is tension.

Unless we have computed the external reactions first, we must next analyze joint C with the known value of F_{EC} and the three unknowns F_{CB} , F_{CA} , and F_{CD} . The procedure is identical with that used for joint E . Joints B , D , and A are then analyzed in the same way and in that order, which limits the scalar unknowns to three for each joint. The external reactions computed from these analyses must, of course, agree with the values which can be determined initially from an analysis of the truss as a whole.



Helpful Hints

- ① *Suggestion:* Draw a free-body diagram of the truss as a whole and verify that the external forces acting on the truss are $\mathbf{A}_x = L\mathbf{i}$, $\mathbf{A}_y = L\mathbf{j}$, $\mathbf{A}_z = (4L/3)\mathbf{k}$, $\mathbf{B}_y = \mathbf{0}$, $\mathbf{D}_y = -L\mathbf{j}$, $\mathbf{D}_z = -(4L/3)\mathbf{k}$.
- ② With this assumption, a negative numerical value for a force indicates compression.

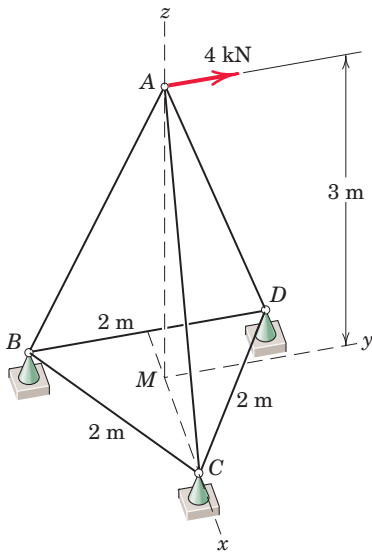


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PROBLEMS

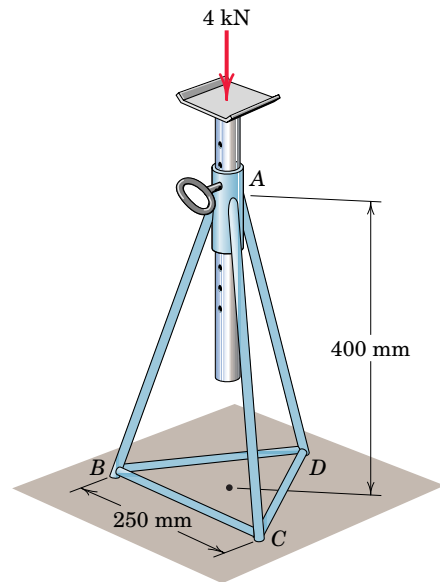
(In the following problems, use plus for tension and minus for compression.)

- 4/55** Determine the forces in members AB , AC , and AD . Point M is the centroid of triangle BCD .
Ans. $T_{AB} = 6.43 \text{ kN}$, $T_{AC} = 0$, $T_{AD} = -6.43 \text{ kN}$



Problem 4/55

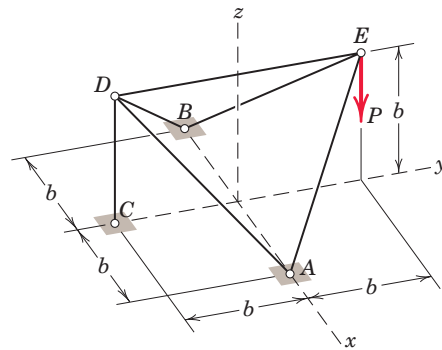
- 4/56** The base of an automobile jackstand forms an equilateral triangle of side length 250 mm and is centered under the collar A . Model the structure as one with a ball and socket at each joint and determine the forces in members BC , BD , and CD . Neglect any horizontal reaction components under the feet B , C , and D .



Problem 4/56

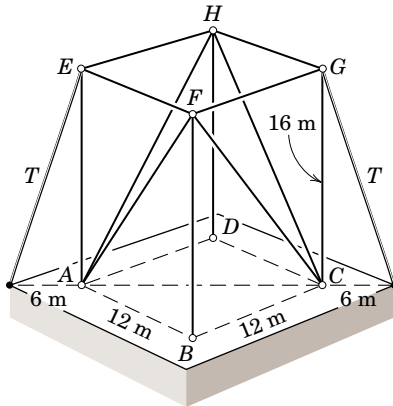
- 4/57** The space truss in the form of a tetrahedron is supported by ball-and-socket connections at its base points A and B and is prevented from rotating about AB by the vertical tie bar CD . After noting the vertical components of the reactions under the symmetrical truss at A and B , draw a free-body diagram of the triangular configuration of links BDE and determine the x -component of the force exerted by the foundation on the truss at B .

Ans. $B_x = P$



Problem 4/57

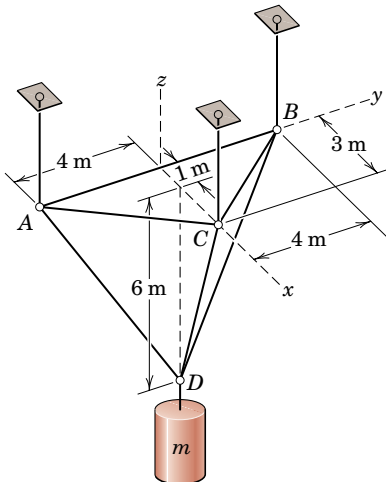
- 4/58** The rectangular space truss 16 m in height is erected on a horizontal square base 12 m on a side. Guy wires are attached to the structure at E and G as shown and are tightened until the tension T in each wire is 9 kN. Calculate the force F in each of the diagonal members.



Problem 4/58

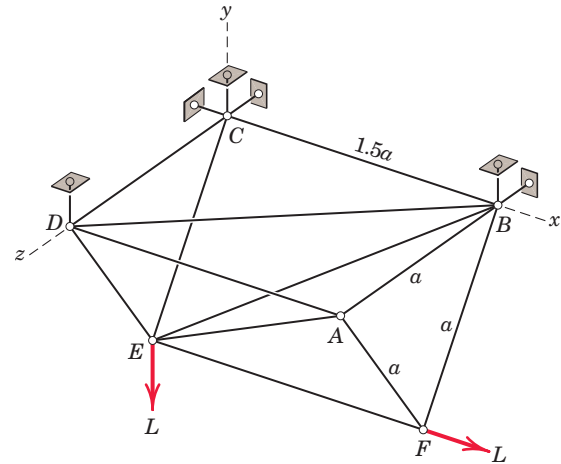
- 4/59** The tetrahedral space truss has a horizontal base ABC in the form of an isosceles triangle and legs AD , BD , and CD which support the mass m from point D . Each vertex of the base is suspended by a vertical wire from an overhead support. Calculate the forces induced in members AC and AB .

$$\text{Ans. } F_{AC} = -\frac{5}{54}mg, F_{AB} = -\frac{4}{27}mg$$



Problem 4/59

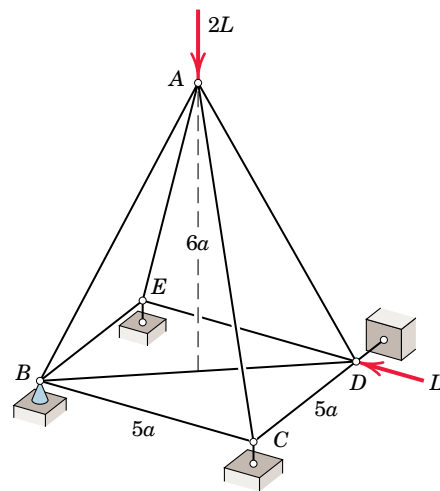
- 4/60** For the space truss shown, check the sufficiency of the supports and also the number of and arrangement of the members to ensure statical determinacy, both external and internal. Determine the forces in members AE , BE , BF , and CE .



Problem 4/60

- 4/61** Determine the force in member BD of the regular pyramid with square base.

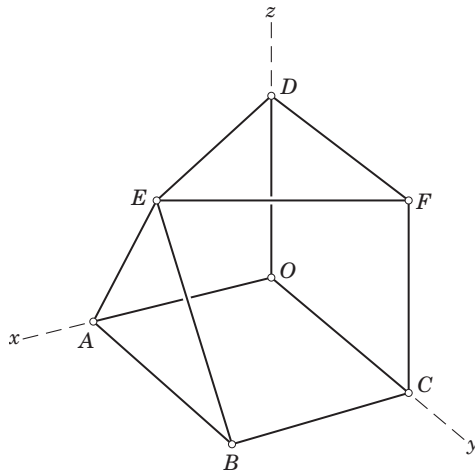
$$\text{Ans. } DB = -2.00L$$



Problem 4/61

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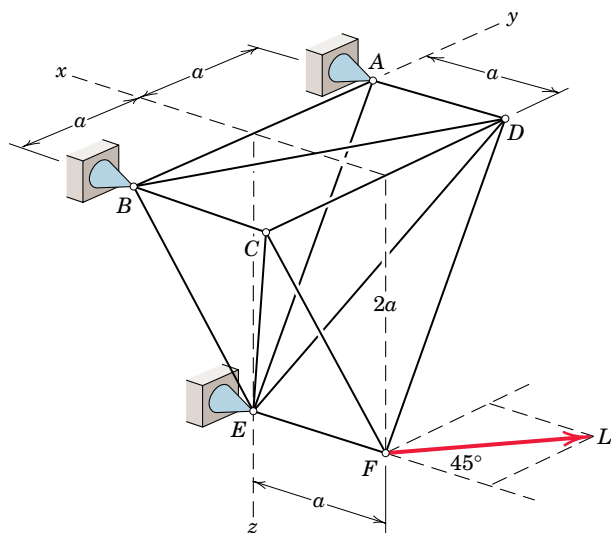
- 4/62** A space truss is being designed with the elements shown. How many more members are needed to make the truss stable internally? Name the members (by specifying their end points) which would produce one possible configuration of internal stability.



Problem 4/62

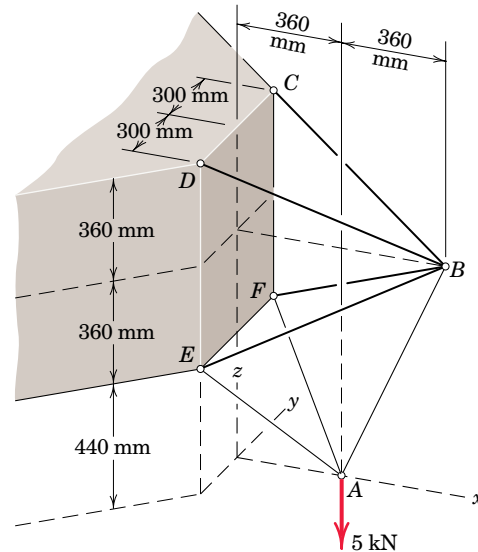
- 4/63** The space truss shown is secured to the fixed supports at A, B, and E and is loaded by the force L which has equal x - and y -components but no vertical z -component. Show that there is a sufficient number of members to provide internal stability and that their placement is adequate for this purpose. Next determine the forces in members CD , BC , and CE .

$$\text{Ans. } F_{BC} = \frac{L\sqrt{2}}{4}, F_{CD} = 0, F_{CE} = -\frac{L\sqrt{3}}{2}$$



Problem 4/63

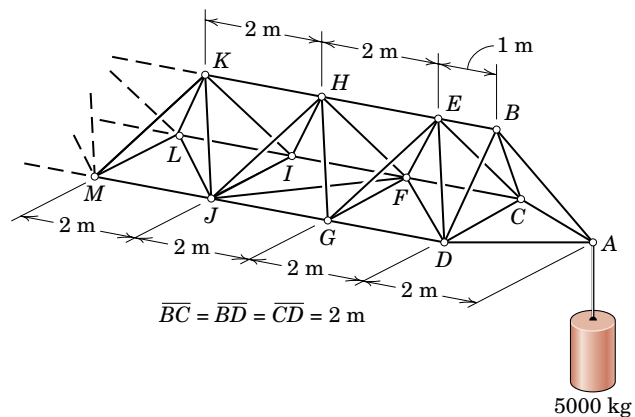
- 4/64** The pyramidal truss section $BCDEF$ is symmetric about the vertical x - z plane as shown. Cables AE , AF , and AB support a 5-kN load. Determine the force in member BE .



Problem 4/64

- 4/65** The lengthy boom of an overhead construction crane, a portion of which is shown, is an example of a periodic structure—one which is composed of repeated and identical structural units. Use the method of sections to find the forces in members FJ and GJ .

$$\text{Ans. } FJ = 0, GJ = -70.8 \text{ kN}$$

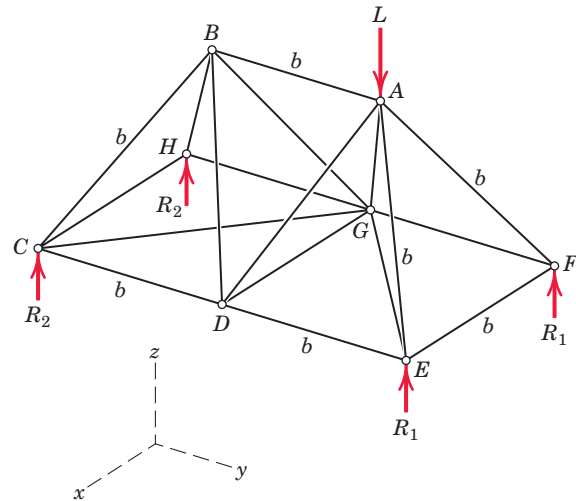


Problem 4/65

- 4/66 A space truss consists of two pyramids on identical square bases in the horizontal x - y plane with common side DG . The truss is loaded at the vertex A by the downward force L and is supported by the vertical reactions shown at its corners. All members except the two base diagonals are of the same length b . Take advantage of the two vertical planes of symmetry and determine the forces in AB and DA . (Note that link AB prevents the two pyramids from hinging about DG .)

$$\text{Ans. } AB = -\frac{\sqrt{2}L}{4}$$

$$DA = -\frac{\sqrt{2}L}{8}$$



Problem 4/66

4/6 FRAMES AND MACHINES

A structure is called a *frame* or *machine* if at least one of its individual members is a *multiforce member*. A multiforce member is defined as one with three or more forces acting on it, or one with two or more forces and one or more couples acting on it. Frames are structures which are designed to support applied loads and are usually fixed in position. Machines are structures which contain moving parts and are designed to transmit input forces or couples to output forces or couples.

Because frames and machines contain multiforce members, the forces in these members in general will *not* be in the directions of the members. Therefore, we cannot analyze these structures by the methods developed in Arts. 4/3, 4/4, and 4/5 because these methods apply to simple trusses composed of two-force members where the forces are in the directions of the members.

Interconnected Rigid Bodies with Multiforce Members

In Chapter 3 we discussed the equilibrium of multiforce bodies, but we concentrated on the equilibrium of a *single* rigid body. In this present article we focus on the equilibrium of *interconnected* rigid bodies which include multiforce members. Although most such bodies may be analyzed as two-dimensional systems, there are numerous examples of frames and machines which are three-dimensional.

The forces acting on each member of a connected system are found by isolating the member with a free-body diagram and applying the equations of equilibrium. The *principle of action and reaction* must be carefully observed when we represent the forces of interaction on the separate free-body diagrams. If the structure contains more members or supports than are necessary to prevent collapse, then, as in the case of trusses, the problem is statically indeterminate, and the principles of equilibrium, although necessary, are not sufficient for solution. Although many frames and machines are statically indeterminate, we will consider in this article only those which are statically determinate.

If the frame or machine constitutes a rigid unit by itself when removed from its supports, like the A-frame in Fig. 4/14a, the analysis is best begun by establishing all the forces external to the structure treated as a single rigid body. We then dismember the structure and consider the equilibrium of each part separately. The equilibrium equations for the several parts will be related through the terms involving

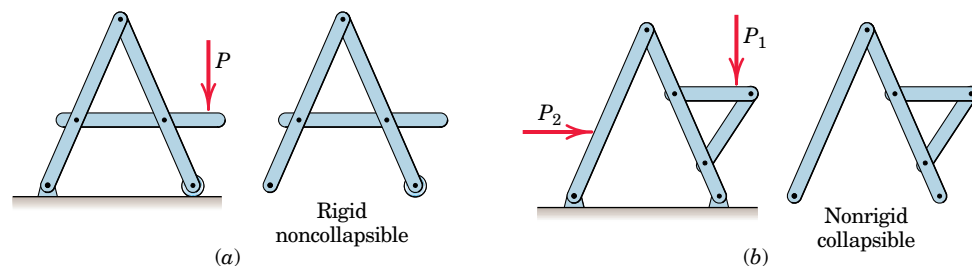


Figure 4/14

the forces of interaction. If the structure is not a rigid unit by itself but depends on its external supports for rigidity, as illustrated in Fig. 4/14b, then the calculation of the external support reactions cannot be completed until the structure is dismembered and the individual parts are analyzed.

Force Representation and Free-Body Diagrams

In most cases the analysis of frames and machines is facilitated by representing the forces in terms of their rectangular components. This is particularly so when the dimensions of the parts are given in mutually perpendicular directions. The advantage of this representation is that the calculation of moment arms is simplified. In some three-dimensional problems, particularly when moments are evaluated about axes which are not parallel to the coordinate axes, use of vector notation is advantageous.

It is not always possible to assign the proper sense to every force or its components when drawing the free-body diagrams, and it becomes necessary to make an arbitrary assignment. In any event, it is *absolutely necessary* that a force be *consistently* represented on the diagrams for interacting bodies which involve the force in question. Thus, for two bodies connected by the pin A, Fig. 4/15a, the force components must be consistently represented in *opposite* directions on the separate free-body diagrams.

For a ball-and-socket connection between members of a space frame, we must apply the action-and-reaction principle to all three components as shown in Fig. 4/15b. The assigned directions may prove to be wrong when the algebraic signs of the components are determined upon calculation. If A_x , for instance, should turn out to be negative, it is actually acting in the direction opposite to that originally represented. Accordingly, we would need to reverse the direction of the force on *both* members and to reverse the sign of its force terms in the equations. Or we may leave the representation as originally made, and the proper sense of the force will be understood from the negative sign. If we choose to use vector notation in labeling the forces, then we must be careful to use a plus sign for an action and a minus sign for the corresponding reaction, as shown in Fig. 4/16.

We may occasionally need to solve two or more equations simultaneously in order to separate the unknowns. In most instances, however, we can avoid simultaneous solutions by careful choice of the member or group of members for the free-body diagram and by a careful choice of moment axes which will eliminate undesired terms from the equations. The method of solution described in the foregoing paragraphs is illustrated in the following sample problems.

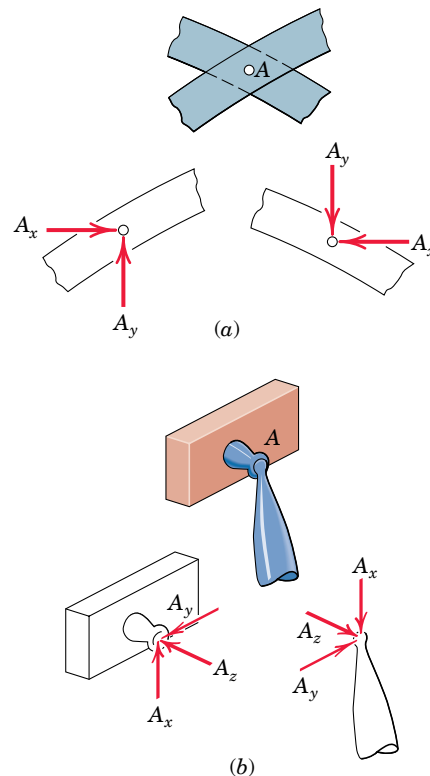


Figure 4/15

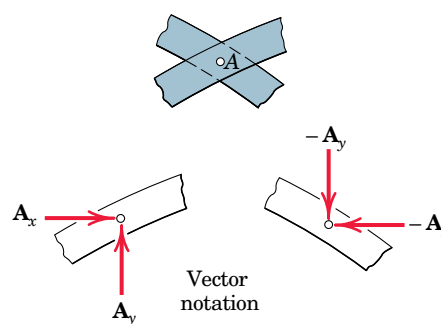


Figure 4/16

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Sample Problem 4/6

The frame supports the 400-kg load in the manner shown. Neglect the weights of the members compared with the forces induced by the load and compute the horizontal and vertical components of all forces acting on each of the members.

- Solution.** We observe first that the three supporting members which constitute the frame form a rigid assembly that can be analyzed as a single unit. We also observe that the arrangement of the external supports makes the frame statically determinate.

From the free-body diagram of the entire frame we determine the external reactions. Thus,

$$[\Sigma M_A = 0] \quad 5.5(0.4)(9.81) - 5D = 0 \quad D = 4.32 \text{ kN}$$

$$[\Sigma F_x = 0] \quad A_x - 4.32 = 0 \quad A_x = 4.32 \text{ kN}$$

$$[\Sigma F_y = 0] \quad A_y - 3.92 = 0 \quad A_y = 3.92 \text{ kN}$$

Next we dismember the frame and draw a separate free-body diagram of each member. The diagrams are arranged in their approximate relative positions to aid in keeping track of the common forces of interaction. The external reactions just obtained are entered onto the diagram for AD . Other known forces are the 3.92-kN forces exerted by the shaft of the pulley on the member BF , as obtained from the free-body diagram of the pulley. The cable tension of 3.92 kN is also shown acting on AD at its attachment point.

- Next, the components of all unknown forces are shown on the diagrams.
- ② Here we observe that CE is a two-force member. The force components on CE have equal and opposite reactions, which are shown on BF at E and on AD at C . We may not recognize the actual sense of the components at B at first glance, so they may be arbitrarily but consistently assigned.

The solution may proceed by use of a moment equation about B or E for member BF , followed by the two force equations. Thus,

$$[\Sigma M_B = 0] \quad 3.92(5) - \frac{1}{2}E_x(3) = 0 \quad E_x = 13.08 \text{ kN} \quad \text{Ans.}$$

$$[\Sigma F_y = 0] \quad B_y + 3.92 - 13.08/2 = 0 \quad B_y = 2.62 \text{ kN} \quad \text{Ans.}$$

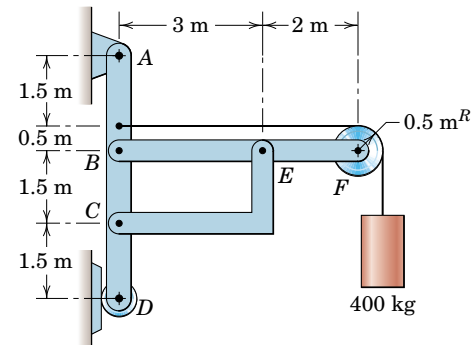
$$[\Sigma F_x = 0] \quad B_x + 3.92 - 13.08 = 0 \quad B_x = 9.15 \text{ kN} \quad \text{Ans.}$$

Positive numerical values of the unknowns mean that we assumed their directions correctly on the free-body diagrams. The value of $C_x = E_x = 13.08$ kN obtained by inspection of the free-body diagram of CE is now entered onto the diagram for AD , along with the values of B_x and B_y just determined. The equations of equilibrium may now be applied to member AD as a check, since all the forces acting on it have already been computed. The equations give

$$[\Sigma M_C = 0] \quad 4.32(3.5) + 4.32(1.5) - 3.92(2) - 9.15(1.5) = 0$$

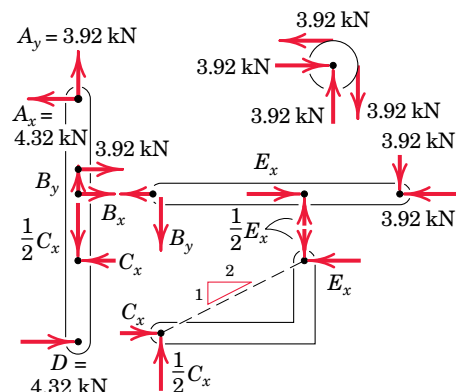
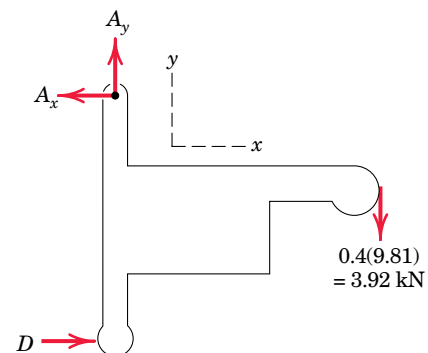
$$[\Sigma F_x = 0] \quad 4.32 - 13.08 + 9.15 + 3.92 + 4.32 = 0$$

$$[\Sigma F_y = 0] \quad -13.08/2 + 2.62 + 3.92 = 0$$



Helpful Hints

- ① We see the frame corresponds to the category illustrated in Fig. 4/14a.
- ② Without this observation, the problem solution would be much longer, because the three equilibrium equations for member BF would contain four unknowns: B_x , B_y , E_x , and E_y . Note that the direction of the line joining the two points of force application, and not the shape of the member, determines the direction of the forces acting on a two-force member.



Sample Problem 4/7

Neglect the weight of the frame and compute the forces acting on all of its members.

- Solution.** We note first that the frame is not a rigid unit when removed from its supports since $BDEF$ is a movable quadrilateral and not a rigid triangle. Consequently the external reactions cannot be completely determined until the individual members are analyzed. However, we can determine the vertical components of the reactions at A and C from the free-body diagram of the frame as a whole. Thus,

$$[\Sigma M_C = 0] \quad 200(0.3) + 120(0.1) - 0.750A_y = 0 \quad A_y = 240 \text{ N} \quad \text{Ans.}$$

$$[\Sigma F_y = 0] \quad C_y - 200(4/5) - 240 = 0 \quad C_y = 400 \text{ N} \quad \text{Ans.}$$

Next we dismember the frame and draw the free-body diagram of each part. Since EF is a two-force member, the direction of the force at E on ED and at F on AB is known. We assume that the 120-N force is applied to the pin as a part of member BC . There should be no difficulty in assigning the correct directions for forces E , F , D , and B_x . The direction of B_y , however, may not be assigned by inspection and therefore is arbitrarily shown as downward on AB and upward on BC .

- Member ED .** The two unknowns are easily obtained by

$$[\Sigma M_D = 0] \quad 200(0.3) - 0.3E = 0 \quad E = 200 \text{ N} \quad \text{Ans.}$$

$$[\Sigma F = 0] \quad D - 200 - 200 = 0 \quad D = 400 \text{ N} \quad \text{Ans.}$$

Member EF . Clearly F is equal and opposite to E with the magnitude of 200 N.

Member AB . Since F is now known, we solve for B_x , A_x , and B_y from

$$[\Sigma M_A = 0] \quad 200(3/5)(0.5) - B_x(1.0) = 0 \quad B_x = 60 \text{ N} \quad \text{Ans.}$$

$$[\Sigma F_x = 0] \quad A_x + 60 - 200(3/5) = 0 \quad A_x = 60 \text{ N} \quad \text{Ans.}$$

$$[\Sigma F_y = 0] \quad 200(4/5) - 240 - B_y = 0 \quad B_y = -80 \text{ N} \quad \text{Ans.}$$

The minus sign shows that we assigned B_y in the wrong direction.

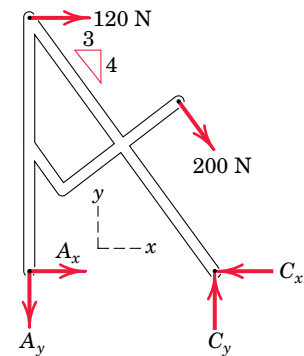
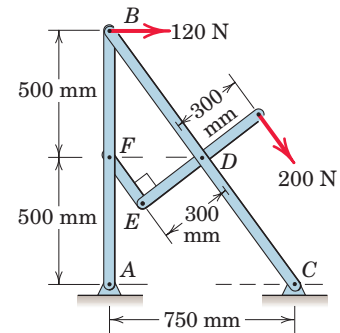
- Member BC .** The results for B_x , B_y , and D are now transferred to BC , and the remaining unknown C_x is found from

$$[\Sigma F_x = 0] \quad 120 + 400(3/5) - 60 - C_x = 0 \quad C_x = 300 \text{ N} \quad \text{Ans.}$$

We may apply the remaining two equilibrium equations as a check. Thus,

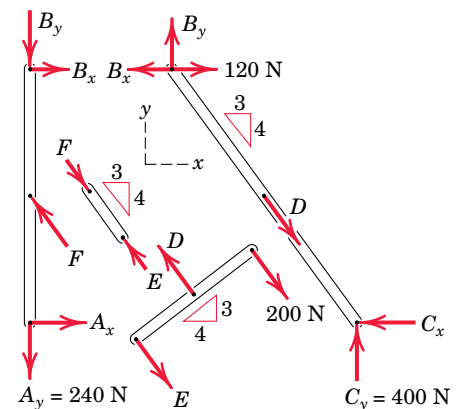
$$[\Sigma F_y = 0] \quad 400 + (-80) - 400(4/5) = 0$$

$$[\Sigma M_C = 0] \quad (120 - 60)(1.0) + (-80)(0.75) = 0$$



Helpful Hints

- ① We see that this frame corresponds to the category illustrated in Fig. 4/14b.
- ② The directions of A_x and C_x are not obvious initially and can be assigned arbitrarily to be corrected later if necessary.
- ③ Alternatively the 120-N force could be applied to the pin considered a part of BA , with a resulting change in the reaction B_x .

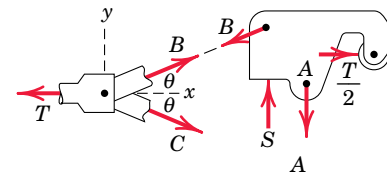
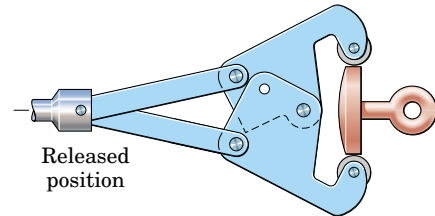
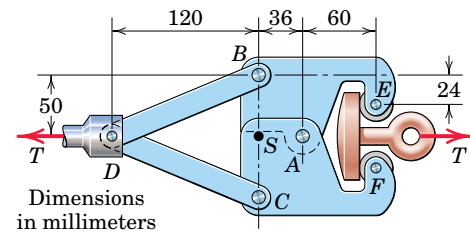


- ④ Alternatively we could have returned to the free-body diagram of the frame as a whole and found C_x .

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Sample Problem 4/8

The machine shown is designed as an overload protection device which releases the load when it exceeds a predetermined value T . A soft metal shear pin S is inserted in a hole in the lower half and is acted on by the upper half. When the total force on the pin exceeds its strength, it will break. The two halves then rotate about A under the action of the tensions in BD and CD , as shown in the second sketch, and rollers E and F release the eye bolt. Determine the maximum allowable tension T if the pin S will shear when the total force on it is 800 N. Also compute the corresponding force on the hinge pin A .



- Solution.** Because of symmetry we analyze only one of the two hinged members. The upper part is chosen, and its free-body diagram along with that for the connection at D is drawn. Because of symmetry the forces at S and A have no x -components. The two-force members BD and CD exert forces of equal magnitude $B = C$ on the connection at D . Equilibrium of the connection gives

$$[\Sigma F_x = 0] \quad B \cos \theta + C \cos \theta - T = 0 \quad 2B \cos \theta = T$$

$$B = T/(2 \cos \theta)$$

From the free-body diagram of the upper part we express the equilibrium of moments about point A . Substituting $S = 800$ N and the expression for B gives

$$② \quad [\Sigma M_A = 0] \quad \frac{T}{2 \cos \theta} (\cos \theta)(50) + \frac{T}{2 \cos \theta} (\sin \theta)(36) - 36(800) - \frac{T}{2} (26) = 0$$

Substituting $\sin \theta / \cos \theta = \tan \theta = 5/12$ and solving for T give

$$T \left(25 + \frac{5(36)}{2(12)} - 13 \right) = 28\,800$$

$$T = 1477 \text{ N} \quad \text{or} \quad T = 1.477 \text{ kN} \quad \text{Ans.}$$

Finally, equilibrium in the y -direction gives us

$$[\Sigma F_y = 0] \quad S - B \sin \theta - A = 0$$

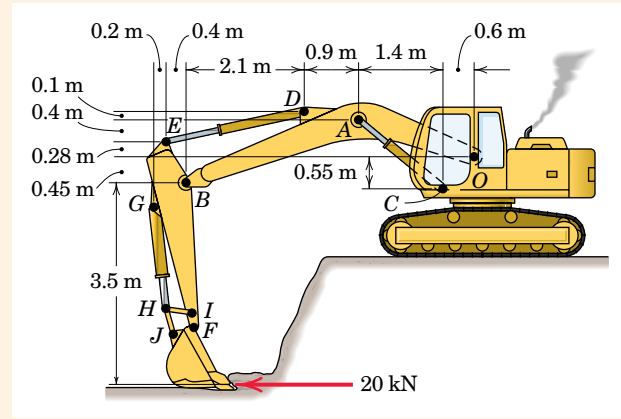
$$800 - \frac{1477}{2(12/13)} \frac{5}{13} - A = 0 \quad A = 492 \text{ N} \quad \text{Ans.}$$

Helpful Hints

- ① It is always useful to recognize symmetry. Here it tells us that the forces acting on the two parts behave as mirror images of each other with respect to the x -axis. Thus, we cannot have an action on one member in the plus x -direction and its reaction on the other member in the negative x -direction. Consequently the forces at S and A have no x -components.
- ② Be careful not to forget the moment of the y component of B . Note that our units here are newton-millimeters.

Sample Problem 4/9

In the particular position shown, the excavator applies a 20-kN force parallel to the ground. There are two hydraulic cylinders AC to control the arm OAB and a single cylinder DE to control arm $EBIF$. (a) Determine the force in the hydraulic cylinders AC and the pressure p_{AC} against their pistons, which have an effective diameter of 95 mm. (b) Also determine the force in hydraulic cylinder DE and the pressure p_{DE} against its 105-mm-diameter piston. Neglect the weights of the members compared with the effects of the 20-kN force.



Solution. (a) We begin by constructing a free-body diagram of the entire arm assembly. Note that we include only the dimensions necessary for this portion of the problem—details of the cylinders DE and GH are unnecessary at this time.

$$[\Sigma M_O = 0] \quad -20\,000(3.95) - 2F_{AC} \cos 41.3^\circ(0.68) + 2F_{AC} \sin 41.3^\circ(2) = 0$$

$$F_{AC} = 48\,800 \text{ N or } 48.8 \text{ kN}$$

Ans.

Helpful Hint

① From $F_{AC} = p_{AC}A_{AC}$, $p_{AC} = \frac{F_{AC}}{A_{AC}} = \frac{48\,800}{\left(\pi \frac{0.095^2}{4}\right)} = 6.89(10^6) \text{ Pa or } 6.89 \text{ MPa}$ Ans.

① Recall that force = (pressure)(area).

(b) For cylinder DE , we “cut” the assembly at a location which makes the desired cylinder force external to our free-body diagram. This means isolating the vertical arm $EBIF$ along with the bucket and its applied force.

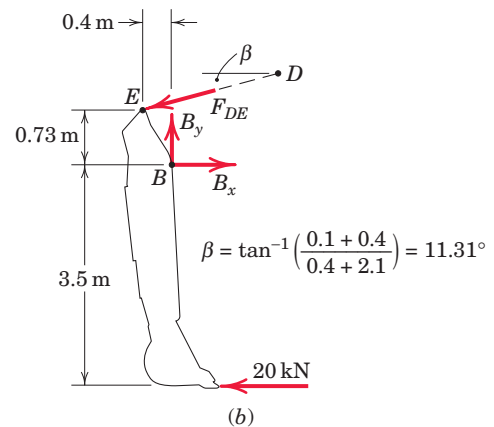
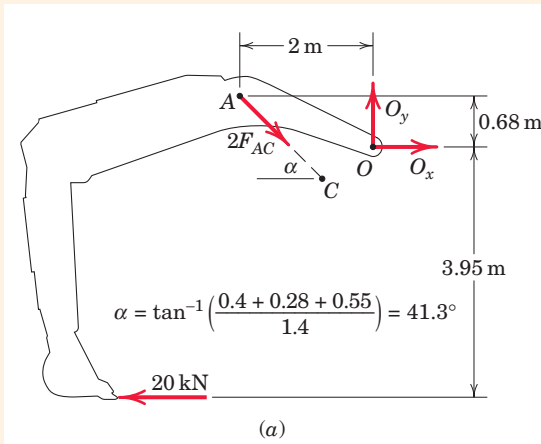
$$[\Sigma M_B = 0] \quad -20\,000(3.5) + F_{DE} \cos 11.31^\circ(0.73) + F_{DE} \sin 11.31^\circ(0.4) = 0$$

$$F_{DE} = 88\,100 \text{ N or } 88.1 \text{ kN}$$

Ans.

$$p_{DE} = \frac{F_{DE}}{A_{DE}} = \frac{88\,100}{\left(\pi \frac{0.105^2}{4}\right)} = 10.18(10^6) \text{ Pa or } 10.18 \text{ MPa}$$

Ans.



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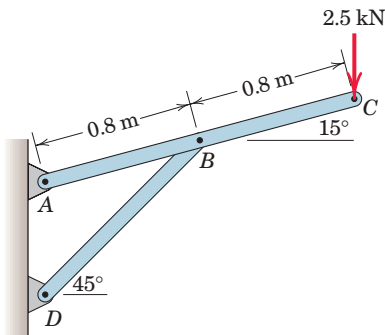
PROBLEMS

Introductory Problems

(Unless otherwise instructed, neglect the mass of the various members and all friction in the problems which follow.)

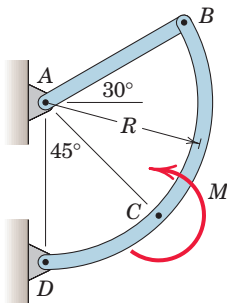
- 4/67** Determine the force in member BD and the forces supported by all pins in the loaded frame.

Ans. $BD = 9.66 \text{ kN}$, $A = 8.09 \text{ kN}$
 $B = D = 9.66 \text{ kN}$



Problem 4/67

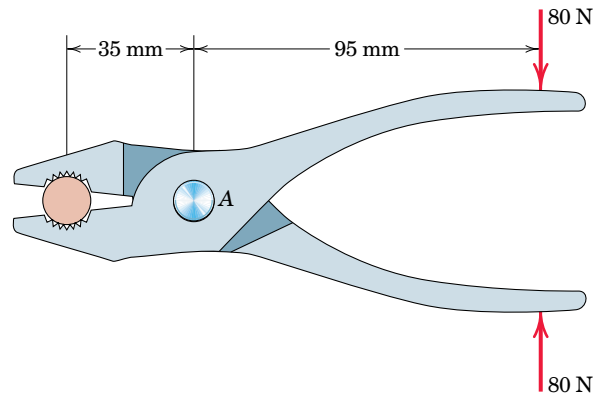
- 4/68** Determine the force supported by each pin of the loaded truss.



Problem 4/68

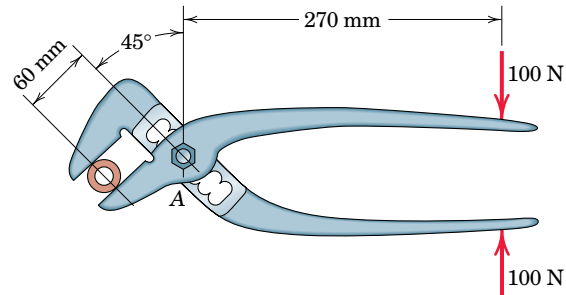
- 4/69** For an 80-N squeeze on the handles of the pliers, determine the force F applied to the round rod by each jaw. In addition, calculate the force supported by the pin at A.

Ans. $F = 217 \text{ N}$, $A = 297 \text{ N}$



Problem 4/69

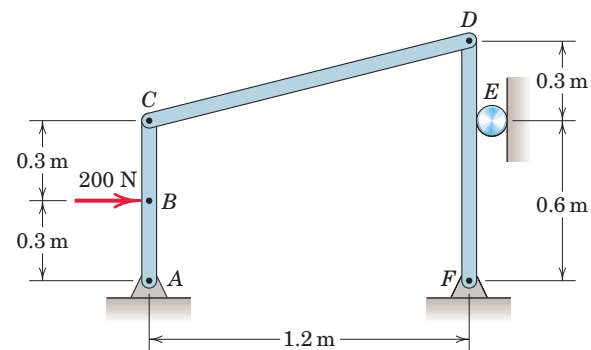
- 4/70** Compute the force supported by the pin at A for the slip-joint pliers under a grip of 100 N.



Problem 4/70

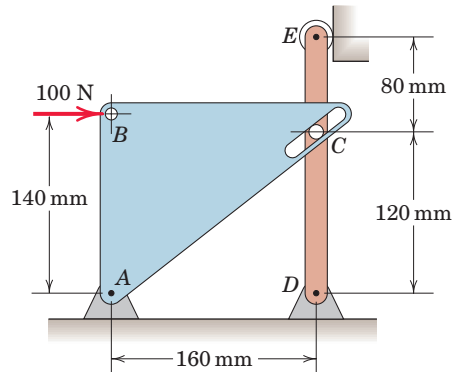
- 4/71** Determine the force supported by the roller at E.

Ans. $E = 150 \text{ N}$



Problem 4/71

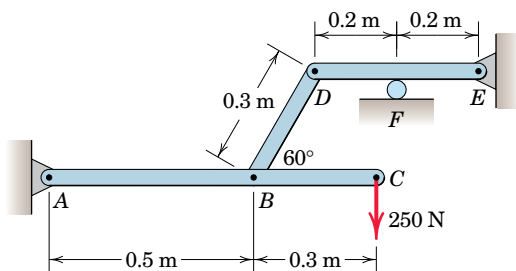
- 4/72** Calculate the magnitude of the force acting on the pin at D . Pin C is fixed in DE and bears against the smooth slot in the triangular plate.



Problem 4/72

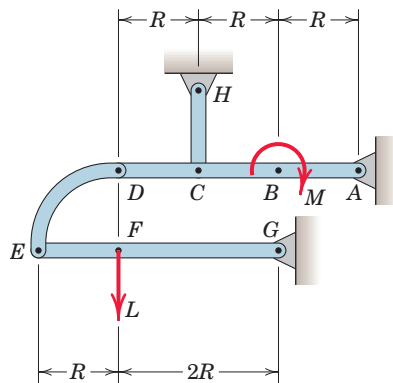
- 4/73** Determine the reaction at the roller F for the frame loaded as shown.

Ans. $F = 800$ N



Problem 4/73

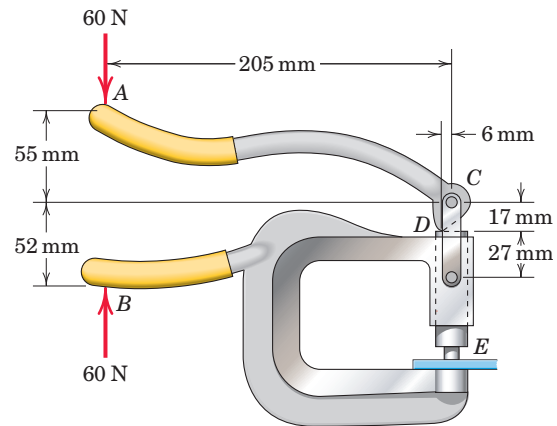
- 4/74** Given the values of the load L and dimension R , for what value of the couple M will the force in link CH be zero?



Problem 4/74

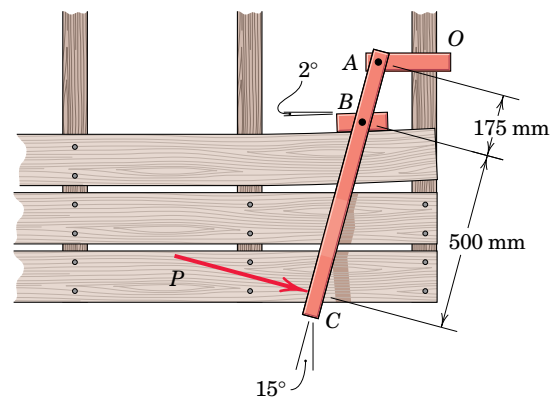
- 4/75** The handheld press is useful for such tasks as squeezing rivets or punching holes. What force P is applied to the sheet metal at E for the 60-N forces applied to the handles?

Ans. $P = 2050$ N



Problem 4/75

- 4/76** The device shown is used to straighten bowed decking boards just prior to final nailing to the joists. There is a lower bracket (not shown) at O which fixes the part OA to a joist, so that the pivot A may be considered fixed. For a given force P exerted perpendicular to the handle ABC as shown, determine the corresponding normal force N applied to the bent board near point B . Neglect friction.

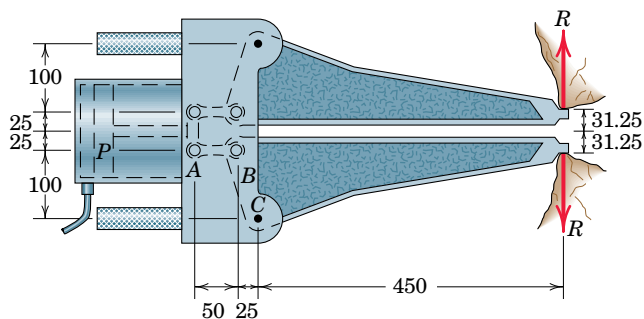


Problem 4/76

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4/77 The “jaws-of-life” device is utilized by rescuers to pry apart wreckage, thus helping to free accident victims. If a pressure of 3.5 MPa is developed behind the piston P of area $13(10^3)$ mm², determine the vertical force R which is exerted by the jaw tips on the wreckage for the position shown. Note that link AB and its counterpart are both horizontal in the figure for this position.

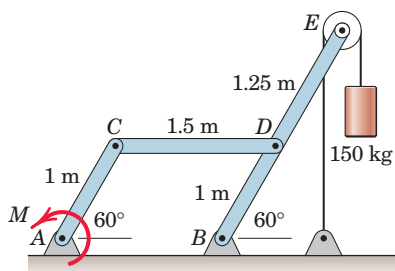
Ans. $R = 5.06 \text{ kN}$



Dimensions in millimeters

Problem 4/77

4/78 Determine the moment M which must be applied at A to keep the frame in static equilibrium in the position shown. Also calculate the magnitude of the pin reaction at A .

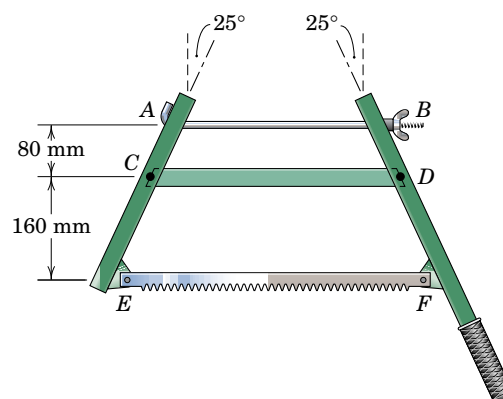


Problem 4/78

Representative Problems

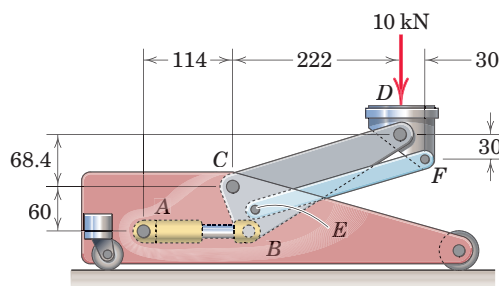
4/79 The wingnut B of the collapsible bucksaw is tightened until the tension in rod AB is 200 N. Determine the force in the saw blade EF and the magnitude F of the force supported by pin C .

Ans. $EF = 100 \text{ N T}, F = 300 \text{ N}$



Problem 4/79

4/80 The elements of a floor jack are shown in the figure. The figure $CDFE$ is a parallelogram. Calculate the force in the hydraulic cylinder AB corresponding to the 10-kN load supported as shown. What is the force in link EF ?

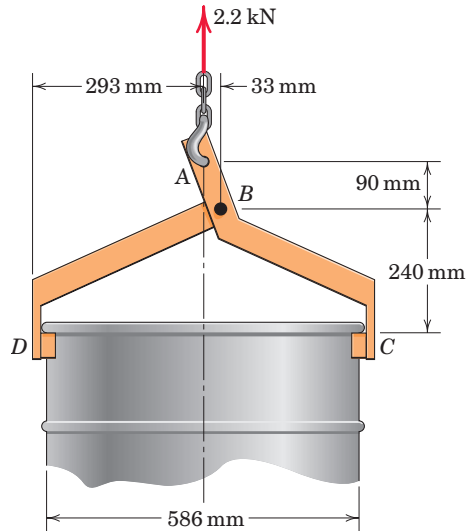


Dimensions in millimeters

Problem 4/80

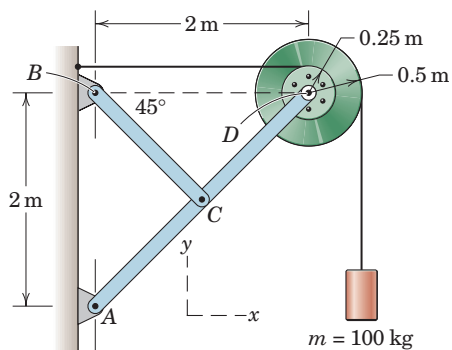
- 4/81** The device shown is used for lifting 55-gallon drums. Determine the magnitude of the force exerted at B .

Ans. $B = 1.855 \text{ kN}$



Problem 4/81

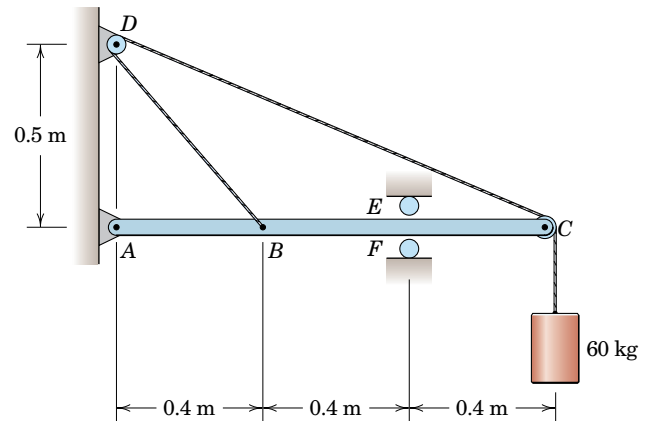
- 4/82** Calculate the x - and y -components of the force \mathbf{C} which member BC exerts on member ACD . The cables are wrapped securely around the two pulleys, which are fastened together.



Problem 4/82

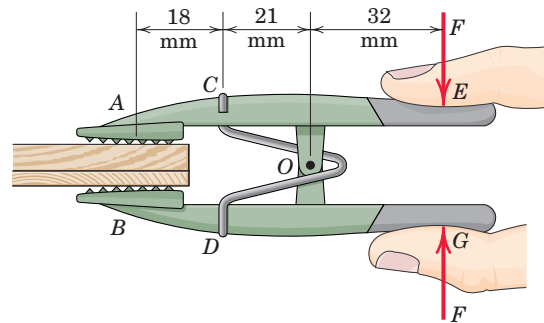
- 4/83** Determine the magnitude of the pin reaction at A and the magnitude and direction of the force reaction at the rollers. The pulleys at C and D are small.

Ans. $A = 999 \text{ N}$, $F = 314 \text{ N up}$



Problem 4/83

- 4/84** If a force $F = 15 \text{ N}$ is required to release the spring-loaded clamps, what are the normal reactions at A and B if $F = 0$?

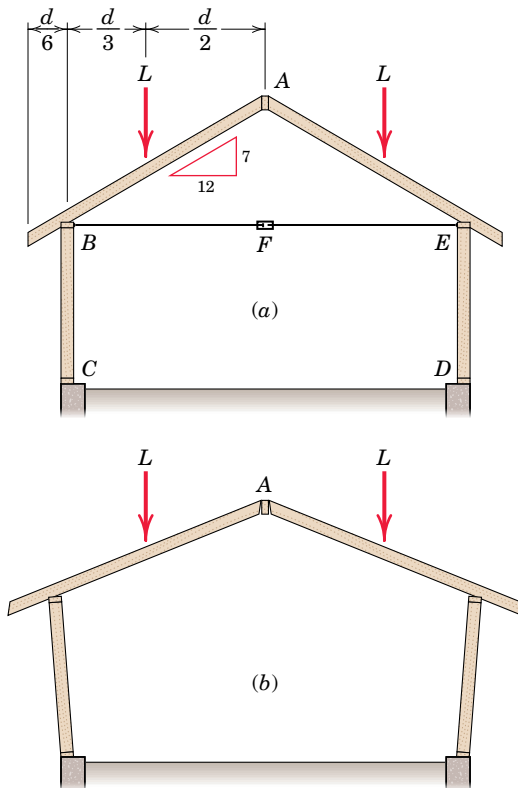


Problem 4/84

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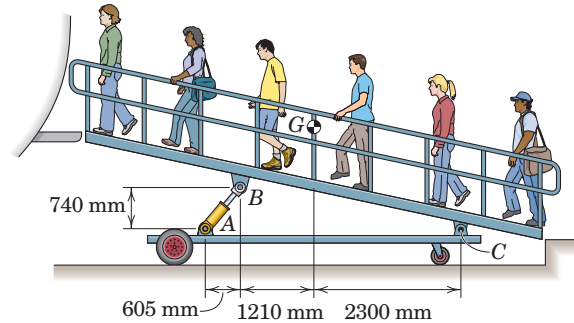
4/85 The figure illustrates a common problem associated with simple structures. Under the loadings L , the rafters can rotate, the ridge beam at A can lower, and the walls BC and DE can rotate outward, as shown in part b of the figure. This phenomenon is sometimes clearly observed in old wooden farm structures as a central sagging of the ridge beam when viewed from the side. A simple remedy is shown in part a of the figure. A chain or cable is stretched between fasteners at B and E , and the turnbuckle F is tightened until a proper tension is achieved, thereby preventing the outward tilting of the walls. For given values of the dimension d and the point loads L (which result from the distributed loads of the rafter and roofing weights and any additional loads such as snow), calculate the tension T required so that there are no outward forces on the walls at B and E . Assume that the support of the rafters at the ridge beam is purely horizontal and that all joints are free to rotate.

$$\text{Ans. } T = \frac{24}{35}L$$



Problem 4/85

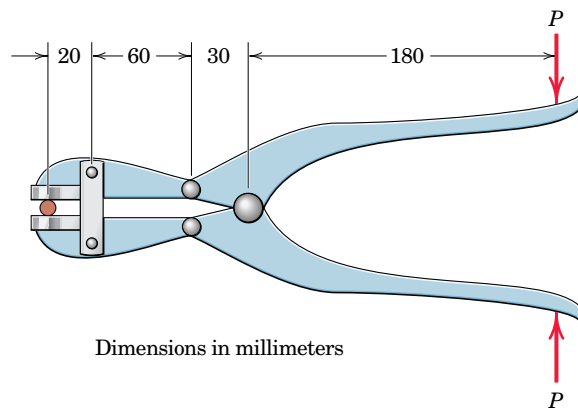
4/86 The ramp is used as passengers board a small commuter airplane. The total mass of the ramp and six passengers is 750 kg with mass center at G . Determine the force in the hydraulic cylinder AB and the magnitude of the pin reaction at C .



