ENGINEERING MECHANICS

工程力学专业英语选编

PROFESSIONAL ENGLISH SELECTIONS

ENGLISH

江理平 编

同济大学出版社

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前　言

本书为工程力学专业英语的教材。为了让学生在短时期内迅速扩大专业词汇与提高阅读翻译能力，本书按照力学课程的先后、由浅入深、适当反复的原则，广泛地摘选力学专业各主要学科的原版教材，其中包括材料力学、弹性力学、塑性力学、结构力学、振动力学、流体力学、结构风振力学、海洋结构动力学、光测力学等内容。另外还附上板壳力学、钢结构、钢筋混凝土结构、土力学等方面的课外阅读材料，以供教学及自学的需要。每课课文后均附有词汇表，并对一些难点作了注解；在书后还附上力学专业常用词汇表。为了使本书更具系统性，编者对原文作了部分必要的删改。多年的教学实践证明，本书对力学专业的英语词汇的典型性较强，覆盖面较宽，经过一学期的学习，学生均能掌握1000个左右的专业词汇。同时对力学专业，包括对工业与民用建筑专业及数学有关专业的英语资料的阅读能力和翻译水平提高有显著的帮助，因而本书也可以作为力学工作者以及相关的工程技术人员学习专业英语的参考材料。

本书在编选过程中，得到毕家驹教授的悉心指导并提供大量的资料，还得到了吴家龙教授、夏志皋教授、唐寿高教授等其他各位教师的鼓励并提供了宝贵的意见和资料。本书的出版得到了上海铁道大学戚震华教授的认真审阅和指教，得到了张若京教授的大力支持，在此一并表示衷心的感谢！编者还要感谢同济大学教材基金会和同济大学教务处、同济大学出版社的支持以及本书责任编辑方芳老师的帮助！

由于编者水平的限制，缺点错误在所难免，恳望有关专家与读者批评指正。

同济大学　江理平
1997年3月于上海

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Lesson 1

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1. Introduction

In all engineering construction the component parts of a structure must be assigned definite physical sizes. Such parts must be properly proportioned to resist the actual or probable forces that may be imposed upon them. Thus, the walls of a pressure vessel must be of adequate strength to withstand the internal pressure: the floors of a building must be sufficiently strong for their intended purpose: the shaft of a machine must be of adequate size to carry the required torque: a wing of an airplane must safely withstand the aerodynamic loads which may come upon it in flight or landing. Likewise, the parts of a composite structure must be rigid enough so as not to deflect or "sag" excessively when in operation under the imposed loads. A floor of a building may be strong enough but yet may deflect excessively, which in some instances may cause misalignment of manufacturing equipment, or in other cases result in the cracking of a plaster ceiling attached underneath. Also a member may be so thin or slender that, upon being subjected to compressive loading, it will collapse through buckling; i.e., the initial configuration of a member may become unstable. Ability to determine the maximum load that a slender column can carry before buckling occurs, or determination of the safe level of vacuum that can be maintained by a vessel is of great practical importance.

In engineering practice, such requirements must be met with minimum expenditure of a given material. Aside from cost, at times—as in the design of satellites—the feasibility and success of the whole mission may depend on the weight of a package. The subject of *mechanics of materials, or the strength of materials*① . as it has been traditionally called in the past. involves analytical methods for determining the **Strength, stiffness** (deformation characteristics), and **stability** of the various load-carrying members. Alternately, the subject may be termed the *mechanics of solid deformable bodies*.

Mechanics of materials is a fairly old subject, generally dated from the work of Galileo in the early part of the seventeenth century. Prior to his investigations into the behavior of solid bodies under loads, constructors followed precedents and empirical rules. Galileo was the first to attempt to explain the behavior of some of the members under load on a rational basis. He studied members in tension and compression, and notably beams used in the construction of hulls of ships for the Italian navy. Of course much progress has been made since that time, but it must be noted in passing that much is owed in the development of this subject to the French investigators, among whom a group of outstanding men such as Coulomb, Poisson, Navier, St. Venant, and Cauchy, who worked at the break of the nineteenth century, has left an indelible impression on this subject.

The subject of mechanics of materials cuts② broadly across all branches of the engineering profession with remarkably many applications. Its methods are needed by designers of offshore structures; by civil engineers in the design of bridges and buildings; by mining engineers and architectural engineers, each of whom is interested in structures; by nuclear engineers in the design of reactor components; by mechanical and chemical engineers, who rely upon the methods of this subject for the design of machinery and pressure vessels, by metallurgists, who need the fundamental concepts of this subject in order to understand how to improve existing materials further; finally, by electrical engineers, who need the methods of this subject because of the importance of the mechanical engineering phases of many portions of electrica equipment. Mechanics of materials has characteristic methods all its own. It is a definite discipline and one of the most fundamental subjects of an engineering curriculum, standing alongside such other basic subjects as fluid mechanics, thermodynamics, and basic electricity.