Problem 4/86

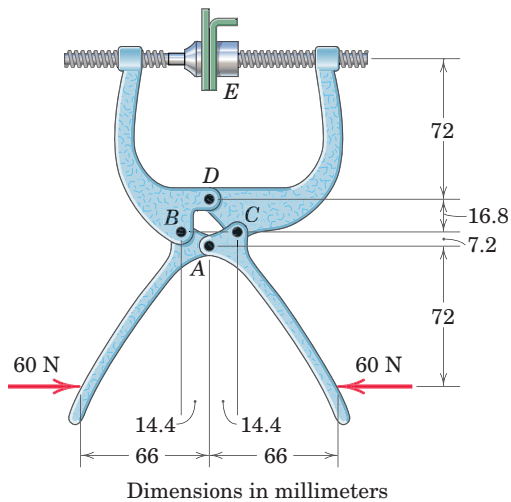
4/87 A small bolt cutter operated by hand for cutting small bolts and rods is shown in the sketch. For a hand grip $P = 150$ N, determine the force Q developed by each jaw on the rod to be cut.

$$\text{Ans. } Q = 2.7 \text{ kN}$$



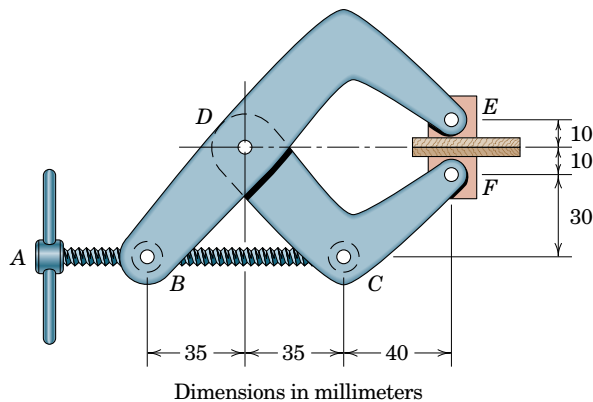
Problem 4/87

- 4/88** The clamp shown in the figure is frequently used in welding operations. Determine the clamping force on the two metal pieces at E and the magnitudes of the forces supported by pins A , B , and D .



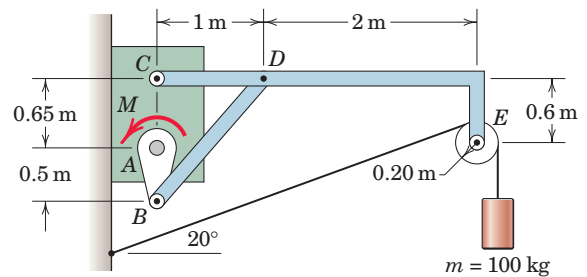
Problem 4/88

- 4/89** The clamp is adjusted so that it exerts a pair of 200-N compressive forces on the boards between its swivel grips. Determine the force in the threaded shaft BC and the magnitude of the pin reaction at D .
Ans. $BC = 375 \text{ N}$, $D = 425 \text{ N}$



Problem 4/89

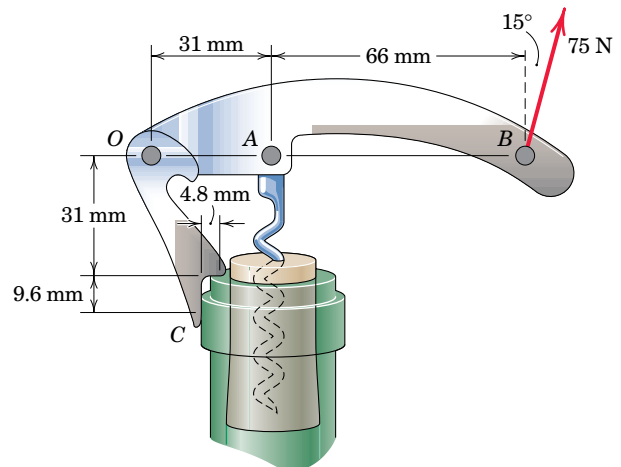
- 4/90** When the crank AB is vertical, the beam CD is horizontal and the cable makes a 20° angle with the horizontal. Compute the moment M required for equilibrium of the frame.



Problem 4/90

- 4/91** A 75-N force is applied to the handle OAB of the cork puller. Determine the extraction force F exerted on the cork.

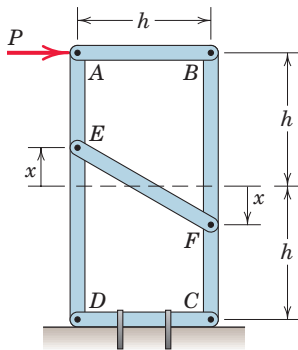
Ans. $F = 227 \text{ N}$



Problem 4/91

216 Chapter 4 Structures

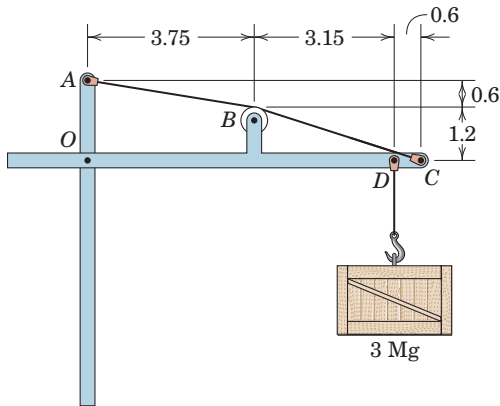
- 4/92** The illustration represents the framework for a storage-shelf unit, with the intermediate shelves not shown. The variable x represents different bracing configurations. Determine and plot, over the range $0 \leq \frac{x}{h} \leq 1$, the force in the brace EF corresponding to the applied force P . What happens if $x = 0$?



Problem 4/92

- 4/93** The simple crane supports the 3-Mg load. Determine the tension T in the cable and the magnitude of the pin reaction at O .

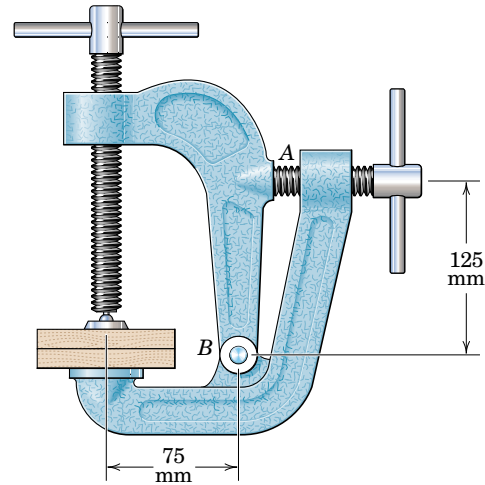
Ans. $T = 114.2 \text{ kN}$, $O = 113.4 \text{ kN}$



Dimensions in meters

Problem 4/93

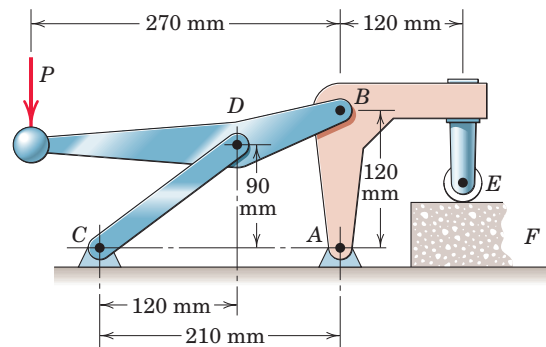
- 4/94** The dual-grip clamp shown in the figure is used to provide added clamping force with a positive action. If the vertical screw is tightened to produce a clamping force of 3 kN and then the horizontal screw is tightened until the force in the screw at A is doubled, find the total reaction R on the pin at B .



Problem 4/94

- 4/95** Determine the vertical clamping force at E in terms of the force P applied to the handle of the toggle clamp, which holds the workpiece F in place.

Ans. $E = 7.2P$

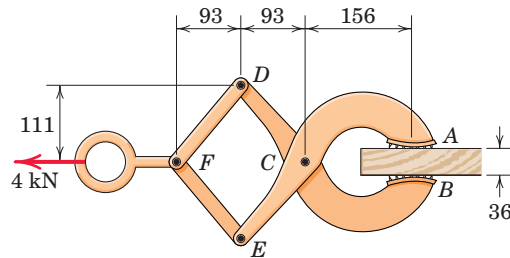


Problem 4/95

218 Chapter 4 Structures

- 4/101** The device shown is used to drag loaded wooden pallets across warehouse floors. The wood board shown is one of several members that comprise the base of the pallet. For the 4-kN force applied by a forklift, determine the magnitude of the force supported by pin C and the normal gripping forces at A and B .

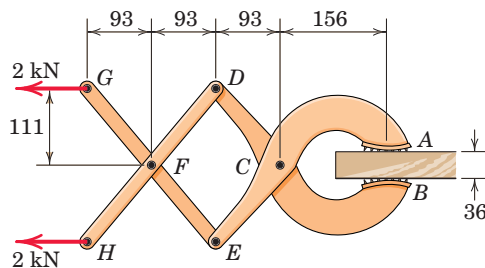
Ans. $C = 5.46 \text{ kN}$, $A_n = B_n = 3.08 \text{ kN}$



Dimensions in millimeters

Problem 4/101

- 4/102** A modification of the pallet puller of Prob. 4/101 is shown here. For the same net 4-kN force as in Prob. 4/101, determine the magnitude of the force supported by pin C and the normal gripping forces at A and B .

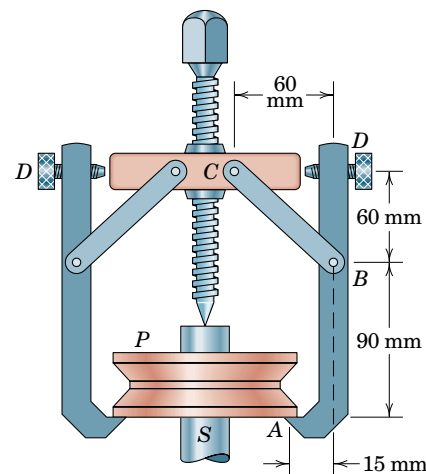


Dimensions in millimeters

Problem 4/102

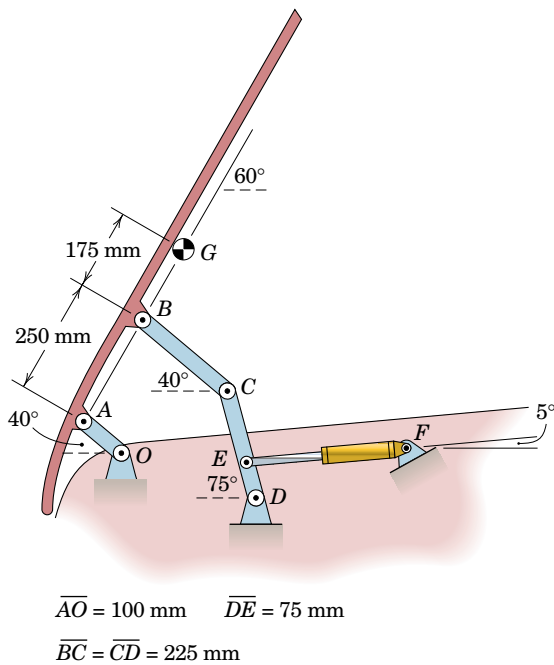
- 4/103** The figure shows a wheel puller which is designed to remove a V-belt pulley P from its tight-fitting shaft S by tightening of the central screw. If the pulley starts to slide off the shaft when the compression in the screw has reached 1.2 kN, calculate the magnitude of the force supported by each jaw at A . The adjusting screws D support horizontal force and keep the side arms parallel with the central screw.

Ans. $A = 0.626 \text{ kN}$



Problem 4/103

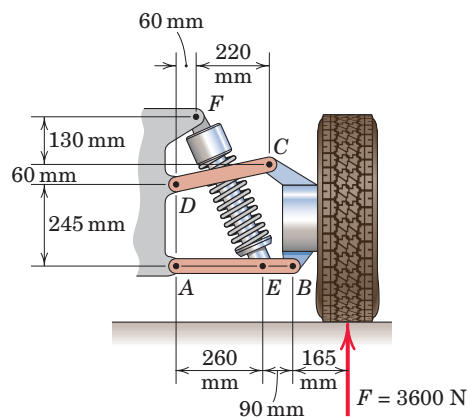
- 4/104** The elements of a front-hinged automobile-hood assembly are shown in the figure. The light linkages BC and CD and the gas-pressurized strut EF hold the hood in the open position shown. In this position, the hood is free to rotate clockwise about pin O ; pin A is locked until the hood has been lowered to a nearly closed horizontal position. For a hood mass of 40 kg with center of mass at G , determine the minimum compression force C in the strut which will maintain the open-hood position. Note that there are two links OA spaced across the front of the car, but only one set of the remaining links, located on the inside of the right-front fender.



Problem 4/104

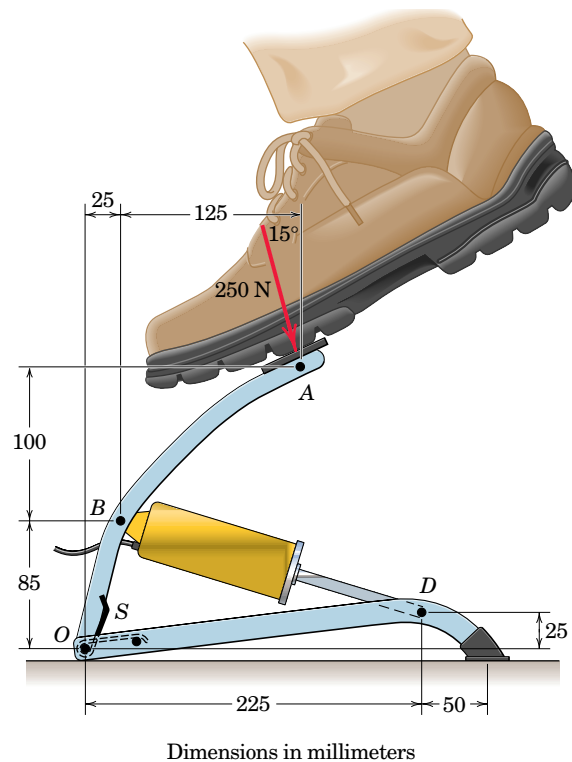
- 4/105** The elements of a rear suspension for a front-wheel-drive car are shown in the figure. Determine the magnitude of the force at each joint if the normal force F exerted on the tire has a magnitude of 3600 N.

Ans. $A = 4550 \text{ N}$, $B = 4410 \text{ N}$
 $C = D = 1898 \text{ N}$, $E = F = 5920 \text{ N}$



Problem 4/105

- 4/106** A 250-N force is applied to the foot-operated air pump. The return spring S exerts a 3-N·m moment on member OBA for this position. Determine the corresponding compression force C in the cylinder BD . If the diameter of the piston in the cylinder is 45 mm, estimate the air pressure generated for these conditions. State any assumptions.

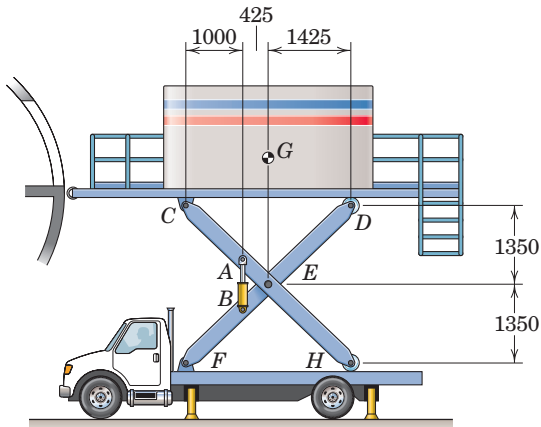


Problem 4/106

220 Chapter 4 Structures

- 4/107** The truck shown is used to deliver food to aircraft. The elevated unit has a mass of 1000 kg with center of mass at G . Determine the required force in the hydraulic cylinder AB .

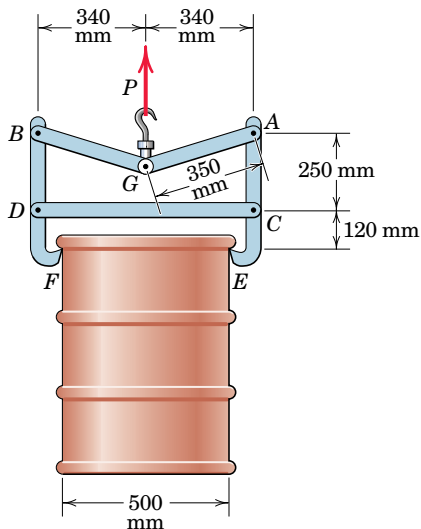
Ans. $F_{AB} = 32.9 \text{ kN}$



Dimensions in millimeters

Problem 4/107

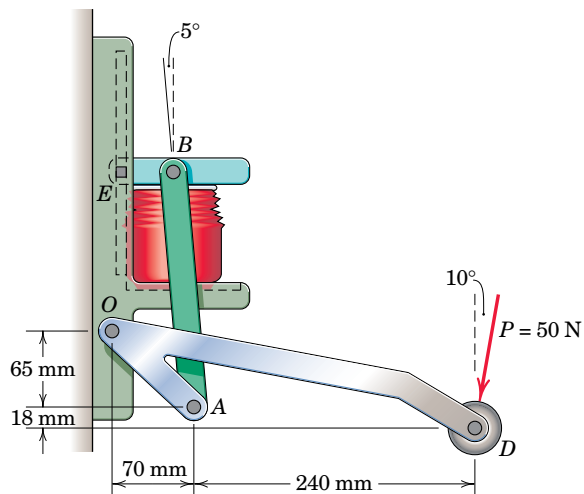
- 4/108** A lifting device for transporting 135-kg steel drums is shown. Calculate the magnitude of the force exerted on the drum at E and F .



Problem 4/108

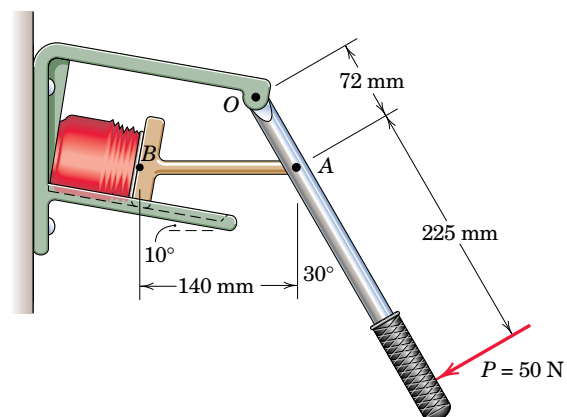
- 4/109** Determine the compression force C exerted on the can for an applied force $P = 50 \text{ N}$ when the can crusher is in the position shown. Note that there are two links AB and two links AOD , with one pair of linkages on each side of the stationary portion of the crusher. Also, pin B is on the vertical centerline of the can. Finally, note that small square projections E of the moving jaw move in recessed slots of the fixed frame.

Ans. $C = 249 \text{ N}$



Problem 4/109

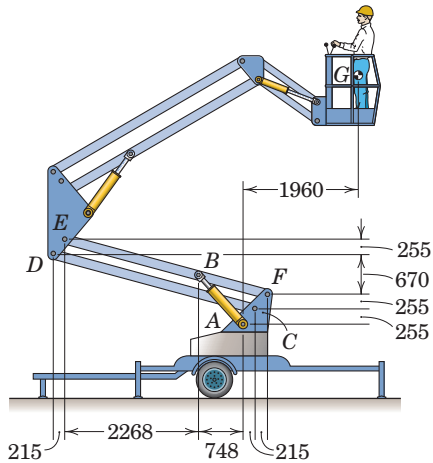
- 4/110** Determine the compression force C exerted on the can for an applied force $P = 50 \text{ N}$ when the can crusher is in the position shown. Point B is centered on the bottom of the can.



Problem 4/110

- 4/111** Determine the force in cylinder AB due to the combined weight of the bucket and operator. The combined mass is 180 kg with mass center at G .

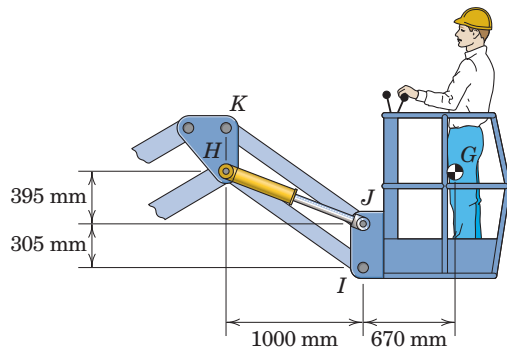
Ans. $F_{AB} = 9200 \text{ N}$



Dimensions in millimeters

Problem 4/111

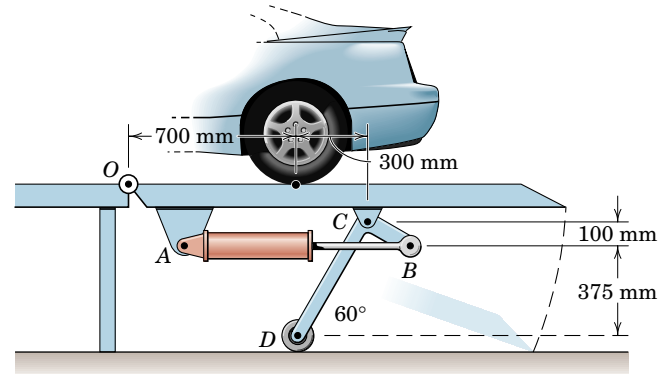
- 4/112** Additional detail in the bucket area of the cherry-picker of Prob. 4/111 is given in the figure. Solve for the force in cylinder HJ due to the effects of the 180-kg combined mass of the bucket and operator, with mass center at G .



Problem 4/112

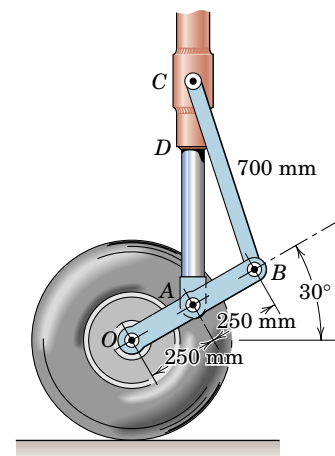
- 4/113** The car hoist allows the car to be driven onto the platform, after which the rear wheels are raised. If the loading from both rear wheels is 6 kN, determine the force in the hydraulic cylinder AB . Neglect the weight of the platform itself. Member BCD is a right-angle bell crank pinned to the ramp at C .

Ans. $AB = 15.87 \text{ kN C}$



Problem 4/113

- 4/114** The aircraft landing gear consists of a spring- and hydraulically-loaded piston and cylinder D and the two pivoted links OB and CB . If the gear is moving along the runway at a constant speed with the wheel supporting a stabilized constant load of 24 kN, calculate the total force which the pin at A supports.

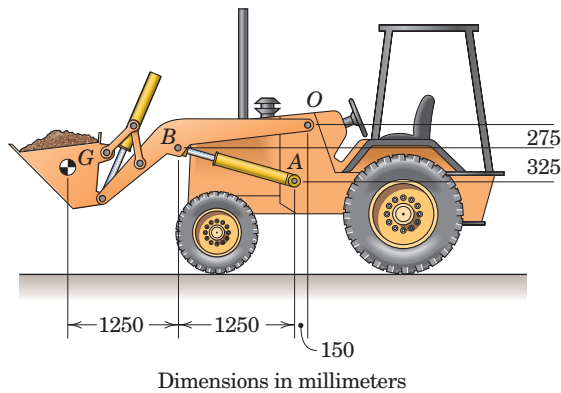


Problem 4/114

222 Chapter 4 Structures

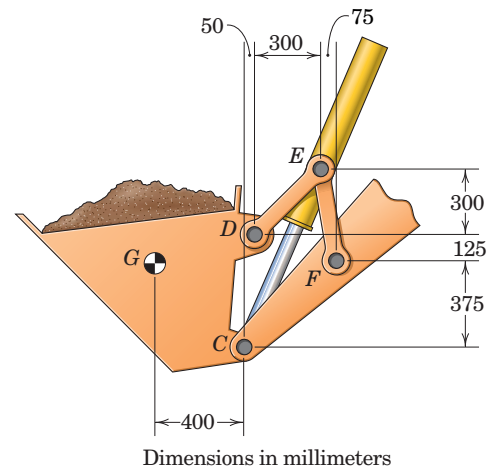
- 4/115** Determine the force in the hydraulic cylinder AB and the magnitude of the pin reaction at O for the position shown. The bucket and its load have a combined mass of 2000 kg with center of mass at G . You may neglect the effect of the weights of the other members.

Ans. $F_{AB} = 84.1 \text{ kN}$, $O = 81.4 \text{ kN}$



Problem 4/115

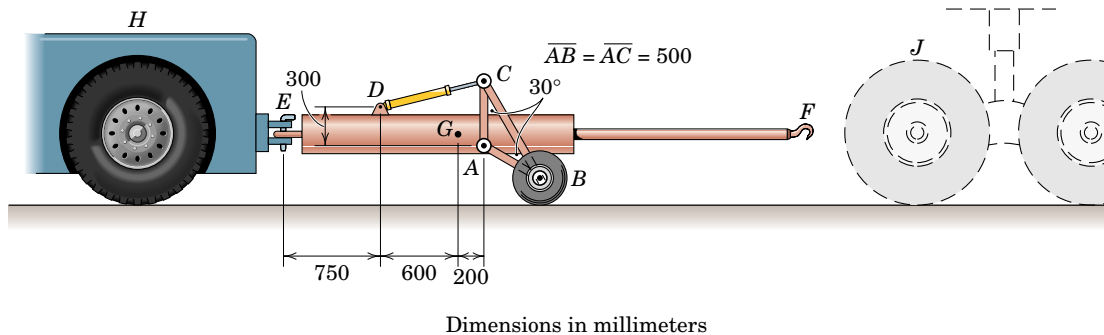
- 4/116** Consider the additional dimensional detail for the front-end loader of Prob. 4/115. Determine the force in the hydraulic cylinder CE . The mass of the bucket and its load is 2000 kg with center of mass at G . You may ignore the effects of the weights of other members.



Problem 4/116

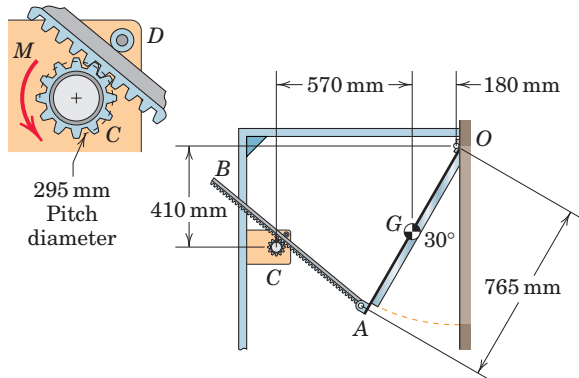
- 4/117** An adjustable tow bar connecting the tractor unit H with the landing gear J of a large aircraft is shown in the figure. Adjusting the height of the hook F at the end of the tow bar is accomplished by the hydraulic cylinder CD activated by a small hand pump (not shown). For the nominal position shown of the triangular linkage ABC , calculate the force P supplied by the cylinder to the pin C to position the tow bar. The rig has a total mass of 50 kg and is supported by the tractor hitch at E .

Ans. $P = 298 \text{ N}$



Problem 4/117

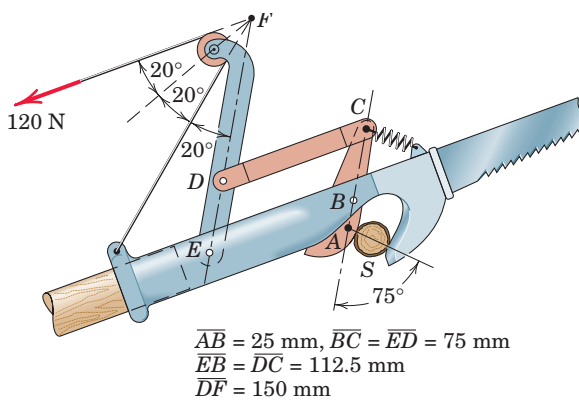
- 4/118** An automatic window-positioning system for a greenhouse is shown in the figure. Window OA has a mass of 50 kg with mass center at G . A motor (not shown) drives the gear C , which in turn moves the positioning rack AB . For the window position shown, determine the motor torque M which must be supplied to the pinion C . Neglect the weight of bar AB .



Problem 4/118

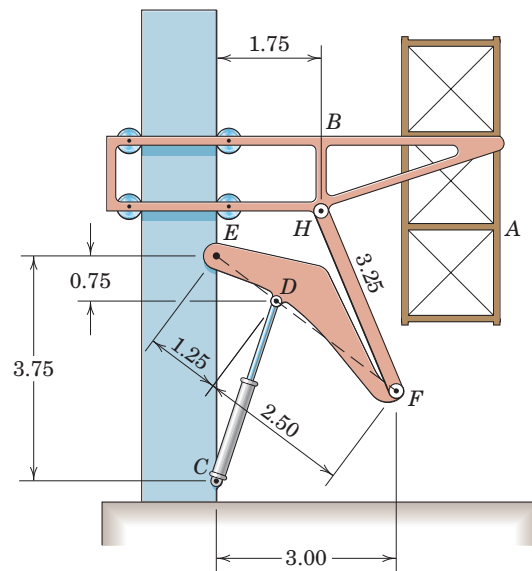
- 4/119** The pruning mechanism of a pole saw is shown as it cuts a branch S . For the particular position drawn, the actuating cord is parallel to the pole and carries a tension of 120 N. Determine the shearing force P applied to the branch by the cutter and the total force supported by the pin at E . The force exerted by the light return spring at C is small and may be neglected.

Ans. $P = 1351$ N, $E = 300$ N



Problem 4/119

- 4/120** The figure shows a special rig designed to erect vertical sections of a construction tower. The assembly A has a mass of 1.5 Mg and is elevated by the platform B , which itself has a mass of 2 Mg. The platform is guided up the fixed vertical column by rollers and is activated by the hydraulic cylinder CD and links EDF and FH . For the particular position shown, calculate the force R exerted by the hydraulic cylinder at D and the magnitude of the force supported by the pin at E .



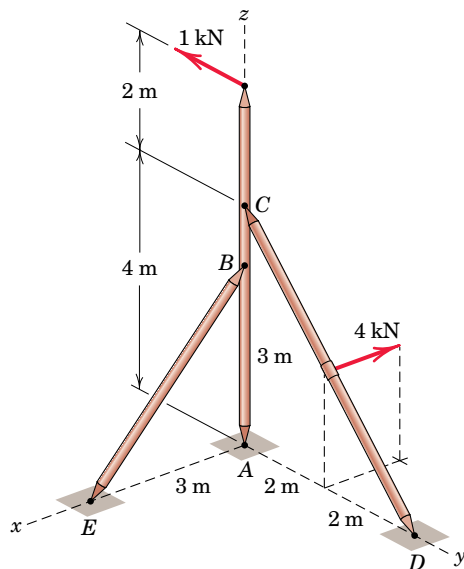
Dimensions in meters

Problem 4/120

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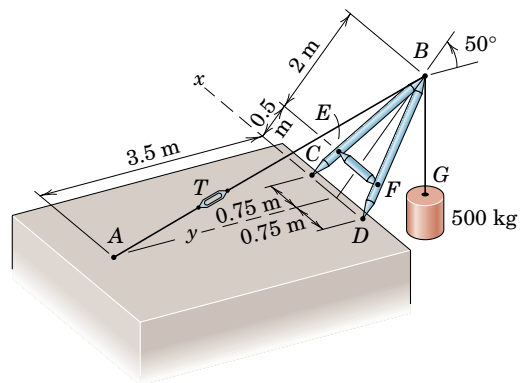
- 4/121** Determine the force acting on member ABC at connection A for the loaded space frame shown. Each connection may be treated as a ball-and-socket joint.

Ans. $A = 4.25 \text{ kN}$



Problem 4/121

- 4/122** In the schematic representation of an actual structure, T represents a turnbuckle, C and D are non-thrust-bearing hinges whose axes are along the line CD , and B , E , and F are ball-and-socket joints. Determine the tension T in the turnbuckle and the force in member EF .



Problem 4/122

4/7 CHAPTER REVIEW

In Chapter 4 we have applied the principles of equilibrium to two classes of problems: (a) simple trusses and (b) frames and machines. No new theory was needed, since we merely drew the necessary free-body diagrams and applied our familiar equations of equilibrium. The structures dealt with in Chapter 4, however, have given us the opportunity to further develop our appreciation for a systematic approach to mechanics problems.

The most essential features of the analysis of these two classes of structures are reviewed in the following statements.

(a) Simple Trusses

1. Simple trusses are composed of two-force members joined at their ends and capable of supporting tension or compression. Each internal force, therefore, is always in the direction of the line joining the endpoints of its member.
2. Simple trusses are built from the basic rigid (noncollapsible) unit of the triangle for plane trusses and the tetrahedron for space trusses. Additional units of a truss are formed by adding new members, two for plane trusses and three for space trusses, attached to existing joints and joined at their ends to form a new joint.
3. The joints of simple trusses are assumed to be pin connections for plane trusses and ball-and-socket connections for space trusses. Thus, the joints can transmit force but not moment.
4. External loads are assumed to be applied only at the joints.
5. Trusses are statically determinate externally when the external constraints are not in excess of those required to maintain an equilibrium position.
6. Trusses are statically determinate internally when constructed in the manner described in item (2), where internal members are not in excess of those required to prevent collapse.
7. The *method of joints* utilizes the force equations of equilibrium for each joint. Analysis normally begins at a joint where at least one force is known and not more than two forces are unknown for plane trusses or not more than three forces are unknown for space trusses.
8. The *method of sections* utilizes a free body of an entire section of a truss containing two or more joints. In general, the method involves the equilibrium of a nonconcurrent system of forces. The moment equation of equilibrium is especially useful when the method of sections is used. In general, the forces acting on a section which cuts more than three unknown members of a plane truss cannot be solved for completely because there are only three independent equations of equilibrium.
9. The vector representing a force acting on a joint or a section is drawn on the same side of the joint or section as the member which transmits the force. With this convention, tension is indicated when the force arrow is away from the joint or section, and compression is indicated when the arrow points toward the joint or section.

10. When the two diagonal members which brace a quadrilateral panel are flexible members incapable of supporting compression, only the one in tension is retained in the analysis, and the panel remains statically determinate.
11. When two joined members under load are collinear and a third member with a different direction is joined with their connection, the force in the third member must be zero unless an external force is applied at the joint with a component normal to the collinear members.

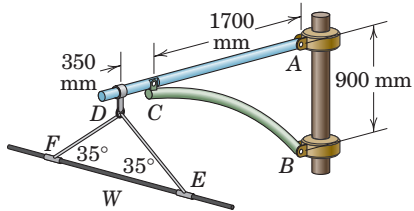
(b) Frames and Machines

1. Frames and machines are structures which contain one or more multiframe members. A multiframe member is one which has acting on it three or more forces, or two or more forces and one or more couples.
2. Frames are structures designed to support loads, generally under static conditions. Machines are structures which transform input forces and moments to output forces and moments and generally involve moving parts. Some structures may be classified as either a frame or a machine.
3. Only frames and machines which are statically determinate externally and internally are considered here.
4. If a frame or machine as a whole is a rigid (noncollapsible) unit when its external supports are removed, then we begin the analysis by computing the external reactions on the entire unit. If a frame or machine as a whole is a nonrigid (collapsible) unit when its external supports are removed, then the analysis of the external reactions cannot be completed until the structure is dismembered.
5. Forces acting in the internal connections of frames and machines are calculated by dismembering the structure and constructing a separate free-body diagram of each part. The principle of action and reaction must be *strictly* observed; otherwise, error will result.
6. The force and moment equations of equilibrium are applied to the members as needed to compute the desired unknowns.

REVIEW PROBLEMS

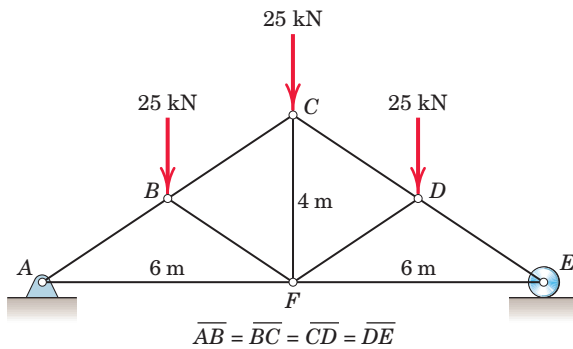
- 4/123** The support apparatus for a power line for a light-rapid-transit train is shown. If a tension of 2.5 kN exists in cable EDF , determine the force F supported by the bolt at B .

Ans. $F = 7.39$ kN



Problem 4/123

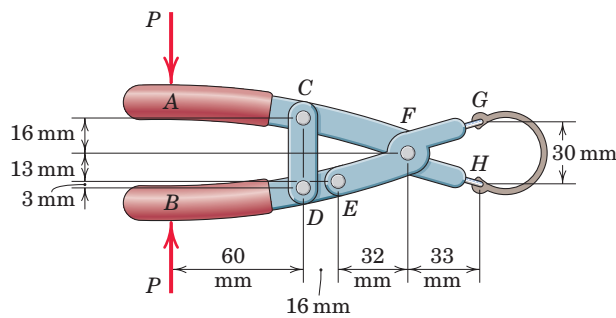
- 4/124** Determine the force in each member of the loaded truss.



Problem 4/124

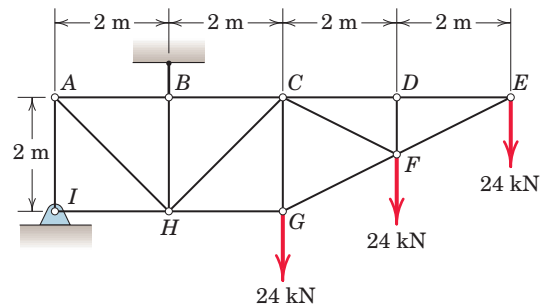
- 4/125** The specialty tool is used for installing and removing snap rings. Determine the spreading force applied at G and H if $P = 50$ N.

Ans. $G = 181.8$ N



Problem 4/125

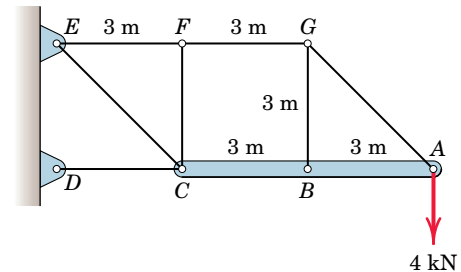
- 4/126** Determine the forces in members CH and CF .



Problem 4/126

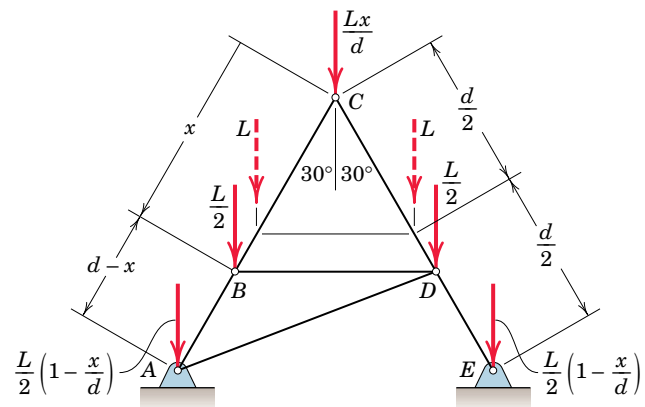
- 4/127** Calculate the force in member BG using a free-body diagram of the rigid member ABC .

Ans. $BG = 8$ kN C



Problem 4/127

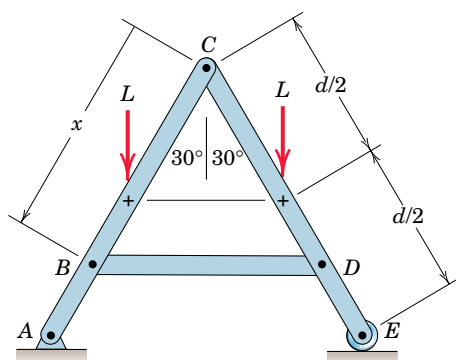
- 4/128** Show that the force in the horizontal member BD is independent of its position x within the triangular truss. Each side of the overall truss supports a centered vertical load L as shown by the two dashed vectors, and their loads are then distributed to the joints as shown by the solid vectors.



Problem 4/128

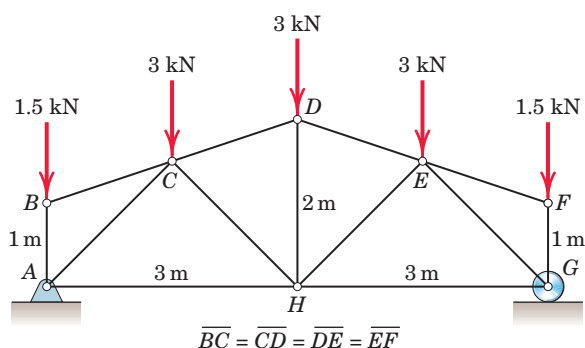
4/129 The basic structural shape and loading of Prob. 4/128 is now treated as the loaded frame shown in the figure. Determine the force in the horizontal member BD as a function of its position x within the frame.

$$Ans. BD = \frac{0.289Ld}{x}$$



Problem 4/129

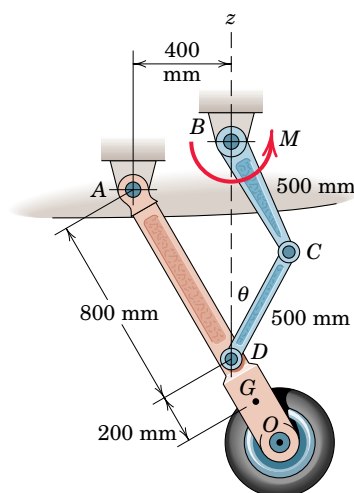
4/130 Determine the forces in members CH , AH , and CD of the loaded truss.



Problem 4/130

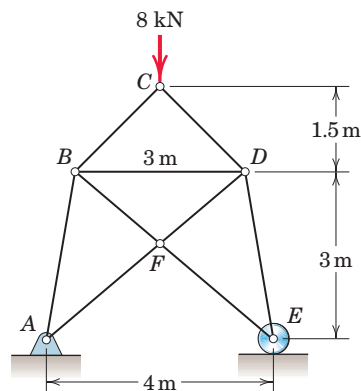
4/131 The nose-wheel assembly is raised by the application of a torque M to link BC through the shaft at B . If the arm and wheel AO have a combined mass of 50 kg with center of mass at G , find the value of M necessary to lift the wheel when D is directly under B , at which position angle θ is 30° .

Ans. $M = 153.3 \text{ N}\cdot\text{m}$



Problem 4/131

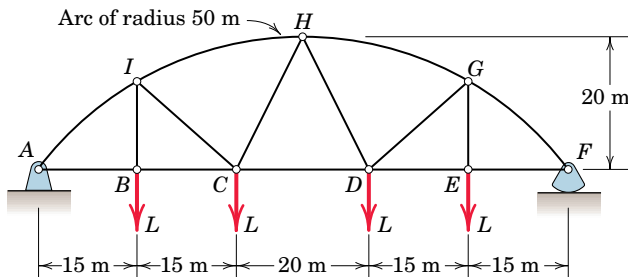
4/132 Determine the force in member BF of the loaded truss.



Problem 4/132

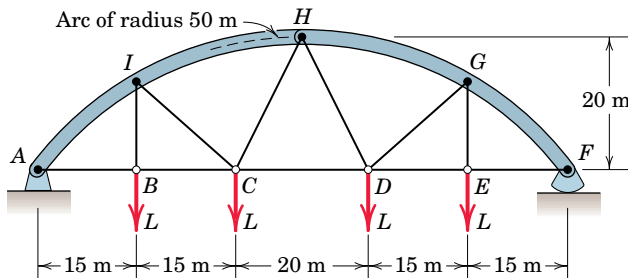
- 4/133** Determine the forces in members AB , BI , and CI of the simple truss. Note that all curved members are two-force members.

Ans. $AB = 2.26L$, $BI = L$, $CI = 0.458L$



Problem 4/133

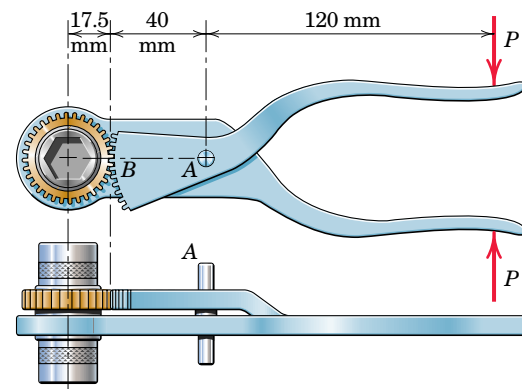
- 4/134** The structure of Prob. 4/133 is modified in that the four curved members are replaced by the two members AIH and HGF . Instrumentation indicates the tension in members CH and DH to be $0.5L$ each. Determine the forces in members AB , BI , and CI . Is the problem solvable without the information about CH ?



Problem 4/134

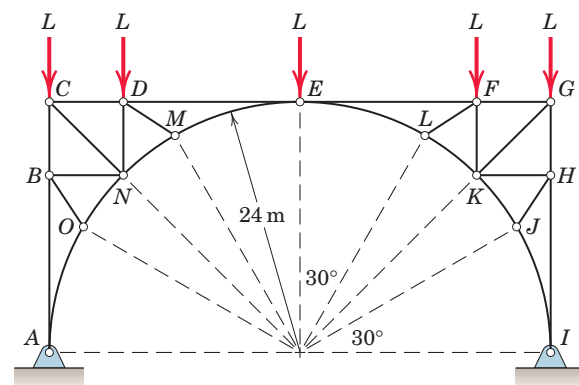
- 4/135** An antitorque wrench is designed for use by a crewman of a spacecraft where he has no stable platform against which to push as he tightens a bolt. The pin A fits into an adjacent hole in the structure which contains the bolt to be turned. Successive oscillations of the gear and handle unit turn the socket in one direction through the action of a ratchet mechanism. The reaction against the pin A provides the "antitorque" characteristic of the tool. For a gripping force $P = 150$ N, determine the torque M transmitted to the bolt and the external reaction R against the pin A normal to the line AB . (One side of the tool is used for tightening and the opposite side for loosening a bolt.)

Ans. $M = 7.88$ N·m, $R = 137.0$ N



Problem 4/135

- 4/136** Determine the forces in members DM and DN of the loaded symmetrical truss.

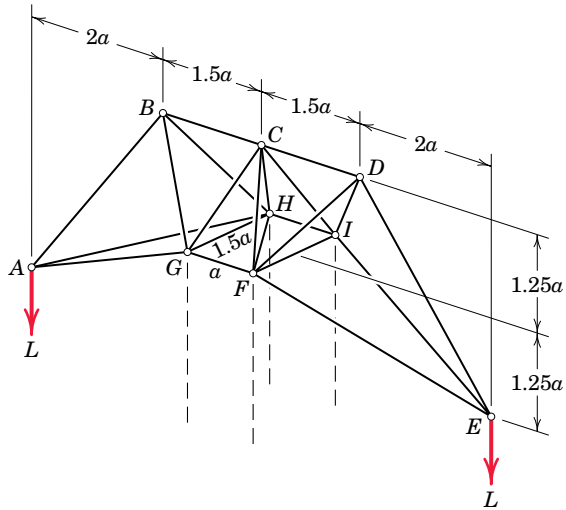


Problem 4/136

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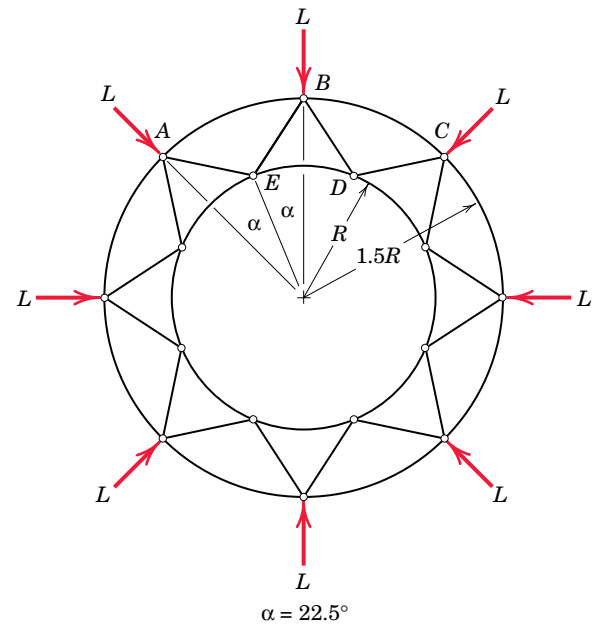
- 4/137** The depicted structure is under consideration as the upper portion of a transmission-line tower and is supported at points F , G , H , and I . Point C is directly above the center of rectangle $FGHI$. Determine the force in member CD .

Ans. $CD = 2.4L$ T



Problem 4/137

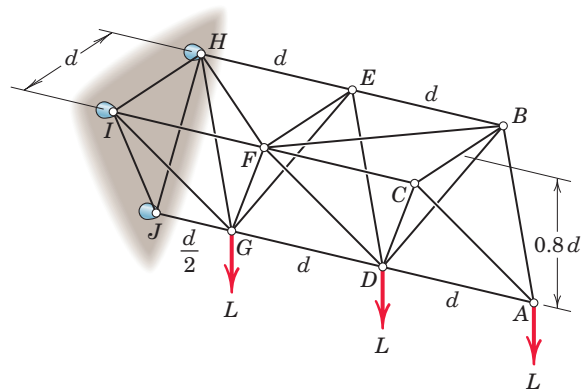
- 4/138** The structure shown is being considered as part of a large cylindrical vessel which must support external loads. Strain-gage instrumentation indicates that the compressive force in member BE is $0.8L$. Determine the forces in members AB and DE . Make use of symmetry.



Problem 4/138

- 4/139** Determine the forces in members AD and DG .

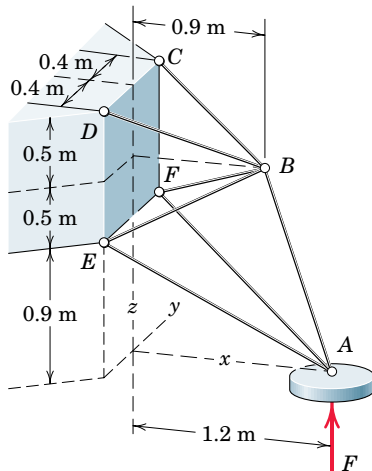
Ans. $AD = 0.625L$ C , $DG = 2.5L$ C



Problem 4/139

- **4/140** Each of the landing struts for a planet exploration spacecraft is designed as a space truss symmetrical about the vertical x - z plane as shown. For a landing force $F = 2.2$ kN, calculate the corresponding force in member BE . The assumption of static equilibrium for the truss is permissible if the mass of the truss is very small. Assume equal loads in the symmetrically placed members.

Ans. $F_{BE} = 1.620$ kN



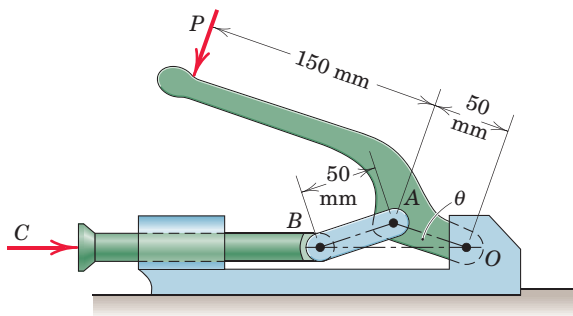
Problem 4/140



Computer-Oriented Problems*

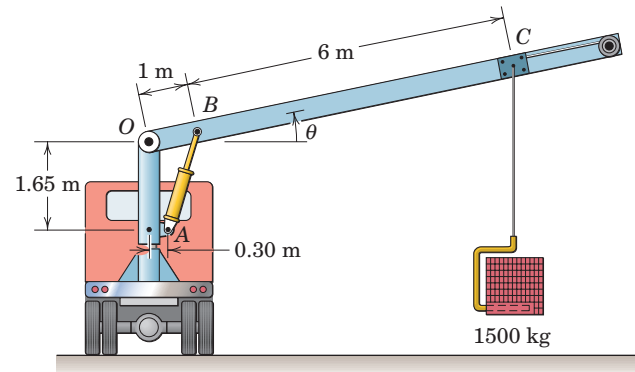
- * **4/141** For a given force P on the handle of the toggle clamp the clamping force C increases to very large values as the angle θ decreases. For $P = 120$ N determine the relationship between C and θ and plot it as a function of θ from $\theta = 2^\circ$ to $\theta = 30^\circ$. Assume that the shaft slides freely in its guide.

Ans. $C = \frac{240}{\sin \theta}$ N



Problem 4/141

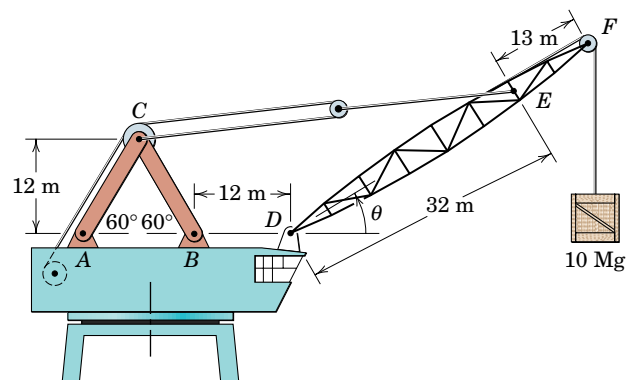
- * **4/142** The truck with bed-mounted crane is used in the delivery of cubes of bricks. For the given position of the carrier C along the boom, determine and plot the force F in the hydraulic cylinder AB as a function of the elevation angle θ for $0 \leq \theta \leq 75^\circ$. Find the maximum value of F and the corresponding angle θ . Neglect the mass of the boom compared with that of the 1500-kg pallet of bricks.



Problem 4/142

- * **4/143** The type of marine crane shown is utilized for both dockside and offshore operations. Determine and plot the force in member BC as a function of the boom angle θ for $0 \leq \theta \leq 80^\circ$ and state the value of this force for $\theta = 40^\circ$. Neglect the radius of all pulleys and the weight of the boom.

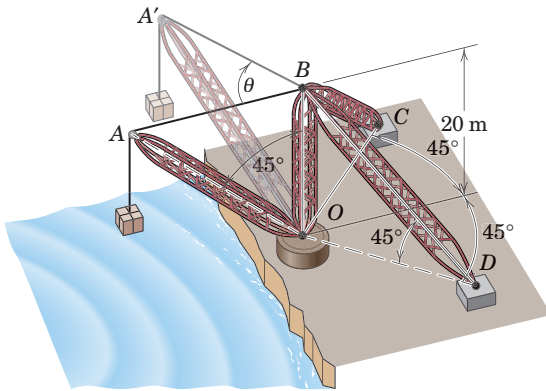
Ans. $BC = 190.5$ kN at $\theta = 40^\circ$



Problem 4/143

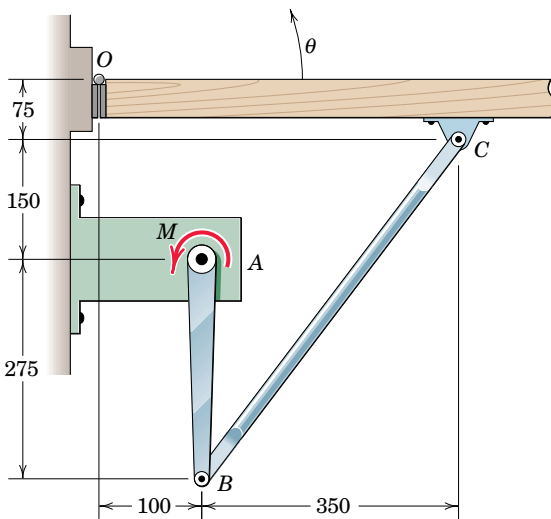
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- *4/144** Boom OA of the large waterfront crane can rotate about the vertical member OB . Determine and plot the forces in members BC and BD as functions of θ over the range $-90^\circ \leq \theta \leq 90^\circ$. The mass of the crate is 3 Mg. The 45° elevation angle of the boom remains constant; the cables for control of boom elevation angle have been omitted for clarity.

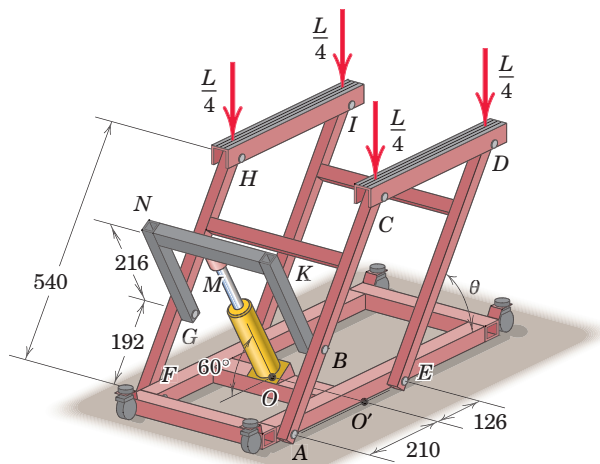

Problem 4/144

- *4/145** A door-opening mechanism is shown in the figure. The spring-loaded hinges at O provide a moment $K_T\theta$ which tends to close the door, where θ is the door-opening angle and the torsional spring constant $K_T = 56.5 \text{ N}\cdot\text{m}/\text{rad}$. The motor unit at A provides a variable moment M so that the slowly opening door is always in quasi-static equilibrium. Determine the moment M and the pin force at B as functions of θ for the range $0 \leq \theta \leq 90^\circ$. State the value of M for $\theta = 45^\circ$.

Ans. $M = 32.2 \text{ N}\cdot\text{m}$ at $\theta = 45^\circ$

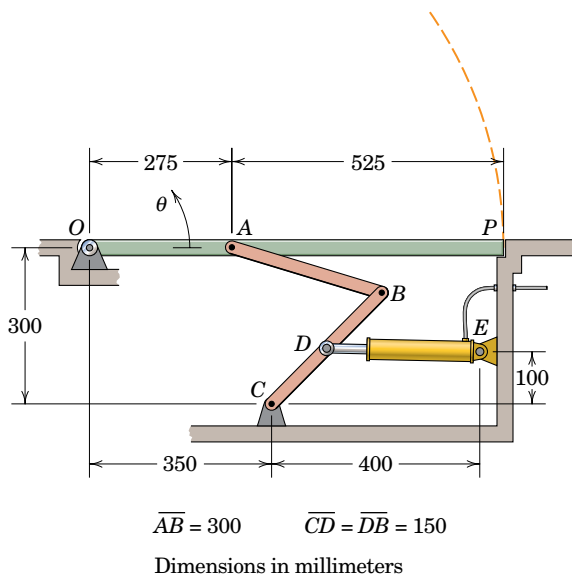

Dimensions in millimeters
Problem 4/145

- *4/146** The lift shown is used to elevate motorcycles during manufacture and repair. The hydraulic cylinder OM remains at the 60° angle as the elevation angle θ varies. The figures $ACDE$ and $FHIJ$ (J hidden) are parallelograms. There is a ball-and-socket joint at M to allow rotation of the frame $BKMNG$. If the total load $L = 4 \text{ kN}$, determine and plot the required axial cylinder force R as a function of θ over the range $0 \leq \theta \leq 90^\circ$. State the maximum value of R and the angle at which the maximum occurs. Over the same range of θ , plot the side force S applied to the cylinder rod at M and state its maximum absolute value.


Dimensions in millimeters
Problem 4/146

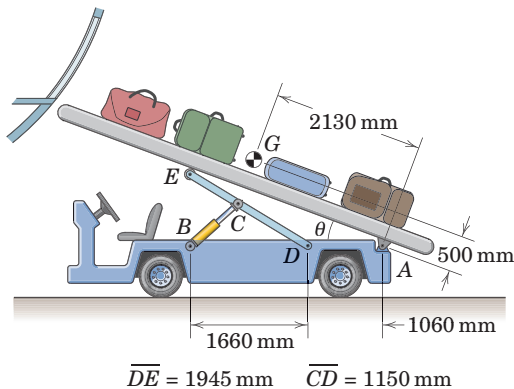
- *4/147** The uniform 30-kg ventilation door OAP is opened by the mechanism shown. Plot the required force in the cylinder DE as a function of the door opening angle θ over the range $0 \leq \theta \leq \theta_{\max}$, where θ_{\max} is the maximum opening. Determine the minimum and maximum values of this force and the angles at which these extremes occur. Note that the cylinder is not horizontal when $\theta = 0$.

Ans. $(F_{DE})_{\max} = 3580 \text{ N at } \theta = 0$
 $(F_{DE})_{\min} = 0 \text{ at } \theta_{\max} = 65.9^\circ$



Problem 4/147

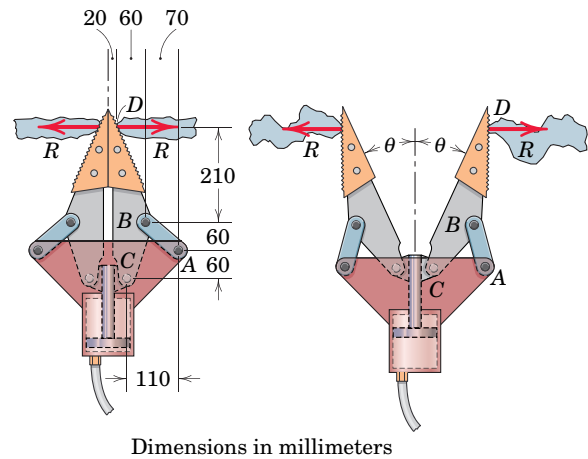
- *4/148** The machine shown is used to help load luggage into airliners. The combined mass of the conveyor and luggage is 100 kg with mass center at G . Determine and plot the force in the hydraulic cylinder as a function of θ over the range $5^\circ \leq \theta \leq 30^\circ$ and state the maximum value over this range.



Problem 4/148

- *4/149** The “jaws of life” device is used by rescuers to pry apart wreckage. A pressure of 35 MPa ($35(10^6) \text{ N/m}^2$) is developed behind the piston of 50-mm radius. Begin by determining the prying force R , the force in link AB , and the horizontal force reaction at C for the condition shown on the left. Then develop expressions for and plot those quantities as functions of the jaw angle θ (shown on the right) over the range $0 \leq \theta \leq 45^\circ$. State the minimum value of R and the value of θ for which this extreme occurs.

Ans. At $\theta = 0$: $R = 75\,000 \text{ N}$, $AB = 211\,000 \text{ N}$
 $C_x = -85\,400 \text{ N}$
 $R_{\min} = 49\,400 \text{ N at } \theta = 23.2^\circ$



Problem 4/149



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When forces are continuously distributed over a region of a structure, the cumulative effect of this distribution must be determined. The designers of high-performance sailboats consider both air-pressure distributions on the sails and water-pressure distributions on the hull.

5

DISTRIBUTED FORCES

CHAPTER OUTLINE

5/1 Introduction

SECTION A CENTERS OF MASS AND CENTROIDS

5/2 Center of Mass

5/3 Centroids of Lines, Areas, and Volumes

5/4 Composite Bodies and Figures; Approximations

5/5 Theorems of Pappus

SECTION B SPECIAL TOPICS

5/6 Beams—External Effects

5/7 Beams—Internal Effects

5/8 Flexible Cables

5/9 Fluid Statics

5/10 Chapter Review

5/1 INTRODUCTION

In the previous chapters we treated all forces as concentrated along their lines of action and at their points of application. This treatment provided a reasonable model for those forces. Actually, “concentrated” forces do not exist in the exact sense, since every external force applied mechanically to a body is distributed over a finite contact area, however small.

The force exerted by the pavement on an automobile tire, for instance, is applied to the tire over its entire area of contact, Fig. 5/1*a*, which may be appreciable if the tire is soft. When analyzing the forces acting on the car as a whole, if the dimension b of the contact area is negligible compared with the other pertinent dimensions, such as the distance between wheels, then we may replace the actual distributed contact forces by their resultant R treated as a concentrated force. Even the force of contact between a hardened steel ball and its race in a loaded ball bearing, Fig. 5/1*b*, is applied over a finite though extremely small contact area. The forces applied to a two-force member of a truss, Fig. 5/1*c*, are applied over an actual area of contact of the pin against the hole and internally across the cut section as shown. In these and other similar examples we may treat the forces as concentrated when analyzing their external effects on bodies as a whole.

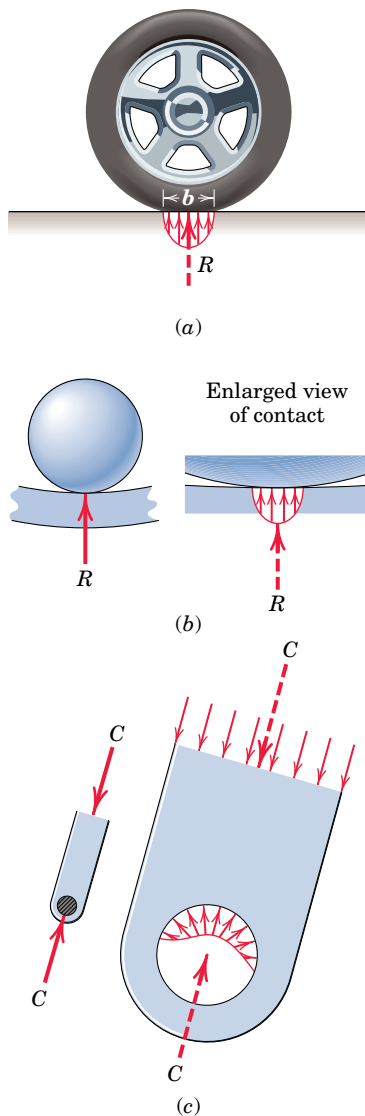


Figure 5/1

If, on the other hand, we want to find the distribution of *internal* forces in the material of the body near the contact location, where the internal stresses and strains may be appreciable, then we must not treat the load as concentrated but must consider the actual distribution. This problem will not be discussed here because it requires a knowledge of the properties of the material and belongs in more advanced treatments of the mechanics of materials and the theories of elasticity and plasticity.

When forces are applied over a region whose dimensions are not negligible compared with other pertinent dimensions, then we must account for the actual manner in which the force is distributed. We do this by summing the effects of the distributed force over the entire region using mathematical integration. This requires that we know the intensity of the force at any location. There are three categories of such problems.

(1) Line Distribution. When a force is distributed along a line, as in the continuous vertical load supported by a suspended cable, Fig. 5/2a, the intensity w of the loading is expressed as force per unit length of line, newtons per meter (N/m) or pounds per foot (lb/ft).

(2) Area Distribution. When a force is distributed over an area, as with the hydraulic pressure of water against the inner face of a section of dam, Fig. 5/2b, the intensity is expressed as force per unit area. This intensity is called *pressure* for the action of fluid forces and *stress* for the internal distribution of forces in solids. The basic unit for pressure or stress in SI is the newton per square meter (N/m^2), which is also called the *pascal* (Pa). This unit, however, is too small for most applications ($6895 \text{ Pa} = 1 \text{ lb/in.}^2$). The kilopascal (kPa), which equals 10^3 Pa , is more commonly used for fluid pressure, and the megapascal, which equals 10^6 Pa , is used for stress. In the U.S. customary system of units, both fluid pressure and mechanical stress are commonly expressed in pounds per square inch (lb/in.^2).

(3) Volume Distribution. A force which is distributed over the volume of a body is called a *body force*. The most common body force is the force of gravitational attraction, which acts on all elements of mass in a body. The determination of the forces on the supports of the heavy cantilevered structure in Fig. 5/2c, for example, would require accounting for the distribution of gravitational force throughout the structure. The intensity of gravitational force is the *specific weight* ρg , where ρ is the density (mass per unit volume) and g is the acceleration due to gravity. The units for ρg are $(\text{kg/m}^3)(\text{m/s}^2) = \text{N/m}^3$ in SI units and lb/ft^3 or lb/in.^3 in the U.S. customary system.

The body force due to the gravitational attraction of the earth (weight) is by far the most commonly encountered distributed force. Section A of this chapter treats the determination of the point in a body through which the resultant gravitational force acts, and discusses the associated geometric properties of lines, areas, and volumes. Section B treats distributed forces which act on and in beams and flexible cables and distributed forces which fluids exert on exposed surfaces.

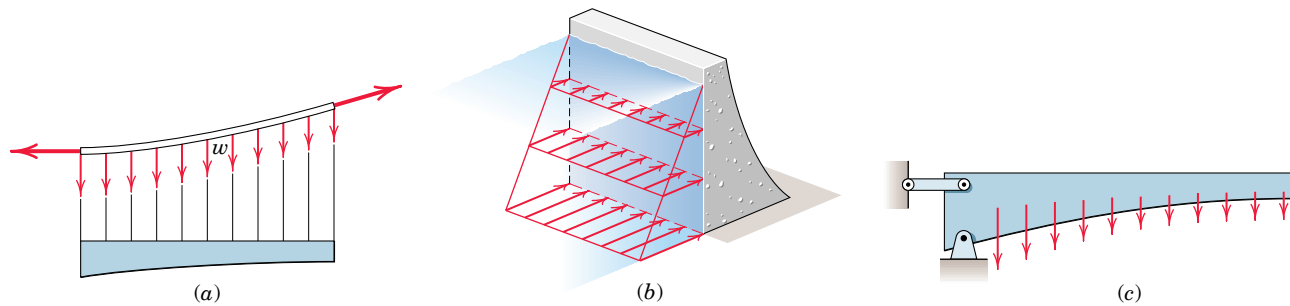


Figure 5/2

SECTION A CENTERS OF MASS AND CENTROIDS

5/2 CENTER OF MASS

Consider a three-dimensional body of any size and shape, having a mass m . If we suspend the body, as shown in Fig. 5/3, from any point such as A , the body will be in equilibrium under the action of the tension in the cord and the resultant W of the gravitational forces acting on all particles of the body. This resultant is clearly collinear with the cord. Assume that we mark its position by drilling a hypothetical hole of negligible size along its line of action. We repeat the experiment by suspending the body from other points such as B and C , and in each instance we mark the line of action of the resultant force. For all practical purposes these lines of action will be concurrent at a single point G , which is called the *center of gravity* of the body.

An exact analysis, however, would account for the slightly differing directions of the gravity forces for the various particles of the body, because those forces converge toward the center of attraction of the earth. Also, because the particles are at different distances from the earth, the intensity of the force field of the earth is not exactly constant over the body. As a result, the lines of action of the gravity-force resultants in the experiments just described will not be quite concurrent, and therefore no unique center of gravity exists in the exact sense. This is of no practical importance as long as we deal with bodies whose dimensions are small compared with those of the earth. We therefore assume a uniform and parallel force field due to the gravitational attraction of the earth, and this assumption results in the concept of a unique center of gravity.

Determining the Center of Gravity

To determine mathematically the location of the center of gravity of any body, Fig. 5/4a, we apply the *principle of moments* (see Art. 2/6) to the parallel system of gravitational forces. The moment of the resultant gravitational force W about any axis equals the sum of the moments

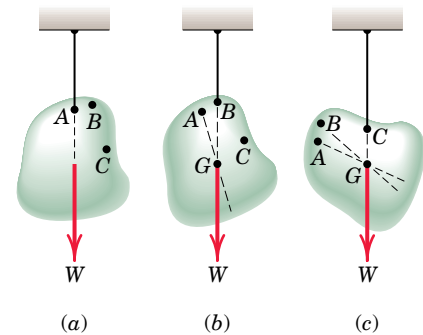


Figure 5/3

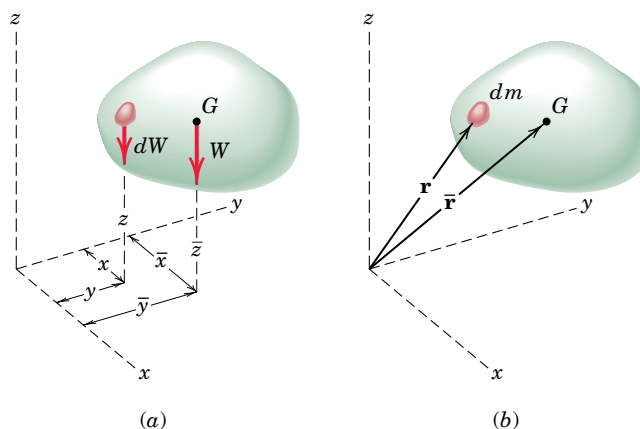


Figure 5/4

about the same axis of the gravitational forces dW acting on all particles treated as infinitesimal elements of the body. The resultant of the gravitational forces acting on all elements is the weight of the body and is given by the sum $W = \int dW$. If we apply the moment principle about the y -axis, for example, the moment about this axis of the elemental weight is $x dW$, and the sum of these moments for all elements of the body is $\int x dW$. This sum of moments must equal $W\bar{x}$, the moment of the sum. Thus, $\bar{x}W = \int x dW$.

With similar expressions for the other two components, we may express the coordinates of the center of gravity G as

$$\bar{x} = \frac{\int x dW}{W} \quad \bar{y} = \frac{\int y dW}{W} \quad \bar{z} = \frac{\int z dW}{W} \quad (5/1a)$$

To visualize the physical moments of the gravity forces appearing in the third equation, we may reorient the body and attached axes so that the z -axis is horizontal. It is essential to recognize that the numerator of each of these expressions represents the *sum of the moments*, whereas the product of W and the corresponding coordinate of G represents the *moment of the sum*. This moment principle finds repeated use throughout mechanics.

With the substitution of $W = mg$ and $dW = g dm$, the expressions for the coordinates of the center of gravity become

$$\bar{x} = \frac{\int x dm}{m} \quad \bar{y} = \frac{\int y dm}{m} \quad \bar{z} = \frac{\int z dm}{m} \quad (5/1b)$$

Equations 5/1b may be expressed in vector form with the aid of Fig. 5/4b, in which the elemental mass and the mass center G are located by

their respective position vectors $\mathbf{r} = x\mathbf{i} + y\mathbf{j} + z\mathbf{k}$ and $\bar{\mathbf{r}} = \bar{x}\mathbf{i} + \bar{y}\mathbf{j} + \bar{z}\mathbf{k}$. Thus, Eqs. 5/1b are the components of the single vector equation

$$\bar{\mathbf{r}} = \frac{\int \mathbf{r} dm}{m} \quad (5/2)$$

The density ρ of a body is its mass per unit volume. Thus, the mass of a differential element of volume dV becomes $dm = \rho dV$. If ρ is not constant throughout the body but can be expressed as a function of the coordinates of the body, we must account for this variation when calculating the numerators and denominators of Eqs. 5/1b. We may then write these expressions as

$$\bar{x} = \frac{\int x\rho dV}{\int \rho dV} \quad \bar{y} = \frac{\int y\rho dV}{\int \rho dV} \quad \bar{z} = \frac{\int z\rho dV}{\int \rho dV} \quad (5/3)$$

Center of Mass versus Center of Gravity

Equations 5/1b, 5/2, and 5/3 are independent of gravitational effects since g no longer appears. They therefore define a unique point in the body which is a function solely of the distribution of mass. This point is called the *center of mass*, and clearly it coincides with the center of gravity as long as the gravity field is treated as uniform and parallel.

It is meaningless to speak of the center of gravity of a body which is removed from the gravitational field of the earth, since no gravitational forces would act on it. The body would, however, still have its unique center of mass. We will usually refer henceforth to the center of mass rather than to the center of gravity. Also, the center of mass has a special significance in calculating the dynamic response of a body to unbalanced forces. This class of problems is discussed at length in *Vol. 2 Dynamics*.

In most problems the calculation of the position of the center of mass may be simplified by an intelligent choice of reference axes. In general the axes should be placed so as to simplify the equations of the boundaries as much as possible. Thus, polar coordinates will be useful for bodies with circular boundaries.

Another important clue may be taken from considerations of symmetry. Whenever there exists a line or plane of symmetry in a homogeneous body, a coordinate axis or plane should be chosen to coincide with this line or plane. The center of mass will always lie on such a line or plane, since the moments due to symmetrically located elements will always cancel, and the body may be considered composed of pairs of these elements. Thus, the center of mass G of the homogeneous right-circular cone of Fig. 5/5a will lie somewhere on its central axis, which is a line of symmetry. The center of mass of the half right-circular cone lies on its plane of symmetry, Fig. 5/5b. The center of mass of the half ring in Fig. 5/5c lies in both of its planes of symmetry and therefore is situated on

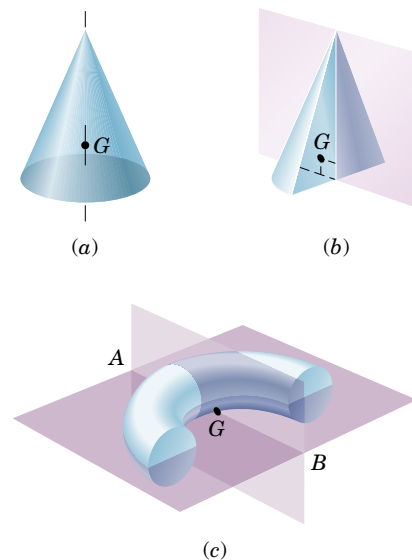


Figure 5/5

line AB . It is easiest to find the location of G by using symmetry when it exists.

5/3 CENTROIDS OF LINES, AREAS, AND VOLUMES

When the density ρ of a body is uniform throughout, it will be a constant factor in both the numerators and denominators of Eqs. 5/3 and will therefore cancel. The remaining expressions define a purely geometrical property of the body, since any reference to its mass properties has disappeared. The term *centroid* is used when the calculation concerns a geometrical shape only. When speaking of an actual physical body, we use the term *center of mass*. If the density is uniform throughout the body, the positions of the centroid and center of mass are identical, whereas if the density varies, these two points will, in general, not coincide.

The calculation of centroids falls within three distinct categories, depending on whether we can model the shape of the body involved as a line, an area, or a volume.

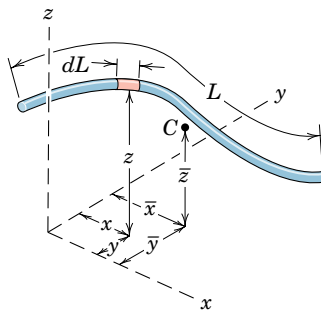


Figure 5/6

(1) Lines. For a slender rod or wire of length L , cross-sectional area A , and density ρ , Fig. 5/6, the body approximates a line segment, and $dm = \rho A dL$. If ρ and A are constant over the length of the rod, the coordinates of the center of mass also become the coordinates of the centroid C of the line segment, which, from Eqs. 5/1b, may be written

$$\bar{x} = \frac{\int x dL}{L} \quad \bar{y} = \frac{\int y dL}{L} \quad \bar{z} = \frac{\int z dL}{L} \quad (5/4)$$

Note that, in general, the centroid C will not lie on the line. If the rod lies on a single plane, such as the x - y plane, only two coordinates need to be calculated.

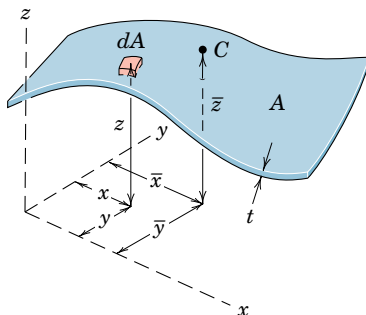


Figure 5/7

(2) Areas. When a body of density ρ has a small but constant thickness t , we can model it as a surface area A , Fig. 5/7. The mass of an element becomes $dm = \rho t dA$. Again, if ρ and t are constant over the entire area, the coordinates of the center of mass of the body also become the coordinates of the centroid C of the surface area, and from Eqs. 5/1b the coordinates may be written

$$\bar{x} = \frac{\int x dA}{A} \quad \bar{y} = \frac{\int y dA}{A} \quad \bar{z} = \frac{\int z dA}{A} \quad (5/5)$$

The numerators in Eqs. 5/5 are called the *first moments of area*.^{*} If the surface is curved, as illustrated in Fig. 5/7 with the shell segment, all three coordinates will be involved. The centroid C for the curved surface will in general not lie on the surface. If the area is a flat surface in,

^{*}Second moments of areas (moments of first moments) appear later in our discussion of area moments of inertia in Appendix A.

say, the x - y plane, only the coordinates of C in that plane need to be calculated.

(3) Volumes. For a general body of volume V and density ρ , the element has a mass $dm = \rho dV$. The density ρ cancels if it is constant over the entire volume, and the coordinates of the center of mass also become the coordinates of the centroid C of the body. From Eqs. 5/3 or 5/1b they become

$$\bar{x} = \frac{\int x dV}{V} \quad \bar{y} = \frac{\int y dV}{V} \quad \bar{z} = \frac{\int z dV}{V} \quad (5/6)$$

Choice of Element for Integration

The principal difficulty with a theory often lies not in its concepts but in the procedures for applying it. With mass centers and centroids the concept of the moment principle is simple enough; the difficult steps are the choice of the differential element and setting up the integrals. The following five guidelines will be useful.

(1) Order of Element. Whenever possible, a first-order differential element should be selected in preference to a higher-order element so that only one integration will be required to cover the entire figure. Thus, in Fig. 5/8a a first-order horizontal strip of area $dA = l dy$ will require only one integration with respect to y to cover the entire figure. The second-order element $dx dy$ will require two integrations, first with respect to x and second with respect to y , to cover the figure. As a further example, for the solid cone in Fig. 5/8b we choose a first-order element in the form of a circular slice of volume $dV = \pi r^2 dy$. This choice requires only one integration, and thus is preferable to choosing a third-order element $dV = dx dy dz$, which would require three awkward integrations.

(2) Continuity. Whenever possible, we choose an element which can be integrated in one continuous operation to cover the figure. Thus, the horizontal strip in Fig. 5/8a would be preferable to the vertical strip in Fig. 5/9, which, if used, would require two separate integrals because of the discontinuity in the expression for the height of the strip at $x = x_1$.

(3) Discarding Higher-Order Terms. Higher-order terms may always be dropped compared with lower-order terms (see Art. 1/7). Thus, the vertical strip of area under the curve in Fig. 5/10 is given by the

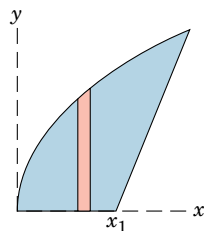


Figure 5/9

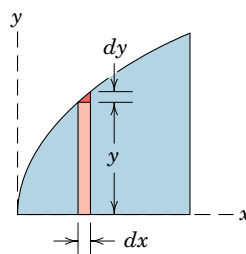
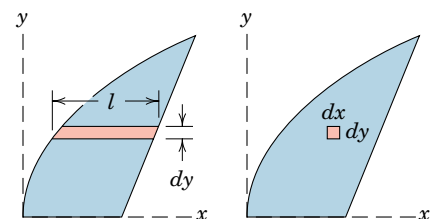
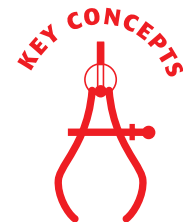
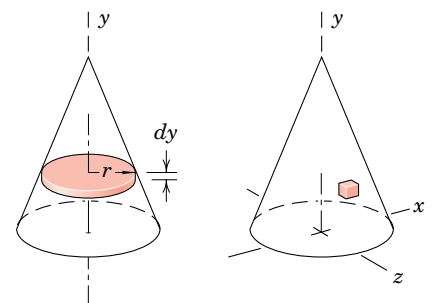


Figure 5/10



(a)



(b)

Figure 5/8

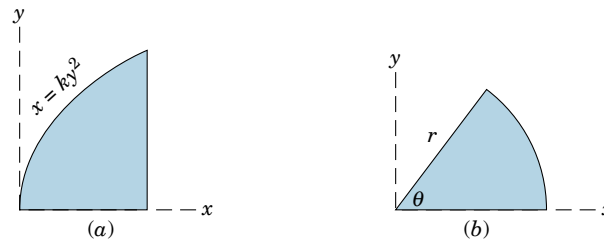


Figure 5/11

first-order term $dA = y dx$, and the second-order triangular area $\frac{1}{2}dx dy$ is discarded. In the limit, of course, there is no error.

(4) Choice of Coordinates. As a general rule, we choose the coordinate system which best matches the boundaries of the figure. Thus, the boundaries of the area in Fig. 5/11a are most easily described in rectangular coordinates, whereas the boundaries of the circular sector of Fig. 5/11b are best suited to polar coordinates.

(5) Centroidal Coordinate of Element. When a first- or second-order differential element is chosen, it is essential to use the *coordinate of the centroid of the element* for the moment arm in expressing the moment of the differential element. Thus, for the horizontal strip of area in Fig. 5/12a, the moment of dA about the y -axis is $x_c dA$, where x_c is the x -coordinate of the centroid C of the element. Note that x_c is *not* the x which describes either boundary of the area. In the y -direction for this element the moment arm y_c of the centroid of the element is the same, in the limit, as the y -coordinates of the two boundaries.

As a second example, consider the solid half-cone of Fig. 5/12b with the semicircular slice of differential thickness as the element of volume. The moment arm for the element in the x -direction is the distance x_c to the centroid of the face of the element and not the x -distance to the boundary of the element. On the other hand, in the z -direction the moment arm z_c of the centroid of the element is the same as the z -coordinate of the element.

With these examples in mind, we rewrite Eqs. 5/5 and 5/6 in the form

$$\bar{x} = \frac{\int x_c dA}{A} \quad \bar{y} = \frac{\int y_c dA}{A} \quad \bar{z} = \frac{\int z_c dA}{A} \quad (5/5a)$$

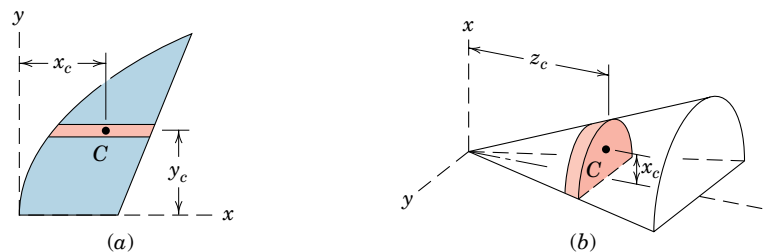


Figure 5/12

and

$$\bar{x} = \frac{\int x_c dV}{V} \quad \bar{y} = \frac{\int y_c dV}{V} \quad \bar{z} = \frac{\int z_c dV}{V} \quad (5/6a)$$

It is *essential* to recognize that the subscript c serves as a reminder that the moment arms appearing in the numerators of the integral expressions for moments are *always* the coordinates of the *centroids* of the particular elements chosen.

At this point you should be certain to understand clearly the principle of moments, which was introduced in Art. 2/4. You should recognize the physical meaning of this principle as it is applied to the system of parallel weight forces depicted in Fig. 5/4a. Keep in mind the equivalence between the moment of the resultant weight W and the sum (integral) of the moments of the elemental weights dW , to avoid mistakes in setting up the necessary mathematics. Recognition of the principle of moments will help in obtaining the correct expression for the moment arm x_c , y_c , or z_c of the centroid of the chosen differential element.

Keeping in mind the physical picture of the principle of moments, we will recognize that Eqs. 5/4, 5/5, and 5/6, which are geometric relationships, are descriptive also of homogeneous physical bodies, because the density ρ cancels. If the density of the body in question is not constant but varies throughout the body as some function of the coordinates, then it will not cancel from the numerator and denominator of the mass-center expressions. In this event, we must use Eqs. 5/3 as explained earlier.

Sample Problems 5/1 through 5/5 which follow have been carefully chosen to illustrate the application of Eqs. 5/4, 5/5, and 5/6 for calculating the location of the centroid for line segments (slender rods), areas (thin flat plates), and volumes (homogeneous solids). The five integration considerations listed above are illustrated in detail in these sample problems.

Section C/10 of Appendix C contains a table of integrals which includes those needed for the problems in this and subsequent chapters. A summary of the centroidal coordinates for some of the commonly used shapes is given in Tables D/3 and D/4, Appendix D.

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Sample Problem 5/1

Centroid of a circular arc. Locate the centroid of a circular arc as shown in the figure.

Solution. Choosing the axis of symmetry as the x -axis makes $\bar{y} = 0$. A differential element of arc has the length $dL = r d\theta$ expressed in polar coordinates, and the x -coordinate of the element is $r \cos \theta$.

- ① Applying the first of Eqs. 5/4 and substituting $L = 2\alpha r$ give

$$[L\bar{x} = \int x dL] \quad (2\alpha r)\bar{x} = \int_{-\alpha}^{\alpha} (r \cos \theta) r d\theta$$

$$2\alpha r\bar{x} = 2r^2 \sin \alpha$$

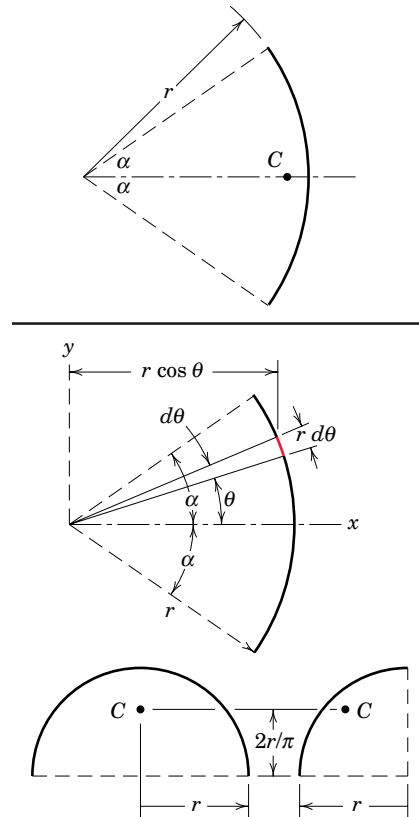
$$\bar{x} = \frac{r \sin \alpha}{\alpha}$$

Ans.

For a semicircular arc $2\alpha = \pi$, which gives $\bar{x} = 2r/\pi$. By symmetry we see immediately that this result also applies to the quarter-circular arc when the measurement is made as shown.

Helpful Hint

- ① It should be perfectly evident that polar coordinates are preferable to rectangular coordinates to express the length of a circular arc.



Sample Problem 5/2

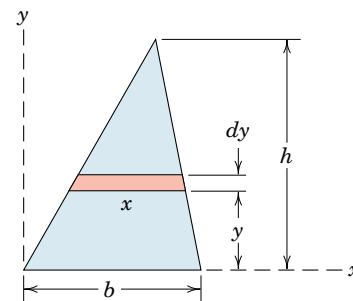
Centroid of a triangular area. Determine the distance \bar{h} from the base of a triangle of altitude h to the centroid of its area.

- ① **Solution.** The x -axis is taken to coincide with the base. A differential strip of area $dA = x dy$ is chosen. By similar triangles $x/(h - y) = b/h$. Applying the second of Eqs. 5/5a gives

$$[A\bar{y} = \int y_c dA] \quad \frac{bh}{2}\bar{y} = \int_0^h y \frac{b(h-y)}{h} dy = \frac{bh^2}{6}$$

and
$$\bar{y} = \frac{h}{3} \quad \text{Ans.}$$

This same result holds with respect to either of the other two sides of the triangle considered a new base with corresponding new altitude. Thus, the centroid lies at the intersection of the medians, since the distance of this point from any side is one-third the altitude of the triangle with that side considered the base.



Helpful Hint

- ① We save one integration here by using the first-order element of area. Recognize that dA must be expressed in terms of the integration variable y ; hence, $x = f(y)$ is required.

Sample Problem 5/3

Centroid of the area of a circular sector. Locate the centroid of the area of a circular sector with respect to its vertex.

Solution I. The x -axis is chosen as the axis of symmetry, and \bar{y} is therefore automatically zero. We may cover the area by moving an element in the form of a partial circular ring, as shown in the figure, from the center to the outer periphery. The radius of the ring is r_0 and its thickness is dr_0 , so that its area is

$$\textcircled{1} \quad dA = 2r_0\alpha \, dr_0.$$

The x -coordinate to the centroid of the element from Sample Problem 5/1 is

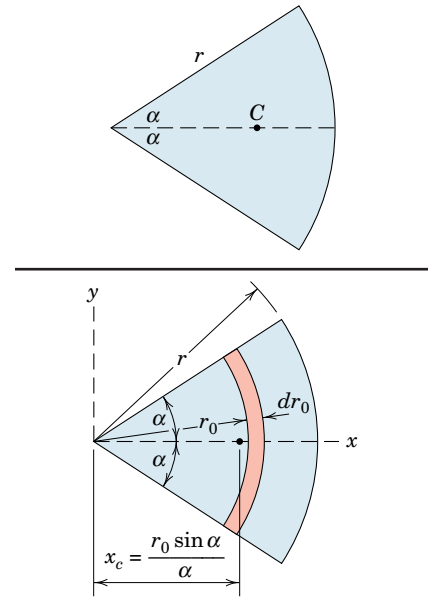
$$\textcircled{2} \quad x_c = r_0 \sin \alpha / \alpha, \text{ where } r_0 \text{ replaces } r \text{ in the formula. Thus, the first of Eqs. 5/5a gives}$$

$$[A\bar{x} = \int x_c \, dA] \quad \frac{2\alpha}{2\pi} (\pi r^2) \bar{x} = \int_0^r \left(\frac{r_0 \sin \alpha}{\alpha} \right) (2r_0\alpha \, dr_0)$$

$$r^2\alpha\bar{x} = \frac{2}{3}r^3 \sin \alpha$$

$$\bar{x} = \frac{2}{3} \frac{r \sin \alpha}{\alpha}$$

Ans.



Solution I

Solution II. The area may also be covered by swinging a triangle of differential area about the vertex and through the total angle of the sector. This triangle, shown in the illustration, has an area $dA = (r/2)(r \, d\theta)$, where higher-order terms are neglected. From Sample Problem 5/2 the centroid of the triangular element of area is two-thirds of its altitude from its vertex, so that the x -coordinate to the centroid of the element is $x_c = \frac{2}{3}r \cos \theta$. Applying the first of Eqs. 5/5a gives

$$[A\bar{x} = \int x_c \, dA] \quad (r^2\alpha)\bar{x} = \int_{-\alpha}^{\alpha} \left(\frac{2}{3}r \cos \theta \right) \left(\frac{1}{2}r^2 \, d\theta \right)$$

$$r^2\alpha\bar{x} = \frac{2}{3}r^3 \sin \alpha$$

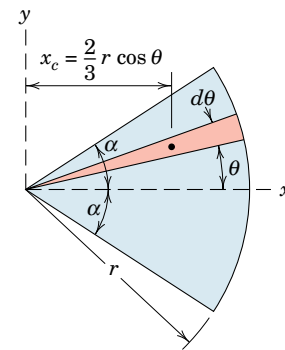
and as before

$$\bar{x} = \frac{2}{3} \frac{r \sin \alpha}{\alpha}$$

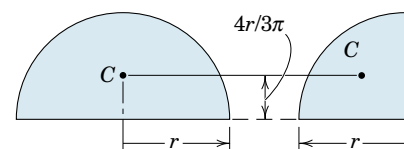
Ans.

Helpful Hints

- ① Note carefully that we must distinguish between the variable r_0 and the constant r .
- ② Be careful not to use r_0 as the centroidal coordinate for the element.



Solution II



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Sample Problem 5/4

Locate the centroid of the area under the curve $x = ky^3$ from $x = 0$ to $x = a$.

Solution I. A vertical element of area $dA = y dx$ is chosen as shown in the figure. The x -coordinate of the centroid is found from the first of Eqs. 5/5a. Thus,

$$\textcircled{1} [A\bar{x} = \int x_c dA] \quad \bar{x} \int_0^a y dx = \int_0^a xy dx$$

Substituting $y = (x/k)^{1/3}$ and $k = a/b^3$ and integrating give

$$\frac{3ab}{4} \bar{x} = \frac{3a^2b}{7} \quad \bar{x} = \frac{4}{7}a \quad \text{Ans.}$$

In the solution for \bar{y} from the second of Eqs. 5/5a, the coordinate to the centroid of the rectangular element is $y_c = y/2$, where y is the height of the strip governed by the equation of the curve $x = ky^3$. Thus, the moment principle becomes

$$[A\bar{y} = \int y_c dA] \quad \frac{3ab}{4} \bar{y} = \int_0^a \left(\frac{y}{2}\right) y dx$$

Substituting $y = b(x/a)^{1/3}$ and integrating give

$$\frac{3ab}{4} \bar{y} = \frac{3ab^2}{10} \quad \bar{y} = \frac{2}{5}b \quad \text{Ans.}$$

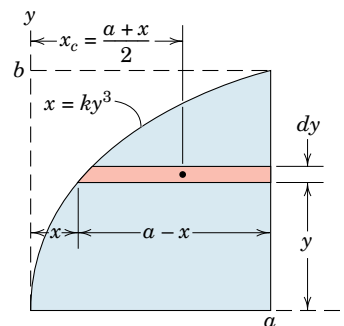
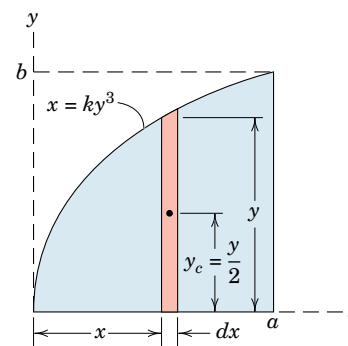
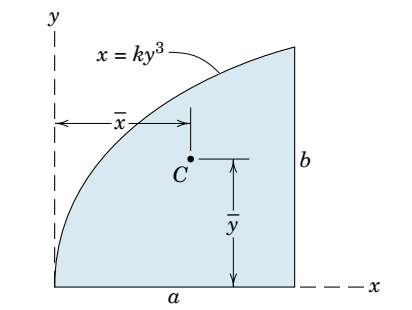
Solution II. The horizontal element of area shown in the lower figure may be employed in place of the vertical element. The x -coordinate to the centroid of the rectangular element is seen to be $x_c = x + \frac{1}{2}(a - x) = (a + x)/2$, which is simply the average of the coordinates a and x of the ends of the strip. Hence,

$$[A\bar{x} = \int x_c dA] \quad \bar{x} \int_0^b (a - x) dy = \int_0^b \left(\frac{a + x}{2}\right)(a - x) dy$$

The value of \bar{y} is found from

$$[A\bar{y} = \int y_c dA] \quad \bar{y} \int_0^b (a - x) dy = \int_0^b y(a - x) dy$$

where $y_c = y$ for the horizontal strip. The evaluation of these integrals will check the previous results for \bar{x} and \bar{y} .



Helpful Hint

- ① Note that $x_c = x$ for the vertical element.

Sample Problem 5/5

Hemispherical volume. Locate the centroid of the volume of a hemisphere of radius r with respect to its base.

Solution I. With the axes chosen as shown in the figure, $\bar{x} = \bar{z} = 0$ by symmetry. The most convenient element is a circular slice of thickness dy parallel to the x - z plane. Since the hemisphere intersects the y - z plane in the circle $y^2 + z^2 = r^2$, the radius of the circular slice is $z = +\sqrt{r^2 - y^2}$. The volume of the elemental slice becomes

$$\textcircled{1} \quad dV = \pi(r^2 - y^2) dy$$

The second of Eqs. 5/6a requires

$$[V\bar{y} = \int y_c dV] \quad \bar{y} \int_0^r \pi(r^2 - y^2) dy = \int_0^r y \pi(r^2 - y^2) dy$$

where $y_c = y$. Integrating gives

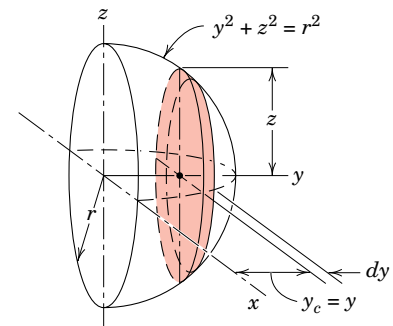
$$\frac{2}{3}\pi r^3 \bar{y} = \frac{1}{4}\pi r^4 \quad \bar{y} = \frac{3}{8}r \quad \text{Ans.}$$

Solution II. Alternatively we may use for our differential element a cylindrical shell of length y , radius z , and thickness dz , as shown in the lower figure. By expanding the radius of the shell from zero to r , we cover the entire volume. By symmetry the centroid of the elemental shell lies at its center, so that $y_c = y/2$. The volume of the element is $dV = (2\pi z dz)(y)$. Expressing y in terms of z from the equation of the circle gives $y = +\sqrt{r^2 - z^2}$. Using the value of $\frac{2}{3}\pi r^3$ computed in *Solution I* for the volume of the hemisphere and substituting in the second of Eqs. 5/6a give us

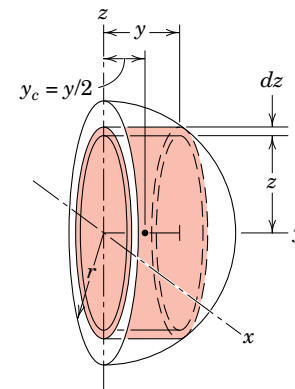
$$\begin{aligned} [V\bar{y} = \int y_c dV] \quad \left(\frac{2}{3}\pi r^3\right)\bar{y} &= \int_0^r \frac{\sqrt{r^2 - z^2}}{2} (2\pi z \sqrt{r^2 - z^2}) dz \\ &= \int_0^r \pi(r^2 z - z^3) dz = \frac{\pi r^4}{4} \\ \bar{y} &= \frac{3}{8}r \quad \text{Ans.} \end{aligned}$$

Solutions I and II are of comparable use since each involves an element of simple shape and requires integration with respect to one variable only.

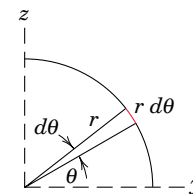
Solution III. As an alternative, we could use the angle θ as our variable with limits of 0 and $\pi/2$. The radius of either element would become $r \sin \theta$, whereas the thickness of the slice in *Solution I* would be $dy = (r d\theta) \sin \theta$ and that of the shell in *Solution II* would be $dz = (r d\theta) \cos \theta$. The length of the shell would be $y = r \cos \theta$.



Solution I



Solution II



Solution III

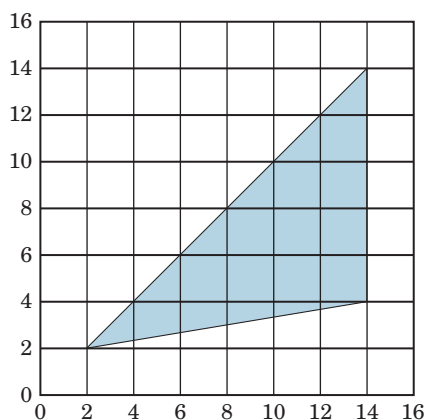
Helpful Hint

- ① Can you identify the higher-order element of volume which is omitted from the expression for dV ?

PROBLEMS

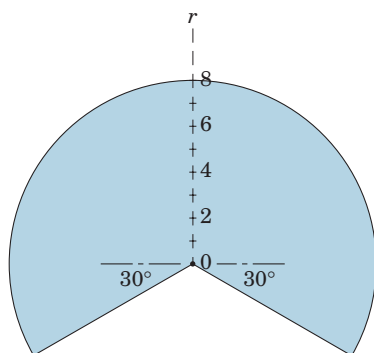
Introductory Problems

- 5/1** Place your pencil on the position of your best visual estimate of the centroid of the triangular area. Check the horizontal position of your estimate by referring to the results of Sample Problem 5/2.



Problem 5/1

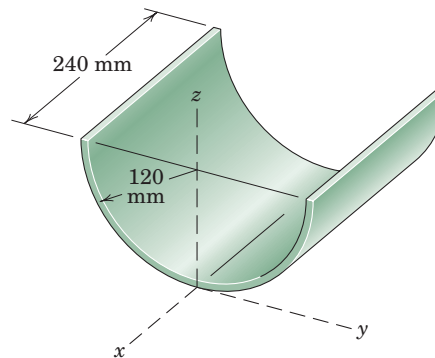
- 5/2** With your pencil make a dot on the position of your best visual estimate of the centroid of the area of the circular sector. Check your estimate by using the results of Sample Problem 5/3.



Problem 5/2

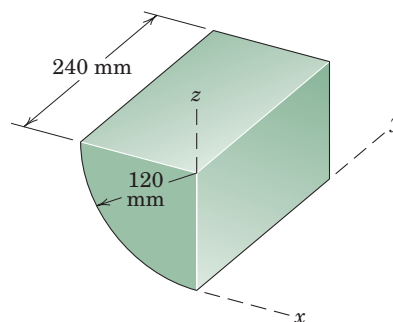
- 5/3** Specify the x - and z -coordinates of the center of mass of the semicylindrical shell.

Ans. $\bar{x} = -120 \text{ mm}$, $\bar{z} = 43.6 \text{ mm}$



Problem 5/3

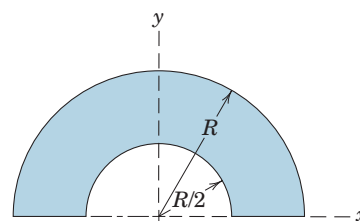
- 5/4** Specify the x -, y -, and z -coordinates of the mass center of the quadrant of the homogeneous solid cylinder.



Problem 5/4

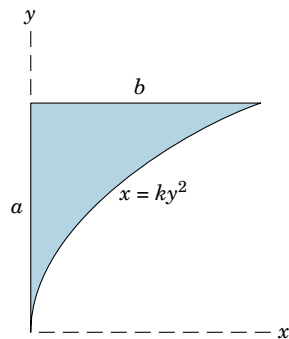
- 5/5** Determine the y -coordinate of the centroid of the area by direct integration.

Ans. $\bar{y} = \frac{14R}{9\pi}$



Problem 5/5

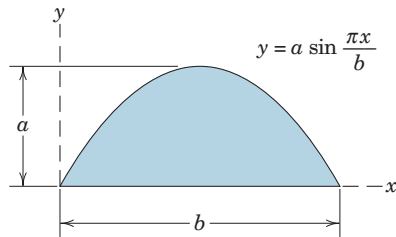
- 5/6** Determine the coordinates of the centroid of the shaded area.



Problem 5/6

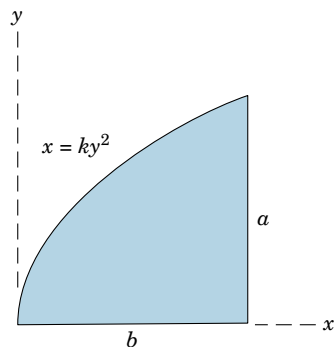
- 5/7** Determine the y -coordinate of the centroid of the area under the sine curve shown.

Ans. $\bar{y} = \frac{\pi a}{8}$



Problem 5/7

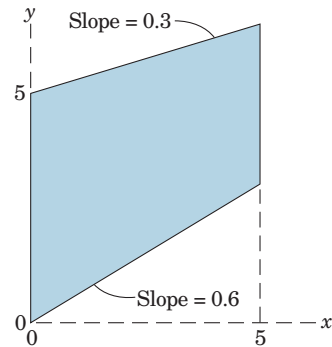
- 5/8** Determine the coordinates of the centroid of the shaded area.



Problem 5/8

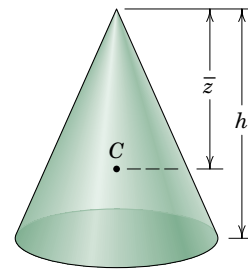
- 5/9** By direct integration, determine the coordinates of the centroid of the trapezoidal area.

Ans. $\bar{x} = 2.35, \bar{y} = 3.56$



Problem 5/9

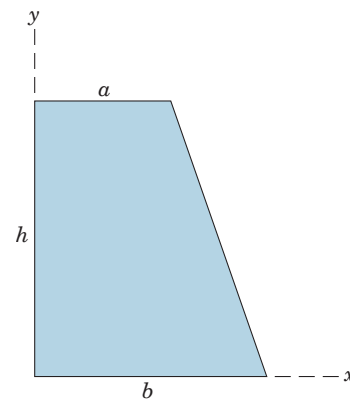
- 5/10** Find the distance \bar{z} from the vertex of the right-circular cone to the centroid of its volume.



Problem 5/10

- 5/11** Determine the x - and y -coordinates of the centroid of the trapezoidal area.

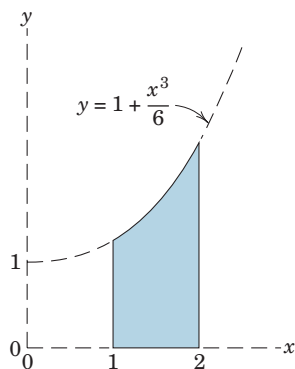
Ans. $\bar{x} = \frac{a^2 + b^2 + ab}{3(a+b)}, \bar{y} = \frac{h(2a+b)}{3(a+b)}$



Problem 5/11

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- 5/12** Determine the x - and y -coordinates of the centroid of the shaded area.

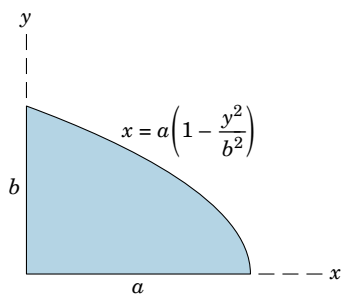


Problem 5/12

Representative Problems

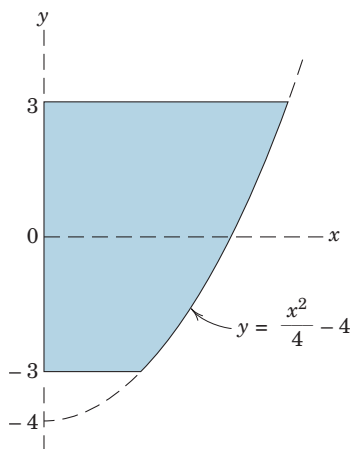
- 5/13** Locate the centroid of the shaded area.

Ans. $\bar{x} = 2a/5$, $\bar{y} = 3b/8$



Problem 5/13

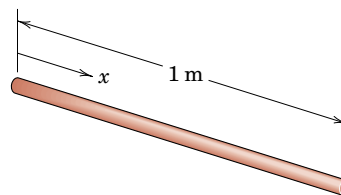
- 5/14** Locate the centroid of the shaded area shown.



Problem 5/14

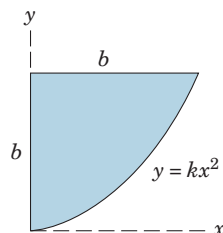
- 5/15** The mass per unit length of the slender rod varies with position according to $\rho = \rho_0(1 - x/2)$, where x is in meters. Determine the location of the center of mass of the rod.

Ans. $\bar{x} = \frac{4}{9}$ m



Problem 5/15

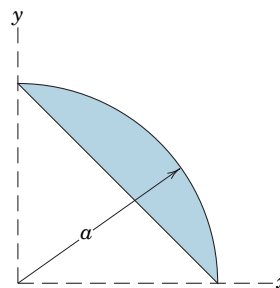
- 5/16** Determine the coordinates of the centroid of the shaded area.



Problem 5/16

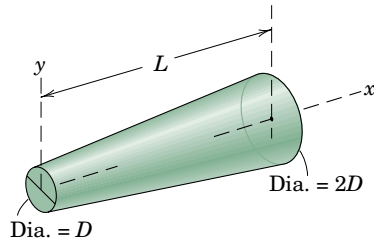
- 5/17** Calculate the coordinates of the centroid of the segment of the circular area.

Ans. $\bar{x} = \bar{y} = \frac{2a}{3(\pi - 2)}$



Problem 5/17

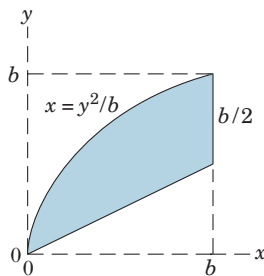
- 5/18** Determine the x -coordinate of the mass center of the tapered steel rod of length L where the diameter at the large end is twice the diameter at the small end.



Problem 5/18

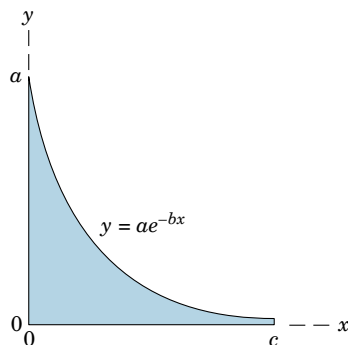
- 5/19** Determine the y -coordinate of the centroid of the shaded area.

Ans. $\bar{y} = \frac{b}{2}$



Problem 5/19

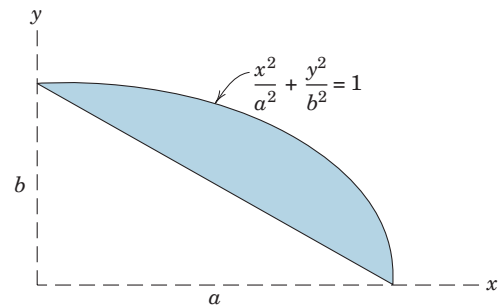
- 5/20** Let $c \rightarrow \infty$ and determine the x - and y -coordinates of the centroid of the shaded area.