The behavior of a member subjected to forces depends not only on the fundamental laws of Newtonian mechanics that govern the equilibrium of the forces but also on the *physical characteristics* of the materials of which the member is fabricated. The necessary information regarding the latter comes from the laboratory where materials are subjected to the action of accurately known forces and the behavior of test specimens is observed with particular regard to such phenomena as the occurrence of breaks, deformations, etc. Determination of such phenomena is a vital part of the subject, but this branch of the subject is left to other books. Here the end results of such investigations are of interest, and this course is concerned with the analytical or mathematical part of the subject in contradistinction to experimentation. For the above reasons, it is seen that mechanics of materials is a blended science of experiment and Newtonian postulates of analytical mechanics. From the latter is borrowed the branch of the science called *statics*, a subject with which the reader of this book is presumed to be familiar, and on which the subject of this book primarily depends.

This text will be limited to the simpler topics of the subject. In spite of the relative simplicity of the methods employed here, however, the resulting techniques are unusually useful as they do apply to a vast number of technically important problems.

The subject matter can be mastered best by solving numerous problems. The number of formulas necessary for the analysis and design of structural and machine members by the methods of mechanics of materials is remarkably small; however, throughout this study the student must develop an ability to *visualize* a problem and the nature of the quantities being computed. *Complete, carefully drawn diagrammatic sketches of problems to be solved will pay large dividends*③ *in a quicker and more complete mastery of this subject.*

2. Method of Sections

One of the main problems of mechanics of materials is the investigation of the internal resistance of a body, that is, *the nature of forces set up within a body to balance the effect of the externally applied forces*. For this purpose, a uniform method of approach is employed. A complete diagrammatic sketch of the member to be investigated is prepared, on which *all* of the external forces acting on a body are shown at their respective points of application. Such a sketch is called a *free-body* diagram④. All forces acting on a body, including the reactive forces caused by the supports and the weight of the body itself, are considered external forces. Moreover, since a stable body at rest is in equilibrium, the forces acting on it satisfy the equations of static equilibrium. Thus, if the forces acting on a body such as shown in Fig. 1.1(a) satisfy the equations of static equilibrium and are all shown acting on it, the sketch represents a freebody diagram. Next, since a determination of the internal forces caused by the external ones is one of the principal concerns of this subject, an arbitrary section is passed through the body, completely separating it into two parts. The result of such a process can be seen in Figs. 1.1(b) and (c) where an arbitrary plane *ABCD* separates the original solid body of Fig. 1.1(a) into two *distinct* parts. This process will be referred to as the *method of sections*. Then, if the body as a whole is in equilibrium, *any part* of it must also be in equilibrium. For such parts of a body, however, some of the forces necessary to maintain equilibrium must act at the cut section. These considerations lead to the following fundamental conclusion: *the externally applied forces to one side of an arbitrary cut must be balanced by the internal forces developed at the cut*, or briefly, the external forces are balanced by the internal forces. Later it will be seen that the cutting planes will be oriented in a particular direction to fit special requirements. However, the above concept will be relied upon as a first step in solving *all* problems where internal forces are being investigated.