Problem 5/20

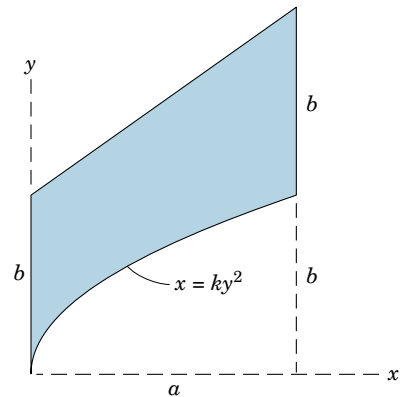
- 5/21** Determine the x - and y -coordinates of the centroid of the shaded area shown.

Ans. $\bar{x} = \frac{a}{3\left(\frac{\pi}{2} - 1\right)}, \bar{y} = \frac{b}{3\left(\frac{\pi}{2} - 1\right)}$



Problem 5/21

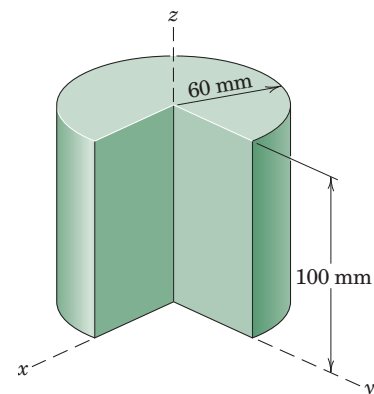
- 5/22** Determine the y -coordinate of the centroid of the shaded area.



Problem 5/22

- 5/23** Use the results of Sample Problem 5/3 to compute the coordinates of the mass center of the portion of the solid homogeneous cylinder shown.

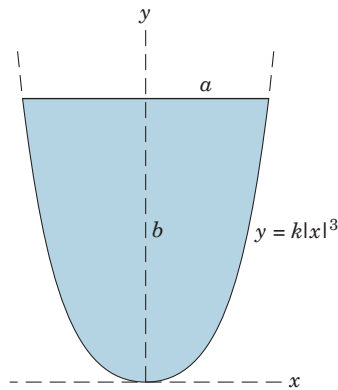
Ans. $\bar{x} = \bar{y} = -8.49 \text{ mm}, \bar{z} = 50 \text{ mm}$



Problem 5/23

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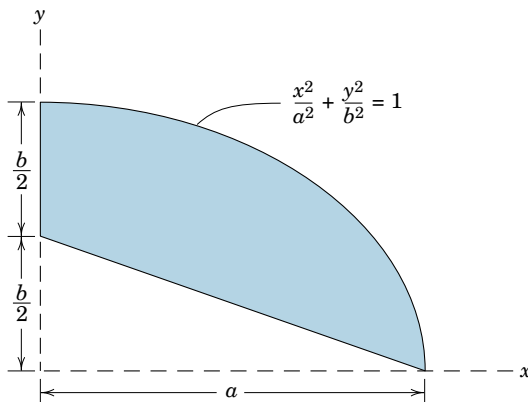
- 5/24** Determine the coordinates of the centroid of the shaded area.



Problem 5/24

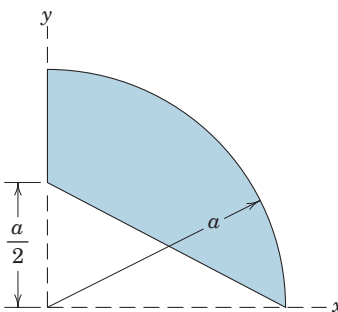
- 5/25** Determine the x - and y -coordinates of the centroid of the shaded area.

$$\text{Ans. } \bar{x} = \frac{a}{\pi - 1}, \bar{y} = \frac{7b}{6(\pi - 1)}$$



Problem 5/25

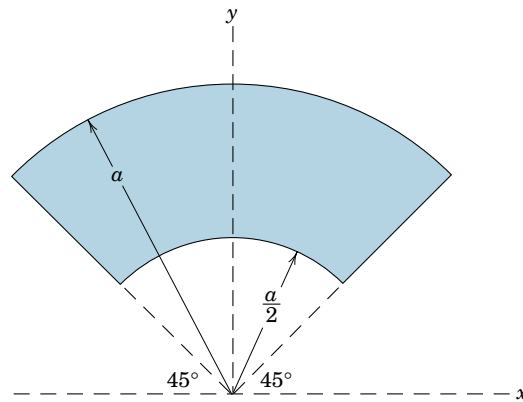
- 5/26** Determine the x - and y -coordinates of the shaded area.



Problem 5/26

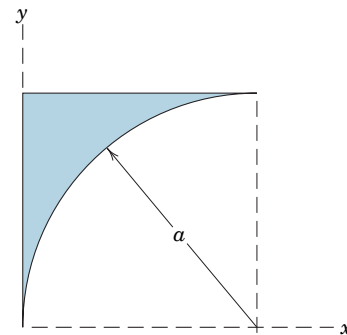
- 5/27** Determine the y -coordinate of the centroid of the shaded area.

$$\text{Ans. } \bar{y} = \frac{14\sqrt{2}}{9\pi} a$$



Problem 5/27

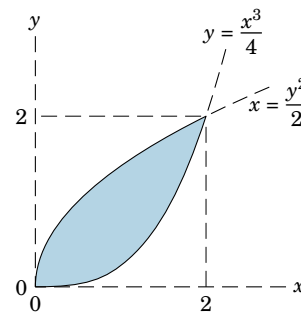
- 5/28** Locate the centroid of the area shown in the figure by direct integration. (*Caution:* Carefully observe the proper sign of the radical involved.)



Problem 5/28

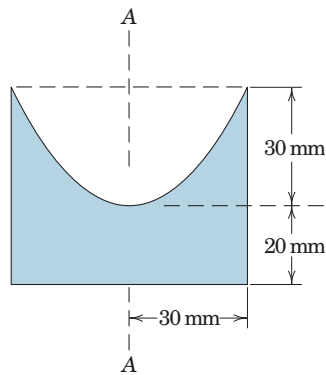
- 5/29** Locate the centroid of the shaded area between the two curves.

$$\text{Ans. } \bar{x} = \frac{24}{25}, \bar{y} = \frac{6}{7}$$



Problem 5/29

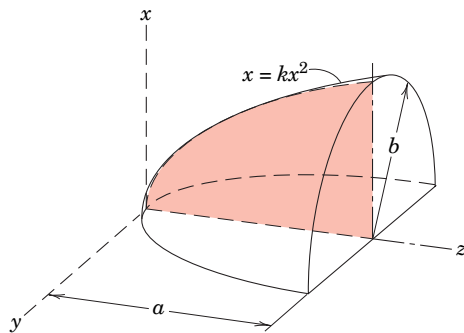
- 5/30** The figure represents a flat piece of sheet metal symmetrical about axis A-A and having a parabolic upper boundary. Choose your own coordinates and calculate the distance \bar{h} from the base to the center of gravity of the piece.



Problem 5/30

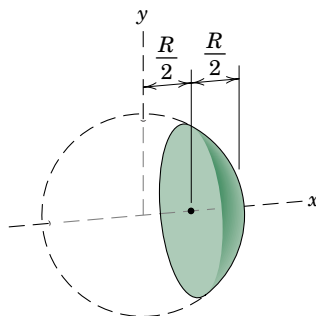
- 5/31** Determine the z -coordinate of the centroid of the volume obtained by revolving the shaded area under the parabola about the z -axis through 180° .

Ans. $\bar{z} = 2a/3$



Problem 5/31

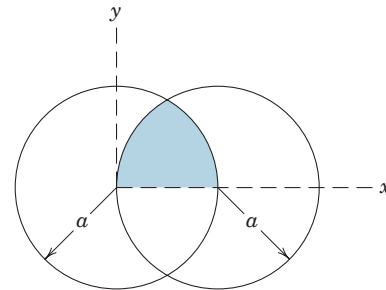
- 5/32** Determine the x -coordinate of the centroid of the solid spherical segment.



Problem 5/32

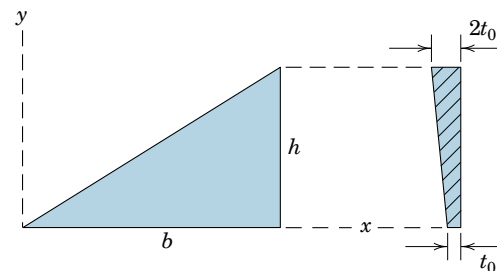
- 5/33** Determine the y -coordinate of the centroid of the shaded area shown. (Observe the caution cited with Prob. 5/28.)

Ans. $\bar{y} = 0.339a$



Problem 5/33

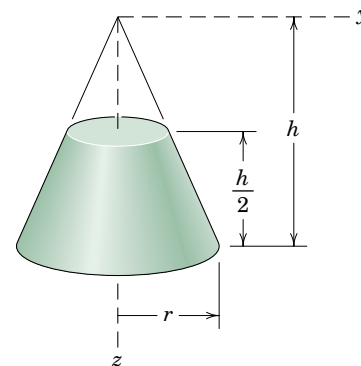
- 5/34** The thickness of the triangular plate varies linearly with y from a value t_0 along its base $y = 0$ to $2t_0$ at $y = h$. Determine the y -coordinate of the center of mass of the plate.



Problem 5/34

- 5/35** Calculate the distance \bar{h} measured from the base to the centroid of the volume of the frustum of the right-circular cone.

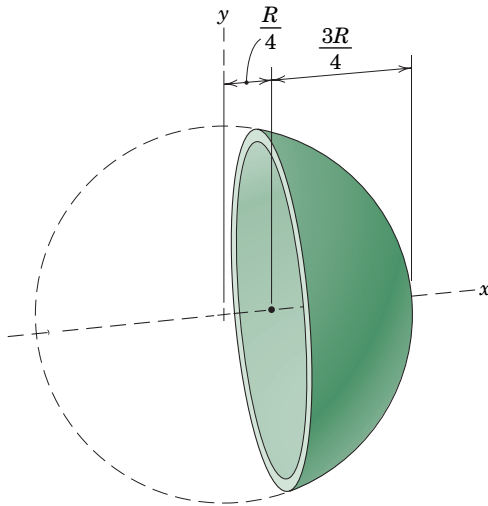
Ans. $\bar{h} = \frac{11}{56}h$



Problem 5/35

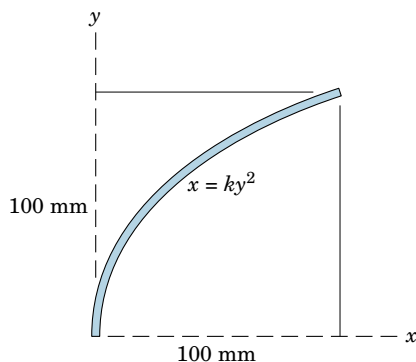
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- 5/36** Determine the x -coordinate of the mass center of the portion of the spherical shell of uniform but small thickness.



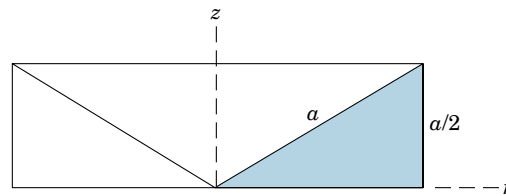
Problem 5/36

- 5/37** The homogeneous slender rod has a uniform cross section and is bent into the shape shown. Calculate the y -coordinate of the mass center of the rod. (Reminder: A differential arc length is $dL = \sqrt{(dx)^2 + (dy)^2} = \sqrt{1 + (dx/dy)^2} dy$.)
- Ans.* $\bar{y} = 57.4 \text{ mm}$



Problem 5/37

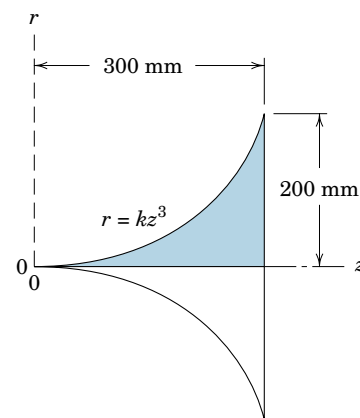
- 5/38** Determine the z -coordinate of the centroid of the volume obtained by revolving the shaded triangular area about the z -axis through 360° .



Problem 5/38

- 5/39** Locate the mass center of the homogeneous solid body whose volume is determined by revolving the shaded area through 360° about the z -axis.

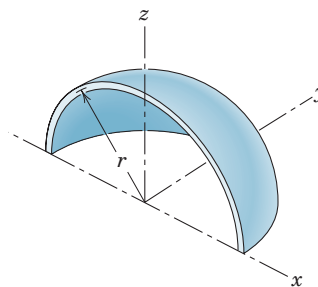
Ans. $\bar{z} = 263 \text{ mm}$



Problem 5/39

- 5/40** Determine the z -coordinate of the mass center of the homogeneous quarter-spherical shell which has a radius r .

Ans. $\bar{z} = \frac{r}{2}$

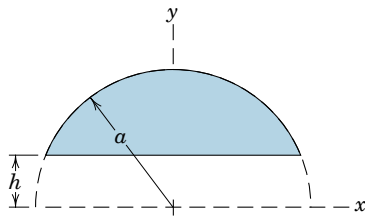


Problem 5/40

- **5/41** Determine the y -coordinate of the centroid of the plane area shown. Set $h = 0$ in your result and compare with the result $\bar{y} = \frac{4a}{3\pi}$ for a full semicircular area (see Sample Problem 5/3 and Table D/3). Also evaluate your result for the conditions $h = \frac{a}{4}$ and $h = \frac{a}{2}$.

$$\text{Ans. } \bar{y} = \frac{\frac{2}{3}[a^2 - h^2]^{3/2}}{a^2 \left[\frac{\pi}{2} - \sin^{-1} \frac{h}{a} \right] - h\sqrt{a^2 - h^2}}$$

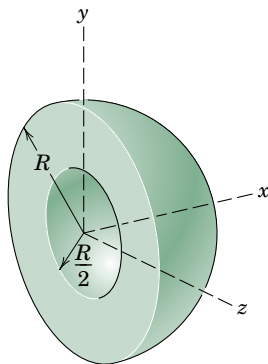
$$h = \frac{a}{4}; \bar{y} = 0.562a, h = \frac{a}{2}; \bar{y} = 0.705a$$



Problem 5/41

- **5/42** Determine the x -coordinate of the mass center of the homogeneous hemisphere with the smaller hemispherical portion removed.

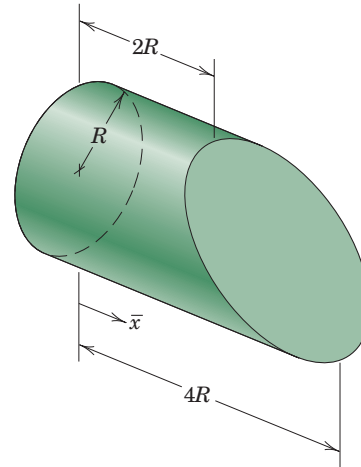
$$\text{Ans. } \bar{x} = \frac{45}{112}R$$



Problem 5/42

- **5/43** Determine the x -coordinate of the mass center of the solid homogeneous body shown.

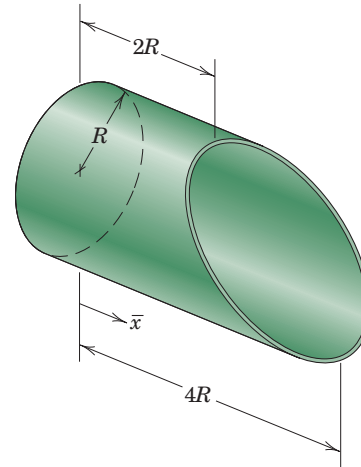
$$\text{Ans. } \bar{x} = 1.542R$$



Problem 5/43

- **5/44** Determine the x -coordinate of the mass center of the cylindrical shell of small uniform thickness.

$$\text{Ans. } \bar{x} = 1.583R$$



Problem 5/44

5/4 COMPOSITE BODIES AND FIGURES; APPROXIMATIONS

When a body or figure can be conveniently divided into several parts whose mass centers are easily determined, we use the principle of moments and treat each part as a finite element of the whole. Such a body is illustrated schematically in Fig. 5/13. Its parts have masses m_1 , m_2 , m_3 with the respective mass-center coordinates \bar{x}_1 , \bar{x}_2 , \bar{x}_3 in the x -direction. The moment principle gives

$$(m_1 + m_2 + m_3)\bar{X} = m_1\bar{x}_1 + m_2\bar{x}_2 + m_3\bar{x}_3$$

where \bar{X} is the x -coordinate of the center of mass of the whole. Similar relations hold for the other two coordinate directions.

We generalize, then, for a body of any number of parts and express the sums in condensed form to obtain the mass-center coordinates

$$\bar{X} = \frac{\Sigma m\bar{x}}{\Sigma m} \quad \bar{Y} = \frac{\Sigma m\bar{y}}{\Sigma m} \quad \bar{Z} = \frac{\Sigma m\bar{z}}{\Sigma m} \quad (5/7)$$

Analogous relations hold for composite lines, areas, and volumes, where the m 's are replaced by L 's, A 's, and V 's, respectively. Note that if a hole or cavity is considered one of the component parts of a composite body or figure, the corresponding mass represented by the cavity or hole is treated as a negative quantity.

An Approximation Method

In practice the boundaries of an area or volume might not be expressible in terms of simple geometrical shapes or as shapes which can be represented mathematically. For such cases we must resort to a method of approximation. As an example, consider the problem of locat-

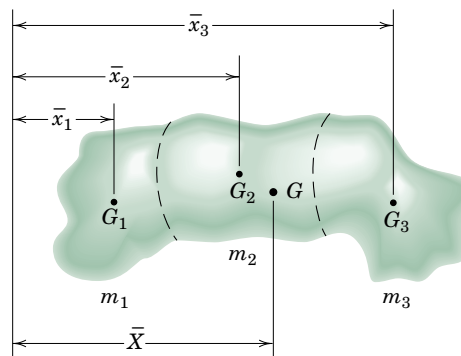


Figure 5/13

ing the centroid C of the irregular area shown in Fig. 5/14. The area is divided into strips of width Δx and variable height h . The area A of each strip, such as the one shown in red, is $h \Delta x$ and is multiplied by the coordinates x_c and y_c of its *centroid* to obtain the moments of the element of area. The sum of the moments for all strips divided by the total area of the strips will give the corresponding centroidal coordinate. A systematic tabulation of the results will permit an orderly evaluation of the total area ΣA , the sums ΣAx_c and ΣAy_c , and the centroidal coordinates

$$\bar{x} = \frac{\Sigma Ax_c}{\Sigma A} \quad \bar{y} = \frac{\Sigma Ay_c}{\Sigma A}$$

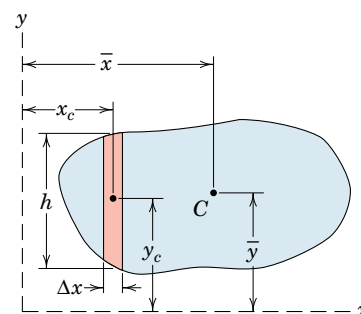


Figure 5/14

We can increase the accuracy of the approximation by decreasing the widths of the strips. In all cases the average height of the strip should be estimated in approximating the areas. Although it is usually advantageous to use elements of constant width, it is not necessary. In fact, we may use elements of any size and shape which approximate the given area to satisfactory accuracy.

Irregular Volumes

To locate the centroid of an irregular volume, we may reduce the problem to one of locating the centroid of an area. Consider the volume shown in Fig. 5/15, where the magnitudes A of the cross-sectional areas normal to the x -direction are plotted against x as shown. A vertical strip of area under the curve is $A \Delta x$, which equals the corresponding element of volume ΔV . Thus, the area under the plotted curve represents the volume of the body, and the x -coordinate of the centroid of the area under the curve is given by

$$\bar{x} = \frac{\Sigma(A \Delta x)x_c}{\Sigma A \Delta x} \quad \text{which equals} \quad \bar{x} = \frac{\Sigma Vx_c}{\Sigma V}$$

for the centroid of the actual volume.

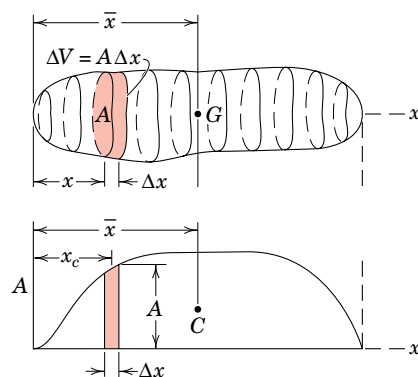


Figure 5/15

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Sample Problem 5/6

Locate the centroid of the shaded area.

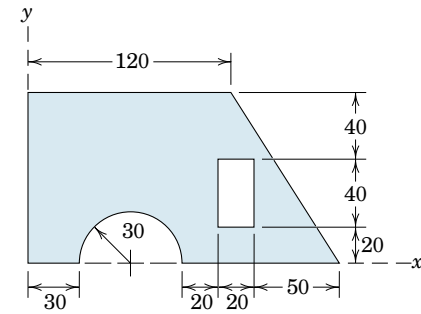
Solution. The composite area is divided into the four elementary shapes shown in the lower figure. The centroid locations of all these shapes may be obtained from Table D/3. Note that the areas of the “holes” (parts 3 and 4) are taken as negative in the following table:

PART	A mm^2	\bar{x} mm	\bar{y} mm	$\bar{x}A$ mm^3	$\bar{y}A$ mm^3
1	12 000	60	50	720 000	600 000
2	3000	140	100/3	420 000	100 000
3	-1414	60	12.73	-84 800	-18 000
4	-800	120	40	-96 000	-32 000
TOTALS	12 790			959 000	650 000

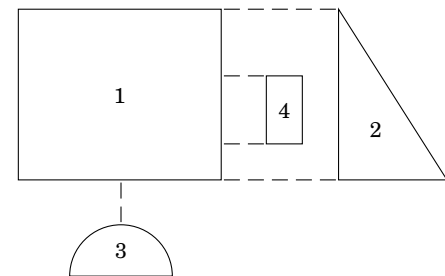
The area counterparts to Eqs. 5/7 are now applied and yield

$$\left[\bar{X} = \frac{\Sigma A\bar{x}}{\Sigma A} \right] \quad \bar{X} = \frac{959\,000}{12\,790} = 75.0 \text{ mm} \quad \text{Ans.}$$

$$\left[\bar{Y} = \frac{\Sigma A\bar{y}}{\Sigma A} \right] \quad \bar{Y} = \frac{650\,000}{12\,790} = 50.8 \text{ mm} \quad \text{Ans.}$$



Dimensions in millimeters



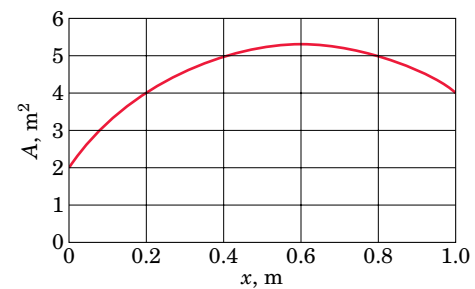
Sample Problem 5/7

Approximate the x -coordinate of the volume centroid of a body whose length is 1 m and whose cross-sectional area varies with x as shown in the figure.

Solution. The body is divided into five sections. For each section, the average area, volume, and centroid location are determined and entered in the following table:

INTERVAL	A_{av} m^2	Volume V m^3	\bar{x} m	$V\bar{x}$ m^4
0–0.2	3	0.6	0.1	0.060
0.2–0.4	4.5	0.90	0.3	0.270
0.4–0.6	5.2	1.04	0.5	0.520
0.6–0.8	5.2	1.04	0.7	0.728
0.8–1.0	4.5	0.90	0.9	0.810
TOTALS		4.48		2.388

$$\textcircled{1} \left[\bar{X} = \frac{\Sigma V\bar{x}}{\Sigma V} \right] \quad \bar{X} = \frac{2.388}{4.48} = 0.533 \text{ m} \quad \text{Ans.}$$



Helpful Hint

① Note that the shape of the body as a function of y and z does not affect \bar{X} .

Sample Problem 5/8

Locate the center of mass of the bracket-and-shaft combination. The vertical face is made from sheet metal which has a mass of 25 kg/m^2 . The material of the horizontal base has a mass of 40 kg/m^2 , and the steel shaft has a density of 7.83 Mg/m^3 .

Solution. The composite body may be considered to be composed of the five elements shown in the lower portion of the illustration. The triangular part will be taken as a negative mass. For the reference axes indicated it is clear by symmetry that the x -coordinate of the center of mass is zero.

The mass m of each part is easily calculated and should need no further explanation. For Part 1 we have from Sample Problem 5/3

$$\bar{z} = \frac{4r}{3\pi} = \frac{4(50)}{3\pi} = 21.2 \text{ mm}$$

For Part 3 we see from Sample Problem 5/2 that the centroid of the triangular mass is one-third of its altitude above its base. Measurement from the coordinate axes becomes

$$\bar{z} = -[150 - 25 - \frac{1}{3}(75)] = -100 \text{ mm}$$

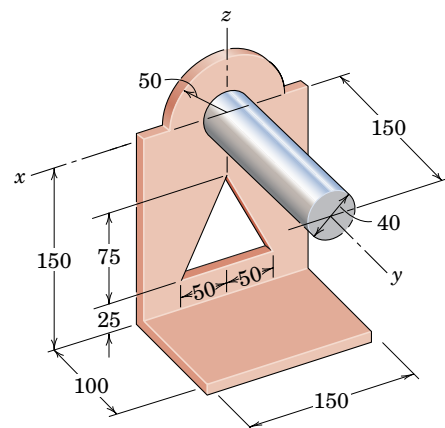
The y - and z -coordinates to the mass centers of the remaining parts should be evident by inspection. The terms involved in applying Eqs. 5/7 are best handled in the form of a table as follows:

PART	m kg	\bar{y} mm	\bar{z} mm	$m\bar{y}$ kg·m	$m\bar{z}$ kg·mm
1	0.098	0	21.2	0	2.08
2	0.562	0	-75.0	0	-42.19
3	-0.094	0	-100.0	0	9.38
4	0.600	50.0	-150.0	30.0	-90.00
5	1.476	75.0	0	110.7	0
TOTALS	2.642			140.7	-120.73

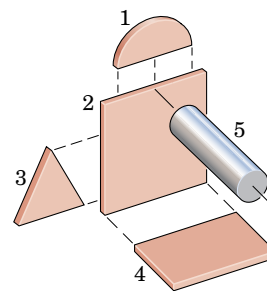
Equations 5/7 are now applied and the results are

$$\left[\bar{Y} = \frac{\Sigma m\bar{y}}{\Sigma m} \right] \quad \bar{Y} = \frac{140.7}{2.642} = 53.3 \text{ mm} \quad \text{Ans.}$$

$$\left[\bar{Z} = \frac{\Sigma m\bar{z}}{\Sigma m} \right] \quad \bar{Z} = \frac{-120.73}{2.642} = -45.7 \text{ mm} \quad \text{Ans.}$$



Dimensions in millimeters

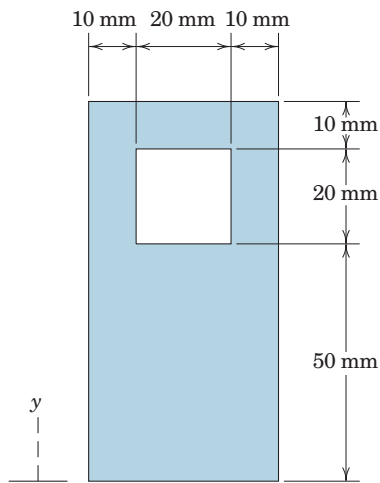


PROBLEMS

Introductory Problems

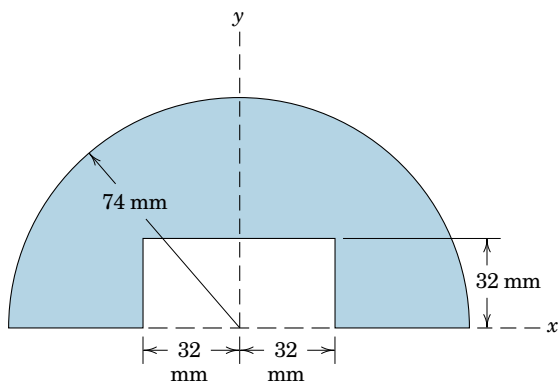
- 5/45** Determine the y -coordinate of the centroid of the shaded area.

Ans. $\bar{Y} = 37.1 \text{ mm}$



Problem 5/45

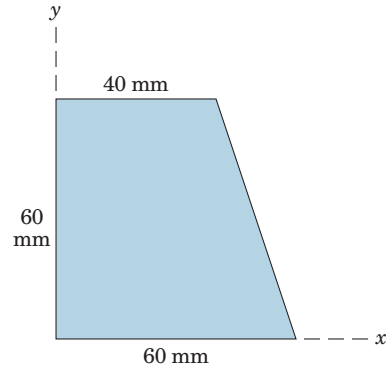
- 5/46** Calculate the y -coordinate of the centroid of the shaded area.



Problem 5/46

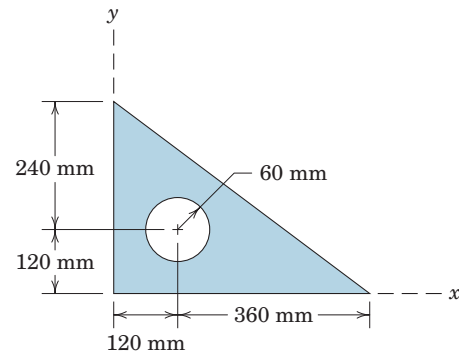
- 5/47** Determine the coordinates of the centroid of the trapezoidal area shown.

Ans. $\bar{X} = 25.3 \text{ mm}$, $\bar{Y} = 28.0 \text{ mm}$



Problem 5/47

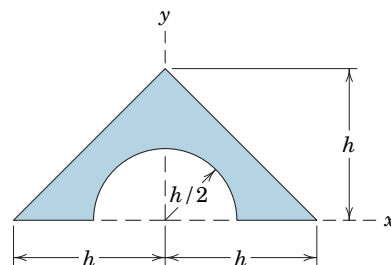
- 5/48** Determine the x - and y -coordinates of the centroid of the shaded area.



Problem 5/48

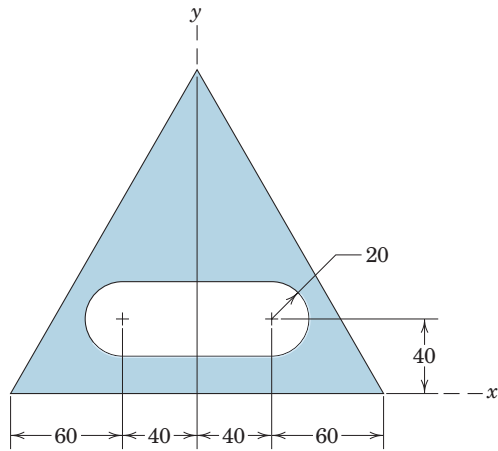
- 5/49** Determine the y -coordinate of the centroid of the shaded area in terms of h .

Ans. $\bar{Y} = 0.412h$



Problem 5/49

- 5/50** Determine the y -coordinate of the centroid of the shaded area. The triangle is equilateral.

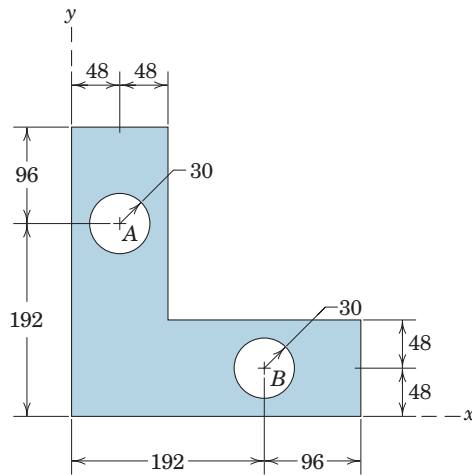


Dimensions in millimeters

Problem 5/50

- 5/51** Determine the x - and y -coordinates of the centroid of the shaded area.

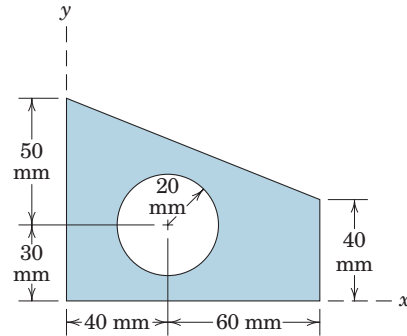
Ans. $\bar{X} = \bar{Y} = 103.6 \text{ mm}$



Dimensions in millimeters

Problem 5/51

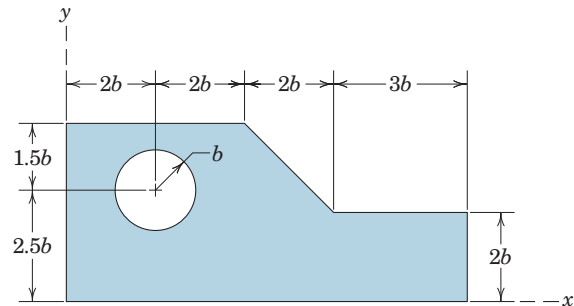
- 5/52** Determine the x - and y -coordinates of the centroid of the shaded area.



Problem 5/52

- 5/53** Determine the x - and y -coordinates of the centroid of the shaded area.

Ans. $\bar{X} = 4.02b$, $\bar{Y} = 1.588b$

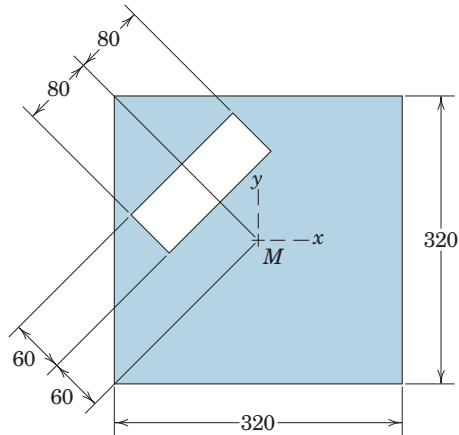


Problem 5/53

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Representative Problems

- 5/54** By inspection, state the quadrant in which the centroid of the shaded area is located. Then determine the coordinates of the centroid. The plate center is M .

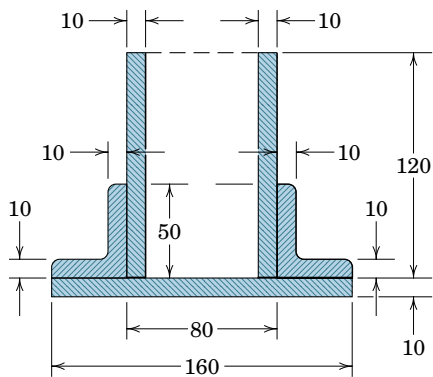


Dimensions in millimeters

Problem 5/54

- 5/55** Determine the distance \bar{H} from the bottom of the base plate to the centroid of the built-up structural section shown.

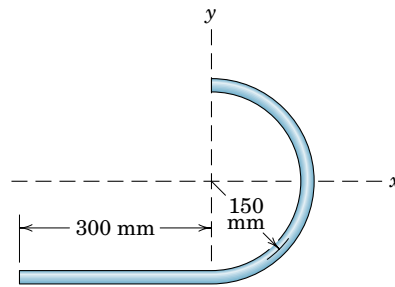
Ans. $\bar{H} = 39.3$ mm



Dimensions in millimeters

Problem 5/55

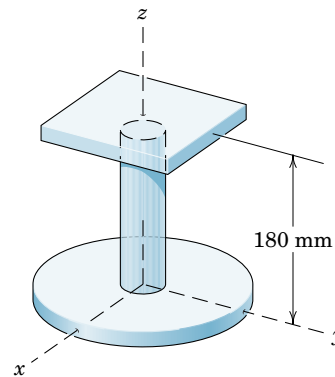
- 5/56** Locate the mass center of the slender rod bent into the shape shown.



Problem 5/56

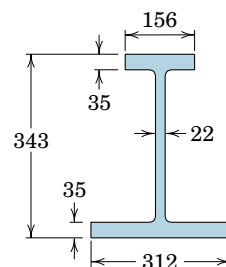
- 5/57** The rigidly connected unit consists of a 2-kg circular disk, a 1.5-kg round shaft, and a 1-kg square plate. Determine the z -coordinate of the mass center of the unit.

Ans. $\bar{Z} = 70$ mm



Problem 5/57

- 5/58** Determine the height above the base of the centroid of the cross-sectional area of the beam. Neglect the fillets.

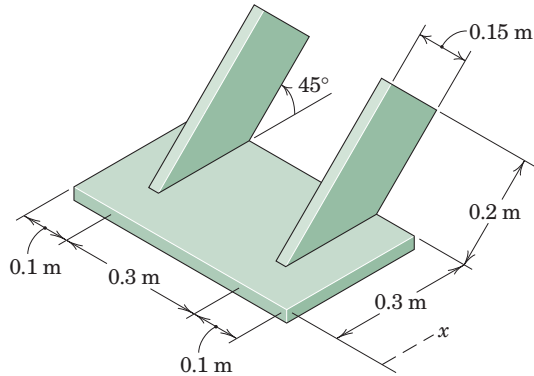


Dimensions in millimeters

Problem 5/58

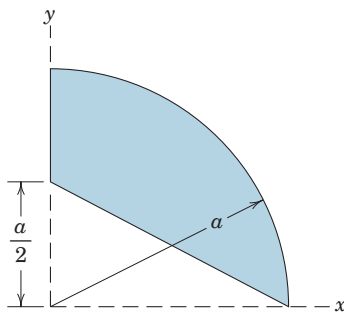
- 5/59** Determine the x -coordinate of the mass center of the bracket constructed of uniform steel plate.

Ans. $\bar{X} = 0.1975 \text{ m}$



Problem 5/59

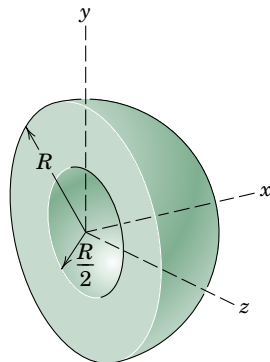
- 5/60** Determine the x - and y -coordinates of the centroid of the area of Prob. 5/26 by the method of this article.



Problem 5/60

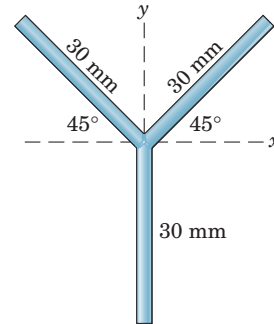
- 5/61** The homogeneous hemisphere with the smaller hemispherical portion removed is repeated here from Prob. 5/42. By the method of this article, determine the x -coordinate of the mass center.

Ans. $\bar{X} = \frac{45}{112}R$



Problem 5/61

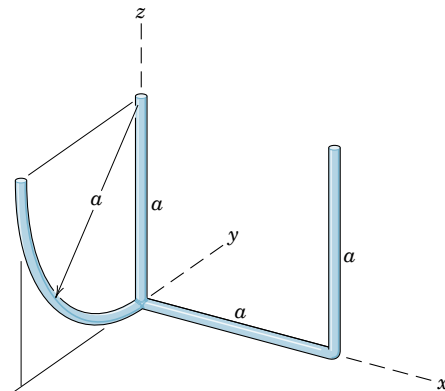
- 5/62** The two upper lengths of the welded Y-shaped assembly of uniform slender rods have a mass per unit length of 0.3 kg/m , while the lower length has a mass of 0.5 kg/m . Locate the mass center of the assembly.



Problem 5/62

- 5/63** Determine the coordinates of the mass center of the welded assembly of uniform slender rods made from the same bar stock.

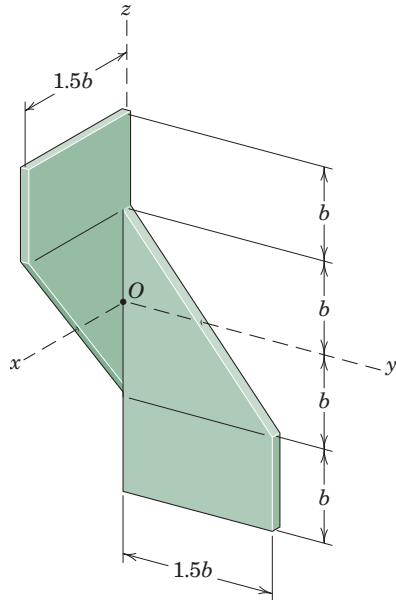
Ans. $\bar{X} = \frac{3a}{6 + \pi}$, $\bar{Y} = -\frac{2a}{6 + \pi}$, $\bar{Z} = \frac{\pi a}{6 + \pi}$



Problem 5/63

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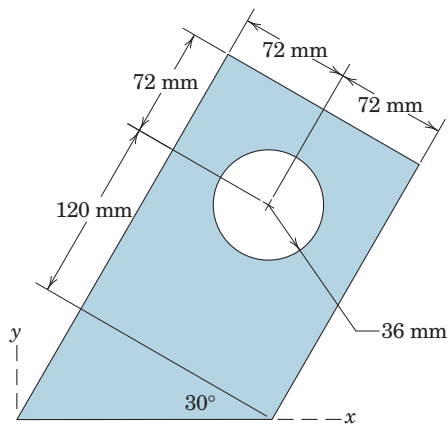
- 5/64** Determine the x -, y -, and z -coordinates of the mass center of the sheet-metal bracket whose thickness is small in comparison with the other dimensions.



Problem 5/64

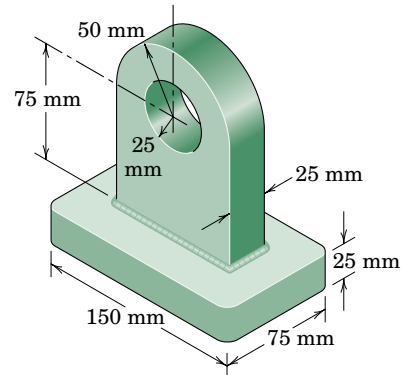
- 5/65** Determine the x - and y -coordinates of the centroid of the shaded area.

Ans. $\bar{X} = 133.5 \text{ mm}$, $\bar{Y} = 97.0 \text{ mm}$



Problem 5/65

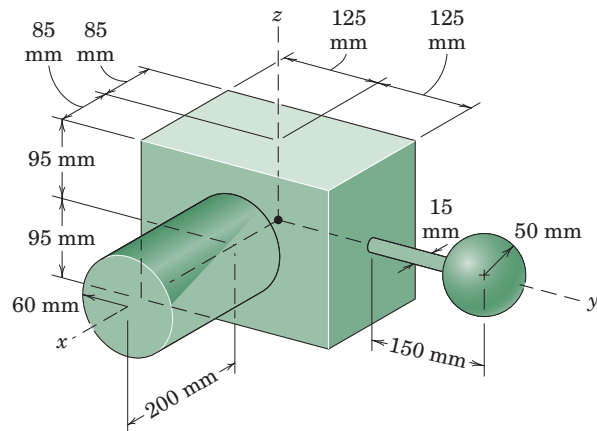
- 5/66** Determine the distance \bar{H} from the bottom of the base to the mass center of the bracket casting.



Problem 5/66

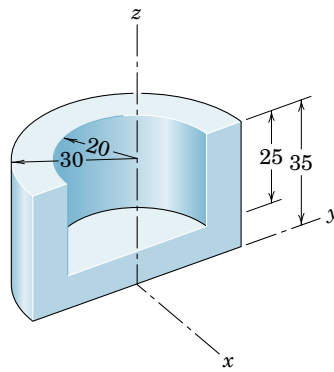
- 5/67** An underwater instrument is modeled as shown in the figure. Determine the coordinates of the centroid of this composite volume.

Ans. $\bar{X} = 38.5 \text{ mm}$, $\bar{Y} = 13.52 \text{ mm}$, $\bar{Z} = 0$



Problem 5/67

- 5/68** Calculate the coordinates of the mass center of the metal die casting shown.

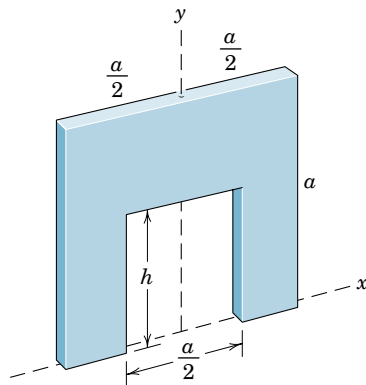


Dimensions in millimeters

Problem 5/68

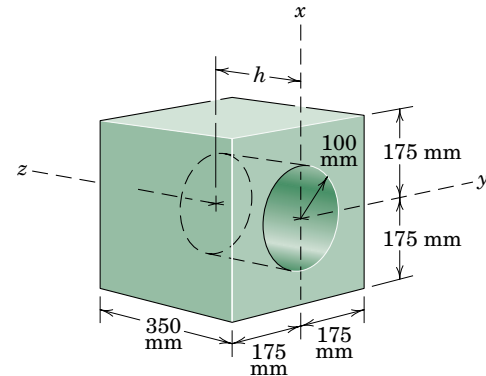
- 5/69** Determine the dimension h of the rectangular opening in the square plate which will result in the mass center of the remaining plate being as close to the upper edge as possible.

Ans. $h = 0.586a$



Problem 5/69

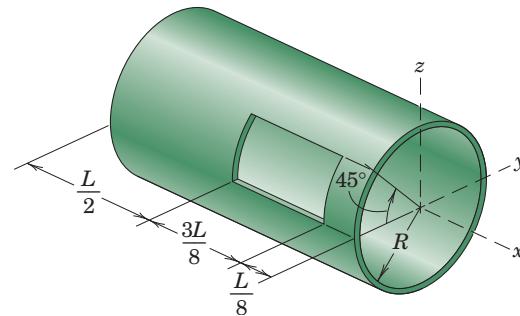
- 5/70** Determine the depth h of the circular hole in the cube for which the z -coordinate of the mass center will have the maximum possible value.



Problem 5/70

- 5/71** An opening is formed in the thin cylindrical shell. Determine the x -, y -, and z -coordinates of the mass center of the homogeneous body.

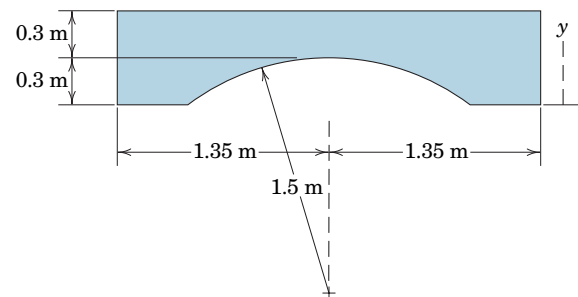
Ans. $\bar{X} = -0.509L$, $\bar{Y} = 0.0443R$, $\bar{Z} = -0.01834R$



Problem 5/71

- 5/72** Determine the y -coordinate of the centroid of the shaded area. Use the result of Prob. 5/41.

Ans. $\bar{Y} = 0.353 \text{ m}$



Problem 5/72

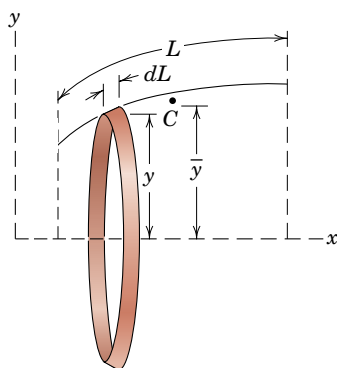


Figure 5/16

5/5 THEOREMS OF PAPPUS*

A very simple method exists for calculating the surface area generated by revolving a plane curve about a nonintersecting axis in the plane of the curve. In Fig. 5/16 the line segment of length L in the x - y plane generates a surface when revolved about the x -axis. An element of this surface is the ring generated by dL . The area of this ring is its circumference times its slant height or $dA = 2\pi y dL$. The total area is then

$$A = 2\pi \int y dL$$

Because $\bar{y}L = \int y dL$, the area becomes

$$A = 2\pi \bar{y}L \quad (5/8)$$

where \bar{y} is the y -coordinate of the centroid C for the line of length L . Thus, the generated area is the same as the lateral area of a right-circular cylinder of length L and radius \bar{y} .

In the case of a volume generated by revolving an area about a nonintersecting line in its plane, an equally simple relation exists for finding the volume. An element of the volume generated by revolving the area A about the x -axis, Fig. 5/17, is the elemental ring of cross-section dA and radius y . The volume of the element is its circumference times dA or $dV = 2\pi y dA$, and the total volume is

$$V = 2\pi \int y dA$$

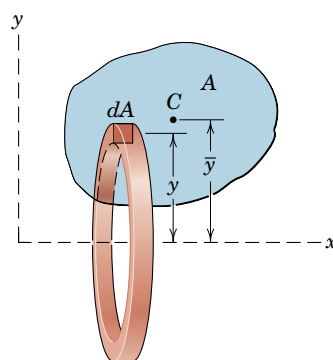


Figure 5/17

*Attributed to Pappus of Alexandria, a Greek geometer who lived in the third century A.D. The theorems often bear the name of Guldinus (Paul Guldin, 1577–1643), who claimed original authorship, although the works of Pappus were apparently known to him.

Because $\bar{y}A = \int y \, dA$, the volume becomes

$$V = 2\pi\bar{y}A \quad (5/9)$$

where \bar{y} is the y -coordinate of the centroid C of the revolved area A . Thus, we obtain the generated volume by multiplying the generating area by the circumference of the circular path described by its centroid.

The two theorems of Pappus, expressed by Eqs. 5/8 and 5/9, are useful for determining areas and volumes of revolution. They are also used to find the centroids of plane curves and plane areas when we know the corresponding areas and volumes created by revolving these figures about a nonintersecting axis. Dividing the area or volume by 2π times the corresponding line segment length or plane area gives the distance from the centroid to the axis.

If a line or an area is revolved through an angle θ less than 2π , we can determine the generated surface or volume by replacing 2π by θ in Eqs. 5/8 and 5/9. Thus, the more general relations are

$$A = \theta\bar{y}L \quad (5/8a)$$

and

$$V = \theta\bar{y}A \quad (5/9a)$$

where θ is expressed in radians.

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Sample Problem 5/9

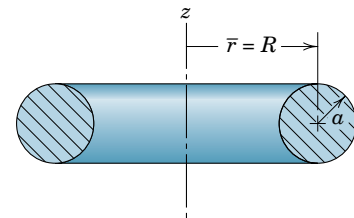
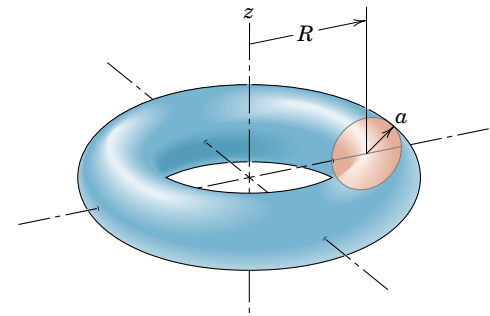
Determine the volume V and surface area A of the complete torus of circular cross section.

Solution. The torus can be generated by revolving the circular area of radius a through 360° about the z -axis. With the use of Eq. 5/9a, we have

$$\textcircled{1} \quad V = \theta \bar{r} A = 2\pi(R)(\pi a^2) = 2\pi^2 R a^2 \quad \text{Ans.}$$

Similarly, using Eq. 5/8a gives

$$A = \theta \bar{r} L = 2\pi(R)(2\pi a) = 4\pi^2 R a \quad \text{Ans.}$$



Helpful Hint

- ① We note that the angle θ of revolution is 2π for the complete ring. This common but special-case result is given by Eq. 5/9.

Sample Problem 5/10

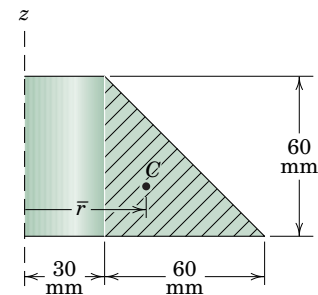
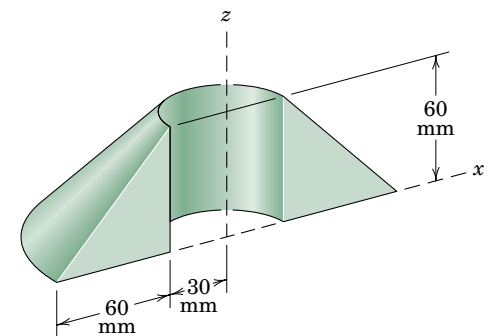
Calculate the volume V of the solid generated by revolving the 60-mm right-triangular area through 180° about the z -axis. If this body were constructed of steel, what would be its mass m ?

Solution. With the angle of revolution $\theta = 180^\circ$, Eq. 5/9a gives

$$\textcircled{1} \quad V = \theta \bar{r} A = \pi \left[30 + \frac{1}{3}(60) \right] \left[\frac{1}{2}(60)(60) \right] = 2.83(10^5) \text{ mm}^3 \quad \text{Ans.}$$

The mass of the body is then

$$m = \rho V = \left[7830 \frac{\text{kg}}{\text{m}^3} \right] [2.83(10^5) \text{ mm}^3] \left[\frac{1 \text{ m}}{1000 \text{ mm}} \right]^3 = 2.21 \text{ kg} \quad \text{Ans.}$$



Helpful Hint

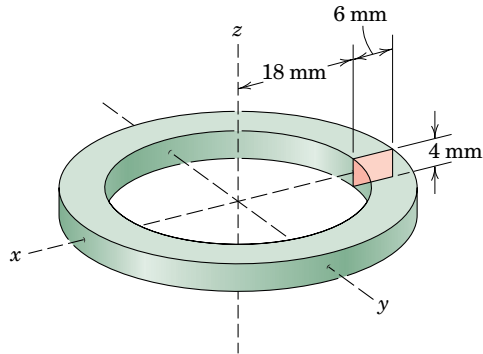
- ① Note that θ must be in radians.

PROBLEMS

Introductory Problems

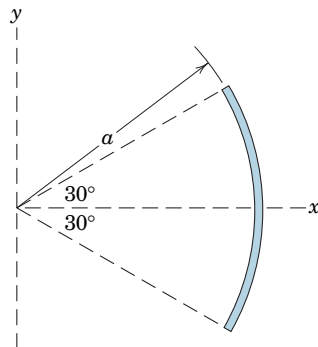
- 5/73** Using the methods of this article, determine the surface area A and volume V of the body formed by revolving the rectangular area through 360° about the z -axis.

Ans. $A = 2640 \text{ mm}^2$, $V = 3170 \text{ mm}^3$



Problem 5/73

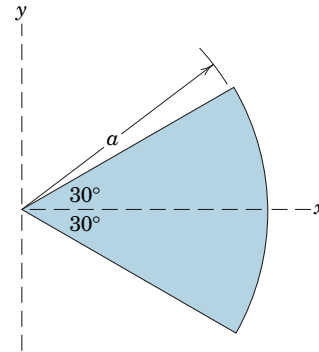
- 5/74** The circular arc is rotated through 360° about the y -axis. Determine the outer surface area S of the resulting body, which is a portion of a sphere.



Problem 5/74

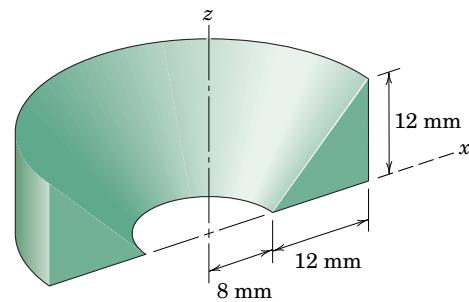
- 5/75** The area of the circular sector is rotated through 180° about the y -axis. Determine the volume of the resulting body, which is a portion of a sphere.

Ans. $V = \frac{\pi a^3}{3}$



Problem 5/75

- 5/76** Compute the volume V of the solid generated by revolving the right triangle about the z -axis through 180° .

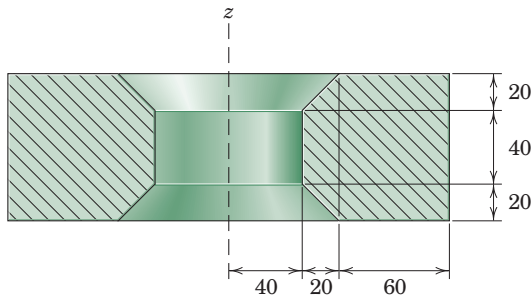


Problem 5/76

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- 5/77** The body shown in cross section is a half-circular ring formed by revolving the cross-hatched area 180° about the z -axis. Determine the surface area A of the body.

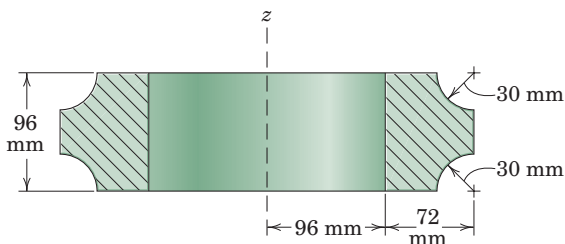
Ans. $A = 90\,000 \text{ mm}^2$



Dimensions in millimeters

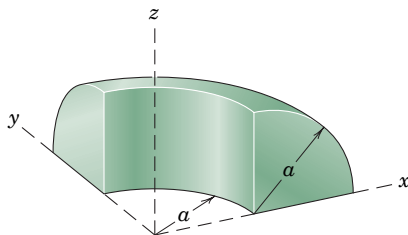
Problem 5/77

- 5/78** Calculate the volume V of the complete ring of cross section shown.

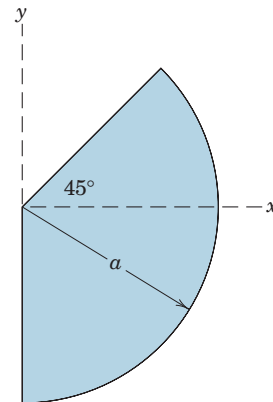

Problem 5/78

- 5/79** Determine the volume V generated by revolving the quarter-circular area about the z -axis through an angle of 90° .

Ans. $V = \left(\frac{\pi a^3}{24} \right) (3\pi + 4)$

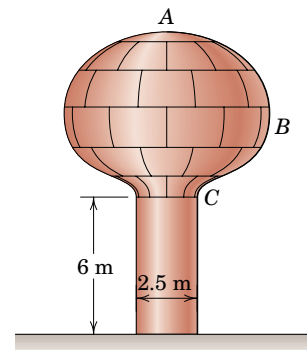

Problem 5/79
Representative Problems

- 5/80** The area shown is rotated through 360° about the y -axis. Determine the volume of the resulting body, which is a sphere with a significant portion removed.

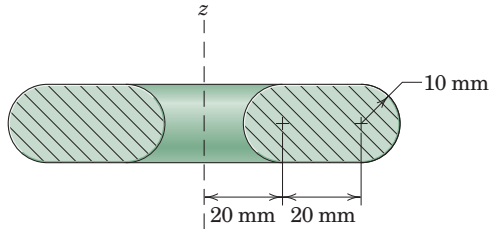

Problem 5/80

- 5/81** The water storage tank is a shell of revolution and is to be sprayed with two coats of paint which has a coverage of 16 m^2 per gallon. The engineer (who remembers mechanics) consults a scale drawing of the tank and determines that the curved line ABC has a length of 10 m and that its centroid is 2.50 m from the centerline of the tank. How many gallons of paint will be used for the tank including the vertical cylindrical column?

Ans. 25.5 liters


Problem 5/81

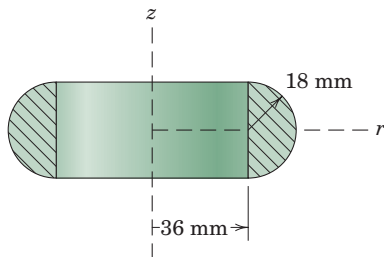
- 5/82** Determine the total surface area A and volume V of the complete solid shown in cross section. Determine the mass of the body if it is constructed of steel.



Problem 5/82

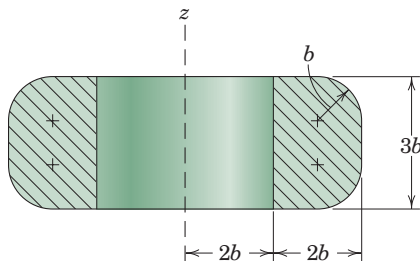
- 5/83** Calculate the volume V of the rubber gasket formed by the complete ring of the semicircular cross section shown. Also compute the surface area A of the outside of the ring.

Ans. $V = 13.95(10^4) \text{ mm}^3$, $A = 1.686(10^4) \text{ mm}^2$



Problem 5/83

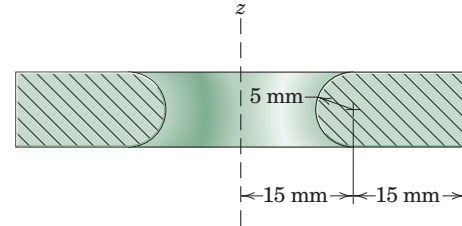
- 5/84** The body shown in cross section is a complete circular ring formed by revolving the cross-hatched area about the z -axis. Determine the surface area A and volume V of the body.



Problem 5/84

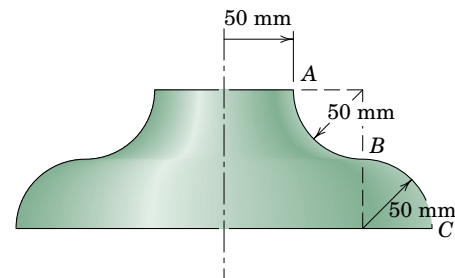
- 5/85** The body shown in cross section is a large neoprene washer. Compute its surface area A and volume V .

Ans. $A = 7290 \text{ mm}^2$, $V = 24\,400 \text{ mm}^3$



Problem 5/85

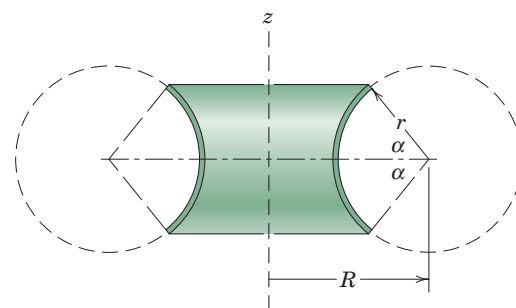
- 5/86** The two circular arcs AB and BC are revolved about the vertical axis to obtain the surface of revolution shown. Compute the area A of the outside of this surface.



Problem 5/86

- 5/87** A thin shell, shown in section, has the form generated by revolving the arc about the z -axis through 360° . Determine the surface area A of one of the two sides of the shell.

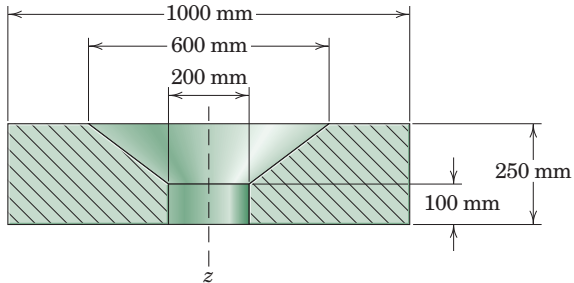
Ans. $A = 4\pi r(R\alpha - r \sin \alpha)$



Problem 5/87

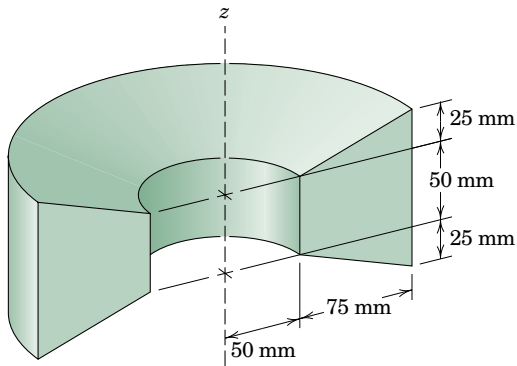
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- 5/88** Calculate the volume formed by completely revolving the cross-sectional area shown about the z -axis of symmetry.



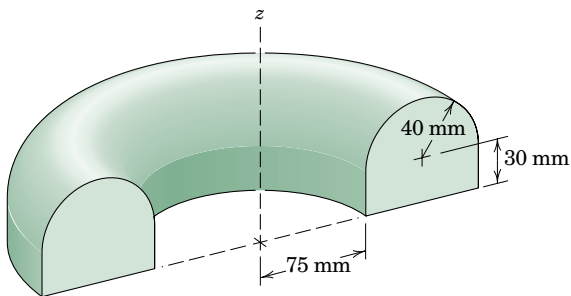
Problem 5/88

- 5/89** Calculate the weight W of the aluminum casting shown. The solid is generated by revolving the trapezoidal area shown about the z -axis through 180° .
Ans. $W = 42.7 \text{ N}$



Problem 5/89

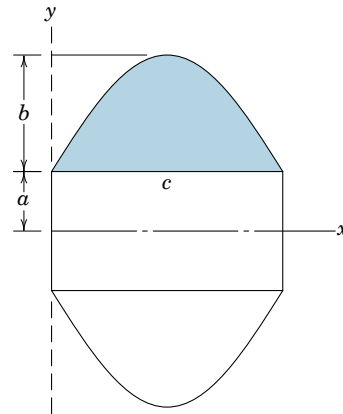
- 5/90** Determine the volume V and total surface area A of the solid generated by revolving the area shown through 180° about the z -axis.



Problem 5/90

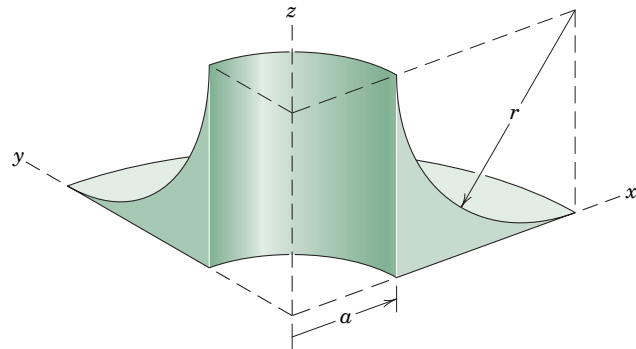
- 5/91** The shaded area is bounded by one half-cycle of a sine wave and the axis of the sine wave. Determine the volume generated by completely revolving the area about the x -axis.

$$\text{Ans. } V = 4bc \left(a + \frac{b\pi}{8} \right)$$



Problem 5/91

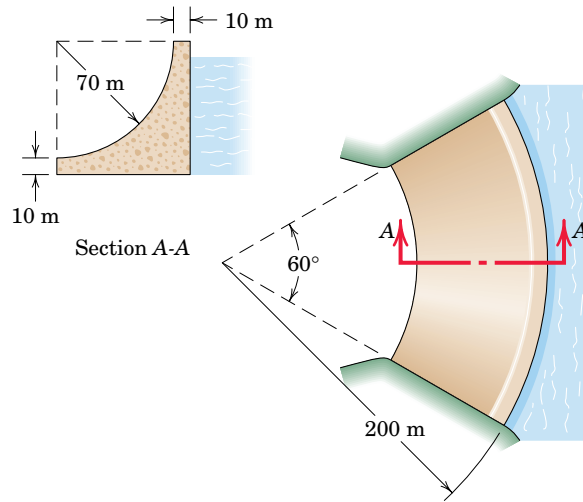
- 5/92** Find the volume V of the solid generated by revolving the shaded area about the z -axis through 90° .



Problem 5/92

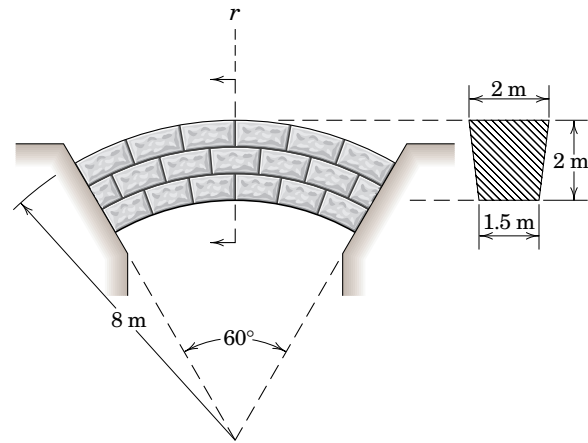
5/93 Calculate the mass m of concrete required to construct the arched dam shown. Concrete has a density of 2.40 Mg/m^3 .

Ans. $m = 1.126(10^6) \text{ Mg}$



Problem 5/93

5/94 In order to provide sufficient support for the stone masonry arch designed as shown, it is necessary to know its total weight W . Use the results of Prob. 5/11 and determine W . The density of stone masonry is 2.40 Mg/m^3 .



Problem 5/94

SECTION B SPECIAL TOPICS

5/6 BEAMS—EXTERNAL EFFECTS

Beams are structural members which offer resistance to bending due to applied loads. Most beams are long prismatic bars, and the loads are usually applied normal to the axes of the bars.

Beams are undoubtedly the most important of all structural members, so it is important to understand the basic theory underlying their design. To analyze the load-carrying capacities of a beam we must first establish the equilibrium requirements of the beam as a whole and any portion of it considered separately. Second, we must establish the relations between the resulting forces and the accompanying internal resistance of the beam to support these forces. The first part of this analysis requires the application of the principles of statics. The second part involves the strength characteristics of the material and is usually treated in studies of the mechanics of solids or the mechanics of materials.

This article is concerned with the *external* loading and reactions acting on a beam. In Art. 5/7 we calculate the distribution along the beam of the *internal* force and moment.

Types of Beams

Beams supported so that their external support reactions can be calculated by the methods of statics alone are called *statically determinate beams*. A beam which has more supports than needed to provide equilibrium is *statically indeterminate*. To determine the support reactions for such a beam we must consider its load-deformation properties in addition to the equations of static equilibrium. Figure 5/18 shows examples

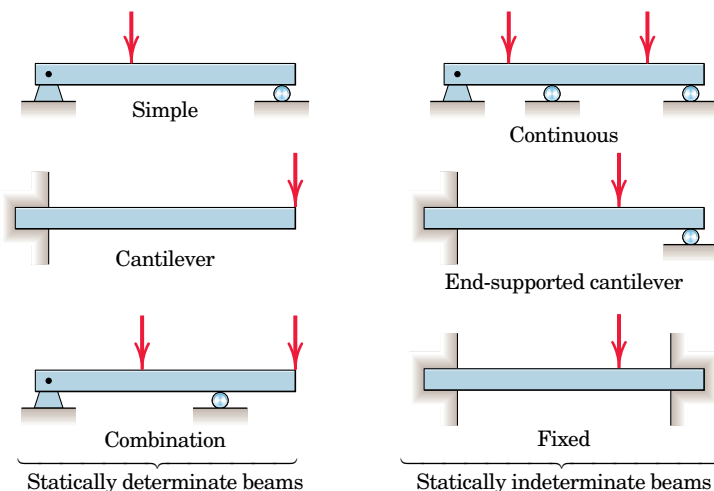


Figure 5/18

of both types of beams. In this article we will analyze statically determinate beams only.

Beams may also be identified by the type of external loading they support. The beams in Fig. 5/18 are supporting concentrated loads, whereas the beam in Fig. 5/19 is supporting a distributed load. The intensity w of a distributed load may be expressed as force per unit length of beam. The intensity may be constant or variable, continuous or discontinuous. The intensity of the loading in Fig. 5/19 is constant from C to D and variable from A to C and from D to B . The intensity is discontinuous at D , where it changes magnitude abruptly. Although the intensity itself is not discontinuous at C , the rate of change of intensity dw/dx is discontinuous.

Distributed Loads

Loading intensities which are constant or which vary linearly are easily handled. Figure 5/20 illustrates the three most common cases and the resultants of the distributed loads in each case.

In cases *a* and *b* of Fig. 5/20, we see that the resultant load R is represented by the area formed by the intensity w (force per unit length of beam) and the length L over which the force is distributed. The resultant passes through the centroid of this area.

In part *c* of Fig. 5/20, the trapezoidal area is broken into a rectangular and a triangular area, and the corresponding resultants R_1 and R_2 of these subareas are determined separately. Note that a single resultant could be determined by using the composite technique for finding centroids, which was discussed in Art. 5/4. Usually, however, the determination of a single resultant is unnecessary.

For a more general load distribution, Fig. 5/21, we must start with a differential increment of force $dR = w dx$. The total load R is then the sum of the differential forces, or

$$R = \int w dx$$

As before, the resultant R is located at the centroid of the area under consideration. The x -coordinate of this centroid is found by the principle of moments $R\bar{x} = \int xw dx$, or

$$\bar{x} = \frac{\int xw dx}{R}$$

For the distribution of Fig. 5/21, the vertical coordinate of the centroid need not be found.

Once the distributed loads have been reduced to their equivalent concentrated loads, the external reactions acting on the beam may be found by a straightforward static analysis as developed in Chapter 3.

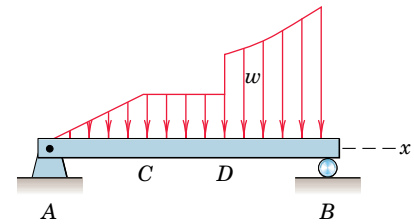


Figure 5/19

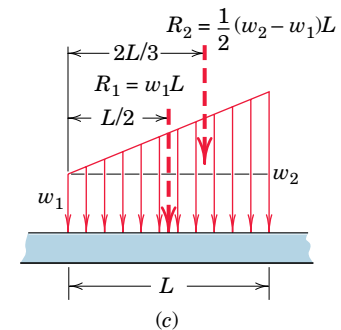
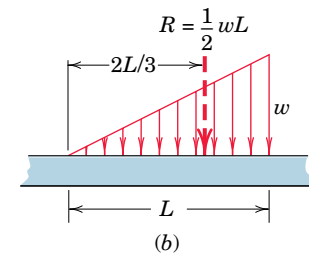
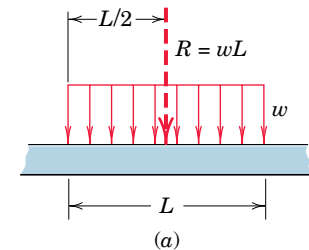


Figure 5/20

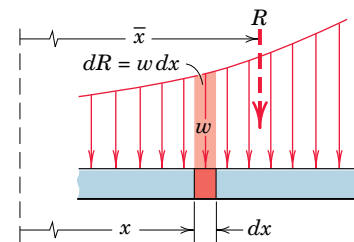
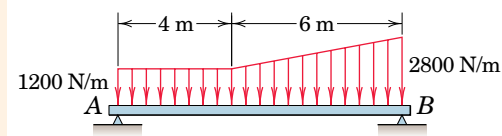


Figure 5/21

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Sample Problem 5/11

Determine the equivalent concentrated load(s) and external reactions for the simply supported beam which is subjected to the distributed load shown.



Solution. The area associated with the load distribution is divided into the rectangular and triangular areas shown. The concentrated-load values are determined by computing the areas, and these loads are located at the centroids of the respective areas.

①

Once the concentrated loads are determined, they are placed on the free-body diagram of the beam along with the external reactions at A and B. Using principles of equilibrium, we have

$$[\Sigma M_A = 0] \quad 12\,000(5) + 4800(8) - R_B(10) = 0$$

$$R_B = 9840 \text{ N or } 9.84 \text{ kN}$$

Ans.

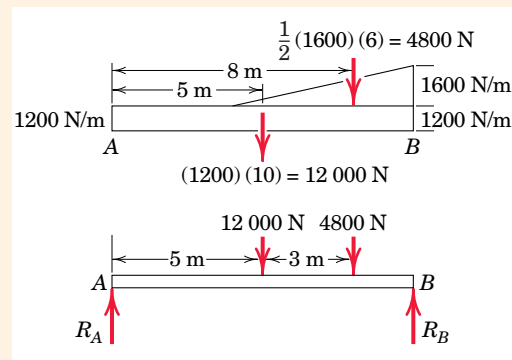
$$[\Sigma M_B = 0] \quad R_A(10) - 12\,000(5) - 4800(2) = 0$$

$$R_A = 6960 \text{ N or } 6.96 \text{ kN}$$

Ans.

Helpful Hint

- ① Note that it is usually unnecessary to reduce a given distributed load to a single concentrated load.



Sample Problem 5/12

Determine the reaction at the support A of the loaded cantilever beam.

Solution. The constants in the load distribution are found to be $w_0 = 1000$ N/m and $k = 2$ N/m⁴. The load R is then

①

$$R = \int w \, dx = \int_0^8 (1000 + 2x^3) \, dx = \left(1000x + \frac{x^4}{2} \right) \Big|_0^8 = 10\,050 \text{ N}$$

② The x -coordinate of the centroid of the area is found by

$$\begin{aligned} \bar{x} &= \frac{\int xw \, dx}{R} = \frac{1}{10\,050} \int_0^8 x(1000 + 2x^3) \, dx \\ &= \frac{1}{10\,050} \left(500x^2 + \frac{2}{5}x^5 \right) \Big|_0^8 = 4.49 \text{ m} \end{aligned}$$

From the free-body diagram of the beam, we have

$$[\Sigma M_A = 0] \quad M_A - (10\,050)(4.49) = 0$$

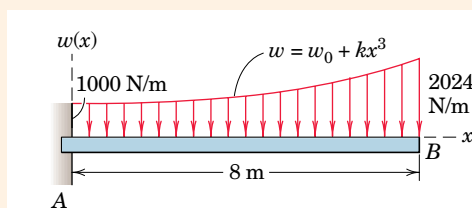
$$M_A = 45\,100 \text{ N} \cdot \text{m}$$

Ans.

$$[\Sigma F_y = 0] \quad A_y = 10\,050 \text{ N}$$

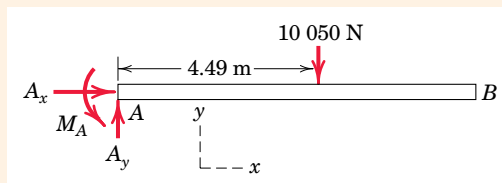
Ans.

Note that $A_x = 0$ by inspection.



Helpful Hints

- ① Use caution with the units of the constants w_0 and k .
- ② The student should recognize that the calculation of R and its location \bar{x} is simply an application of centroids as treated in Art. 5/3.

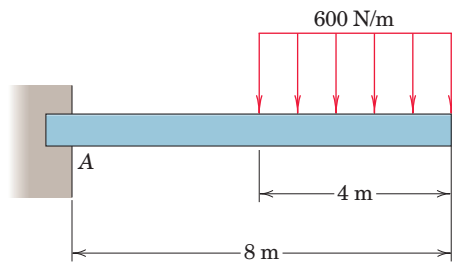


PROBLEMS

Introductory Problems

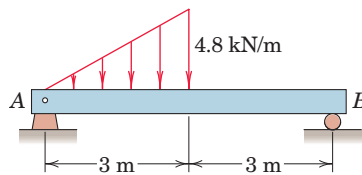
- 5/95** Calculate the supporting force R_A and moment M_A at A for the loaded cantilever beam.

Ans. $R_A = 2.4 \text{ kN}$, $M_A = 14.4 \text{ kN}\cdot\text{m}$ CCW



Problem 5/95

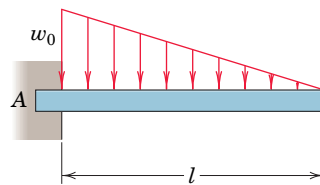
- 5/96** Calculate the reactions at A and B for the beam subjected to the triangular load distribution.



Problem 5/96

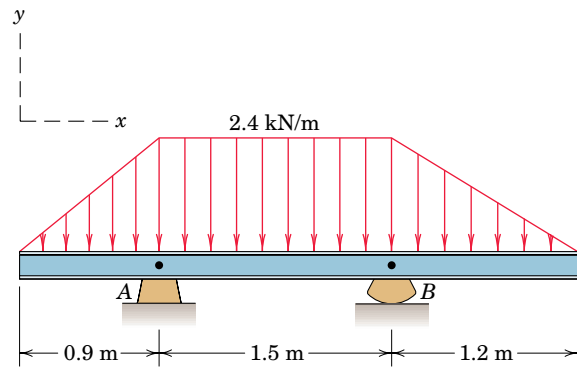
- 5/97** Determine the reactions at the built-in end of the beam subjected to the triangular load distribution.

Ans. $R_A = \frac{w_0 l}{2}$, $M_A = \frac{w_0 l^2}{6}$ CCW



Problem 5/97

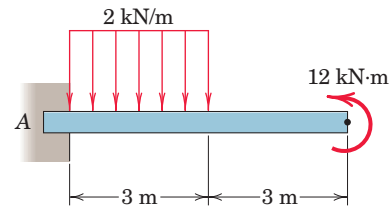
- 5/98** Determine the reactions at A and B for the loaded beam.



Problem 5/98

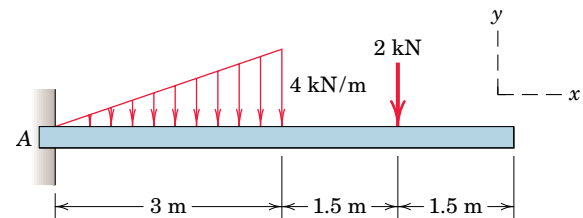
- 5/99** Find the reaction at A due to the uniform loading and the applied couple.

Ans. $R_A = 6 \text{ kN}$, $M_A = 3 \text{ kN}\cdot\text{m}$ CW



Problem 5/99

- 5/100** Determine the reactions at A for the cantilever beam subjected to the distributed and concentrated loads.

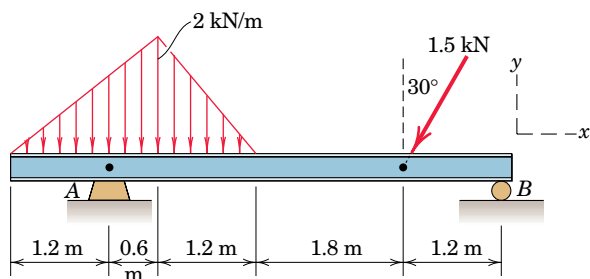


Problem 5/100

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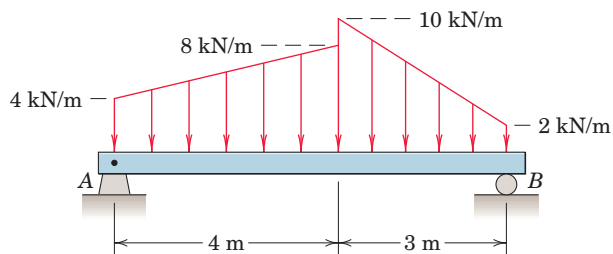
- 5/101** Determine the reactions at A and B for the beam subjected to a combination of distributed and point loads.

Ans. $A_x = 750 \text{ N}$, $A_y = 3.07 \text{ kN}$, $B_y = 1.224 \text{ kN}$



Problem 5/101

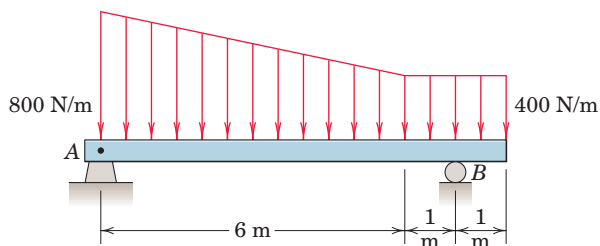
- 5/102** Calculate the supporting reactions at A and B for the beam subjected to the two linearly distributed loads.



Problem 5/102

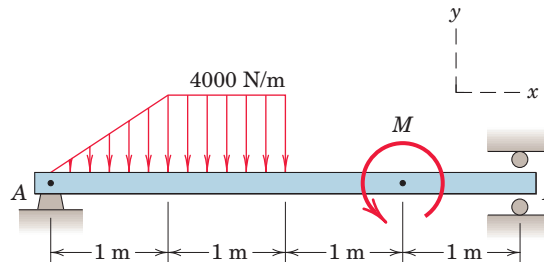
- 5/103** Determine the reactions at the supports of the beam which is loaded as shown.

Ans. $R_A = 2230 \text{ N}$, $R_B = 2170 \text{ N}$



Problem 5/103

- 5/104** The beam is subjected to the distributed load and the couple shown. If M is slowly increased starting from zero, at what value M_0 will contact at B change from the lower surface to the upper surface?

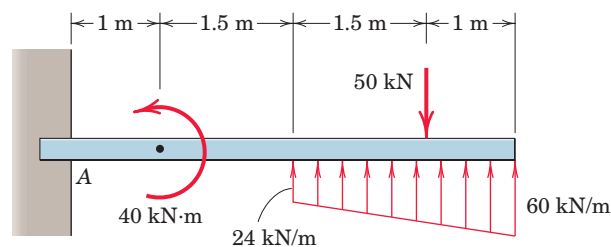


Problem 5/104

Representative Problems

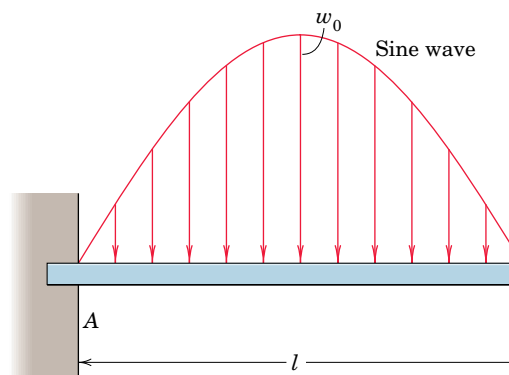
- 5/105** Determine the force and moment reactions at A for the beam which is subjected to the load combination shown.

Ans. $R_A = 55 \text{ kN}$, $M_A = 253 \text{ kN} \cdot \text{m CW}$



Problem 5/105

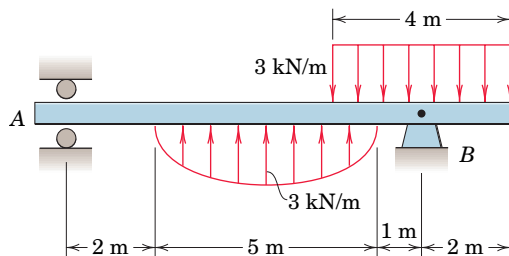
- 5/106** Determine the force and moment reactions at the support A of the built-in beam which is subjected to the sine-wave load distribution.



Problem 5/106

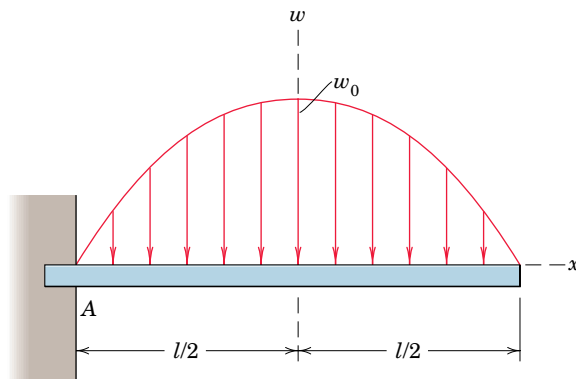
- 5/107** Determine the reactions at points A and B of the beam subjected to the elliptical and uniform load distributions. At which surface, upper or lower, is the reaction at A exerted?

Ans. $A = 5.15 \text{ kN}$, $B = 5.37 \text{ kN}$, upper



Problem 5/107

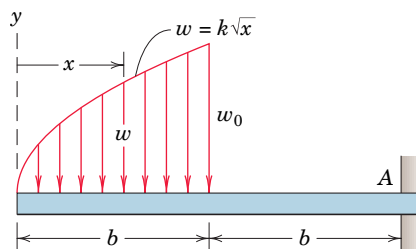
- 5/108** The cantilever beam is subjected to a parabolic distribution of load symmetrical about the middle of the beam. Determine the supporting force R_A and moment M_A acting on the beam at A.



Problem 5/108

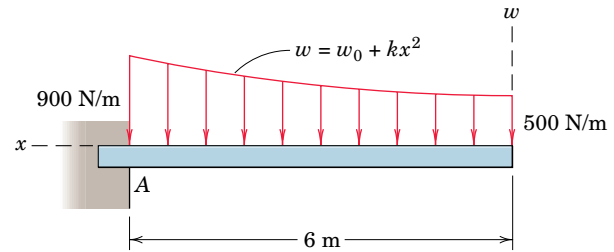
- 5/109** Determine the force and moment reactions at the support A of the cantilever beam subjected to the load distribution shown.

Ans. $R_A = \frac{2}{3}w_0b$, $M_A = \frac{14}{15}w_0b^2 \text{ CW}$



Problem 5/109

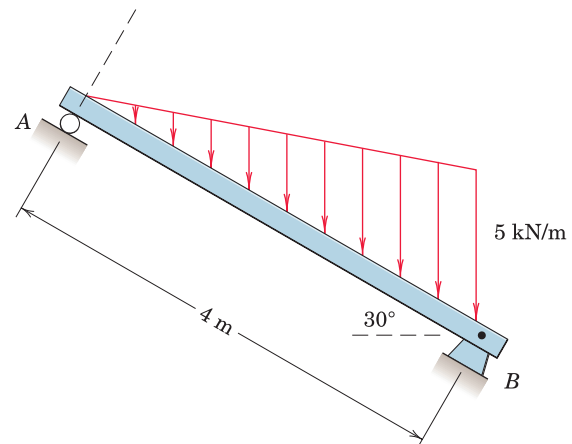
- 5/110** A cantilever beam supports the variable load shown. Calculate the supporting force R_A and moment M_A at A.



Problem 5/110

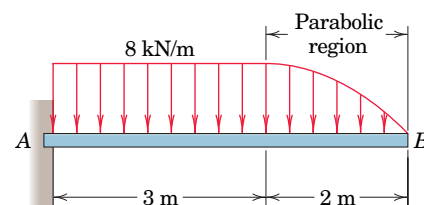
- 5/111** Determine the reactions at points A and B of the inclined beam subjected to the vertical load distribution shown. The value of the load distribution at the right end of the beam is 5 kN per horizontal meter.

Ans. $A = 2.5 \text{ kN}$, $B = 6.61 \text{ kN}$



Problem 5/111

- 5/112** Determine the reactions at the support for the beam which is subjected to the combination of uniform and parabolic loading distributions.

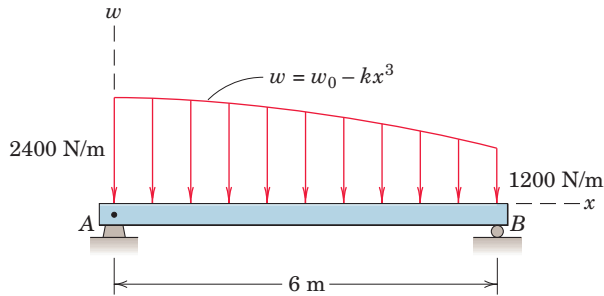


Problem 5/112

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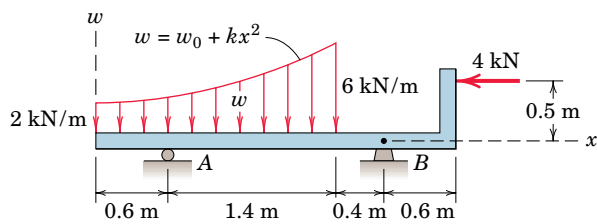
- 5/113** A beam is subjected to the variable loading shown. Calculate the support reactions at A and B.

Ans. $R_A = 6.84 \text{ kN}$, $R_B = 5.76 \text{ kN}$



Problem 5/113

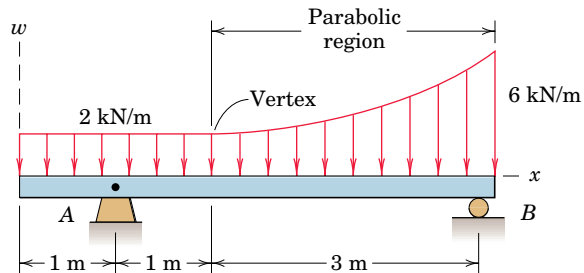
- 5/114** Determine the reactions at A and B for the beam subjected to the distributed and concentrated loads.



Problem 5/114

- 5/115** Determine the reactions at the supports of the beam which is acted on by the combination of uniform and parabolic loading distributions.

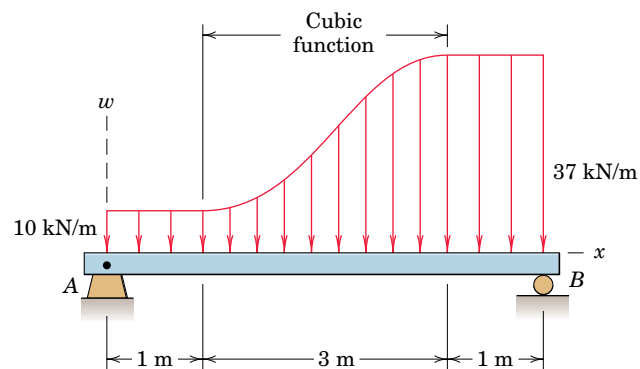
Ans. $R_A = R_B = 7 \text{ kN}$



Problem 5/115

- 5/116** The transition between the loads of 10 kN/m and 37 kN/m is accomplished by means of a cubic function of form $w = k_0 + k_1x + k_2x^2 + k_3x^3$, the slope of which is zero at its end points $x = 1 \text{ m}$ and $x = 4 \text{ m}$. Determine the reactions at A and B.

Ans. $R_A = 43.1 \text{ kN}$, $R_B = 74.4 \text{ kN}$



Problem 5/116

5/7 BEAMS—INTERNAL EFFECTS

The previous article treated the reduction of a distributed force to one or more equivalent concentrated forces and the subsequent determination of the external reactions acting on the beam. In this article we introduce internal beam effects and apply principles of statics to calculate the internal shear force and bending moment as functions of location along the beam.

Shear, Bending, and Torsion

In addition to supporting tension or compression, a beam can resist shear, bending, and torsion. These three effects are illustrated in Fig. 5/22. The force V is called the *shear force*, the couple M is called the *bending moment*, and the couple T is called a *torsional moment*. These effects represent the vector components of the resultant of the forces acting on a transverse section of the beam as shown in the lower part of the figure.

Consider the shear force V and bending moment M caused by forces applied to the beam in a single plane. The conventions for positive values of shear V and bending moment M shown in Fig. 5/23 are the ones generally used. From the principle of action and reaction we can see that the directions of V and M are reversed on the two sections. It is frequently impossible to tell without calculation whether the shear and moment at a particular section are positive or negative. For this reason it is advisable to represent V and M in their positive directions on the free-body diagrams and let the algebraic signs of the calculated values indicate the proper directions.

As an aid to the physical interpretation of the bending couple M , consider the beam shown in Fig. 5/24 bent by the two equal and opposite positive moments applied at the ends. The cross section of the beam is treated as an H-section with a very narrow center web and heavy top and bottom flanges. For this beam we may neglect the load carried by the small web compared with that carried by the two flanges. The upper flange of the beam clearly is shortened and is under compression, whereas the lower flange is lengthened and is under tension. The resultant of the two forces, one tensile and the other compressive, acting on any section is a couple and has the value of the bending moment on the section. If a beam having some other cross-sectional shape were loaded

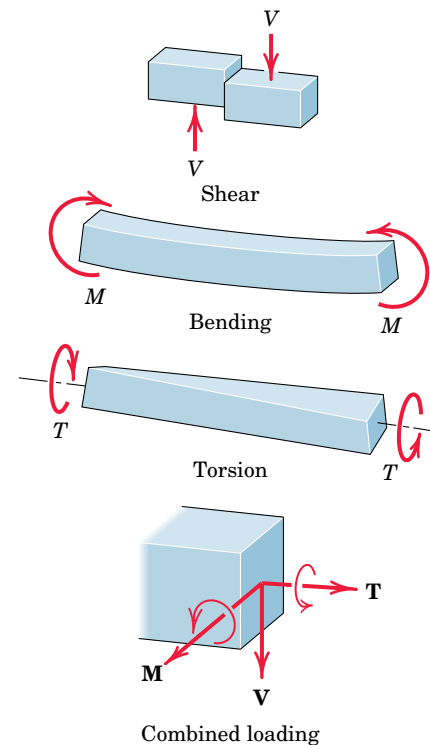


Figure 5/22

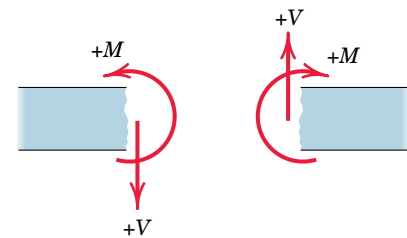


Figure 5/23

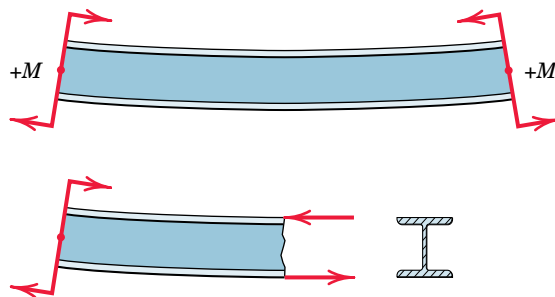


Figure 5/24

in the same way, the distribution of force over the cross section would be different, but the resultant would be the same couple.

Shear-Force and Bending-Moment Diagrams

The variation of shear force V and bending moment M over the length of a beam provides information necessary for the design analysis of the beam. In particular, the maximum magnitude of the bending moment is usually the primary consideration in the design or selection of a beam, and its value and position should be determined. The variations in shear and moment are best shown graphically, and the expressions for V and M when plotted against distance along the beam give the *shear-force* and *bending-moment diagrams* for the beam.

The first step in the determination of the shear and moment relations is to establish the values of all external reactions on the beam by applying the equations of equilibrium to a free-body diagram of the beam as a whole. Next, we isolate a portion of the beam, either to the right or to the left of an arbitrary transverse section, with a free-body diagram, and apply the equations of equilibrium to this isolated portion of the beam. These equations will yield expressions for the shear force V and bending moment M acting at the cut section on the part of the beam isolated. The part of the beam which involves the smaller number of forces, either to the right or to the left of the arbitrary section, usually yields the simpler solution.

We should avoid using a transverse section which coincides with the location of a concentrated load or couple, as such a position represents a point of discontinuity in the variation of shear or bending moment. Finally, it is important to note that the calculations for V and M on each section chosen should be consistent with the positive convention illustrated in Fig. 5/23.

General Loading, Shear, and Moment Relationships

For any beam with distributed loads we can establish certain general relationships which will aid greatly in the determination of the shear and moment distributions along the beam. Figure 5/25 represents a portion of a loaded beam, where an element dx of the beam is isolated. The loading w represents the force per unit length of beam. At the location x the shear V and moment M acting on the element are drawn in their positive directions. On the opposite side of the element where the

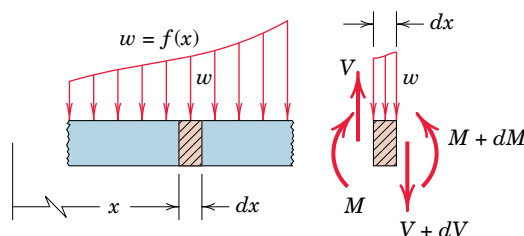


Figure 5/25

coordinate is $x + dx$, these quantities are also shown in their positive directions. They must, however, be labeled $V + dV$ and $M + dM$, since V and M change with x . The applied loading w may be considered constant over the length of the element, since this length is a differential quantity and the effect of any change in w disappears in the limit compared with the effect of w itself.

Equilibrium of the element requires that the sum of the vertical forces be zero. Thus, we have

$$V - w dx - (V + dV) = 0$$

or

$$w = -\frac{dV}{dx} \quad (5/10)$$

We see from Eq. 5/10 that the slope of the shear diagram must everywhere be equal to the negative of the value of the applied loading. Equation 5/10 holds on either side of a concentrated load but not at the concentrated load because of the discontinuity produced by the abrupt change in shear.

We may now express the shear force V in terms of the loading w by integrating Eq. 5/10. Thus,

$$\int_{V_0}^V dV = -\int_{x_0}^x w dx$$

or

$$V = V_0 + (\text{the negative of the area under the loading curve from } x_0 \text{ to } x)$$

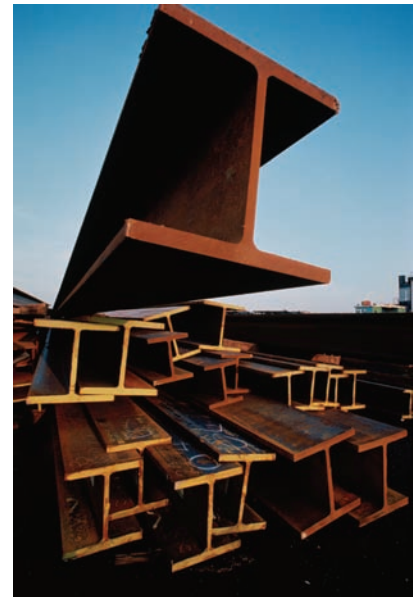
In this expression V_0 is the shear force at x_0 and V is the shear force at x . Summing the area under the loading curve is usually a simple way to construct the shear-force diagram.

Equilibrium of the element in Fig. 5/25 also requires that the moment sum be zero. Summing moments about the left side of the element gives

$$M + w dx \frac{dx}{2} + (V + dV) dx - (M + dM) = 0$$

The two M 's cancel, and the terms $w(dx)^2/2$ and $dV dx$ may be dropped, since they are differentials of higher order than those which remain. This leaves

$$V = \frac{dM}{dx} \quad (5/11)$$



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Because of its economical use of material in achieving bending stiffness, the I-beam is a very common structural element.

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which expresses the fact that the shear everywhere is equal to the slope of the moment curve. Equation 5/11 holds on either side of a concentrated couple but not at the concentrated couple because of the discontinuity caused by the abrupt change in moment.

We may now express the moment M in terms of the shear V by integrating Eq. 5/11. Thus,

$$\int_{M_0}^M dM = \int_{x_0}^x V dx$$

or

$$M = M_0 + (\text{area under the shear diagram from } x_0 \text{ to } x)$$

In this expression M_0 is the bending moment at x_0 and M is the bending moment at x . For beams where there is no externally applied moment M_0 at $x_0 = 0$, the total moment at any section equals the area under the shear diagram up to that section. Summing the area under the shear diagram is usually the simplest way to construct the moment diagram.

When V passes through zero and is a continuous function of x with $dV/dx \neq 0$, the bending moment M will be a maximum or a minimum, since $dM/dx = 0$ at such a point. Critical values of M also occur when V crosses the zero axis discontinuously, which occurs for beams under concentrated loads.

We observe from Eqs. 5/10 and 5/11 that the degree of V in x is one higher than that of w . Also M is of one higher degree in x than is V . Consequently, M is two degrees higher in x than w . Thus for a beam loaded by $w = kx$, which is of the first degree in x , the shear V is of the second degree in x and the bending moment M is of the third degree in x .

Equations 5/10 and 5/11 may be combined to yield

$$\frac{d^2M}{dx^2} = -w \quad (5/12)$$

Thus, if w is a known function of x , the moment M can be obtained by two integrations, provided that the limits of integration are properly evaluated each time. This method is usable only if w is a continuous function of x .*

When bending in a beam occurs in more than a single plane, we may perform a separate analysis in each plane and combine the results vectorially.

*When w is a discontinuous function of x , it is possible to introduce a special set of expressions called *singularity functions* which permit writing analytical expressions for shear V and moment M over an interval which includes discontinuities. These functions are not discussed in this book.

Sample Problem 5/13

Determine the shear and moment distributions produced in the simple beam by the 4-kN concentrated load.

Solution. From the free-body diagram of the entire beam we find the support reactions, which are

$$R_1 = 1.6 \text{ kN} \quad R_2 = 2.4 \text{ kN}$$

A section of the beam of length x is next isolated with its free-body diagram on which we show the shear V and the bending moment M in their positive directions. Equilibrium gives

$$[\Sigma F_y = 0] \quad 1.6 - V = 0 \quad V = 1.6 \text{ kN}$$

$$[\Sigma M_{R_1} = 0] \quad M - 1.6x = 0 \quad M = 1.6x$$

- ① These values of V and M apply to all sections of the beam to the left of the 4-kN load.

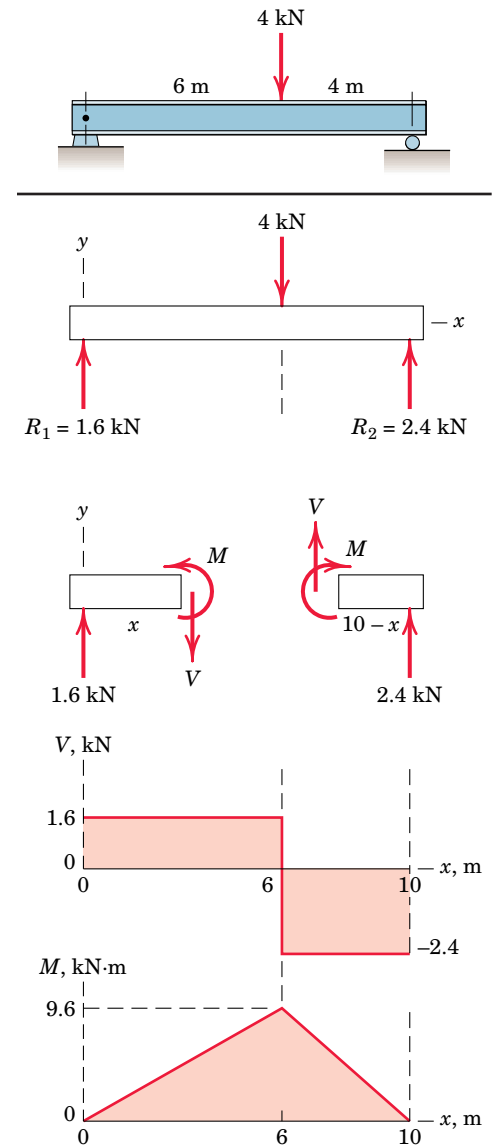
A section of the beam to the right of the 4-kN load is next isolated with its free-body diagram on which V and M are shown in their positive directions. Equilibrium requires

$$[\Sigma F_y = 0] \quad V + 2.4 = 0 \quad V = -2.4 \text{ kN}$$

$$[\Sigma M_{R_2} = 0] \quad -(2.4)(10 - x) + M = 0 \quad M = 2.4(10 - x)$$

These results apply only to sections of the beam to the right of the 4-kN load.

The values of V and M are plotted as shown. The maximum bending moment occurs where the shear changes direction. As we move in the positive x -direction starting with $x = 0$, we see that the moment M is merely the accumulated area under the shear diagram.

**Helpful Hint**

- ① We must be careful not to take our section at a concentrated load (such as $x = 6 \text{ m}$) since the shear and moment relations involve discontinuities at such positions.

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Sample Problem 5/14

The cantilever beam is subjected to the load intensity (force per unit length) which varies as $w = w_0 \sin(\pi x/l)$. Determine the shear force V and bending moment M as functions of the ratio x/l .

Solution. The free-body diagram of the entire beam is drawn first so that the shear force V_0 and bending moment M_0 which act at the supported end at $x = 0$ can be computed. By convention V_0 and M_0 are shown in their positive mathematical senses. A summation of vertical forces for equilibrium gives

$$[\Sigma F_y = 0] \quad V_0 - \int_0^l w \, dx = 0 \quad V_0 = \int_0^l w_0 \sin \frac{\pi x}{l} \, dx = \frac{2w_0 l}{\pi}$$

- ① A summation of moments about the left end at $x = 0$ for equilibrium gives

$$[\Sigma M = 0] \quad -M_0 - \int_0^l x(w \, dx) = 0 \quad M_0 = -\int_0^l w_0 x \sin \frac{\pi x}{l} \, dx$$

$$M_0 = \frac{-w_0 l^2}{\pi^2} \left[\sin \frac{\pi x}{l} - \frac{\pi x}{l} \cos \frac{\pi x}{l} \right]_0^l = -\frac{w_0 l^2}{\pi}$$

From a free-body diagram of an arbitrary section of length x , integration of Eq. 5/10 permits us to find the shear force internal to the beam. Thus,

② $[dV = -w \, dx] \quad \int_{V_0}^V dV = -\int_0^x w_0 \sin \frac{\pi x}{l} \, dx$

$$V - V_0 = \left[\frac{w_0 l}{\pi} \cos \frac{\pi x}{l} \right]_0^x \quad V - \frac{2w_0 l}{\pi} = \frac{w_0 l}{\pi} \left(\cos \frac{\pi x}{l} - 1 \right)$$

or in dimensionless form

$$\frac{V}{w_0 l} = \frac{1}{\pi} \left(1 + \cos \frac{\pi x}{l} \right)$$

The bending moment is obtained by integration of Eq. 5/11, which gives

$$[dM = V \, dx] \quad \int_{M_0}^M dM = \int_0^x \frac{w_0 l}{\pi} \left(1 + \cos \frac{\pi x}{l} \right) dx$$

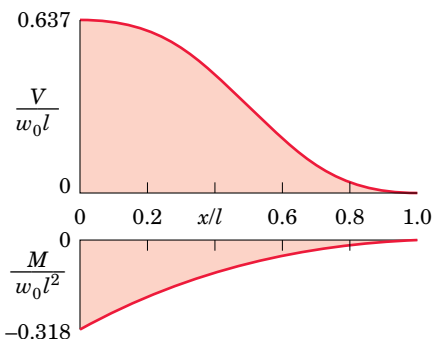
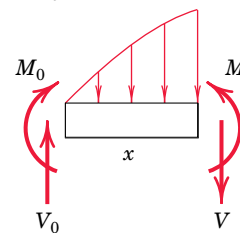
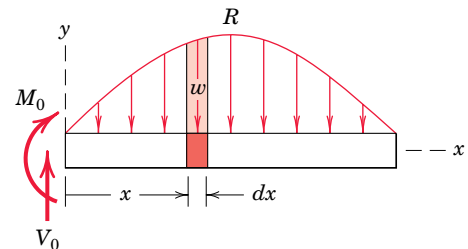
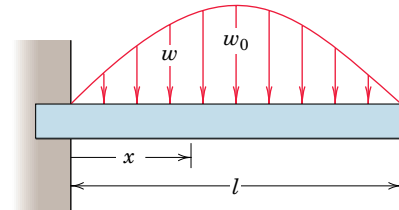
$$M - M_0 = \frac{w_0 l}{\pi} \left[x + \frac{l}{\pi} \sin \frac{\pi x}{l} \right]_0^x$$

$$M = -\frac{w_0 l^2}{\pi} + \frac{w_0 l}{\pi} \left[x + \frac{l}{\pi} \sin \frac{\pi x}{l} - 0 \right]$$

or in dimensionless form

$$\frac{M}{w_0 l^2} = \frac{1}{\pi} \left(\frac{x}{l} - 1 + \frac{1}{\pi} \sin \frac{\pi x}{l} \right) \quad \text{Ans.}$$

The variations of $V/w_0 l$ and $M/w_0 l^2$ with x/l are shown in the bottom figures. The negative values of $M/w_0 l^2$ indicate that physically the bending moment is in the direction opposite to that shown.



Ans.

Helpful Hints

- ① In this case of symmetry it is clear that the resultant $R = V_0 = 2w_0 l/\pi$ of the load distribution acts at midspan, so that the moment requirement is simply $M_0 = -Rl/2 = -w_0 l^2/\pi$. The minus sign tells us that physically the bending moment at $x = 0$ is opposite to that represented on the free-body diagram.
- ② The free-body diagram serves to remind us that the integration limits for V as well as for x must be accounted for. We see that the expression for V is positive, so that the shear force is as represented on the free-body diagram.

Sample Problem 5/15

Draw the shear-force and bending-moment diagrams for the loaded beam and determine the maximum moment M and its location x from the left end.

Solution. The support reactions are most easily obtained by considering the resultants of the distributed loads as shown on the free-body diagram of the beam as a whole. The first interval of the beam is analyzed from the free-body diagram of the section for $0 < x < 2$ m. A summation of vertical forces and a moment summation about the cut section yield

$$\begin{aligned} [\Sigma F_y = 0] \quad V &= 1.233 - 0.25x^2 \\ [\Sigma M = 0] \quad M + (0.25x^2) \frac{x}{3} - 1.233x &= 0 \quad M = 1.233x - 0.0833x^3 \end{aligned}$$

These values of V and M hold for $0 < x < 2$ m and are plotted for that interval in the shear and moment diagrams shown.