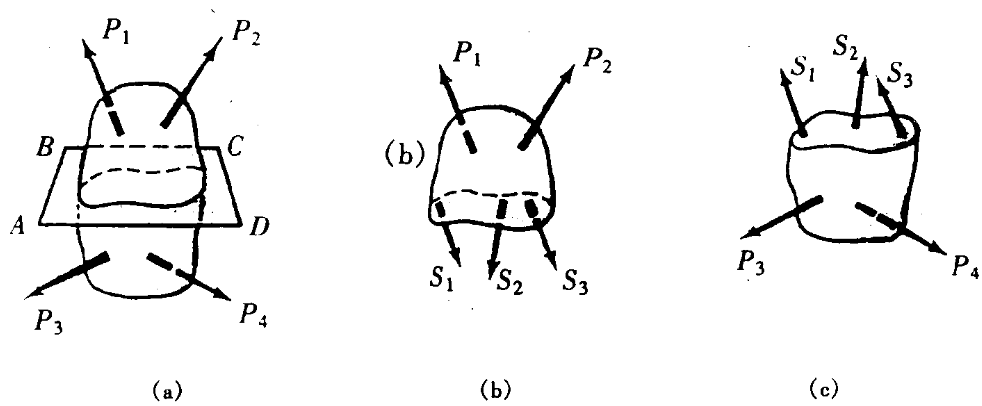

Fig. 1.1 Sectioning of a body

In discussing the method of sections, it is significant to note that some bodies, although not in static equilibrium, may be in dynamic equilibrium. These problems can be reduced to problems of static equilibrium. First, the acceleration of the part in question is computed, then it is multiplied by the mass of the body, giving a force $F = ma$. If the force so computed is applied to the body at its mass center in a direction opposite to the acceleration, the dynamic problem is reduced to one of statics. This is the so-called *d'Alembert principle*[5]. With this point of view, all bodies can be thought of as being instantaneously in a state of static equilibrium. Hence for any body, whether in static or dynamic equilibrium, a free-body diagram can be prepared on which the necessary forces to maintain the body as a whole in equilibrium can be shown. From then on the problem is the same as discussed above.

3. Stress

In general, the internal forces acting on infinitesimal areas of a cut may be of varying magnitudes and directions, as is shown diagrammatically in Figs. 1.1(b) and (c). These internal forces are vectorial in nature and maintain in equilibrium the externally applied forces. In mechanics of materials it is particularly significant to determine the *intensity* of these forces on the various portions of the cut, as resistance to deformation and the capacity of materials to resist forces depend on these intensities. In general, these intensities of force acting on infinitesimal areas of the cut vary from point to point, and, in general, they are inclined with respect to the plane of the cut. In engineering practice it is customary to resolve this intensity of force prependicular and parallel to the section investigated. Such resolution of the intensity of a force on an infinitesimal area is shown in Fig. 1.2. The intensity of the force *perpendicular or normal to the section* is called the *normal stress* at a point. In this book it will be designated by the Greek letter σ (sigma). As a particular stress generally holds true only at a point, it is defined mathematically as

$$\sigma = \lim_{\Delta A \to 0} \frac{\Delta F}{\Delta A}$$

where F is a force acting *normal* to the cut, while A is the corresponding area. It is customary to refer to the normal stresses that cause traction or tension on the surface of a cut as *tensile* stresses. On the other hand, those that are pushing against the cut are *compressive* stresses.

The other component of the intensity of force acts *pa rallel to the plane of the elementary area*, as in Fig. 1.2. This component of the intensity of force is called the *shearing stress*. It will be designated by the Greek letter τ(tau). Mathematically it is defined as

$$\tau = \lim_{\Delta A \to 0} \frac{\Delta V}{\Delta A}$$

where A represents area, and V is the component of the force parallel to the cut. It should be noted that these definitions of stresses at a point involve the concept of letting $\Delta A \to 0$ and may be questionable from a strictly atomic view of matter. However, the homogeneous model implied by these equations has been a good approximation to inhomogenous matter on the macroscopic level. Therefore, this so-called phenomenological approach is used.

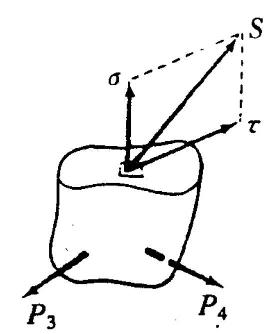

Fig. 1.2 The normal and shearing components of stress

The student should form a clear mental picture of the stresses called normal and those called shearing. To repeat, normal stresses result from force components perpendicular to the plane of the cut, while shearing stresses result from components parallel to the plane of the cut.

It is seen from the above definitions of normal and shearing stresses that, since they represent the intensity of force on an area, stresses* are measured in units of force divided by units of area. Since a force is a vector and an area is a scalar, their ratio, which represents the component of stress in a given direction, is a vectorial quantity.

It should be noted that *stresses multiplied by the respective areas on which they act give forces, and it is the sum of these forces at an imaginary cut that keeps a body in equilibrium.*

A metric system of units, referred to as the International System of Units and abbreviated SI, from the French *Systeme International d Unites*, is used in this text. A change to this modernized metric system of measurement is taking place throughout the world. In the United States a number of maior industries have announced plans to convert to SI units. Among these are the automotive, agricultural equipment, and business machine industries. These changes make it inevitable that SI units will become the predominant system of measurement in the United States.