From the free-body diagram of the section for which $2 < x < 4$ m, equilibrium in the vertical direction and a moment sum about the cut section give

$$\begin{aligned} [\Sigma F_y = 0] \quad V + 1(x - 2) + 1 - 1.233 &= 0 \quad V = 2.23 - x \\ [\Sigma M = 0] \quad M + 1(x - 2) \frac{x - 2}{2} + 1[x - \frac{2}{3}(2)] - 1.233x &= 0 \\ M &= -0.667 + 2.23x - 0.50x^2 \end{aligned}$$

These values of V and M are plotted on the shear and moment diagrams for the interval $2 < x < 4$ m.

The analysis of the remainder of the beam is continued from the free-body diagram of the portion of the beam to the right of a section in the next interval. It should be noted that V and M are represented in their positive directions. A vertical-force summation and a moment summation about the section yield

$$V = -1.767 \text{ kN} \quad \text{and} \quad M = 7.33 - 1.767x$$

These values of V and M are plotted on the shear and moment diagrams for the interval $4 < x < 5$ m.

The last interval may be analyzed by inspection. The shear is constant at +1.5 kN, and the moment follows a straight-line relation beginning with zero at the right end of the beam.

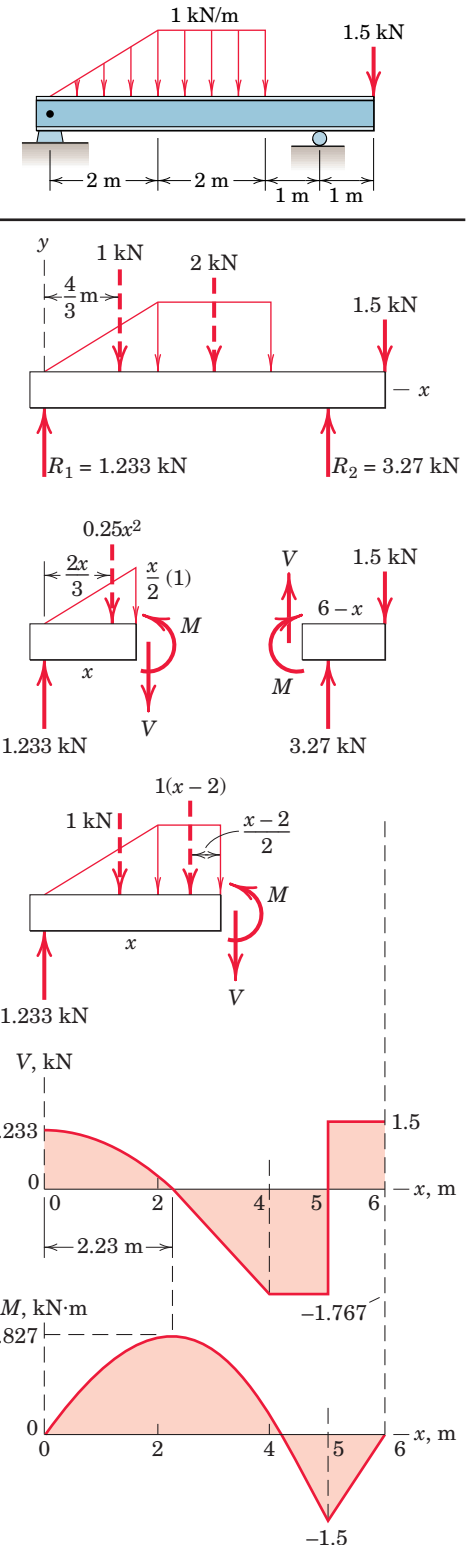
The maximum moment occurs at $x = 2.23$ m, where the shear curve crosses the zero axis, and the magnitude of M is obtained for this value of x by substitution into the expression for M for the second interval. The maximum moment is

$$M = 1.827 \text{ kN} \cdot \text{m} \quad \text{Ans.}$$

As before, note that the change in moment M up to any section equals the area under the shear diagram up to that section. For instance, for $x < 2$ m,

$$[\Delta M = \int V dx] \quad M - 0 = \int_0^x (1.233 - 0.25x^2) dx$$

$$\text{and, as above,} \quad M = 1.233x - 0.0833x^3$$



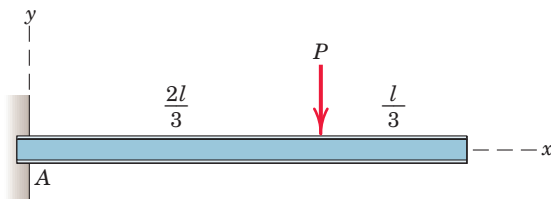
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PROBLEMS

Introductory Problems

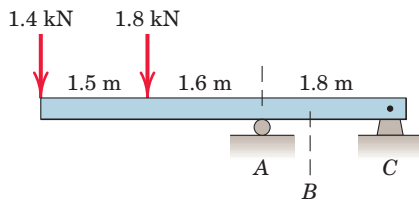
- 5/117** Determine the shear-force and bending-moment distributions produced in the beam by the concentrated load. What are the values of the shear and moment at $x = l/2$?

$$\text{Ans. } V = P, M = -\frac{Pl}{6}$$



Problem 5/117

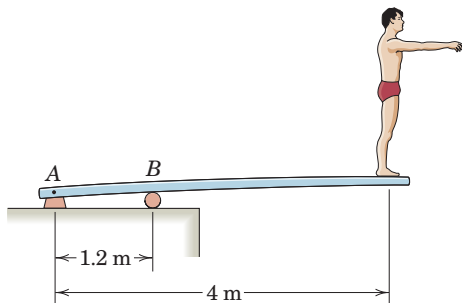
- 5/118** Determine the shear V at a section B between A and C and the moment M at the support A .



Problem 5/118

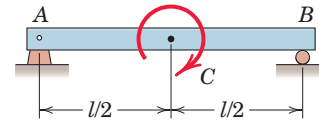
- 5/119** Draw the shear and moment diagrams for the diving board, which supports the 80-kg man poised to dive. Specify the bending moment with the maximum magnitude.

$$\text{Ans. } M_B = -2200 \text{ N}\cdot\text{m}$$



Problem 5/119

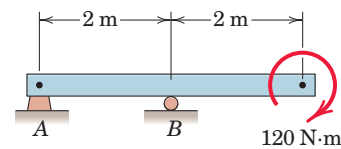
- 5/120** Draw the shear and moment diagrams for the beam loaded at its center by the couple C .



Problem 5/120

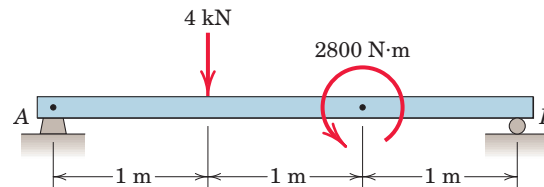
- 5/121** Draw the shear and moment diagrams for the beam subjected to the end couple. What is the moment M at a section 0.5 m to the right of B ?

$$\text{Ans. } M = -120 \text{ N}\cdot\text{m}$$



Problem 5/121

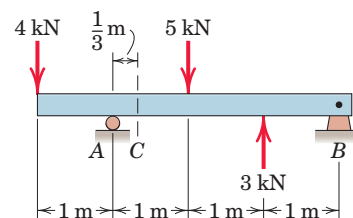
- 5/122** Draw the shear and moment diagrams for the loaded beam. What are the values of the shear and moment at the middle of the beam?



Problem 5/122

- 5/123** Draw the shear and moment diagrams for the beam shown and find the bending moment M at section C .

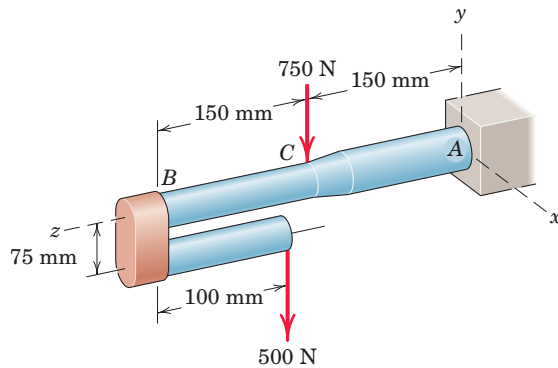
$$\text{Ans. } M_C = -2.78 \text{ kN}\cdot\text{m}$$



Problem 5/123

Representative Problems

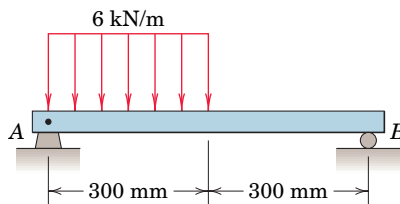
- 5/124** Construct the bending-moment diagram for the cantilevered shaft AB of the rigid unit shown.



Problem 5/124

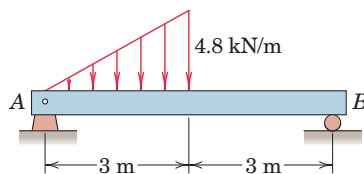
- 5/125** Determine the shear V and moment M at a section of the loaded beam 200 mm to the right of A .

Ans. $V = 0.15 \text{ kN}$, $M = 0.15 \text{ kN} \cdot \text{m}$



Problem 5/125

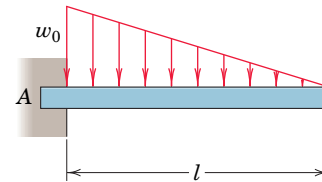
- 5/126** Determine the shear V and moment M in the beam at a section 2 m to the right of end A .



Problem 5/126

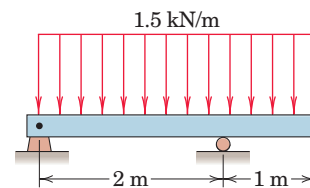
- 5/127** Draw the shear and moment diagrams for the cantilever beam with the linear loading, repeated here from Prob. 5/97. Find the maximum magnitude of the bending moment M .

$$\text{Ans. } |M|_{\max} = \frac{w_0 l^2}{6}$$



Problem 5/127

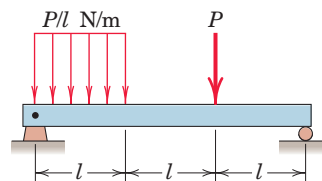
- 5/128** Draw the shear and moment diagrams for the beam shown. Determine the distance b , measured from the left end, to the point where the bending moment is zero between the supports.



Problem 5/128

- 5/129** Draw the shear and moment diagrams for the loaded beam and find the maximum magnitude M of the bending moment.

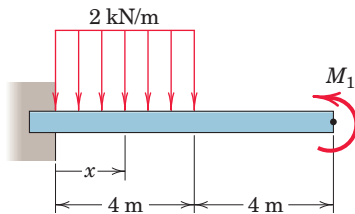
$$\text{Ans. } M = \frac{5}{6}Pl$$



Problem 5/129

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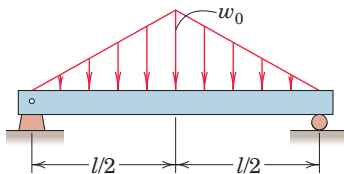
- 5/130** Draw the shear and moment diagrams for the loaded cantilever beam where the end couple M_1 is adjusted so as to produce zero moment at the fixed end of the beam. Find the bending moment M at $x = 2$ m.



Problem 5/130

- 5/131** Draw the shear and moment diagrams for the linearly loaded simple beam shown. Determine the maximum magnitude of the bending moment M .

$$\text{Ans. } |M|_{\max} = \frac{w_0 l^2}{12}$$

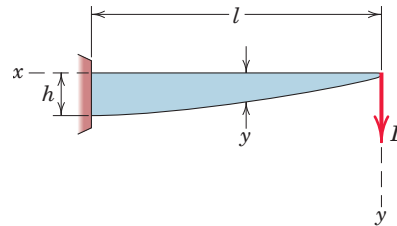


Problem 5/131

- 5/132** The shear force in kilonewtons in a certain beam is given by $V = 33x - 7x^3$ where x is the distance in meters measured along the beam. Determine the corresponding variation with x of the normal loading w in kilonewtons per meter of length. Also determine the bending moment M at $x = 1.5$ m if the bending moment at $x = 0.5$ m is $0.4 \text{ kN} \cdot \text{m}$.

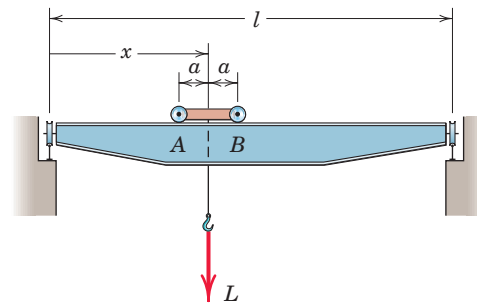
- 5/133** The resistance of a beam of uniform width to bending is found to be proportional to the square of the beam depth y . For the cantilever beam shown the depth is h at the support. Find the required depth y as a function of the length x in order for all sections to be equally effective in their resistance to bending.

$$\text{Ans. } y = h\sqrt{x/l}$$



Problem 5/133

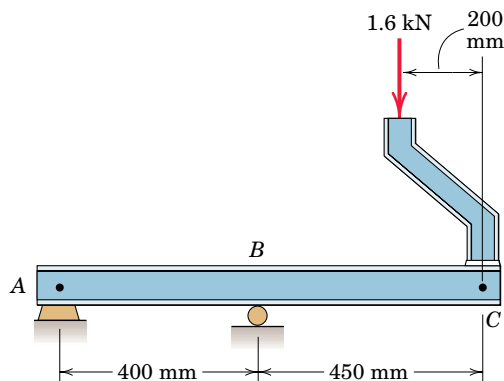
- 5/134** Determine the maximum bending moment M and the corresponding value of x in the crane beam and indicate the section where this moment acts.



Problem 5/134

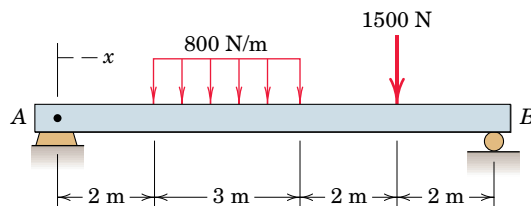
- 5/135** The angle strut is welded to the end C of the I-beam and supports the 1.6-kN vertical force. Determine the bending moment at B and the distance x to the left of C at which the bending moment is zero. Also construct the moment diagram for the beam.

Ans. $M_B = -0.40 \text{ kN}\cdot\text{m}$, $x = 0.2 \text{ m}$



Problem 5/135

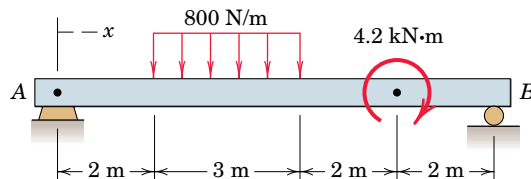
- 5/136** Plot the shear and moment diagrams for the beam loaded with both the distributed and point loads. What are the values of the shear and moment at $x = 6 \text{ m}$? Determine the maximum bending moment M_{\max} .



Problem 5/136

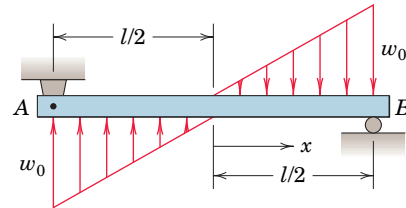
- 5/137** Repeat Prob. 5/136, where the 1500-N load has been replaced by the 4.2-kN·m couple.

Ans. $V = -1400 \text{ N}$, $M = 0$, $M_{\max} = 2800 \text{ N}\cdot\text{m}$



Problem 5/137

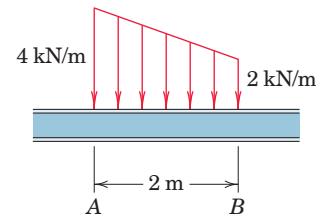
- 5/138** The beam is subjected to the two similar loadings shown where the maximum intensity of loading, in force per unit length, is w_0 . Derive expressions for the shear V and moment M in the beam in terms of the distance x measured from the center of the beam.



Problem 5/138

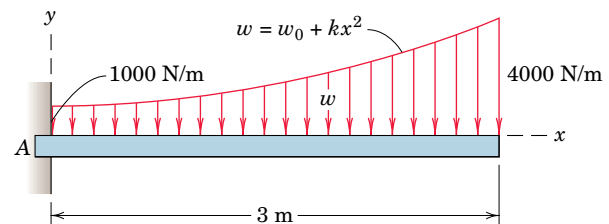
- 5/139** The distributed load decreases linearly from 4 to 2 kN/m in a distance of 2 m along a certain beam in equilibrium. If the shear force and bending moment at section A are +3 kN and +2 kN·m, respectively, calculate the shear force and bending moment at section B.

Ans. $V_B = -3 \text{ kN}$, $M_B = 4/3 \text{ kN}\cdot\text{m}$



Problem 5/139

- 5/140** Derive expressions for the shear force V and bending moment M as functions of x in the cantilever beam loaded as shown.



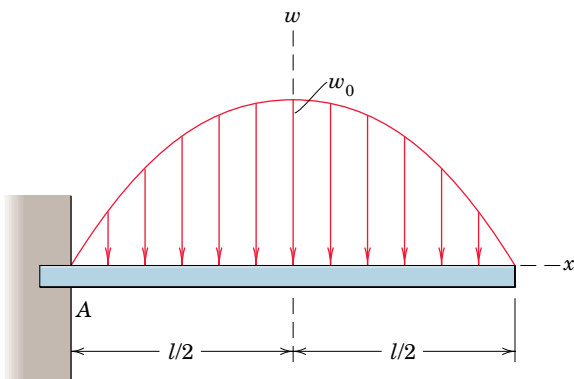
Problem 5/140

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- 5/141** Derive expressions for the shear V and moment M in terms of x for the cantilever beam of Prob. 5/108 shown again here.

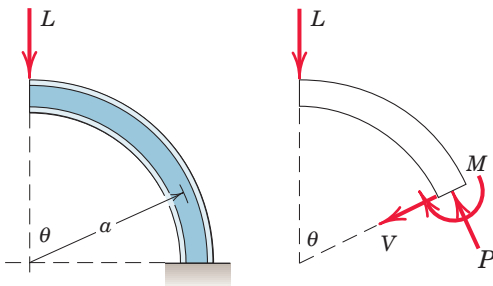
$$\text{Ans. } V = w_0 \left(\frac{l}{3} - x + \frac{4x^3}{3l^2} \right)$$

$$M = w_0 \left(-\frac{l^2}{16} + \frac{x l}{3} - \frac{x^2}{2} + \frac{x^4}{3l^2} \right)$$



Problem 5/141

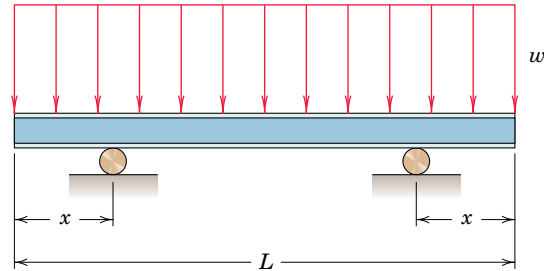
- 5/142** A curved cantilever beam has the form of a quarter circular arc. Determine the expressions for the shear V and the bending moment M as functions of θ .



Problem 5/142

- 5/143** The beam supports a uniform unit load w . Determine the location x of the two supports so as to minimize the maximum bending moment M_{\max} in the beam. Specify M_{\max} .

$$\text{Ans. } x = 0.207L, M_{\max} = 0.0214wL^2$$

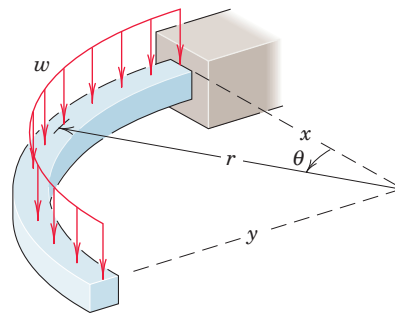


Problem 5/143

- 5/144** The curved cantilever beam in the form of a quarter-circular arc supports a load of w N/m applied along the curve of the beam on its upper surface. Determine the magnitudes of the torsional moment T and bending moment M in the beam as functions of θ .

$$\text{Ans. } T = wr^2 \left(\frac{\pi}{2} - \theta - \cos \theta \right)$$

$$M = wr^2 (1 - \sin \theta)$$



Problem 5/144

5/8 FLEXIBLE CABLES

One important type of structural member is the flexible cable which is used in suspension bridges, transmission lines, messenger cables for supporting heavy trolley or telephone lines, and many other applications. To design these structures we must know the relations involving the tension, span, sag, and length of the cables. We determine these quantities by examining the cable as a body in equilibrium. In the analysis of flexible cables we assume that any resistance offered to bending is negligible. This assumption means that the force in the cable is always in the direction of the cable.

Flexible cables may support a series of distinct concentrated loads, as shown in Fig. 5/26*a*, or they may support loads continuously distributed over the length of the cable, as indicated by the variable-intensity loading w in 5/26*b*. In some instances the weight of the cable is negligible compared with the loads it supports. In other cases the weight of the cable may be an appreciable load or the sole load and cannot be neglected. Regardless of which of these conditions is present, the equilibrium requirements of the cable may be formulated in the same manner.

General Relationships

If the intensity of the variable and continuous load applied to the cable of Fig. 5/26*b* is expressed as w units of force per unit of horizontal length x , then the resultant R of the vertical loading is

$$R = \int dR = \int w \, dx$$

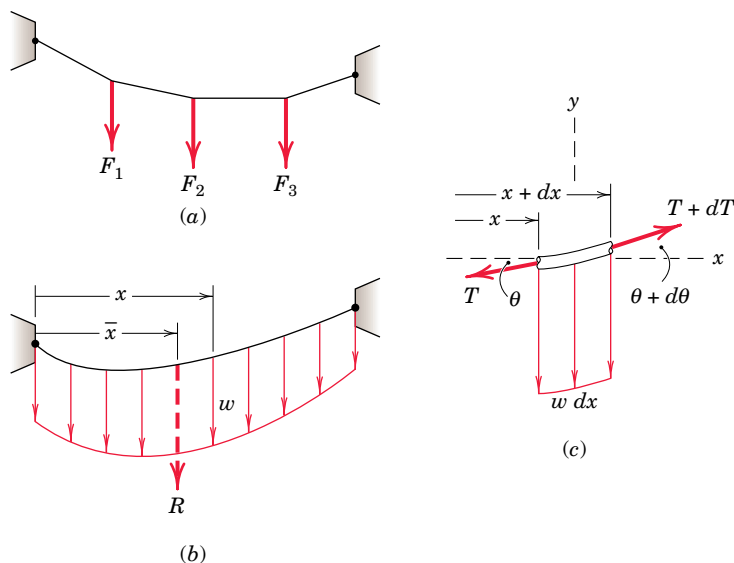


Figure 5/26

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where the integration is taken over the desired interval. We find the position of R from the moment principle, so that

$$R\bar{x} = \int x dR \quad \bar{x} = \frac{\int x dR}{R}$$

The elemental load $dR = w dx$ is represented by an elemental strip of vertical length w and width dx of the shaded area of the loading diagram, and R is represented by the total area. It follows from the foregoing expressions that R passes through the *centroid* of the shaded area.

The equilibrium condition of the cable is satisfied if each infinitesimal element of the cable is in equilibrium. The free-body diagram of a differential element is shown in Fig. 5/26c. At the general position x the tension in the cable is T , and the cable makes an angle θ with the horizontal x -direction. At the section $x + dx$ the tension is $T + dT$, and the angle is $\theta + d\theta$. Note that the changes in both T and θ are taken to be positive with a positive change in x . The vertical load $w dx$ completes the free-body diagram. The equilibrium of vertical and horizontal forces requires, respectively, that

$$(T + dT) \sin(\theta + d\theta) = T \sin \theta + w dx$$

$$(T + dT) \cos(\theta + d\theta) = T \cos \theta$$

The trigonometric expansion for the sine and cosine of the sum of two angles and the substitutions $\sin d\theta = d\theta$ and $\cos d\theta = 1$, which hold in the limit as $d\theta$ approaches zero, yield

$$(T + dT)(\sin \theta + \cos \theta d\theta) = T \sin \theta + w dx$$

$$(T + dT)(\cos \theta - \sin \theta d\theta) = T \cos \theta$$

Dropping the second-order terms and simplifying give us

$$T \cos \theta d\theta + dT \sin \theta = w dx$$

$$-T \sin \theta d\theta + dT \cos \theta = 0$$

which we write as

$$d(T \sin \theta) = w dx \quad \text{and} \quad d(T \cos \theta) = 0$$

The second relation expresses the fact that the horizontal component of T remains unchanged, which is clear from the free-body diagram. If we introduce the symbol $T_0 = T \cos \theta$ for this constant horizontal force, we may then substitute $T = T_0 / \cos \theta$ into the first of the two equations just derived and obtain $d(T_0 \tan \theta) = w dx$. Because $\tan \theta = dy/dx$, the equilibrium equation may be written in the form

$$\frac{d^2y}{dx^2} = \frac{w}{T_0} \quad (5/13)$$

Equation 5/13 is the differential equation for the flexible cable. The solution to the equation is that functional relation $y = f(x)$ which satisfies the equation and also satisfies the conditions at the fixed ends of the cable, called *boundary conditions*. This relationship defines the shape of the cable, and we will use it to solve two important and limiting cases of cable loading.

Parabolic Cable

When the intensity of vertical loading w is constant, the condition closely approximates that of a suspension bridge where the uniform weight of the roadway may be expressed by the constant w . The mass of the cable itself is not distributed uniformly with the horizontal but is relatively small, and thus we neglect its weight. For this limiting case we will prove that the cable hangs in a *parabolic arc*.

We start with a cable suspended from two points A and B which are not on the same horizontal line, Fig. 5/27a. We place the coordinate origin at the lowest point of the cable, where the tension is horizontal and is T_0 . Integration of Eq. 5/13 once with respect to x gives

$$\frac{dy}{dx} = \frac{wx}{T_0} + C$$

where C is a constant of integration. For the coordinate axes chosen, $dy/dx = 0$ when $x = 0$, so that $C = 0$. Thus,

$$\frac{dy}{dx} = \frac{wx}{T_0}$$

which defines the slope of the curve as a function of x . One further integration yields

$$\int_0^y dy = \int_0^x \frac{wx}{T_0} dx \quad \text{or} \quad y = \frac{wx^2}{2T_0} \quad (5/14)$$

Alternatively, you should be able to obtain the identical results with the indefinite integral together with the evaluation of the constant of in-

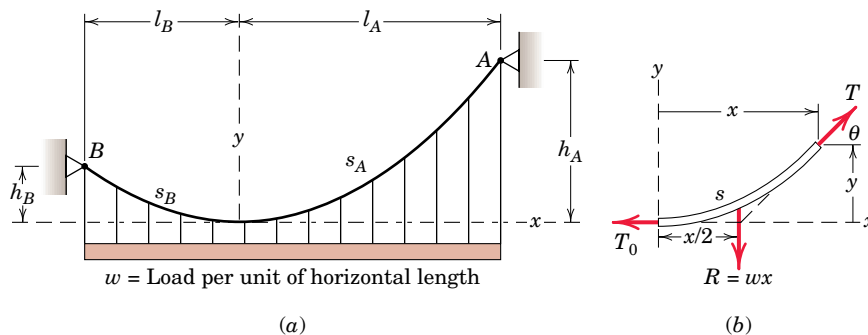


Figure 5/27

tegration. Equation 5/14 gives the shape of the cable, which we see is a vertical parabola. The constant horizontal component of cable tension becomes the cable tension at the origin.

Inserting the corresponding values $x = l_A$ and $y = h_A$ in Eq. 5/14 gives

$$T_0 = \frac{wl_A^2}{2h_A} \quad \text{so that} \quad y = h_A(x/l_A)^2$$

The tension T is found from a free-body diagram of a finite portion of the cable, shown in Fig. 5/27b. From the Pythagorean theorem

$$T = \sqrt{T_0^2 + w^2x^2}$$

Elimination of T_0 gives

$$T = w\sqrt{x^2 + (l_A^2/2h_A)^2} \quad (5/15)$$

The maximum tension occurs where $x = l_A$ and is

$$T_{\max} = wl_A\sqrt{1 + (l_A/2h_A)^2} \quad (5/15a)$$

We obtain the length s_A of the cable from the origin to point A by integrating the expression for a differential length $ds = \sqrt{(dx)^2 + (dy)^2}$. Thus,

$$\int_0^{s_A} ds = \int_0^{l_A} \sqrt{1 + (dy/dx)^2} dx = \int_0^{l_A} \sqrt{1 + (wx/T_0)^2} dx$$

Although we can integrate this expression in closed form, for computational purposes it is more convenient to express the radical as a convergent series and then integrate it term by term. For this purpose we use the binomial expansion

$$(1 + x)^n = 1 + nx + \frac{n(n-1)}{2!}x^2 + \frac{n(n-1)(n-2)}{3!}x^3 + \dots$$

which converges for $x^2 < 1$. Replacing x in the series by $(wx/T_0)^2$ and setting $n = \frac{1}{2}$ give the expression

$$\begin{aligned} s_A &= \int_0^{l_A} \left(1 + \frac{w^2x^2}{2T_0^2} - \frac{w^4x^4}{8T_0^4} + \dots \right) dx \\ &= l_A \left[1 + \frac{2}{3} \left(\frac{h_A}{l_A} \right)^2 - \frac{2}{5} \left(\frac{h_A}{l_A} \right)^4 + \dots \right] \end{aligned} \quad (5/16)$$

This series is convergent for values of $h_A/l_A < \frac{1}{2}$, which holds for most practical cases.

The relationships which apply to the cable section from the origin to point B can be easily obtained by replacing h_A , l_A , and s_A by h_B , l_B , and s_B , respectively.

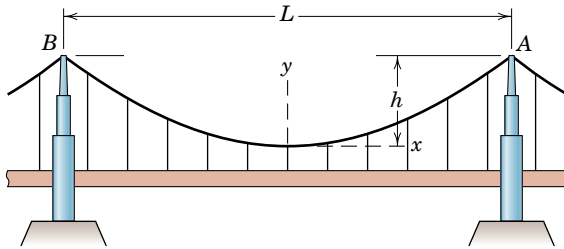


Figure 5/28

For a suspension bridge where the supporting towers are on the same horizontal line, Fig. 5/28, the total span is $L = 2l_A$, the sag is $h = h_A$, and the total length of the cable is $S = 2s_A$. With these substitutions, the maximum tension and the total length become

$$T_{\max} = \frac{wL}{2} \sqrt{1 + (L/4h)^2} \quad (5/15b)$$

$$S = L \left[1 + \frac{8}{3} \left(\frac{h}{L} \right)^2 - \frac{32}{5} \left(\frac{h}{L} \right)^4 + \dots \right] \quad (5/16a)$$

This series converges for all values of $h/L < \frac{1}{4}$. In most cases h is much smaller than $L/4$, so that the three terms of Eq. 5/16a give a sufficiently accurate approximation.

Catenary Cable

Consider now a uniform cable, Fig. 5/29a, suspended from two points A and B and hanging under the action of its own weight only. We will show in this limiting case that the cable assumes a curved shape known as a *catenary*.

The free-body diagram of a finite portion of the cable of length s measured from the origin is shown in part b of the figure. This free-body diagram differs from the one in Fig. 5/27b in that the total vertical force supported is equal to the weight of the cable section of length s rather

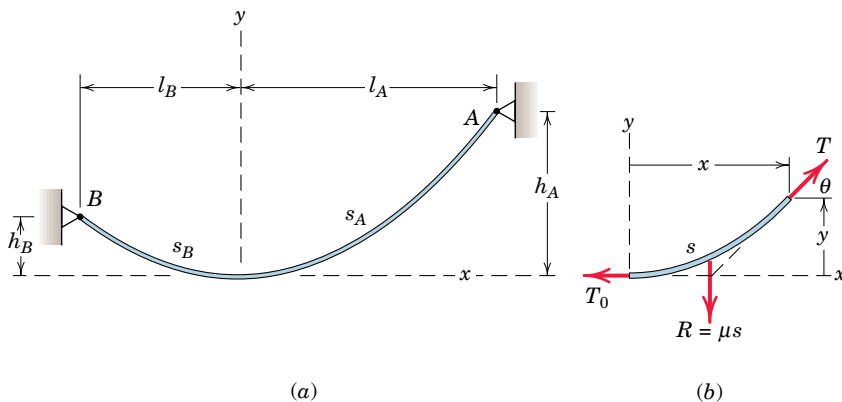


Figure 5/29

than the load distributed uniformly with respect to the horizontal. If the cable has a weight μ per unit of its length, the resultant R of the load is $R = \mu s$, and the incremental vertical load $w \, dx$ of Fig. 5/26c is replaced by $\mu \, ds$. With this replacement the differential relation, Eq. 5/13, for the cable becomes

$$\frac{d^2y}{dx^2} = \frac{\mu}{T_0} \frac{ds}{dx} \quad (5/17)$$

Because $s = f(x, y)$, we must change this equation to one containing only the two variables.

We may substitute the identity $(ds)^2 = (dx)^2 + (dy)^2$ to obtain

$$\frac{d^2y}{dx^2} = \frac{\mu}{T_0} \sqrt{1 + \left(\frac{dy}{dx}\right)^2} \quad (5/18)$$

Equation 5/18 is the differential equation of the curve (catenary) formed by the cable. This equation is easier to solve if we substitute $p = dy/dx$ to obtain

$$\frac{dp}{\sqrt{1 + p^2}} = \frac{\mu}{T_0} dx$$

Integrating this equation gives us

$$\ln(p + \sqrt{1 + p^2}) = \frac{\mu}{T_0} x + C$$

The constant C is zero because $dy/dx = p = 0$ when $x = 0$. Substituting $p = dy/dx$, changing to exponential form, and clearing the equation of the radical give

$$\frac{dy}{dx} = \frac{e^{\mu x/T_0} - e^{-\mu x/T_0}}{2} = \sinh \frac{\mu x}{T_0}$$

where the hyperbolic function* is introduced for convenience. The slope may be integrated to obtain

$$y = \frac{T_0}{\mu} \cosh \frac{\mu x}{T_0} + K$$

The integration constant K is evaluated from the boundary condition $x = 0$ when $y = 0$. This substitution requires that $K = -T_0/\mu$, and hence,

$$y = \frac{T_0}{\mu} \left(\cosh \frac{\mu x}{T_0} - 1 \right) \quad (5/19)$$

*See Arts. C/8 and C/10, Appendix C, for the definition and integral of hyperbolic functions.

Equation 5/19 is the equation of the curve (catenary) formed by the cable hanging under the action of its weight only.

From the free-body diagram in Fig. 5/29b we see that $dy/dx = \tan \theta = \mu s/T_0$. Thus, from the previous expression for the slope,

$$s = \frac{T_0}{\mu} \sinh \frac{\mu x}{T_0} \quad (5/20)$$

We obtain the tension T in the cable from the equilibrium triangle of the forces in Fig. 5/29b. Thus,

$$T^2 = \mu^2 s^2 + T_0^2$$

which, when combined with Eq. 5/20, becomes

$$T^2 = T_0^2 \left(1 + \sinh^2 \frac{\mu x}{T_0} \right) = T_0^2 \cosh^2 \frac{\mu x}{T_0}$$

or

$$T = T_0 \cosh \frac{\mu x}{T_0} \quad (5/21)$$

We may also express the tension in terms of y with the aid of Eq. 5/19, which, when substituted into Eq. 5/21, gives

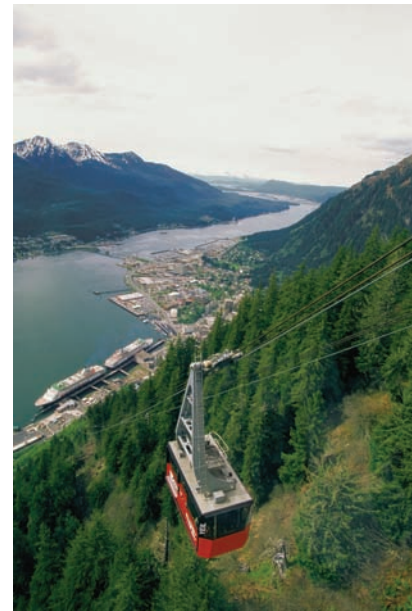
$$T = T_0 + \mu y \quad (5/22)$$

Equation 5/22 shows that the change in cable tension from that at the lowest position depends only on μy .

Most problems dealing with the catenary involve solutions of Eqs. 5/19 through 5/22, which can be handled by a graphical approximation or solved by computer. The procedure for a graphical or computer solution is illustrated in Sample Problem 5/17 following this article.

The solution of catenary problems where the sag-to-span ratio is small may be approximated by the relations developed for the parabolic cable. A small sag-to-span ratio means a tight cable, and the uniform distribution of weight along the cable is not very different from the same load intensity distributed uniformly along the horizontal.

Many problems dealing with both the catenary and parabolic cables involve suspension points which are not on the same level. In such cases we may apply the relations just developed to the part of the cable on each side of the lowest point.



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Tramway in Juneau, Alaska

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Sample Problem 5/16

The light cable supports a mass of 12 kg per meter of horizontal length and is suspended between the two points on the same level 300 m apart. If the sag is 60 m, find the tension at midlength, the maximum tension, and the total length of the cable.

Solution. With a uniform horizontal distribution of load, the solution of part (b) of Art. 5/8 applies, and we have a parabolic shape for the cable. For $h = 60$ m, $L = 300$ m, and $w = 12(9.81)(10^{-3})$ kN/m the relation following Eq. 5/14 with $l_A = L/2$ gives for the midlength tension

$$\left[T_0 = \frac{wL^2}{8h} \right] \quad T_0 = \frac{0.1177(300)^2}{8(60)} = 22.1 \text{ kN} \quad \text{Ans.}$$

The maximum tension occurs at the supports and is given by Eq. 5/15b. Thus,

$$\textcircled{1} \left[T_{\max} = \frac{wL}{2} \sqrt{1 + \left(\frac{L}{4h} \right)^2} \right]$$

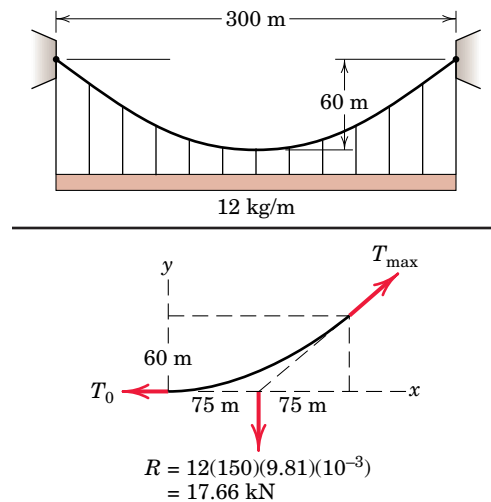
$$T_{\max} = \frac{12(9.81)(10^{-3})(300)}{2} \sqrt{1 + \left(\frac{300}{4(60)} \right)^2} = 28.3 \text{ kN} \quad \text{Ans.}$$

The sag-to-span ratio is $60/300 = 1/5 < 1/4$. Therefore, the series expression developed in Eq. 5/16a is convergent, and we may write for the total length

$$S = 300 \left[1 + \frac{8}{3} \left(\frac{1}{5} \right)^2 - \frac{32}{5} \left(\frac{1}{5} \right)^4 + \cdots \right]$$

$$= 300[1 + 0.1067 - 0.01024 + \cdots]$$

$$= 329 \text{ m} \quad \text{Ans.}$$



Helpful Hint

① *Suggestion:* Check the value of T_{\max} directly from the free-body diagram of the right-hand half of the cable, from which a force polygon may be drawn.

Sample Problem 5/17

Replace the cable of Sample Problem 5/16, which is loaded uniformly along the horizontal, by a cable which has a mass of 12 kg per meter of its own length and supports its own weight only. The cable is suspended between two points on the same level 300 m apart and has a sag of 60 m. Find the tension at midlength, the maximum tension, and the total length of the cable.

Solution. With a load distributed uniformly along the length of the cable, the solution of part (c) of Art. 5/8 applies, and we have a catenary shape of the cable. Equations 5/20 and 5/21 for the cable length and tension both involve the minimum tension T_0 at midlength, which must be found from Eq. 5/19. Thus, for $x = 150$ m, $y = 60$ m, and $\mu = 12(9.81)(10^{-3}) = 0.1177$ kN/m, we have

$$60 = \frac{T_0}{0.1177} \left[\cosh \frac{(0.1177)(150)}{T_0} - 1 \right]$$

or
$$\frac{7.06}{T_0} = \cosh \frac{17.66}{T_0} - 1$$

This equation can be solved graphically. We compute the expression on each side of the equals sign and plot it as a function of T_0 . The intersection of the two curves establishes the equality and determines the correct value of T_0 . This plot is shown in the figure accompanying this problem and yields the solution

$$T_0 = 23.2 \text{ kN}$$

Alternatively, we may write the equation as

$$f(T_0) = \cosh \frac{17.66}{T_0} - \frac{7.06}{T_0} - 1 = 0$$

and set up a computer program to calculate the value(s) of T_0 which renders $f(T_0) = 0$. See Art. C/11 of Appendix C for an explanation of one applicable numerical method.

The maximum tension occurs for maximum y and from Eq. 5/22 is

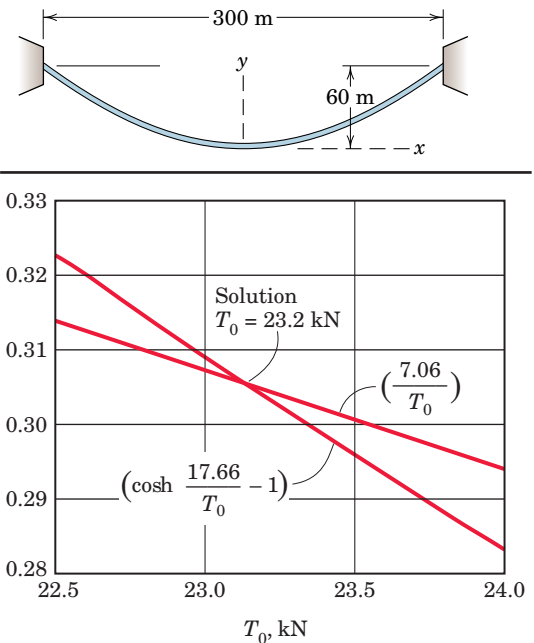
$$T_{\max} = 23.2 + (0.1177)(60) = 30.2 \text{ kN} \quad \text{Ans.}$$

- ① From Eq. 5/20 the total length of the cable becomes

$$2s = 2 \frac{23.2}{0.1177} \sinh \frac{(0.1177)(150)}{23.2} = 330 \text{ m} \quad \text{Ans.}$$

Helpful Hint

- ① Note that the solution of Sample Problem 5/16 for the parabolic cable gives a very close approximation to the values for the catenary even though we have a fairly large sag. The approximation is even better for smaller sag-to-span ratios.



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PROBLEMS

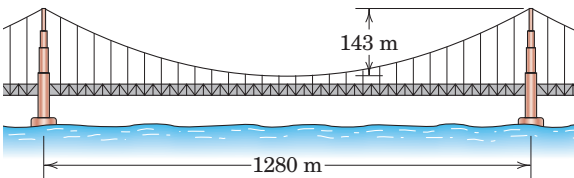
(The problems marked with an asterisk (*) involve transcendental equations which may be solved with a computer or by graphical methods.)

Introductory Problems

- 5/145** A coil of surveyor's tape 30 m in length has a mass of 0.283 kg. When the tape is stretched between two points on the same level by a tension of 42 N at each end, calculate the sag h of the tape in the middle.

Ans. $h = 248$ mm

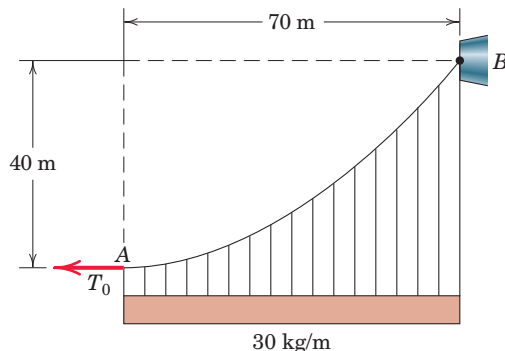
- 5/146** The Golden Gate Bridge in San Francisco has a main span of 1280 m, a sag of 143 m, and a total static loading of 310.8 kN per lineal meter of horizontal measurement. The weight of both of the main cables is included in this figure and is assumed to be uniformly distributed along the horizontal. The angle made by the cable with the horizontal at the top of the tower is the same on each side of each tower. Calculate the midspan tension T_0 in each of the main cables and the compressive force C exerted by each cable on the top of each tower.



Problem 5/146

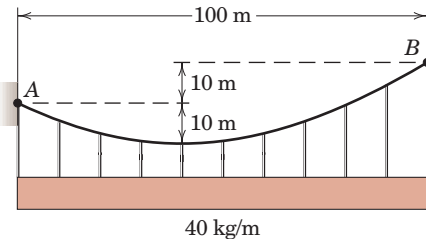
- 5/147** Calculate the tension T_0 in the cable at A necessary to support the load distributed uniformly with respect to the horizontal. Also find the angle θ made by the cable with the horizontal at the attachment point B.

Ans. $T_0 = 18.03$ kN, $\theta = 48.8^\circ$



Problem 5/147

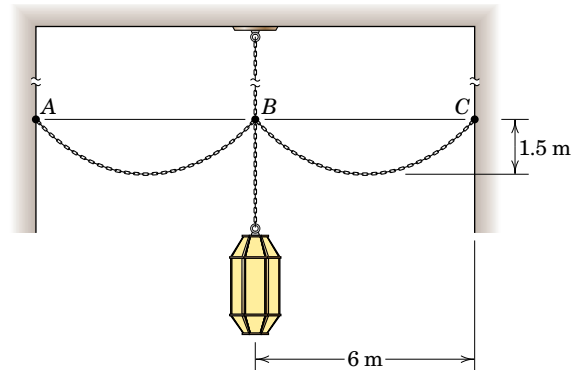
- 5/148** A cable supports a load of 40 kg/m uniformly distributed along the horizontal and is suspended from two fixed points A and B located as shown. Calculate the cable tensions at A and B and the minimum tension T_0 .



Problem 5/148

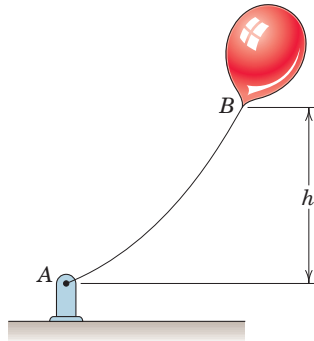
- *5/149** A light fixture is suspended from the ceiling of an outside portico. Four chains, two of which are shown, prevent excessive motion of the fixture during windy conditions. If the chains weigh 200 newtons per meter of length, determine the chain tension at C and the length L of chain BC.

Ans. $T_C = 945$ N, $L = 6.90$ m



Problem 5/149

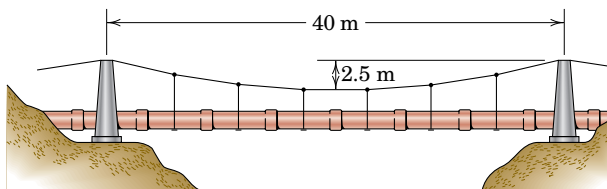
- 5/150** An advertising balloon is moored to a post with a cable which has a mass of 0.12 kg/m . In a wind the cable tensions at A and B are 110 N and 230 N , respectively. Determine the height h of the balloon.



Problem 5/150

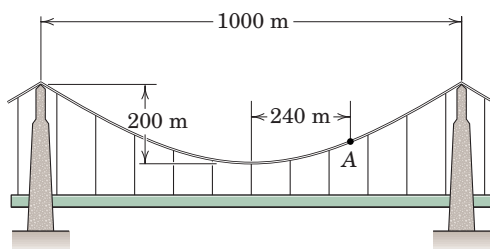
- 5/151** A horizontal 350-mm -diameter water pipe is supported over a ravine by the cable shown. The pipe and the water within it have a combined mass of 1400 kg per meter of its length. Calculate the compression C exerted by the cable on each support. The angles made by the cable with the horizontal are the same on both sides of each support.

Ans. $C = 549 \text{ kN}$



Problem 5/151

- 5/152** Strain-gage measurements made on the cables of the suspension bridge at position A indicate an increase of 2.14 MN of tension in *each* of the two main cables because the bridge has been repaved. Determine the total mass m' of added paving material used per foot of roadway.

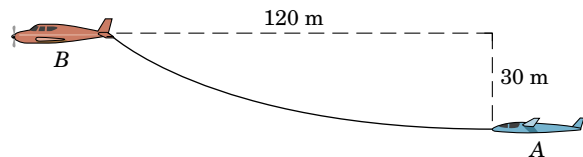


Problem 5/152

Representative Problems

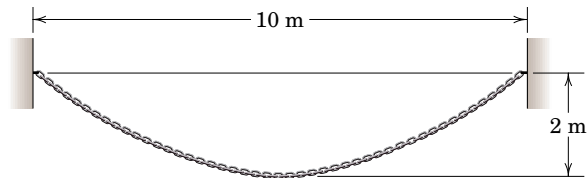
- *5/153** The glider A is being towed in level flight and is 120 m behind and 30 m below the tow plane B . The tangent to the cable at the glider is horizontal. The cable has a mass of 0.750 kg per meter of length. Calculate the horizontal tension T_0 in the cable at the glider. Neglect air resistance and compare your result with that obtained by approximating the cable shape by a parabola.

Ans. $T_0 = 1801 \text{ N}$, $(T_0)_{\text{par}} = 1766 \text{ N}$



Problem 5/153

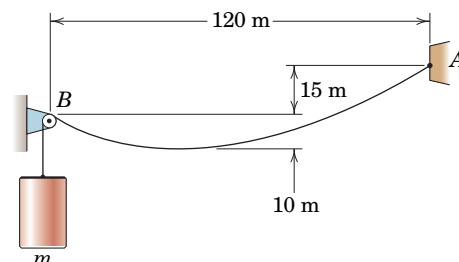
- *5/154** Find the total length L of chain which will have a sag of 2 m when suspended from two points on the same horizontal line 10 m apart.



Problem 5/154

- 5/155** A cable weighing 40 newtons per meter of length is suspended from point A and passes over the small pulley at B . Determine the mass m of the attached cylinder which will produce a sag of 10 m . With the small sag-to-span ratio, approximation as a parabolic cable may be used.

Ans. $m = 480 \text{ kg}$



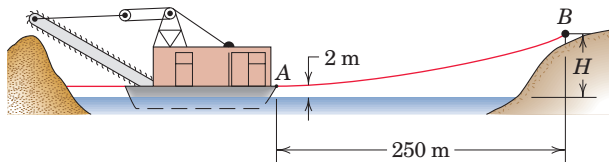
Problem 5/155

- *5/156** Repeat Prob. 5/155, but do not use the approximation of a parabolic cable. Compare your results with the printed answer for Prob. 5/155.

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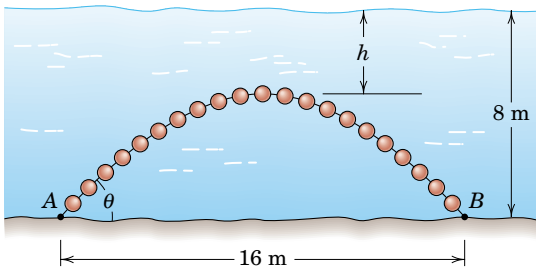
5/157 A floating dredge is anchored in position with a single stern cable which has a horizontal direction at the attachment A and extends a horizontal distance of 250 m to an anchorage B on shore. A tension of 300 kN is required in the cable at A . If the cable has a mass of 22 kg per meter of its length, compute the required height H of the anchorage above water level and find the length s of cable between A and B .

Ans. $H = 24.5$ m, $s = 251$ m



Problem 5/157

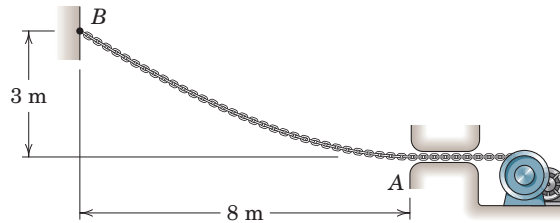
***5/158** A series of spherical floats are equally spaced and securely fastened to a flexible cable of length 20 m. Ends A and B are anchored 16 m apart to the bottom of a fresh-water lake at a depth of 8 m. The floats and cable have a combined weight of 100 N per meter of cable length, and the buoyancy of the water produces an upward force of 560 N per meter of cable length. Calculate the depth h below the surface to the top of the line of floats. Also find the angle θ made by the line of floats with the horizontal at A .



Problem 5/158

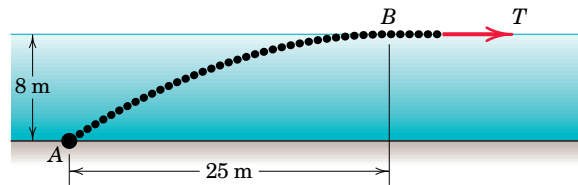
***5/159** Determine the length L of chain required from B to A and the corresponding tension at A if the slope of the chain is to be horizontal as it enters the guide at A . The weight of the chain is 140 N per meter of its length.

Ans. $L = 8.71$ m, $T_A = 1559$ N



Problem 5/159

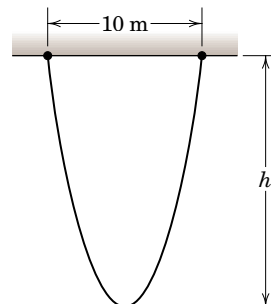
***5/160** Numerous small flotation devices are attached to the cable, and the difference between buoyancy and weight results in a net upward force of 30 newtons per meter of cable length. Determine the force T which must be applied to cause the cable configuration shown.



Problem 5/160

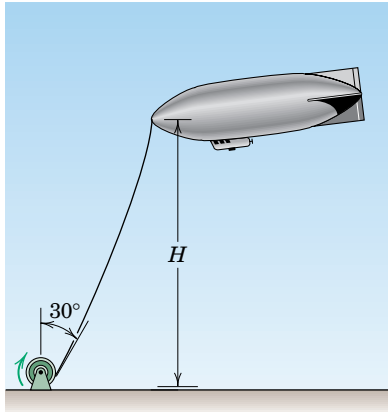
***5/161** A rope 40 m in length is suspended between two points which are separated by a horizontal distance of 10 m. Compute the distance h to the lowest part of the loop.

Ans. $h = 18.53$ m



Problem 5/161

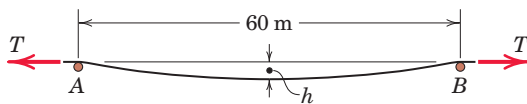
- 5/162** The blimp is moored to the ground winch in a gentle wind with 100 m of 12-mm cable which has a mass of 0.51 kg/m. A torque of 400 N·m on the drum is required to start winding in the cable. At this condition the cable makes an angle of 30° with the vertical as it approaches the winch. Calculate the height H of the blimp. The diameter of the drum is 0.5 m.



Problem 5/162

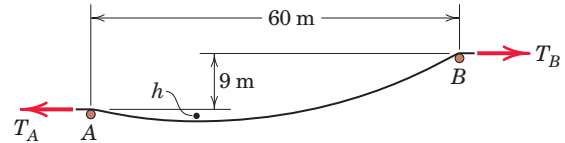
- *5/163** A cable installation crew wishes to establish the dependence of the tension T on the cable sag h . Plot both the minimum tension T_0 and the tension T at the supports A and B as functions of h for $1 \leq h \leq 10$ m. The cable mass per unit length is 3 kg/m. State the values of T_0 and T for $h = 2$ m.

Ans. $T_0 = 6630$ N, $T = 6690$ N



Problem 5/163

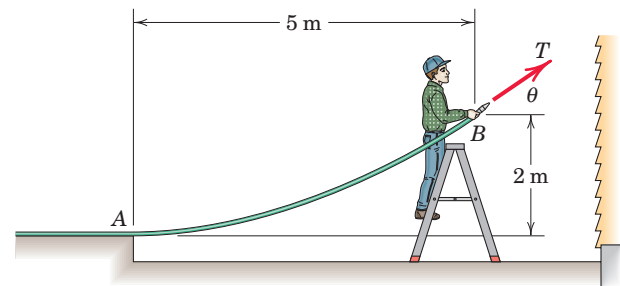
- *5/164** The cable of Prob. 5/163 is now placed on supports A and B whose elevation differs by 9 m as shown. Plot the minimum tension T_0 , the tension T_A at support A , and the tension T_B at support B as functions of h for $1 \leq h \leq 10$ m, where h is the sag below point A . State all three tensions for $h = 2$ m and compare with the results of Prob. 5/163. The cable mass per unit length is 3 kg/m.



Problem 5/164

- *5/165** In preparing to spray-clean a wall, a person arranges a hose as shown in the figure. The hose is horizontal at A and has a mass of 0.75 kg/m when empty and 1.25 kg/m when full of water. Determine the necessary tension T and angle θ for both the empty and full hose.

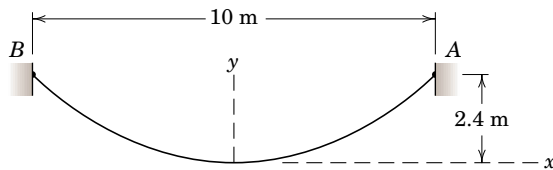
Ans. $T = 63.0$ N (empty), $T = 105.0$ N (full)
 $\theta = 40.0^\circ$ in both cases



Problem 5/165

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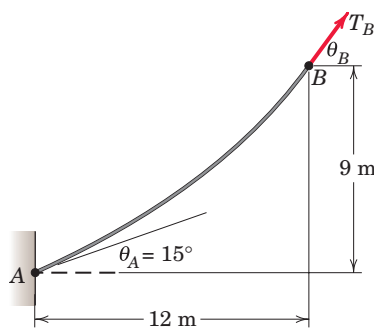
- *5/166** A length of cable which has a mass of 1.2 kg/m is to have a sag of 2.4 m when suspended from the two points A and B on the same horizontal line 10 m apart. For comparison purposes, determine the length L of cable required and plot its configuration for the two cases of (a) assuming a parabolic shape and (b) using the proper catenary model. In order to more clearly distinguish between the two cases, also plot the difference $(y_C - y_P)$ as a function of x , where C and P refer to catenary and parabola, respectively.



Problem 5/166

- *5/167** A power line is suspended from two towers 200 m apart on the same horizontal line. The cable has a mass of 18.2 kg per meter of length and has a sag of 32 m at midspan. If the cable can support a maximum tension of 60 kN, determine the mass ρ of ice per meter which can form on the cable without exceeding the maximum cable tension.
- Ans.* $\rho = 13.44$ kg of ice per meter

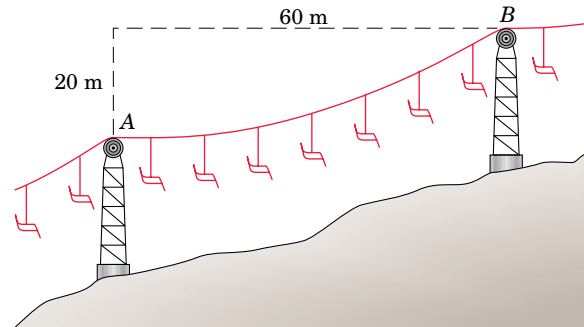
- *5/168** A cable which has a mass of 0.5 kg per meter of length is attached to point A. A tension T_B is applied to point B, causing the angle θ_A to be 15° . Determine T_B and θ_B .



Problem 5/168

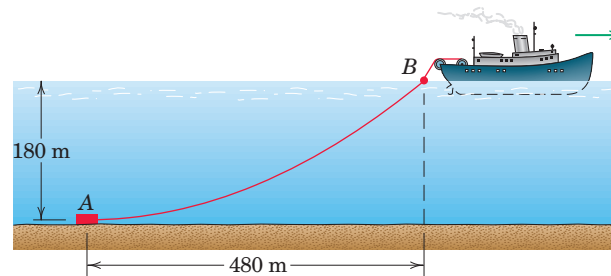
- *5/169** The moving cable for a ski lift has a mass of 10 kg/m and carries equally spaced chairs and passengers, whose added mass is 20 kg/m when averaged over the length of the cable. The cable leads horizontally from the supporting guide wheel at A. Calculate the tensions in the cable at A and B and the length s of the cable between A and B.

Ans. $T_A = 27.4$ kN, $T_B = 33.3$ kN, $s = 64.2$ m



Problem 5/169

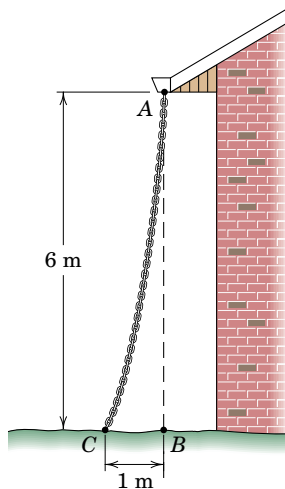
- *5/170** A cable ship tows a plow A during a survey of the ocean floor for later burial of a telephone cable. The ship maintains a constant low speed with the plow at a depth of 180 m and with a sufficient length of cable so that it leads horizontally from the plow, which is 480 m astern of the ship. The tow cable has an effective weight of 45.2 N/m when the buoyancy of the water is accounted for. Also, the forces on the cable due to movement through the water are neglected at the low speed. Compute the horizontal force T_0 applied to the plow and the maximum tension in the cable. Also find the length of the tow cable from point A to point B.



Problem 5/170

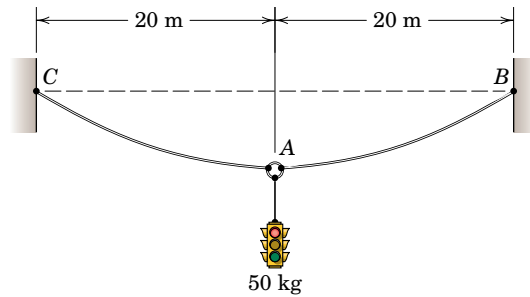
- *5/171** For aesthetic reasons, chains are sometimes used instead of downspouts on small buildings in order to direct roof runoff water from the gutter down to ground level. The architect of the illustrated building specified a 6-m vertical chain from A to B , but the builder decided to use a 6.1-m chain from A to C as shown in order to place the water farther from the structure. By what percentage n did the builder increase the magnitude of the force exerted on the gutter at A over that figured by the architect? The chain weighs 100 N per meter of its length.

Ans. $n = 29.0\%$



Problem 5/171

- *5/172** A 50-kg traffic signal is suspended by two 21-m cables which have a mass of 1.2 kg per meter of length. Determine the vertical deflection δ of the junction ring A relative to its position before the signal is added.



Problem 5/172

5/9 FLUID STATICS

So far in this chapter we have treated the action of forces on and between *solid* bodies. In this article we consider the equilibrium of bodies subjected to forces due to fluid pressures. A *fluid* is any continuous substance which, when at rest, is unable to support shear force. A shear force is one tangent to the surface on which it acts and is developed when differential velocities exist between adjacent layers of fluids. Thus, a fluid at rest can exert only normal forces on a bounding surface. Fluids may be either gaseous or liquid. The statics of fluids is generally called *hydrostatics* when the fluid is a liquid and *aerostatics* when the fluid is a gas.

Fluid Pressure

The pressure at any given point in a fluid is the same in all directions (Pascal's law). We may prove this by considering the equilibrium of an infinitesimal triangular prism of fluid as shown in Fig. 5/30. The fluid pressures normal to the faces of the element are p_1 , p_2 , p_3 , and p_4 as shown. With force equal to pressure times area, the equilibrium of forces in the x - and y -directions gives

$$p_1 dy dz = p_3 ds dz \sin \theta \quad p_2 dx dz = p_3 ds dz \cos \theta$$

Since $ds \sin \theta = dy$ and $ds \cos \theta = dx$, these equations require that

$$p_1 = p_2 = p_3 = p$$

By rotating the element through 90° , we see that p_4 is also equal to the other pressures. Thus, the pressure at any point in a fluid at rest is the same in all directions. In this analysis we need not account for the weight of the fluid element because, when the weight per unit volume (density ρ times g) is multiplied by the volume of the element, a differential quantity of third order results which disappears in the limit compared with the second-order pressure-force terms.

In all fluids at rest the pressure is a function of the vertical dimension. To determine this function, we consider the forces acting on a differential element of a vertical column of fluid of cross-sectional area dA , as shown in Fig. 5/31. The positive direction of vertical measurement h is taken downward. The pressure on the upper face is p , and that on the lower face is p plus the change in p , or $p + dp$. The weight of the element equals ρg multiplied by its volume. The normal forces on the lateral surface, which are horizontal and do not affect the balance of forces in the vertical direction, are not shown. Equilibrium of the fluid element in the h -direction requires

$$p dA + \rho g dA dh - (p + dp) dA = 0$$

$$dp = \rho g dh \quad (5/23)$$

This differential relation shows us that the pressure in a fluid increases with depth or decreases with increased elevation. Equation 5/23 holds

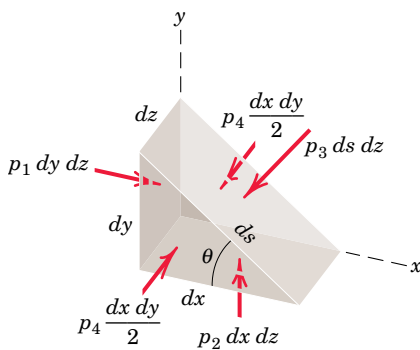


Figure 5/30

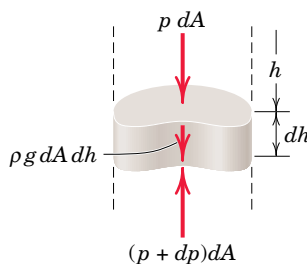


Figure 5/31

for both liquids and gases, and agrees with our common observations of air and water pressures.

Fluids which are essentially incompressible are called *liquids*, and for most practical purposes we may consider their density ρ constant for every part of the liquid.* With ρ a constant, integration of Eq. 5/23 gives

$$p = p_0 + \rho gh \quad (5/24)$$

The pressure p_0 is the pressure on the surface of the liquid where $h = 0$. If p_0 is due to atmospheric pressure and the measuring instrument records only the increment above atmospheric pressure,[†] the measurement gives what is called *gage pressure*. It is computed from $p = \rho gh$.

The common unit for pressure in SI units is the kilopascal (kPa), which is the same as a kilonewton per square meter (10^3 N/m^2). In computing pressure, if we use Mg/m^3 for ρ , m/s^2 for g , and m for h , then the product ρgh gives us pressure in kPa directly. For example, the pressure at a depth of 10 m in fresh water is

$$\begin{aligned} p &= \rho gh = \left(1.0 \frac{\text{Mg}}{\text{m}^3}\right) \left(9.81 \frac{\text{m}}{\text{s}^2}\right) (10 \text{ m}) = 98.1 \left(10^3 \frac{\text{kg} \cdot \text{m}}{\text{s}^2} \frac{1}{\text{m}^2}\right) \\ &= 98.1 \text{ kN/m}^2 = 98.1 \text{ kPa} \end{aligned}$$

In the U.S. customary system, fluid pressure is generally expressed in pounds per square inch (lb/in.^2) or occasionally in pounds per square foot (lb/ft^2). Thus, at a depth of 10 ft in fresh water the pressure is

$$p = \rho gh = \left(62.4 \frac{\text{lb}}{\text{ft}^3}\right) \left(\frac{1}{1728} \frac{\text{ft}^3}{\text{in.}^3}\right) (120 \text{ in.}) = 4.33 \text{ lb/in.}^2$$

Hydrostatic Pressure on Submerged Rectangular Surfaces

A body submerged in a liquid, such as a gate valve in a dam or the wall of a tank, is subjected to fluid pressure acting normal to its surface and distributed over its area. In problems where fluid forces are appreciable, we must determine the resultant force due to the distribution of pressure on the surface and the position at which this resultant acts. For systems open to the atmosphere, the atmospheric pressure p_0 acts over all surfaces and thus yields a zero resultant. In such cases, then, we need to consider only the gage pressure $p = \rho gh$, which is the increment above atmospheric pressure.

Consider the special but common case of the action of hydrostatic pressure on the surface of a rectangular plate submerged in a liquid. Figure 5/32a shows such a plate 1-2-3-4 with its top edge horizontal and with the plane of the plate making an arbitrary angle θ with the vertical plane. The horizontal surface of the liquid is represented by the x - y' plane. The fluid pressure (gage) acting normal to the plate at point 2 is

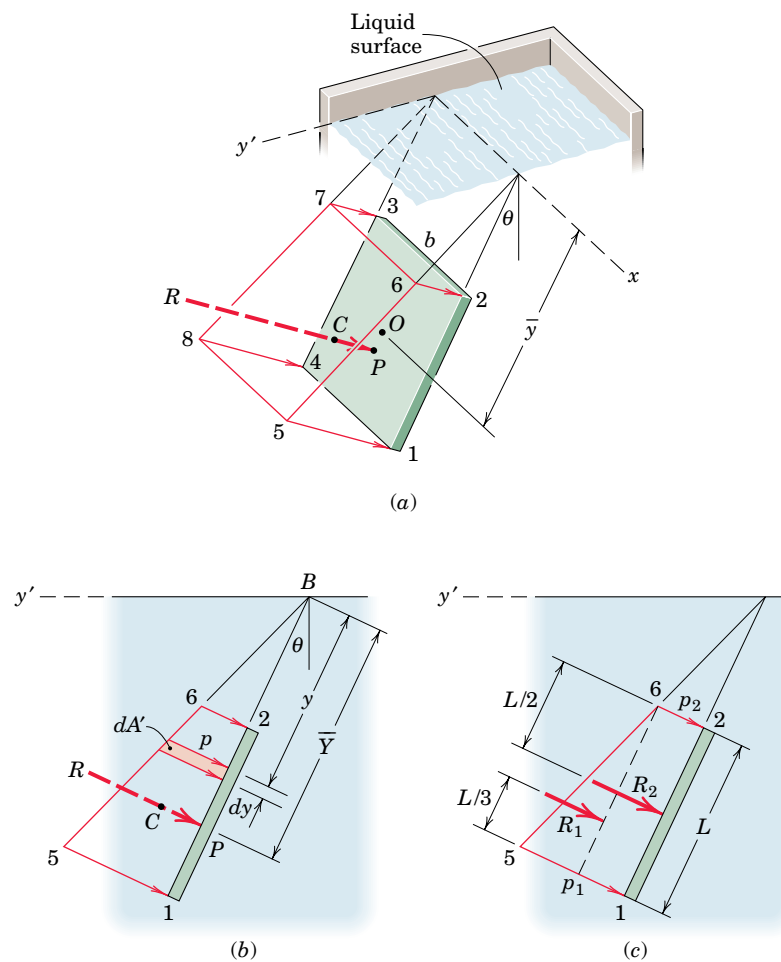
*See Table D/1, Appendix D, for table of densities.

[†]Atmospheric pressure at sea level may be taken to be 101.3 kPa or 14.7 lb/in.^2

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represented by the arrow 6-2 and equals ρg times the vertical distance from the liquid surface to point 2. This same pressure acts at all points along the edge 2-3. At point 1 on the lower edge, the fluid pressure equals ρg times the depth of point 1, and this pressure is the same at all points along edge 1-4. The variation of pressure p over the area of the plate is governed by the linear depth relationship and therefore it is represented by the arrow p , shown in Fig. 5/32*b*, which varies linearly from the value 6-2 to the value 5-1. The resultant force produced by this pressure distribution is represented by R , which acts at some point P called the *center of pressure*.

The conditions which prevail at the vertical section 1-2-6-5 in Fig. 5/32*a* are identical to those at section 4-3-7-8 and at every other vertical section normal to the plate. Thus, we may analyze the problem from the two-dimensional view of a vertical section as shown in Fig. 5/32*b* for section 1-2-6-5. For this section the pressure distribution is trapezoidal. If b is the horizontal width of the plate measured normal to the plane of the figure (dimension 2-3 in Fig. 5/32*a*), an element of plate area over which


Figure 5/32

the pressure $p = \rho gh$ acts is $dA = b dy$, and an increment of the resultant force is $dR = p dA = bp dy$. But $p dy$ is merely the shaded increment of trapezoidal area dA' , so that $dR = b dA'$. We may therefore express the resultant force acting on the entire plate as the trapezoidal area 1-2-6-5 times the width b of the plate,

$$R = b \int dA' = bA'$$

Be careful not to confuse the physical area A of the plate with the geometrical area A' defined by the trapezoidal distribution of pressure.

The trapezoidal area representing the pressure distribution is easily expressed by using its average altitude. The resultant force R may therefore be written in terms of the average pressure $p_{av} = \frac{1}{2}(p_1 + p_2)$ times the plate area A . The average pressure is also the pressure which exists at the average depth, measured to the centroid O of the plate. An alternative expression for R is therefore

$$R = p_{av}A = \rho g \bar{h}A$$

where $\bar{h} = \bar{y} \cos \theta$.

We obtain the line of action of the resultant force R from the principle of moments. Using the x -axis (point B in Fig. 5/32b) as the moment axis yields $R\bar{Y} = \int y(pb dy)$. Substituting $p dy = dA'$ and $R = bA'$ and canceling b give

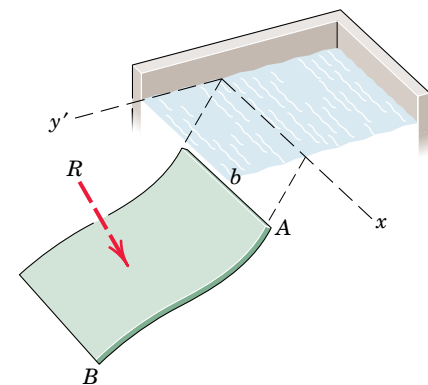
$$\bar{Y} = \frac{\int y dA'}{\int dA'}$$

which is simply the expression for the centroidal coordinate of the trapezoidal area A' . In the two-dimensional view, therefore, the resultant R passes through the centroid C of the trapezoidal area defined by the pressure distribution in the vertical section. Clearly \bar{Y} also locates the centroid C of the truncated prism 1-2-3-4-5-6-7-8 in Fig. 5/32a through which the resultant passes.

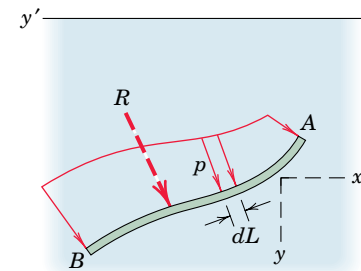
For a trapezoidal distribution of pressure, we may simplify the calculation by dividing the trapezoid into a rectangle and a triangle, Fig. 5/32c, and separately considering the force represented by each part. The force represented by the rectangular portion acts at the center O of the plate and is $R_2 = p_2A$, where A is the area 1-2-3-4 of the plate. The force R_1 represented by the triangular increment of pressure distribution is $\frac{1}{2}(p_1 - p_2)A$ and acts through the centroid of the triangular portion shown.

Hydrostatic Pressure on Cylindrical Surfaces

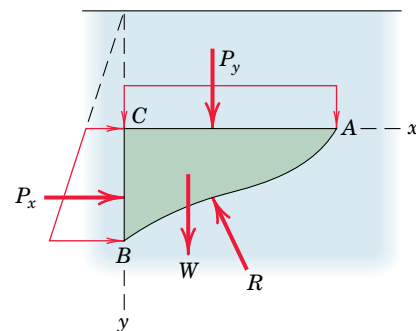
The determination of the resultant R due to distributed pressure on a submerged curved surface involves more calculation than for a flat surface. For example, consider the submerged cylindrical surface shown in Fig. 5/33a where the elements of the curved surface are parallel to the horizontal surface x - y' of the liquid. Vertical sections perpendicular to the surface all disclose the same curve AB and the same pressure distri-



(a)



(b)



(c)

Figure 5/33

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bution. Thus, the two-dimensional representation in Fig. 5/33b may be used. To find R by a direct integration, we need to integrate the x - and y -components of dR along the curve AB , since dR continuously changes direction. Thus,

$$R_x = b \int (p \, dL)_x = b \int p \, dy \quad \text{and} \quad R_y = b \int (p \, dL)_y = b \int p \, dx$$

A moment equation would now be required if we wished to establish the position of R .

A second method for finding R is usually much simpler. Consider the equilibrium of the block of liquid ABC directly above the surface, shown in Fig. 5/33c. The resultant R then appears as the equal and opposite reaction of the surface on the block of liquid. The resultants of the pressures along AC and CB are P_y and P_x , respectively, and are easily obtained. The weight W of the liquid block is calculated from the area ABC of its section multiplied by the constant dimension b and by ρg . The weight W passes through the centroid of area ABC . The equilibrant R is then determined completely from the equilibrium equations which we apply to the free-body diagram of the fluid block.

Hydrostatic Pressure on Flat Surfaces of Any Shape

Figure 5/34a shows a flat plate of any shape submerged in a liquid. The horizontal surface of the liquid is the plane $x-y'$, and the plane of the plate makes an angle θ with the vertical. The force acting on a differential strip of area dA parallel to the surface of the liquid is $dR = p \, dA = \rho g h \, dA$. The pressure p has the same magnitude throughout the length of the strip, because there is no change of

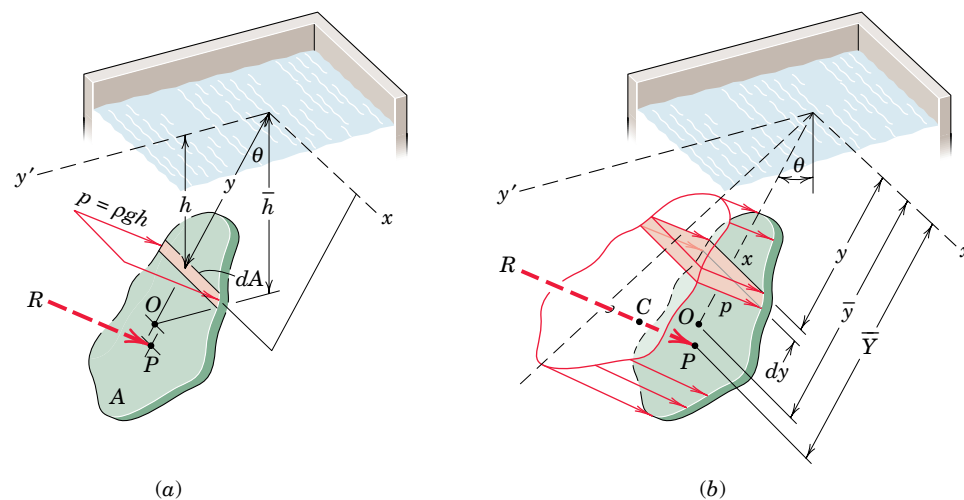


Figure 5/34

depth along the strip. We obtain the total force acting on the exposed area A by integration, which gives

$$R = \int dR = \int p \, dA = \rho g \int h \, dA$$

Substituting the centroidal relation $\bar{h}A = \int h \, dA$ gives us

$$R = \rho g \bar{h}A \quad (5/25)$$

The quantity $\rho g \bar{h}$ is the pressure which exists at the depth of the centroid O of the area and is the average pressure over the area.

We may also represent the resultant R geometrically by the volume V' of the figure shown in Fig. 5/34b. Here the fluid pressure p is represented as a dimension normal to the plate regarded as a base. We see that the resulting volume is a truncated right cylinder. The force dR acting on the differential area $dA = x \, dy$ is represented by the elemental volume $p \, dA$ shown by the shaded slice, and the total force is represented by the total volume of the cylinder. We see from Eq. 5/25 that the average altitude of the truncated cylinder is the average pressure $\rho g \bar{h}$ which exists at a depth corresponding to the centroid O of the area exposed to pressure.

For problems where the centroid O or the volume V' is not readily apparent, a direct integration may be performed to obtain R . Thus,

$$R = \int dR = \int p \, dA = \int \rho g h x \, dy$$

where the depth h and the length x of the horizontal strip of differential area must be expressed in terms of y to carry out the integration.

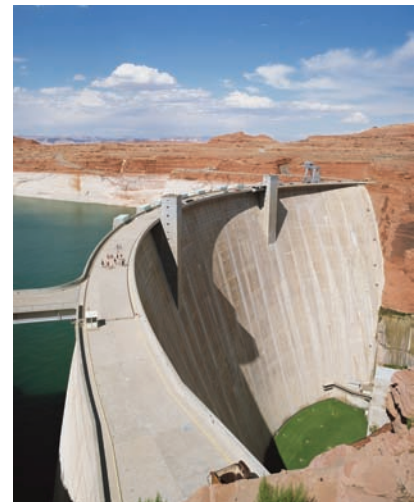
After the resultant is obtained, we must determine its location. Using the principle of moments with the x -axis of Fig. 5/34b as the moment axis, we obtain

$$R\bar{Y} = \int y \, dR \quad \text{or} \quad \bar{Y} = \frac{\int y(p x \, dy)}{\int p x \, dy} \quad (5/26)$$

This second relation satisfies the definition of the coordinate \bar{Y} to the centroid of the volume V' of the pressure-area truncated cylinder. We conclude, therefore, that the resultant R passes through the centroid C of the volume described by the plate area as base and the linearly varying pressure as the perpendicular coordinate. The point P at which R is applied to the plate is the center of pressure. Note that the center of pressure P and the centroid O of the plate area are *not* the same.

Buoyancy

Archimedes is credited with discovering the *principle of buoyancy*. This principle is easily explained for any fluid, gaseous or liquid, in equilibrium. Consider a portion of the fluid defined by an imaginary closed



Glen Canyon Dam at Lake Powell, Arizona

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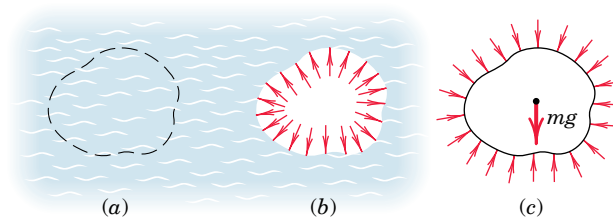


Figure 5/35

surface, as illustrated by the irregular dashed boundary in Fig. 5/35*a*. If the body of the fluid could be sucked out from within the closed cavity and replaced simultaneously by the forces which it exerted on the boundary of the cavity, Fig. 5/35*b*, the equilibrium of the surrounding fluid would not be disturbed. Furthermore, a free-body diagram of the fluid portion before removal, Fig. 5/35*c*, shows that the resultant of the pressure forces distributed over its surface must be equal and opposite to its weight mg and must pass through the center of mass of the fluid element. If we replace the fluid element by a body of the same dimensions, the surface forces acting on the body held in this position will be identical to those acting on the fluid element. Thus, the resultant force exerted on the surface of an object immersed in a fluid is equal and opposite to the weight of fluid displaced and passes through the center of mass of the displaced fluid. This resultant force is called the force of *buoyancy*



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The design of ship hulls must take into account fluid dynamics as well as fluid statics.

$$F = \rho g V$$

(5/27)

where ρ is the density of the fluid, g is the acceleration due to gravity, and V is the volume of the fluid displaced. In the case of a liquid whose density is constant, the center of mass of the displaced liquid coincides with the centroid of the displaced volume.

Thus when the density of an object is less than the density of the fluid in which it is fully immersed, there is an imbalance of force in the vertical direction, and the object rises. When the immersing fluid is a liquid, the object continues to rise until it comes to the surface of the liquid and then comes to rest in an equilibrium position, assuming that the density of the new fluid above the surface is less than the density of the object. In the case of the surface boundary between a liquid and a gas, such as water and air, the effect of the gas pressure on that portion of the floating object above the liquid is balanced by the added pressure in the liquid due to the action of the gas on its surface.

An important problem involving buoyancy is the determination of the stability of a floating object, such as a ship hull shown in cross section in an upright position in Fig. 5/36*a*. Point B is the centroid of the displaced volume and is called the *center of buoyancy*. The resultant of the forces exerted on the hull by the water pressure is the buoyancy force F which passes through B and is equal and opposite to the weight W of the ship. If the ship is caused to list through an angle α , Fig. 5/36*b*,

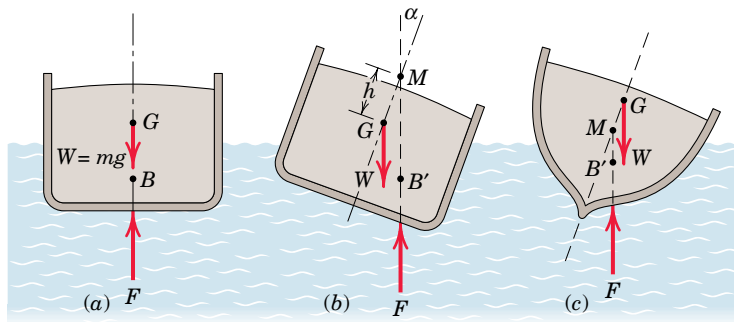
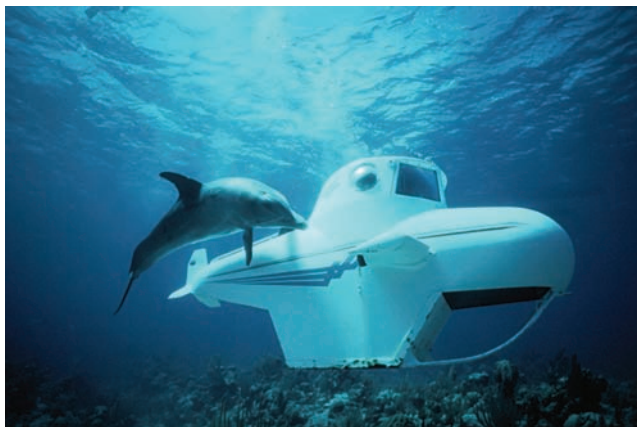


Figure 5/36

the shape of the displaced volume changes, and the center of buoyancy shifts to B' .

The point of intersection of the vertical line through B' with the centerline of the ship is called the *metacenter* M , and the distance h of M from the center of mass G is called the *metacentric height*. For most hull shapes h remains practically constant for angles of list up to about 20° . When M is above G , as in Fig. 5/36b, there is a righting moment which tends to bring the ship back to its upright position. If M is below G , as for the hull of Fig. 5/36c, the moment accompanying the list is in the direction to increase the list. This is clearly a condition of instability and must be avoided in the design of any ship.



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Submersible vessels must be designed with extremely large external pressures in mind.

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Sample Problem 5/18

A rectangular plate, shown in vertical section AB , is 4 m high and 6 m wide (normal to the plane of the paper) and blocks the end of a fresh-water channel 3 m deep. The plate is hinged about a horizontal axis along its upper edge through A and is restrained from opening by the fixed ridge B which bears horizontally against the lower edge of the plate. Find the force B exerted on the plate by the ridge.

Solution. The free-body diagram of the plate is shown in section and includes the vertical and horizontal components of the force at A , the unspecified weight $W = mg$ of the plate, the unknown horizontal force B , and the resultant R of the triangular distribution of pressure against the vertical face.

The density of fresh water is $\rho = 1.000 \text{ Mg/m}^3$ so that the average pressure is

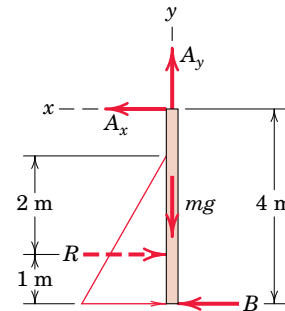
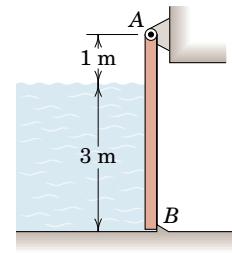
$$\textcircled{1} [p_{\text{av}} = \rho g \bar{h}] \quad p_{\text{av}} = 1.000(9.81)\left(\frac{3}{2}\right) = 14.72 \text{ kPa}$$

The resultant R of the pressure forces against the plate becomes

$$[R = p_{\text{av}}A] \quad R = (14.72)(3)(6) = 265 \text{ kN}$$

This force acts through the centroid of the triangular distribution of pressure, which is 1 m above the bottom of the plate. A zero moment summation about A establishes the unknown force B . Thus,

$$[\Sigma M_A = 0] \quad 3(265) - 4B = 0 \quad B = 198.7 \text{ kN} \quad \text{Ans.}$$



Helpful Hint

① Note that the units of pressure ρgh are

$$\left(10^3 \frac{\text{kg}}{\text{m}^3}\right)\left(\frac{\text{m}}{\text{s}^2}\right)(\text{m}) = \left(10^3 \frac{\text{kg} \cdot \text{m}}{\text{s}^2}\right)\left(\frac{1}{\text{m}^2}\right) = \text{kN/m}^2 = \text{kPa}.$$

Sample Problem 5/19

The air space in the closed fresh-water tank is maintained at a pressure of 5.5 kPa (above atmospheric). Determine the resultant force R exerted by the air and water on the end of the tank.

Solution. The pressure distribution on the end surface is shown, where $p_0 = 5.5 \text{ kPa}$. The specific weight of fresh water is $\mu = \rho g = 1000(9.81) = 9.81 \text{ kN/m}^3$ so that the increment of pressure Δp due to the water is

$$\Delta p = \mu \Delta h = 9.81(0.6) = 5.89 \text{ kPa}$$

① The resultant forces R_1 and R_2 due to the rectangular and triangular distributions of pressure, respectively, are

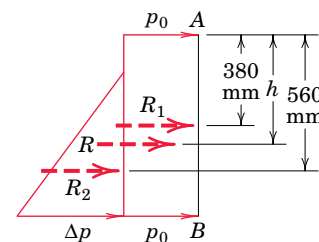
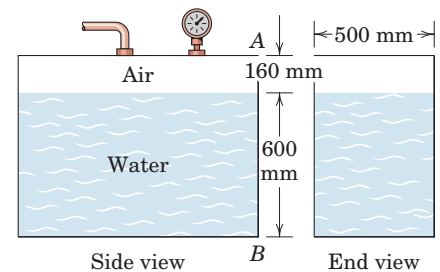
$$R_1 = p_0 A_1 = 5.5(0.760)(0.5) = 2.09 \text{ kN}$$

$$R_2 = \Delta p_{\text{av}} A_2 = \frac{5.89}{2} (0.6)(0.5) = 0.883 \text{ kN}$$

The resultant is then $R = R_1 + R_2 = 2.09 + 0.883 = 2.97 \text{ kN}$. Ans.

We locate R by applying the moment principle about A noting that R_1 acts through the center of the 760-mm depth and that R_2 acts through the centroid of the triangular pressure distribution 400 mm below the surface of the water and $400 + 160 = 560 \text{ mm}$ below A . Thus,

$$[Rh = \Sigma M_A] \quad 2.97h = 2.09(380) + 0.833(560) \quad h = 433 \text{ mm} \quad \text{Ans.}$$



Helpful Hint

① Dividing the pressure distribution into these two parts is decidedly the simplest way in which to make the calculation.

Sample Problem 5/20

Determine completely the resultant force R exerted on the cylindrical dam surface by the water. The density of fresh water is 1.000 Mg/m^3 , and the dam has a length b , normal to the paper, of 30 m.

Solution. The circular block of water BDO is isolated and its free-body diagram is drawn. The force P_x is

$$\textcircled{1} \quad P_x = \rho g \bar{h} A = \frac{\rho g r}{2} b r = \frac{(1.000)(9.81)(4)}{2} (30)(4) = 2350 \text{ kN}$$

The weight W of the water passes through the mass center G of the quarter-circular section and is

$$mg = \rho g V = (1.000)(9.81) \frac{\pi(4)^2}{4} (30) = 3700 \text{ kN}$$

Equilibrium of the section of water requires

$$[\Sigma F_x = 0] \quad R_x = P_x = 2350 \text{ kN}$$

$$[\Sigma F_y = 0] \quad R_y = mg = 3700 \text{ kN}$$

The resultant force R exerted by the fluid on the dam is equal and opposite to that shown acting on the fluid and is

$$[R = \sqrt{R_x^2 + R_y^2}] \quad R = \sqrt{(2350)^2 + (3700)^2} = 4380 \text{ kN} \quad \text{Ans.}$$

The x -coordinate of the point A through which R passes may be found from the principle of moments. Using B as a moment center gives

$$P_x \frac{r}{3} + mg \frac{4r}{3\pi} - R_y x = 0, \quad x = \frac{2350\left(\frac{4}{3}\right) + 3700\left(\frac{16}{3\pi}\right)}{3700} = 2.55 \text{ m} \quad \text{Ans.}$$

- Alternative Solution.** The force acting on the dam surface may be obtained by a direct integration of the components

$$dR_x = p \, dA \cos \theta \quad \text{and} \quad dR_y = p \, dA \sin \theta$$

where $p = \rho gh = \rho gr \sin \theta$ and $dA = b(r \, d\theta)$. Thus,

$$R_x = \int_0^{\pi/2} \rho gr^2 b \sin \theta \cos \theta \, d\theta = -\rho gr^2 b \left[\frac{\cos 2\theta}{4} \right]_0^{\pi/2} = \frac{1}{2} \rho gr^2 b$$

$$R_y = \int_0^{\pi/2} \rho gr^2 b \sin^2 \theta \, d\theta = \rho gr^2 b \left[\frac{\theta}{2} - \frac{\sin 2\theta}{4} \right]_0^{\pi/2} = \frac{1}{4} \pi \rho gr^2 b$$

Thus, $R = \sqrt{R_x^2 + R_y^2} = \frac{1}{2} \rho gr^2 b \sqrt{1 + \pi^2/4}$. Substituting the numerical values gives

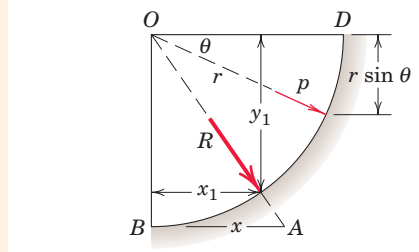
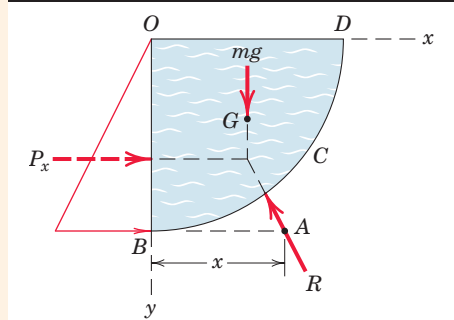
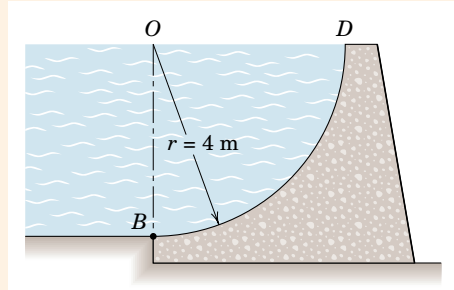
$$R = \frac{1}{2} (1.000)(9.81)(4^2)(30) \sqrt{1 + \pi^2/4} = 4380 \text{ kN} \quad \text{Ans.}$$

Since dR always passes through point O , we see that R also passes through O and, therefore, the moments of R_x and R_y about O must cancel. So we write $R_x y_1 = R_y x_1$, which gives us

$$x_1/y_1 = R_x/R_y = (\frac{1}{2} \rho gr^2 b) / (\frac{1}{4} \pi \rho gr^2 b) = 2/\pi$$

By similar triangles we see that

$$x/r = x_1/y_1 = 2/\pi \quad \text{and} \quad x = 2r/\pi = 2(4)/\pi = 2.55 \text{ m} \quad \text{Ans.}$$



Helpful Hints

- $\textcircled{1}$ See note $\textcircled{1}$ in Sample Problem 5/18 if there is any question about the units for ρgh .
- $\textcircled{2}$ This approach by integration is feasible here mainly because of the simple geometry of the circular arc.

318 Chapter 5 Distributed Forces

Sample Problem 5/21

Determine the resultant force R exerted on the semicircular end of the water tank shown in the figure if the tank is filled to capacity. Express the result in terms of the radius r and the water density ρ .

Solution I. We will obtain R first by a direct integration. With a horizontal strip of area $dA = 2x \, dy$ acted on by the pressure $p = \rho gy$, the increment of the resultant force is $dR = p \, dA$ so that

$$R = \int p \, dA = \int \rho gy(2x \, dy) = 2\rho g \int_0^r y\sqrt{r^2 - y^2} \, dy$$

Integrating gives $R = \frac{2}{3}\rho g r^3$ *Ans.*

The location of R is determined by using the principle of moments. Taking moments about the x -axis gives

$$[R\bar{Y} = \int y \, dR] \quad \frac{2}{3}\rho g r^3 \bar{Y} = 2\rho g \int_0^r y^2 \sqrt{r^2 - y^2} \, dy$$

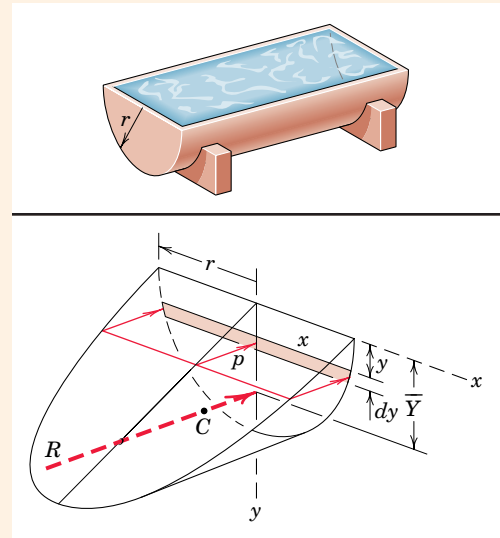
Integrating gives $\frac{2}{3}\rho g r^3 \bar{Y} = \frac{\rho g r^4}{4} \frac{\pi}{2}$ and $\bar{Y} = \frac{3\pi r}{16}$ *Ans.*

Solution II. We may use Eq. 5/25 directly to find R , where the average pressure is $\rho g \bar{h}$ and \bar{h} is the coordinate to the centroid of the area over which the pressure acts. For a semicircular area $\bar{h} = 4r/(3\pi)$.

$$[R = \rho g \bar{h} A] \quad R = \rho g \frac{4r}{3\pi} \frac{\pi r^2}{2} = \frac{2}{3}\rho g r^3 \quad \text{Ans.}$$

which is the volume of the pressure-area figure.

- ① The resultant R acts through the centroid C of the volume defined by the pressure-area figure. Calculation of the centroidal distance \bar{Y} involves the same integral obtained in *Solution I*.



Helpful Hint

- ① Be very careful not to make the mistake of assuming that R passes through the centroid of the area over which the pressure acts.

Sample Problem 5/22

A buoy in the form of a uniform 8-m pole 0.2 m in diameter has a mass of 200 kg and is secured at its lower end to the bottom of a fresh-water lake with 5 m of cable. If the depth of the water is 10 m, calculate the angle θ made by the pole with the horizontal.

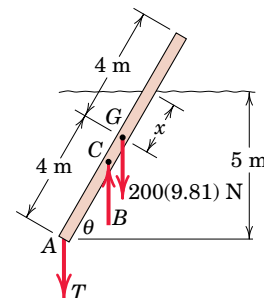
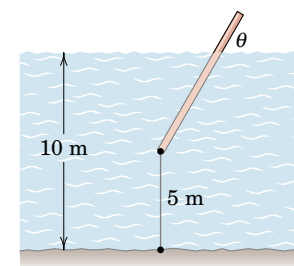
Solution. The free-body diagram of the buoy shows its weight acting through G , the vertical tension T in the anchor cable, and the buoyancy force B which passes through centroid C of the submerged portion of the buoy. Let x be the distance from G to the waterline. The density of fresh water is $\rho = 10^3 \, \text{kg/m}^3$, so that the buoyancy force is

$$[B = \rho g V] \quad B = 10^3(9.81)\pi(0.1)^2(4 + x) \, \text{N}$$

Moment equilibrium, $\Sigma M_A = 0$, about A gives

$$200(9.81)(4 \cos \theta) - [10^3(9.81)\pi(0.1)^2(4 + x)] \frac{4 + x}{2} \cos \theta = 0$$

Thus, $x = 3.14 \, \text{m}$ and $\theta = \sin^{-1}\left(\frac{5}{4 + 3.14}\right) = 44.5^\circ$ *Ans.*

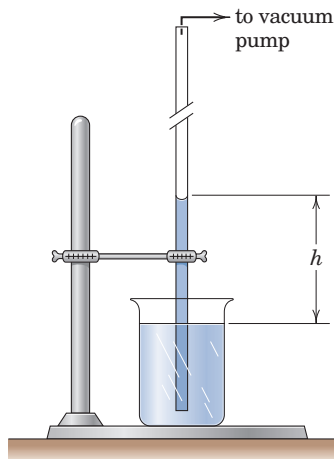


PROBLEMS

Introductory Problems

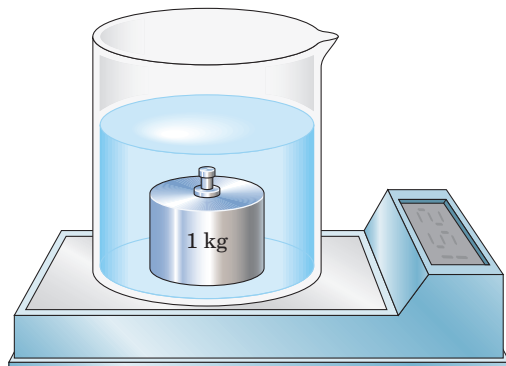
- 5/173** Determine the maximum height h to which a vacuum pump can cause the fresh water to rise. Assume standard atmospheric pressure of $1.0133(10^5)$ Pa. Repeat your calculations for mercury.

Ans. $h = 10.33$ m (water)
 $h = 0.761$ m (mercury)



Problem 5/173

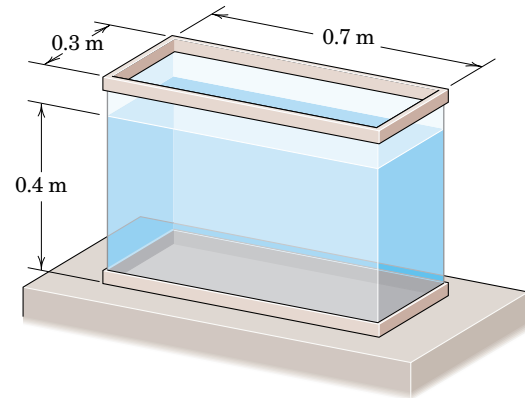
- 5/174** A beaker of fresh water is in place on the scale when the 1-kg stainless-steel weight is added to the beaker. What is the normal force which the weight exerts on the bottom of the beaker? By how much does the scale reading increase as the weight is added? Explain your answer.



Problem 5/174

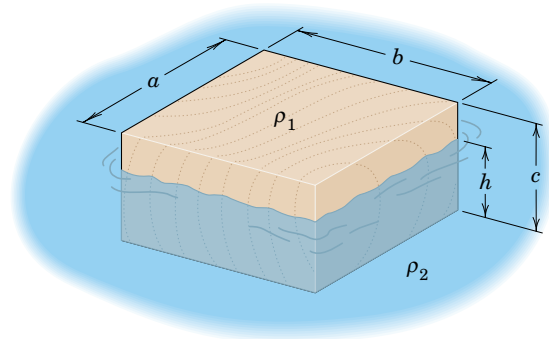
- 5/175** Specify the magnitude and location of the resultant force which acts on each side and the bottom of the aquarium due to the fresh water inside it.

Ans. Bottom force = 824 N
 Side forces = 235 N, 549 N
 All four side forces at $\frac{2}{3}$ depth



Problem 5/175

- 5/176** A rectangular block of density ρ_1 floats in a liquid of density ρ_2 . Determine the ratio $r = h/c$, where h is the submerged depth of block. Evaluate r for an oak block floating in fresh water and for steel floating in mercury.

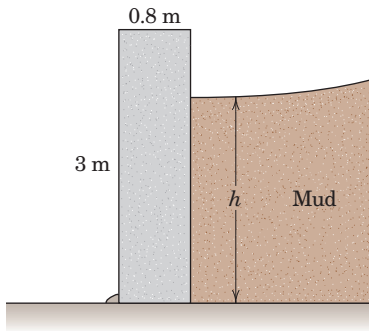


Problem 5/176

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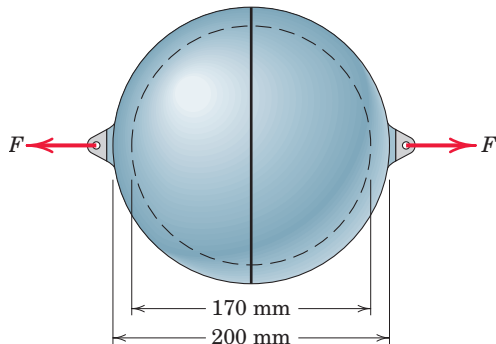
- 5/177** Calculate the depth h of mud for which the 3-m concrete retaining wall is on the verge of tipping about its forward edge A. The density of mud may be taken to be 1760 kg/m^3 and that of concrete to be 2400 kg/m^3 .

Ans. $h = 1.99 \text{ m}$



Problem 5/177

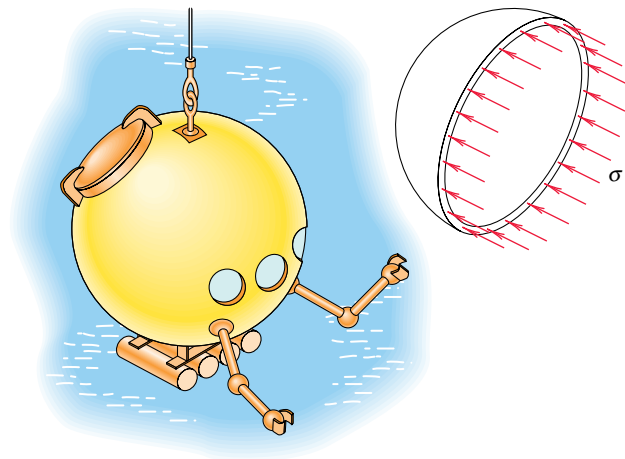
- 5/178** The two hemispherical shells are perfectly sealed together over their mating surfaces, and the air inside is partially evacuated to a pressure of 14 kPa. Atmospheric pressure is 101.3 kPa. Determine the force F required to separate the shells.



Problem 5/178

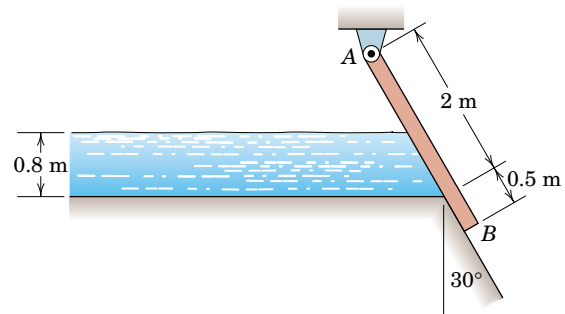
- 5/179** A deep-submersible diving chamber designed in the form of a spherical shell 1500 mm in diameter is ballasted with lead so that its weight slightly exceeds its buoyancy. Atmospheric pressure is maintained within the sphere during an ocean dive to a depth of 3 km. The thickness of the shell is 25 mm. For this depth calculate the compressive stress σ which acts on a diametral section of the shell, as indicated in the right-hand view.

Ans. $\sigma = 463 \text{ MPa}$



Problem 5/179

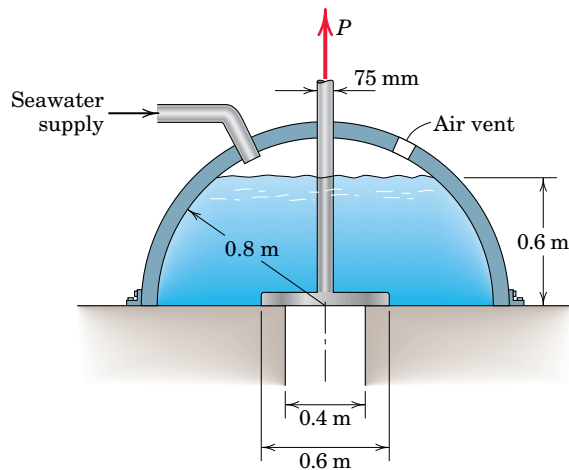
- 5/180** Fresh water in a channel is contained by the uniform 2.5-m plate freely hinged at A. If the gate is designed to open when the depth of the water reaches 0.8 m as shown in the figure, what must be the weight w (in newtons per meter of horizontal length into the paper) of the gate?



Problem 5/180

- 5/181** When the seawater level inside the hemispherical chamber reaches the 0.6-m level shown in the figure, the plunger is lifted, allowing a surge of seawater to enter the vertical pipe. For this fluid level (a) determine the average pressure σ supported by the seal area of the valve before force is applied to lift the plunger and (b) determine the force P (in addition to the force needed to support its weight) required to lift the plunger. Assume atmospheric pressure in all airspaces and in the seal area when contact ceases under the action of P .

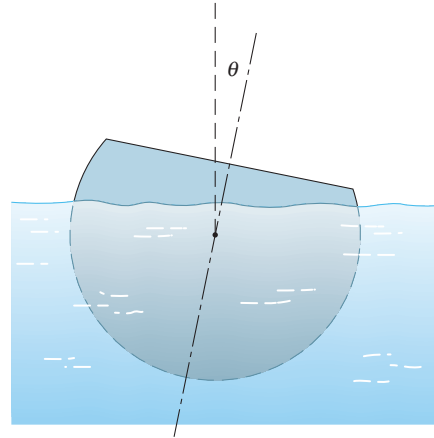
Ans. $\sigma = 10.74 \text{ kPa}$, $P = 1.687 \text{ kN}$



Problem 5/181

Representative Problems

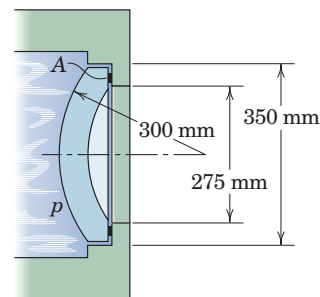
- 5/182** The figure shows the end view of a long homogeneous solid cylinder which floats in a liquid and has a removed segment. Show that $\theta = 0$ and $\theta = 180^\circ$ are the two values of the angle between its center-line and the vertical for which the cylinder floats in stable positions.



Problem 5/182

- 5/183** One of the critical problems in the design of deep-submergence vehicles is to provide viewing ports which will withstand tremendous hydrostatic pressures without fracture or leakage. The figure shows the cross section of an experimental acrylic window with spherical surfaces under test in a high-pressure liquid chamber. If the pressure p is raised to a level that simulates the effect of a dive to a depth of 1 km in sea water, calculate the average pressure σ supported by the gasket A.

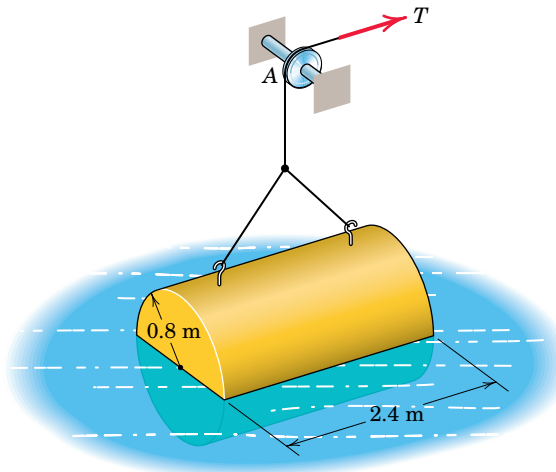
Ans. $\sigma = 26.4 \text{ MPa}$



Problem 5/183

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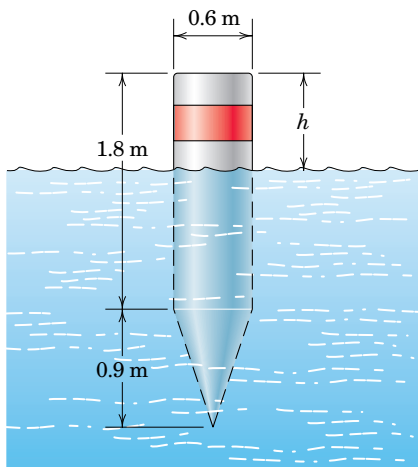
- 5/184** The solid concrete cylinder 2.4 m long and 1.6 m in diameter is supported in a half-submerged position in fresh water by a cable which passes over a fixed pulley at A. Compute the tension T in the cable. The cylinder is waterproofed by a plastic coating. (Consult Table D/1, Appendix D, as needed.)



Problem 5/184

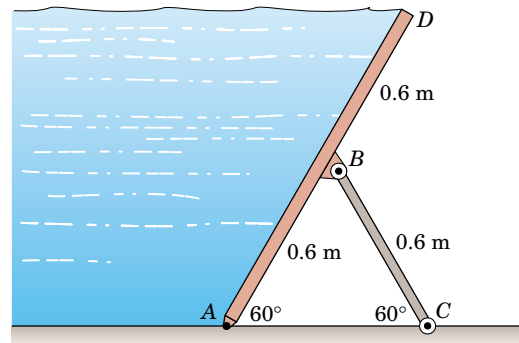
- 5/185** A marker buoy consisting of a cylinder and cone has the dimensions shown and has a mass of 285 kg. Determine the protrusion h when the buoy is floating in salt water. The buoy is weighted so that a low center of mass ensures stability.

Ans. $h = 1.121$ m



Problem 5/185

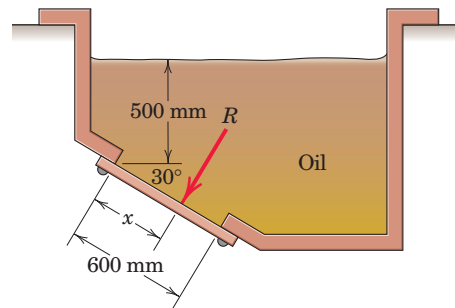
- 5/186** A fresh-water channel 3 m wide (normal to the plane of the paper) is blocked at its end by a rectangular barrier, shown in section ABD . Supporting struts BC are spaced every 0.6 m along the 3-m width. Determine the compression C in each strut. Neglect the weights of the members.



Problem 5/186

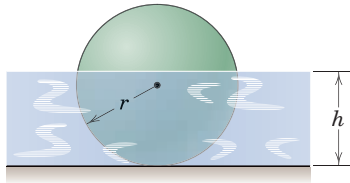
- 5/187** A vertical section of an oil sump is shown. The access plate covers a rectangular opening which has a dimension of 400 mm normal to the plane of the paper. Calculate the total force R exerted by the oil on the plate and the location x of R . The oil has a density of 900 kg/m^3 .

Ans. $R = 1377$ N, $x = 323$ mm



Problem 5/187

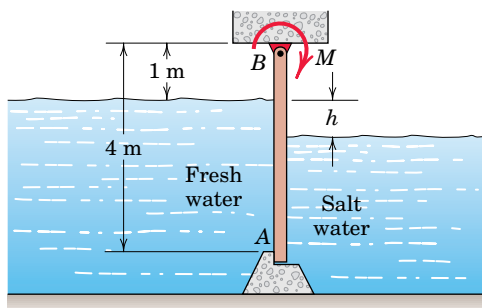
- 5/188** A homogeneous solid sphere of radius r is resting on the bottom of a tank containing a liquid of density ρ_l , which is greater than the density ρ_s of the sphere. As the tank is filled, a depth h is reached at which the sphere begins to float. Determine the expression for the density ρ_s of the sphere.



Problem 5/188

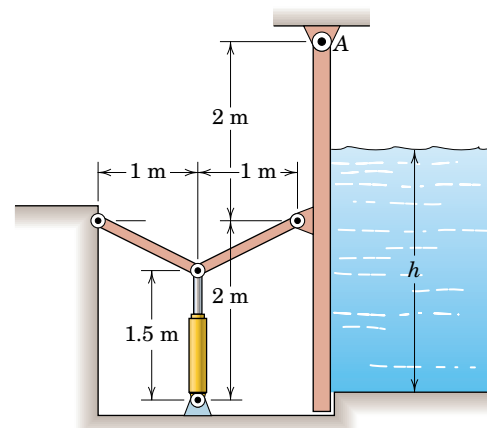
- 5/189** The rectangular gate shown in section is 3 m long (perpendicular to the paper) and is hinged about its upper edge B . The gate divides a channel leading to a fresh-water lake on the left and a salt-water tidal basin on the right. Calculate the torque M on the shaft of the gate at B required to prevent the gate from opening when the salt-water level drops to $h = 1$ m.

Ans. $M = 195.2 \text{ kN}\cdot\text{m}$



Problem 5/189

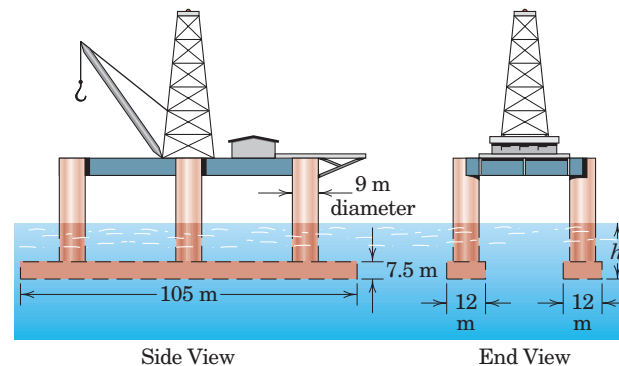
- 5/190** The hydraulic cylinder operates the toggle which closes the vertical gate against the pressure of fresh water on the opposite side. The gate is rectangular with a horizontal width of 2 m perpendicular to the paper. For a depth $h = 3$ m of water, calculate the required oil pressure p which acts on the 150-mm-diameter piston of the hydraulic cylinder.



Problem 5/190

- 5/191** The design of a floating oil-drilling platform consists of two rectangular pontoons and six cylindrical columns which support the working platform. When ballasted, the entire structure has a displacement of 26,000 metric tons (1 metric ton equals 1000 kg). Calculate the total draft h of the structure when it is moored in the ocean. The density of salt water is 1030 kg/m^3 . Neglect the vertical components of the mooring forces.

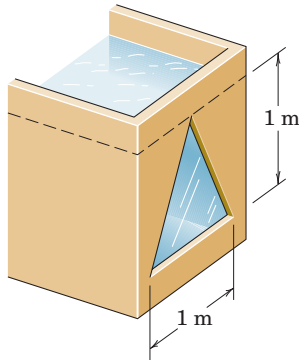
Ans. $h = 24.1 \text{ m}$



Problem 5/191

324 Chapter 5 Distributed Forces

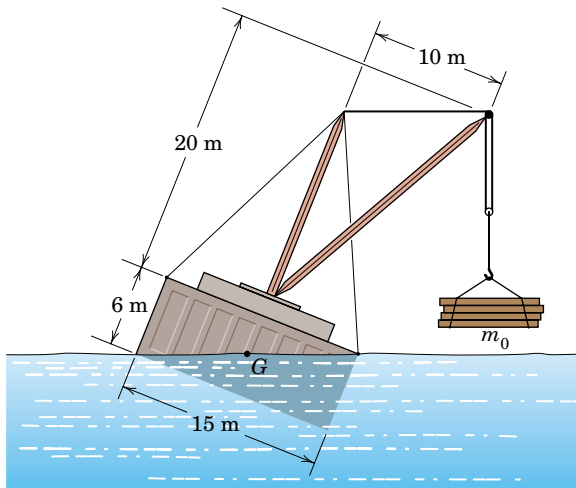
- 5/192** Determine the total force R exerted on the triangular window by the fresh water in the tank. The water level is even with the top of the window. Also determine the distance H from R to the water level.



Problem 5/192

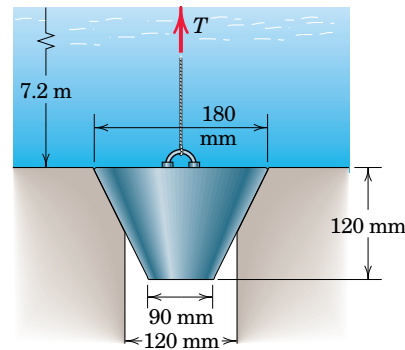
- 5/193** The barge crane of rectangular proportions has a 6-m by 15-m cross section over its entire length of 40 m. If the maximum permissible submergence and list in sea water are represented by the position shown, determine the corresponding maximum safe mass m_0 which the barge can handle at the 10-m extended position of the boom. Also find the total displacement m in metric tons of the unloaded barge. The distribution of machinery and ballast places the center of gravity G of the barge, minus the mass m_0 , at the center of the hull.

Ans. $m_0 = 203 \text{ Mg}$, $m = 1651 \text{ Mg}$



Problem 5/193

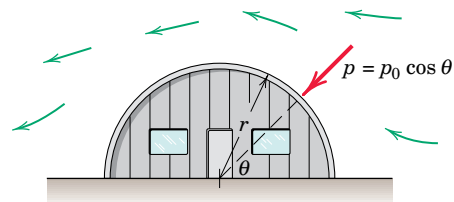
- 5/194** The cast-iron plug seals the drainpipe of an open fresh-water tank which is filled to a depth of 7.2 m. Determine the tension T required to remove the plug from its tapered hole. Atmospheric pressure exists in the drainpipe and in the seal area as the plug is being removed. Neglect mechanical friction between the plug and its supporting surface.



Problem 5/194

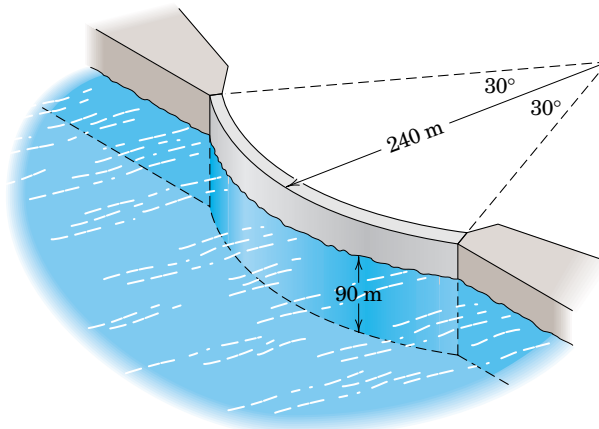
- 5/195** The Quonset hut is subjected to a horizontal wind, and the pressure p against the circular roof is approximated by $p_0 \cos \theta$. The pressure is positive on the windward side of the hut and is negative on the leeward side. Determine the total horizontal shear Q on the foundation per unit length of roof measured normal to the paper.

Ans. $Q = \frac{1}{2} \pi r p_0$



Problem 5/195

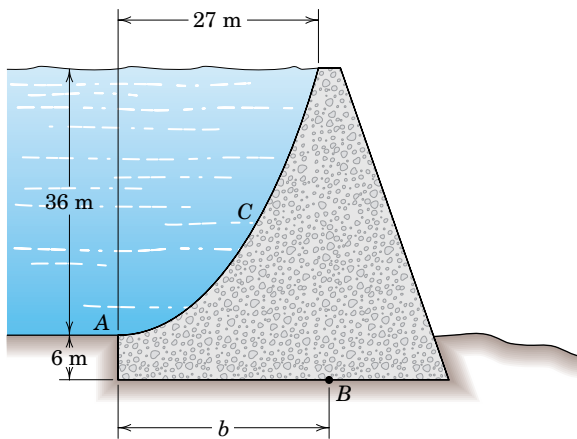
- 5/196** The upstream side of an arched dam has the form of a vertical cylindrical surface of 240-m radius and subtends an angle of 60° . If the fresh water is 90 m deep, determine the total force R exerted by the water on the dam face.



Problem 5/196

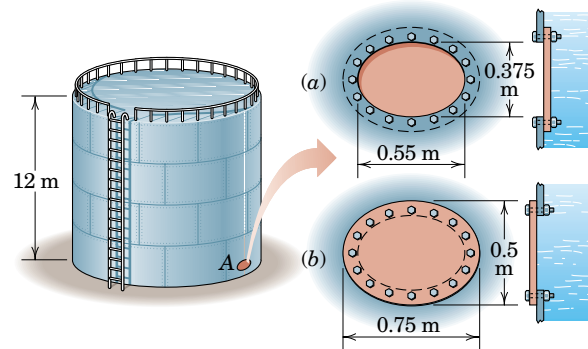
- 5/197** The fresh-water side of a concrete dam has the shape of a vertical parabola with vertex at A . Determine the position b of the base point B through which acts the resultant force of the water against the dam face C .

Ans. $b = 28.1$ m



Problem 5/197

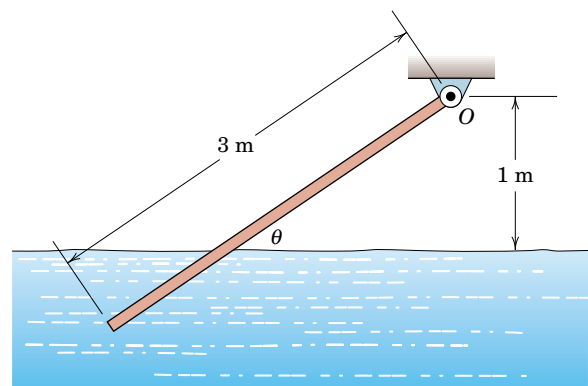
- 5/198** The small access hole A allows maintenance workers to enter the storage tank at ground level when it is empty. Two designs, (a) and (b), are shown for the hole cover. If the tank is full of fresh water, estimate the average pressure σ in the seal area of design (a) and the average increase ΔT in the initial tension in each of the 16 bolts of design (b). You may take the pressure over the hole area to be constant, and the pressure in the seal area of design (b) may be assumed to be atmospheric.



Problem 5/198

- 5/199** The 3-m plank shown in section has a density of 800 kg/m^3 and is hinged about a horizontal axis through its upper edge O . Calculate the angle θ assumed by the plank with the horizontal for the level of fresh water shown.

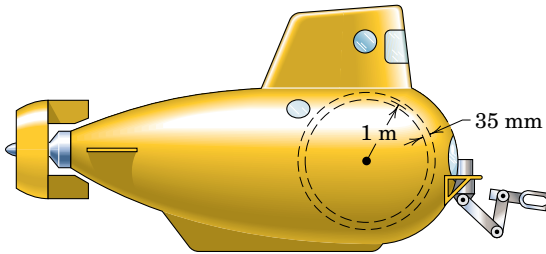
Ans. $\theta = 48.2^\circ$



Problem 5/199

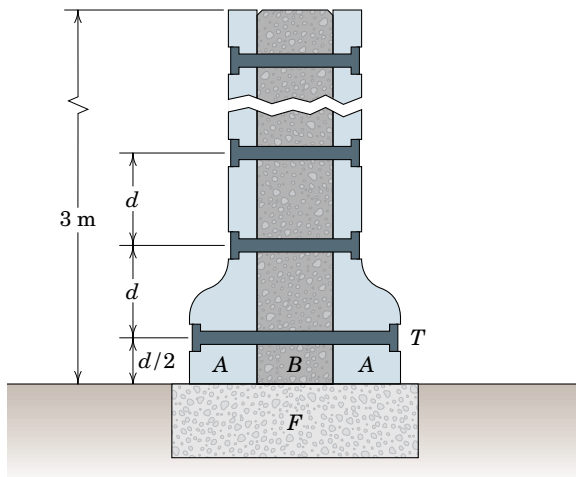
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- 5/200** The deep-submersible research vessel has a passenger compartment in the form of a spherical steel shell with a mean radius of 1.000 m and a thickness of 35 mm. Calculate the mass of lead ballast which the vessel must carry so that the combined weight of the steel shell and lead ballast exactly cancels the combined buoyancy of these two parts alone. (Consult Table D/1, Appendix D, as needed.)

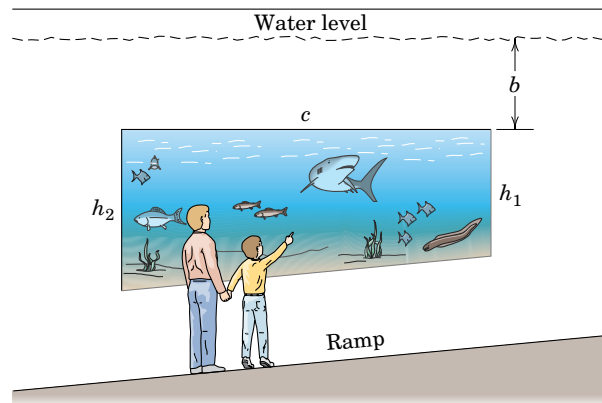

Problem 5/200

- 5/201** The elements of a new method for constructing concrete foundation walls for new houses are shown in the figure. Once the footing F is in place, polystyrene forms A are erected and a thin concrete mixture B is poured between the forms. Ties T prevent the forms from separating. After the concrete cures, the forms are left in place for insulation. As a design exercise, make a conservative estimate for the uniform tie spacing d if the tension in each tie is not to exceed 6.5 kN. The horizontal tie spacing is the same as the vertical spacing. State any assumptions. The density of wet concrete is 2400 kg/m^3 .

Ans. $d = 0.300 \text{ m}$

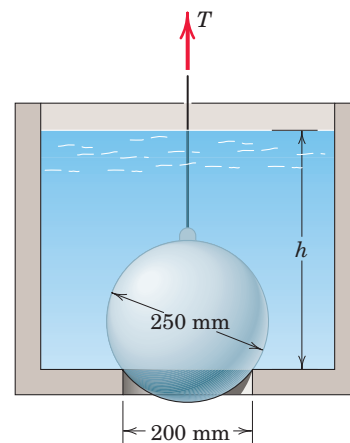

Problem 5/201

- 5/202** The trapezoidal viewing window in a sea-life aquarium has the dimensions shown. With the aid of appropriate diagrams and coordinates, describe two methods by which the resultant force R on the glass due to water pressure, and the vertical location of R , could be found if numerical values were supplied.


Problem 5/202

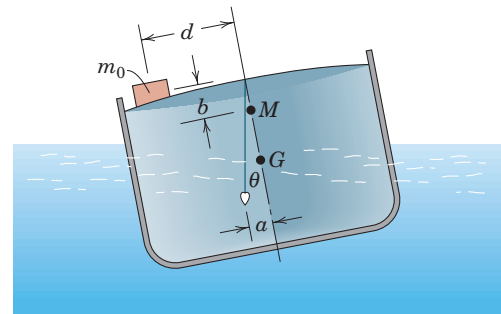
- 5/203** The sphere is used as a valve to close the hole in the fresh-water tank. As the depth h decreases, the tension T required to open the valve decreases because the downward force on the sphere decreases with less pressure. Determine the depth h for which T equals the weight of the sphere.

Ans. $h = 0.233 \text{ m}$


Problem 5/203

- **5/204** The accurate determination of the vertical position of the center of mass G of a ship is difficult to achieve by calculation. It is more easily obtained by a simple inclining experiment on the loaded ship. With reference to the figure, a known external mass m_0 is placed a distance d from the centerline, and the angle of list θ is measured by means of the deflection of a plumb bob. The displacement of the ship and the location of the metacenter M are known. Calculate the metacentric height \overline{GM} for a 12 000-t ship inclined by a 27-t mass placed 7.8 m from the centerline if a 6-m plumb line is deflected a distance $a = 0.2$ m. The mass m_0 is at a distance $b = 1.8$ m above M . [Note that the metric ton (t) equals 1000 kg and is the same as the megagram (Mg).]

Ans. $\overline{GM} = 0.530$ m



Problem 5/204

5/10 CHAPTER REVIEW

In Chapter 5 we have studied various common examples of forces distributed throughout volumes, over areas, and along lines. In all these problems we often need to determine the resultant of the distributed forces and the location of the resultant.

Finding Resultants of Distributed Forces

To find the resultant and line of action of a distributed force:

1. Begin by multiplying the intensity of the force by the appropriate element of volume, area, or length in terms of which the intensity is expressed. Then sum (integrate) the incremental forces over the region involved to obtain their resultant.
2. To locate the line of action of the resultant, use the principle of moments. Evaluate the sum of the moments, about a convenient axis, of all of the increments of force. Equate this sum to the moment of the resultant about the same axis. Then solve for the unknown moment arm of the resultant.

Gravitational Forces

When force is distributed throughout a mass, as in the case of gravitational attraction, the intensity is the force of attraction ρg per unit of volume, where ρ is the density and g is the gravitational acceleration. For bodies whose density is constant, we saw in Section A that ρg cancels when the moment principle is applied. This leaves us with a strictly geometric problem of finding the centroid of the figure, which coincides with the mass center of the physical body whose boundary defines the figure.

1. For flat plates and shells which are homogeneous and have constant thickness, the problem becomes one of finding the properties of an area.
2. For slender rods and wires of uniform density and constant cross section, the problem becomes one of finding the properties of a line segment.

Integration of Differential Relationships

For problems which require the integration of differential relationships, keep in mind the following considerations.

1. Select a coordinate system which provides the simplest description of the boundaries of the region of integration.
2. Eliminate higher-order differential quantities whenever lower-order differential quantities will remain.
3. Choose a first-order differential element in preference to a second-order element and a second-order element in preference to a third-order element.

4. Whenever possible, choose a differential element which avoids discontinuities within the region of integration.

Distributed Forces in Beams, Cables, and Fluids

In Section B we used these guidelines along with the principles of equilibrium to solve for the effects of distributed forces in beams, cables, and fluids. Remember that:

1. For beams and cables the force intensity is expressed as force per unit length.
2. For fluids the force intensity is expressed as force per unit area, or pressure.

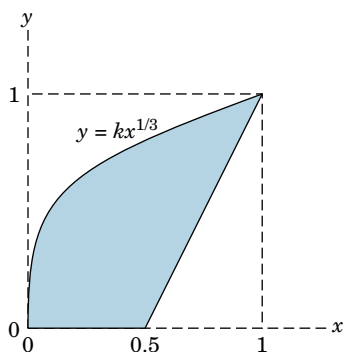
Although beams, cables, and fluids are physically quite different applications, their problem formulations share the common elements cited above.

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REVIEW PROBLEMS

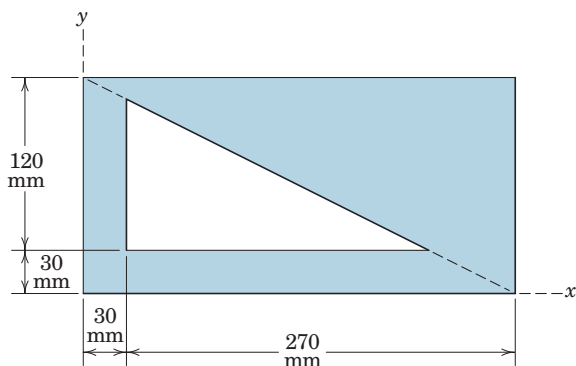
- 5/205** Determine the x - and y -coordinates of the centroid of the shaded area.

$$\text{Ans. } \bar{x} = \frac{37}{84}, \bar{y} = \frac{13}{30}$$



Problem 5/205

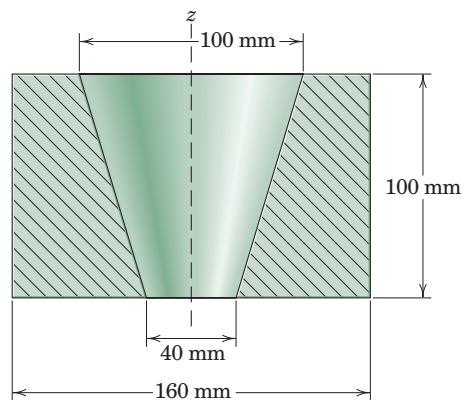
- 5/206** Determine the x - and y -coordinates of the centroid of the shaded area.



Problem 5/206

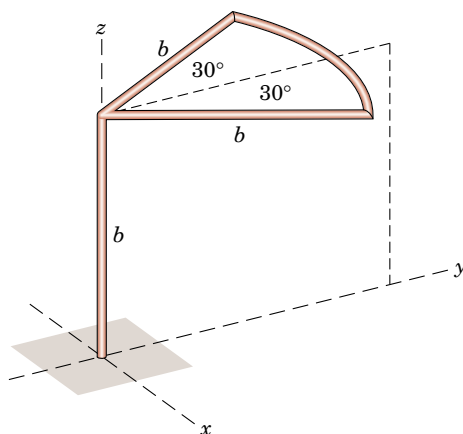
- 5/207** The cross section shown is for a complete cast-iron body of revolution about the z -axis. Compute its mass m .

$$\text{Ans. } m = 11.55 \text{ kg}$$



Problem 5/207

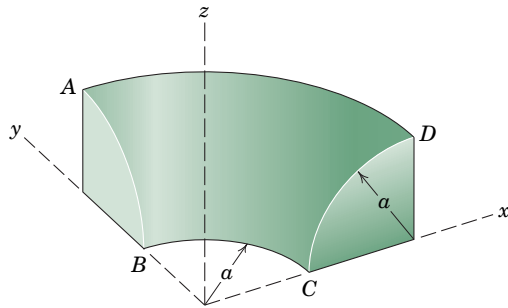
- 5/208** The assembly consists of four rods cut from the same bar stock. The curved member is a circular arc of radius b . Determine the y - and z -coordinates of the mass center of the assembly.



Problem 5/208

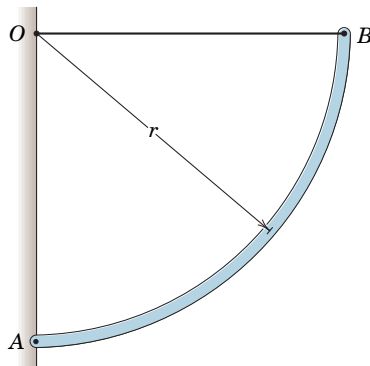
- 5/209** Determine the area A of the curved surface $ABCD$ of the solid of revolution shown.

$$\text{Ans. } A = \frac{\pi a^2}{2} (\pi - 1)$$



Problem 5/209

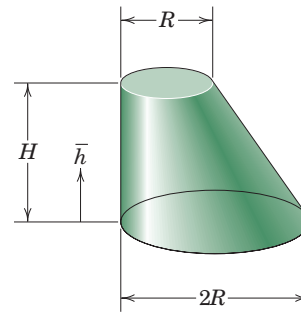
- 5/210** The uniform bar of mass m is bent into the quarter-circular arc in the vertical plane and is hinged freely at A . The bar is held in position by the horizontal wire from O to B . Determine the magnitude R of the force on the pin at A .



Problem 5/210

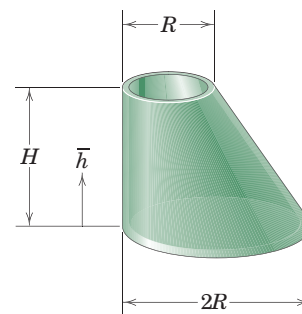
- 5/211** The tapered body has a horizontal cross section which is circular. Determine the height \bar{h} of its mass center above the base of the homogeneous body.

$$\text{Ans. } \bar{h} = \frac{11}{28}H$$



Problem 5/211

- 5/212** Shown in the figure is a body similar to that of Prob. 5/211, only now the body is a thin shell with open top and bottom. Determine the height \bar{h} of its mass center above the base of the homogeneous body and compare your result with the printed answer for Prob. 5/211.

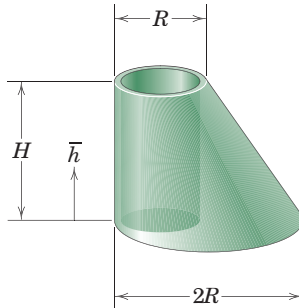


Problem 5/212

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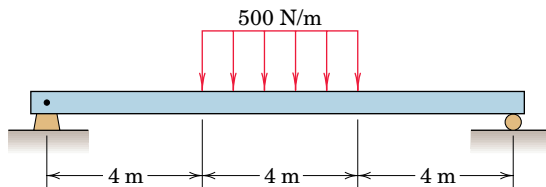
- 5/213** The homogeneous body shown is constructed by beginning with the solid body of Prob. 5/211 and then removing a cylindrical volume of diameter just less than R as shown, leaving a thin wall at the left portion of the body as well as at the very top. Determine the height \bar{h} of the mass center above the base of the body.

Ans. $\bar{h} = \frac{5}{16}H$



Problem 5/213

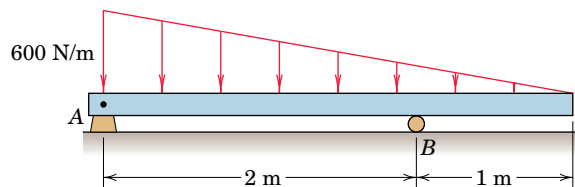
- 5/214** Draw the shear and moment diagrams for the beam, which supports the uniform load of 500 N/m per meter of beam length distributed over its mid-section. Determine the maximum bending moment and its location.



Problem 5/214

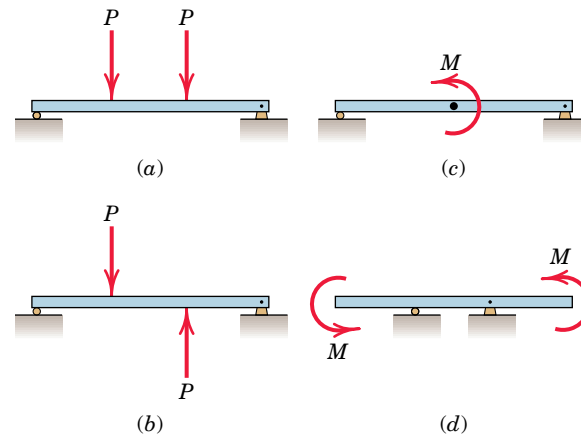
- 5/215** Determine the maximum bending moment M_{\max} for the loaded beam and specify the distance x to the right of end A where M_{\max} exists.

Ans. $M_{\max} = 186.4 \text{ N} \cdot \text{m}$ at $x = 0.879 \text{ m}$



Problem 5/215

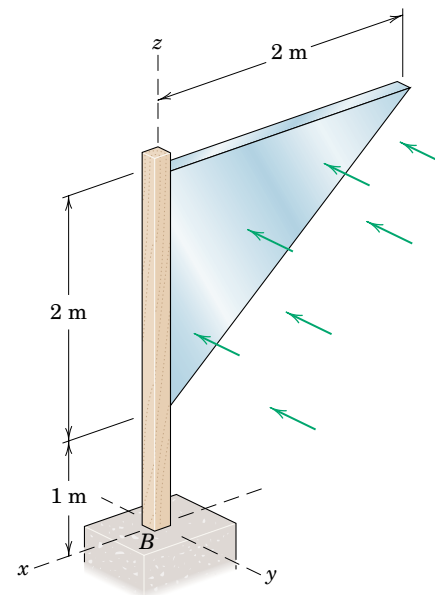
- 5/216** Sketch the shear and moment diagrams for each of the four beams loaded and supported as shown.



Problem 5/216

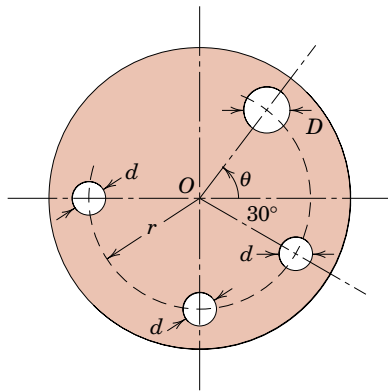
- 5/217** The triangular sign is attached to the post embedded in the concrete base at B. Calculate the shear force V , the bending moment M , and the torsional moment T in the post at B during a storm where the wind velocity normal to the sign reaches 100 km/h. The air pressure (called stagnation pressure) against the vertical surface corresponding to this velocity is 1.4 kPa.

Ans. $V = 2.8 \text{ kN}$, $M = 6.53 \text{ kN} \cdot \text{m}$, $T = 1.867 \text{ kN} \cdot \text{m}$



Problem 5/217

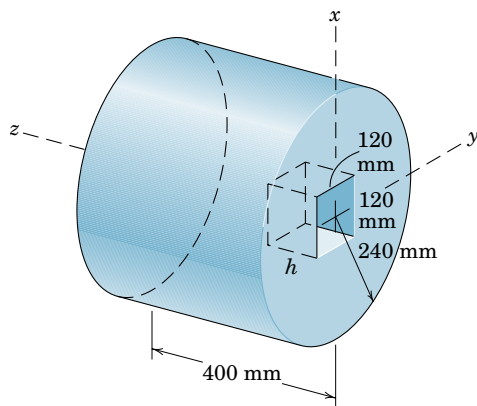
- 5/218** The circular disk rotates about an axis through its center O and has three holes of diameter d positioned as shown. A fourth hole is to be drilled in the disk at the same radius r so that the disk will be in balance (mass center at O). Determine the required diameter D of the new hole and its angular position.



Problem 5/218

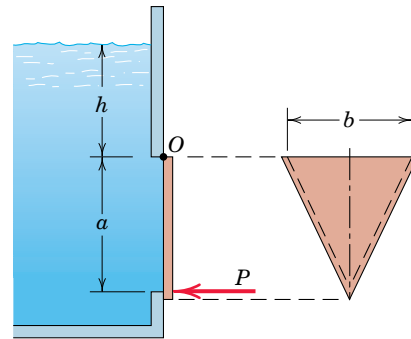
- 5/219** Determine the depth h of the square hole in the solid circular cylinder for which the z -coordinate of the mass center will have the maximum possible value.

Ans. $h = 204 \text{ mm}$



Problem 5/219

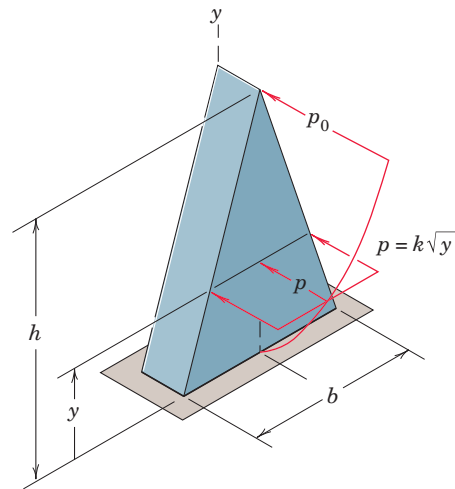
- 5/220** A flat plate seals a triangular opening in the vertical wall of a tank of liquid of density ρ . The plate is hinged about the upper edge O of the triangle. Determine the force P required to hold the gate in a closed position against the pressure of the liquid.



Problem 5/220

- 5/221** A prismatic structure of height h and base b is subjected to a horizontal wind load whose pressure p increases from zero at the base to p_0 at the top according to $p = k\sqrt{y}$. Determine the resisting moment M at the base of the structure.

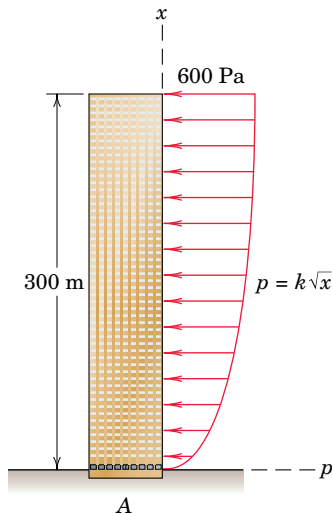
Ans. $M = \frac{4}{35} p_0 b h^2$



Problem 5/221

334 Chapter 5 Distributed Forces

5/222 As part of a preliminary design study, the effects of wind loads on a 300-m building are investigated. For the parabolic distribution of wind pressure shown in the figure, compute the force and moment reactions at the base A of the building due to the wind load. The depth of the building (perpendicular to the paper) is 60 m.



Problem 5/222

► 5/223 Regard the tall building of Prob. 5/222 as a uniform upright beam. Determine and plot the shear force and bending moment in the structure as functions of the height x above the ground. Evaluate your expressions at $x = 150$ m.

$$\text{Ans. } V = 7.2(10^6) - 1386 x^{3/2} \text{ N}$$

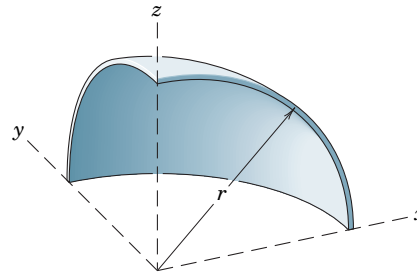
$$M = 1296 - 7.2x + 5.54(10^{-4}) x^{5/2} \text{ MN}\cdot\text{m}$$

$$V|_{x=150 \text{ m}} = 4.65 \text{ MN}$$

$$M|_{x=150 \text{ m}} = 369 \text{ MN}\cdot\text{m}$$

► 5/224 Locate the center of mass of the thin spherical shell which is formed by cutting out one-eighth of a complete shell.

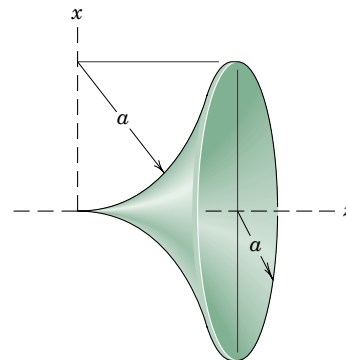
$$\text{Ans. } \bar{x} = \bar{y} = \bar{z} = \frac{r}{2}$$



Problem 5/224

► 5/225 Locate the center of mass of the homogeneous bell-shaped shell of uniform but negligible thickness.

$$\text{Ans. } \bar{z} = \frac{a}{\pi - 2}$$

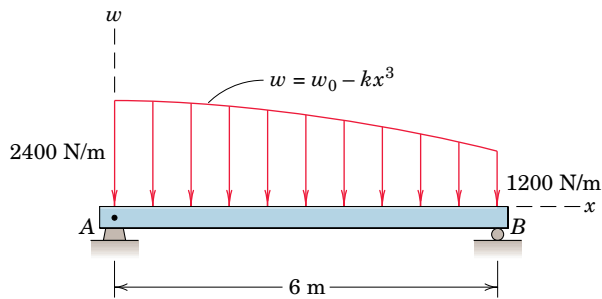


Problem 5/225



Computer-Oriented Problems*

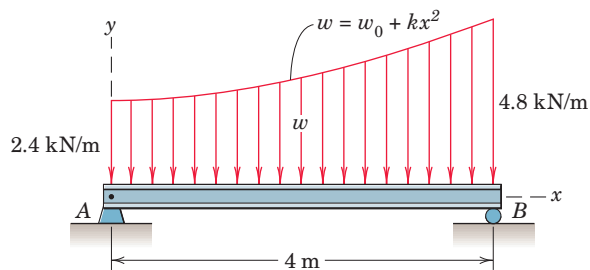
- *5/226** Construct the shear and moment diagrams for the loaded beam of Prob. 5/113, repeated here. Determine the maximum values of the shear and moment and their locations on the beam.



Problem 5/226

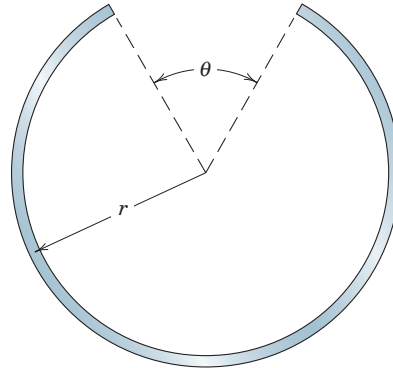
- *5/227** Construct the shear and moment diagrams for the loaded beam shown. Determine the maximum bending moment and its location.

Ans. $M_{\max} = 6.23 \text{ kN}\cdot\text{m}$ at $x = 2.13 \text{ m}$



Problem 5/227

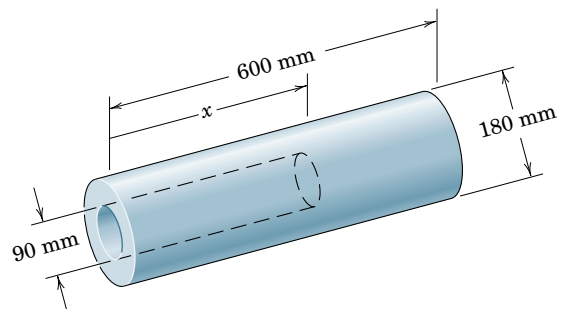
- *5/228** Find the angle θ which will place the mass center of the thin ring a distance $r/10$ from the center of the arc.



Problem 5/228

- *5/229** A homogeneous charge of solid propellant for a rocket is in the shape of the circular cylinder formed with a concentric hole of depth x . For the dimensions shown, plot \bar{X} , the x -coordinate of the mass center of the propellant, as a function of the depth x of the hole from $x = 0$ to $x = 600 \text{ mm}$. Determine the maximum value of \bar{X} and show that it is equal to the corresponding value of x .

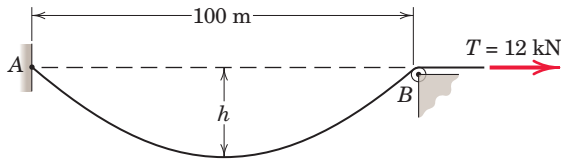
Ans. $\bar{X}_{\max} = 322 \text{ mm}$



Problem 5/229

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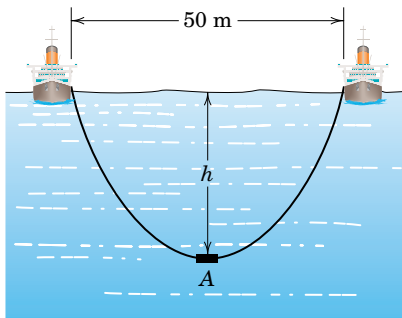
- *5/230** A cable with a mass of 15 kg/m is attached to point A and passes over the small pulley at B on the same horizontal line with A . Determine the sag h and length S of the cable between A and B if a tension of 12 kN is applied to the cable over the pulley.



Problem 5/230

- *5/231** An underwater detection instrument A is attached to the midpoint of a 100-m cable suspended between two ships 50 m apart. Determine the depth h of the instrument, which has negligible mass. Does the result depend on the mass of the cable or on the density of the water?

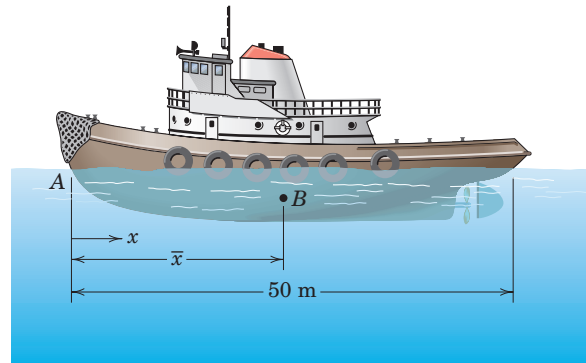
Ans. $h = 39.8$ m



Problem 5/231

- *5/232** The center of buoyancy B of a ship's hull is the centroid of its submerged volume. The underwater cross-sectional areas A of the transverse sections of the tugboat hull shown are tabulated for every five meters of waterline length. With an appropriate computer program, determine to the nearest 0.5 m the distance \bar{x} of B aft of point A .

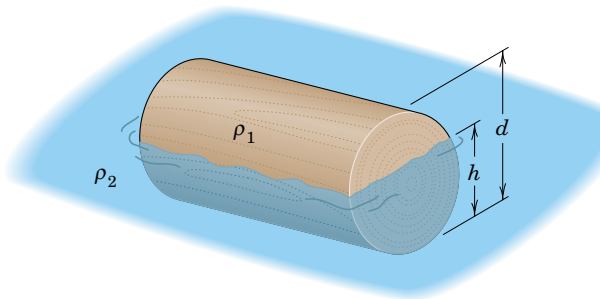
x , m	A , m ²	x , m	A , m ²
0	0	30	23.8
5	7.1	35	19.5
10	15.8	40	12.5
15	22.1	45	5.1
20	24.7	50	0
25	25.1		



Problem 5/232

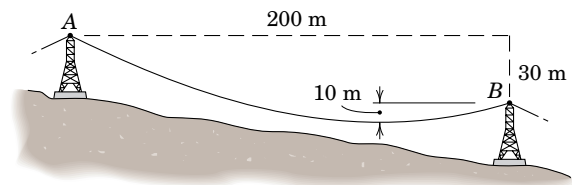
- *5/233** A right-circular cylinder of density ρ_1 floats in a liquid of density ρ_2 . If d is the diameter of the cylinder and h is the submerged depth, plot the ratio $r = \frac{h}{d}$ as a function of $\frac{\rho_1}{\rho_2}$ over the range $0 \leq \frac{\rho_1}{\rho_2} \leq 1$. Evaluate r for a pine cylinder floating in sea water.

Ans. $r = 0.473$



Problem 5/233

- *5/234** A length of power cable is suspended from the two towers as shown. The cable has a mass of 20 kg per meter of cable. If the maximum allowable cable tension is 75 kN, determine the mass ρ of ice per meter which can form on the cable without the maximum allowable tension being exceeded. If additional stretch in the cable is neglected, does the addition of the ice change the cable configuration?



Problem 5/234

ch06 Friction, ch07 Virtual Work, and appendices
are missing. See the accompanying replacement
doc drawn from the 5th edition.

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Conversion Factors
U.S. Customary Units to SI Units

To convert from	To	Multiply by
<i>(Acceleration)</i>		
foot/second ² (ft/sec ²)	meter/second ² (m/s ²)	$3.048 \times 10^{-1*}$
inch/second ² (in./sec ²)	meter/second ² (m/s ²)	$2.54 \times 10^{-2*}$
<i>(Area)</i>		
foot ² (ft ²)	meter ² (m ²)	9.2903×10^{-2}
inch ² (in. ²)	meter ² (m ²)	$6.4516 \times 10^{-4*}$
<i>(Density)</i>		
pound mass/inch ³ (lbm/in. ³)	kilogram/meter ³ (kg/m ³)	2.7680×10^4
pound mass/foot ³ (lbm/ft ³)	kilogram/meter ³ (kg/m ³)	1.6018×10
<i>(Force)</i>		
kip (1000 lb)	newton (N)	4.4482×10^3
pound force (lb)	newton (N)	4.4482
<i>(Length)</i>		
foot (ft)	meter (m)	$3.048 \times 10^{-1*}$
inch (in.)	meter (m)	$2.54 \times 10^{-2*}$
mile (mi), (U.S. statute)	meter (m)	1.6093×10^3
mile (mi), (international nautical)	meter (m)	$1.852 \times 10^3*$
<i>(Mass)</i>		
pound mass (lbm)	kilogram (kg)	4.5359×10^{-1}
slug (lb-sec ² /ft)	kilogram (kg)	1.4594×10
ton (2000 lbm)	kilogram (kg)	9.0718×10^2
<i>(Moment of force)</i>		
pound-foot (lb-ft)	newton-meter (N · m)	1.3558
pound-inch (lb-in.)	newton-meter (N · m)	0.1129 8
<i>(Moment of inertia, area)</i>		
inch ⁴	meter ⁴ (m ⁴)	41.623×10^{-8}
<i>(Moment of inertia, mass)</i>		
pound-foot-second ² (lb-ft-sec ²)	kilogram-meter ² (kg · m ²)	1.3558
<i>(Momentum, linear)</i>		
pound-second (lb-sec)	kilogram-meter/second (kg · m/s)	4.4482
<i>(Momentum, angular)</i>		
pound-foot-second (lb-ft-sec)	newton-meter-second (kg · m ² /s)	1.3558
<i>(Power)</i>		
foot-pound/minute (ft-lb/min)	watt (W)	2.2597×10^{-2}
horsepower (550 ft-lb/sec)	watt (W)	7.4570×10^2
<i>(Pressure, stress)</i>		
atmosphere (std)(14.7 lb/in. ²)	newton/meter ² (N/m ² or Pa)	1.0133×10^5
pound/foot ² (lb/ft ²)	newton/meter ² (N/m ² or Pa)	4.7880×10
pound/inch ² (lb/in. ² or psi)	newton/meter ² (N/m ² or Pa)	6.8948×10^3
<i>(Spring constant)</i>		
pound/inch (lb/in.)	newton/meter (N/m)	1.7513×10^2
<i>(Velocity)</i>		
foot/second (ft/sec)	meter/second (m/s)	$3.048 \times 10^{-1*}$
knot (nautical mi/hr)	meter/second (m/s)	5.1444×10^{-1}
mile/hour (mi/hr)	meter/second (m/s)	$4.4704 \times 10^{-1*}$
mile/hour (mi/hr)	kilometer/hour (km/h)	1.6093
<i>(Volume)</i>		
foot ³ (ft ³)	meter ³ (m ³)	2.8317×10^{-2}
inch ³ (in. ³)	meter ³ (m ³)	1.6387×10^{-5}
<i>(Work, Energy)</i>		
British thermal unit (BTU)	joule (J)	1.0551×10^3
foot-pound force (ft-lb)	joule (J)	1.3558
kilowatt-hour (kw-h)	joule (J)	$3.60 \times 10^6*$
*Exact value		

SI Units Used in Mechanics

Quantity	Unit	SI Symbol
<i>(Base Units)</i>		
Length	meter*	m
Mass	kilogram	kg
Time	second	s
<i>(Derived Units)</i>		
Acceleration, linear	meter/second ²	m/s ²
Acceleration, angular	radian/second ²	rad/s ²
Area	meter ²	m ²
Density	kilogram/meter ³	kg/m ³
Force	newton	N (= kg · m/s ²)
Frequency	hertz	Hz (= 1/s)
Impulse, linear	newton-second	N · s
Impulse, angular	newton-meter-second	N · m · s
Moment of force	newton-meter	N · m
Moment of inertia, area	meter ⁴	m ⁴
Moment of inertia, mass	kilogram-meter ²	kg · m ²
Momentum, linear	kilogram-meter/second	kg · m/s (= N · s)
Momentum, angular	kilogram-meter ² /second	kg · m ² /s (= N · m · s)
Power	watt	W (= J/s = N · m/s)
Pressure, stress	pascal	Pa (= N/m ²)
Product of inertia, area	meter ⁴	m ⁴
Product of inertia, mass	kilogram-meter ²	kg · m ²
Spring constant	newton/meter	N/m
Velocity, linear	meter/second	m/s
Velocity, angular	radian/second	rad/s
Volume	meter ³	m ³
Work, energy	joule	J (= N · m)
<i>(Supplementary and Other Acceptable Units)</i>		
Distance (navigation)	nautical mile	(= 1,852 km)
Mass	ton (metric)	t (= 1000 kg)
Plane angle	degrees (decimal)	°
Plane angle	radian	—
Speed	knot	(1.852 km/h)
Time	day	d
Time	hour	h
Time	minute	min

*Also spelled *metre*.

SI Unit Prefixes

Multiplication Factor	Prefix	Symbol
1 000 000 000 000 = 10 ¹²	tera	T
1 000 000 000 = 10 ⁹	giga	G
1 000 000 = 10 ⁶	mega	M
1 000 = 10 ³	kilo	k
100 = 10 ²	hecto	h
10 = 10 ¹	deka	da
0.1 = 10 ⁻¹	deci	d
0.01 = 10 ⁻²	centi	c
0.001 = 10 ⁻³	milli	m
0.000 001 = 10 ⁻⁶	micro	μ
0.000 000 001 = 10 ⁻⁹	nano	n
0.000 000 000 001 = 10 ⁻¹²	pico	p

Selected Rules for Writing Metric Quantities

- Use prefixes to keep numerical values generally between 0.1 and 1000.
 - Use of the prefixes hecto, deka, deci, and centi should generally be avoided except for certain areas or volumes where the numbers would be awkward otherwise.
 - Use prefixes only in the numerator of unit combinations. The one exception is the base unit kilogram. (*Example:* write kN/m not N/mm; J/kg not mJ/g)
 - Avoid double prefixes. (*Example:* write GN not kMN)
- Unit designations
 - Use a dot for multiplication of units. (*Example:* write N · m not Nm)
 - Avoid ambiguous double solidus. (*Example:* write N/m² not N/m/m)
 - Exponents refer to entire unit. (*Example:* mm² means (mm)²)
- Number grouping

Use a space rather than a comma to separate numbers in groups of three, counting from the decimal point in both directions. *Example:* 4 607 321.048 72)

Space may be omitted for numbers of four digits. (*Example:* 4296 or 0.0476)

ENGINEERING MECHANICS

VOLUME 1

STATICS

SIXTH EDITION

SI VERSION

J. L. MERIAM

L. G. KRAIGE

*Virginia Polytechnic Institute
and State University*



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On the Cover: The Gateway Arch in St. Louis was first conceived by architect Eero Saarinen in the late 1940s. Later, a team of engineers led by John Dinkeloo devised further design details, and construction was completed in 1965. The shape of the 630-ft-high arch is that of a *weighted catenary*. This configuration could be formed by suspending a nonuniform flexible cable (symmetrically heavier near its ends) from two points on a horizontal line, “freezing” that shape, and inverting.

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FOREWORD

This series of textbooks was begun in 1951 by the late Dr. James L. Meriam. At that time, the books represented a revolutionary transformation in undergraduate mechanics education. They became the definitive textbooks for the decades that followed as well as models for other engineering mechanics texts that have subsequently appeared. Published under slightly different titles prior to the 1978 First Editions, this textbook series has always been characterized by logical organization, clear and rigorous presentation of the theory, instructive sample problems, and a rich collection of real-life problems, all with a high standard of illustration. In addition to the U.S. versions, the books have appeared in SI versions and have been translated into many foreign languages. These texts collectively represent an international standard for undergraduate texts in mechanics.

The innovations and contributions of Dr. Meriam (1917–2000) to the field of engineering mechanics cannot be overstated. He was one of the premier engineering educators of the second half of the twentieth century. Dr. Meriam earned his B.E., M. Eng., and Ph.D. degrees from Yale University. He had early industrial experience with Pratt and Whitney Aircraft and the General Electric Company. During the Second World War he served in the U.S. Coast Guard. He was a member of the faculty of the University of California–Berkeley, Dean of Engineering at Duke University, a faculty member at the California Polytechnic State University–San Luis Obispo, and visiting professor at the University of California–Santa Barbara, finally retiring in 1990. Professor Meriam always placed great emphasis on teaching, and this trait was recognized by his students wherever he taught. At Berkeley in 1963, he was the first recipient of the Outstanding Faculty Award of Tau Beta Pi, given primarily for excellence in teaching. In 1978, he received the Distinguished Educator Award for Outstanding Service to Engineering Mechanics Education from the American Society for Engineering Education, and in 1992 was the Society’s recipient of the Benjamin Garver Lamme Award, which is ASEE’s highest annual national award.

Dr. L. Glenn Kraige, coauthor of the *Engineering Mechanics* series since the early 1980s, has also made significant contributions to mechanics education. Dr. Kraige earned his B.S., M.S., and Ph.D. degrees at the University of Virginia, principally in aerospace engi-

v

vi Foreword

neering, and he currently serves as Professor of Engineering Science and Mechanics at Virginia Polytechnic Institute and State University. During the mid 1970s, I had the singular pleasure of chairing Professor Kraige's graduate committee and take particular pride in the fact that he was the first of my three dozen Ph.D. graduates. Professor Kraige was invited by Professor Meriam to team with him and thereby ensure that the Meriam legacy of textbook authorship excellence was carried forward to future generations. For the past two and a half decades, this highly successful team of authors has made an enormous and global impact on the education of several generations of engineers.

In addition to his widely recognized research and publications in the field of spacecraft dynamics, Professor Kraige has devoted his attention to the teaching of mechanics at both introductory and advanced levels. His outstanding teaching has been widely recognized and has earned him teaching awards at the departmental, college, university, state, regional, and national levels. These include the Francis J. Maher Award for excellence in education in the Department of Engineering Science and Mechanics, the Wine Award for excellence in university teaching, and the Outstanding Educator Award from the State Council of Higher Education for the Commonwealth of Virginia. In 1996, the Mechanics Division of ASEE bestowed upon him the Archie Higdon Distinguished Educator Award. The Carnegie Foundation for the Advancement of Teaching and the Council for Advancement and Support of Education awarded him the distinction of Virginia Professor of the Year for 1997. In his teaching, Professor Kraige stresses the development of analytical capabilities along with the strengthening of physical insight and engineering judgment. Since the early 1980s, he has worked on personal-computer software designed to enhance the teaching/learning process in statics, dynamics, strength of materials, and higher-level areas of dynamics and vibrations.

The Sixth Edition of *Engineering Mechanics* continues the same high standards set by previous editions and adds new features of help and interest to students. It contains a vast collection of interesting and instructive problems. The faculty and students privileged to teach or study from Professors Meriam and Kraige's *Engineering Mechanics* will benefit from the several decades of investment by two highly accomplished educators. Following the pattern of the previous editions, this textbook stresses the application of theory to actual engineering situations, and at this important task it remains the best.



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