The base units in the SI are *meter* or *metre* (m) for length, *kilogram* (kg) for mass, and *second* (s) for time. The derived unit for area is a *square meter* (m^2), and for acceleration a *meter per second squared* (m/s^2). The unit of force is defined as a unit mass subjected to a unit acceleration, i.e., *kilogrammeter per second squared* ($kg \cdot m/s^2$), and is designated a *newton* (N). The unit of stress is the *newton per square meter* (N/m^2), also designated a *pascal* (Pa). Multiple and submultiple prefixes representing steps of 1000 are recommended. For example, force can be shown in *millinewtons* (1 mN = 0.001 N), *newtons*, or *kilonewtons* (1 kN = 1000N), length in *millimeters* (1 mm = 0.001m), *meters*, or *kilometers* (1 km = 1000m), stresses in *kilopascals* (1 kPa = 10^6Pa), *megapascals* (1 MPa = 10^6Pa), *gigapascals* (1 GPa = 10^9Pa), etc.

The stress expressed numerically in units of N/m^2 may appear to be unusually small to those accustomed to the English system of units. This is because the force of one newton is small in relation to a pound-force, and a square meter is associated with a much larger area than one square inch. Therefore, it may be more acceptable to think in terms of a force of one newton acting on one square millimeter. This leads to the notation N/mm^2, a notation which initially was not recommended. However, since this is precisely equivalent to the megapascal (MPa), it is gaining wide acceptance.

If in addition to a plane such as *ABCD* in Fig. 1.1(a) another plane an infinitesimal distance away and parallel to the first were passed through the body, a thin element of the body would be isolated. Then, if an additional two pairs of planes were passed normal to the first pair, an elementary cube of infinitesimal dimensions would be isolated from the body. Such a cube is shown in Fig. 1.3. Here, for identification purposes, the process of resolution of stresses into components has been carried further than discussed above. At each surface the shearing stress τ has been resolved into two components parallel to a particular set of axes. The subscripts of the σ's designate the direction of the normal stress along a particular axis, while the stress itself acts on a plane perpendicular to the same axis. The first subscripts of the τ's associate the shearing stress with a plane that is perpendicular to a given axis, while the second designate the direction of the shearing stress.

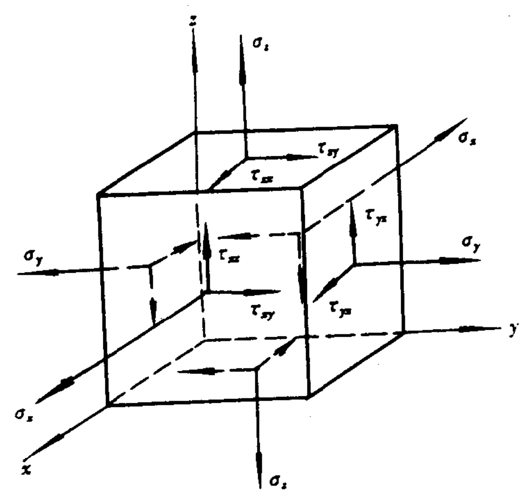

Fig. 1.3 The most general state of stress acting on an element

An infinitesimal cube, as shown in Fig. 1.3. could be used as the basis for an exact formulation of the problem in mechanics of materials. However, the methods for the study of the behavior of such a cube (which involve the writing of an equation for its equilibrium and making certain that such a cube, after deformations caused in it by the action of forces will be geometrically compatible with the adjoining infinitesimal cubes) are beyond the scope of this course. They are in the realm of the mathematical theory of elasticity. The procedures used in this text do not resort to the generality implied in Fig. 1.3. The methods used here will be much simpler.

4. Axial Load; Normal Stress

In many practical situations, if the direction of the imaginary plane cutting a member is judiciously selected, the stresses that act on the cut will be found both particularly significant and simple to determine. One such important case occurs in a straight *axially* loaded rod in tension, *provided a plane is passed perpendicular to the axis of the rod*. The tensile stress acting on such a cut is the *maximum* stress, as any other cut not perpendicular to the axis of the rod provides a larger surface for resisting the applied force. The maximum stress is the most significant one, as it tends to cause the failure of the material.

To obtain an algebraic expression for this maximum stress, consider the case illustrated in Fig. 1.4(a). If the rod is assumed weightless, two equal and opposite forces P are necessary, one at each end to maintain equilibrium. Then, as stated in Art. 1-2, since the body as a whole is in equilibrium, any part of it is also in equilibrium. A part of the rod to either side of the cut x-x is in equilibrium. At the cut, where the cross-sectional area of the rod is A, a force equivalent to P, as shown in Figs. 1.4(b) and (c), must be developed. Whereupon, from the definition of stress, the normal stress, or the stress that acts perpendicularly to the cut, is

$$\sigma = \frac{P}{A} \quad \text{or} \quad \frac{force}{area} \quad \left[\frac{\mathrm{N}}{\mathrm{m}^2}\right] \tag{1-1}$$

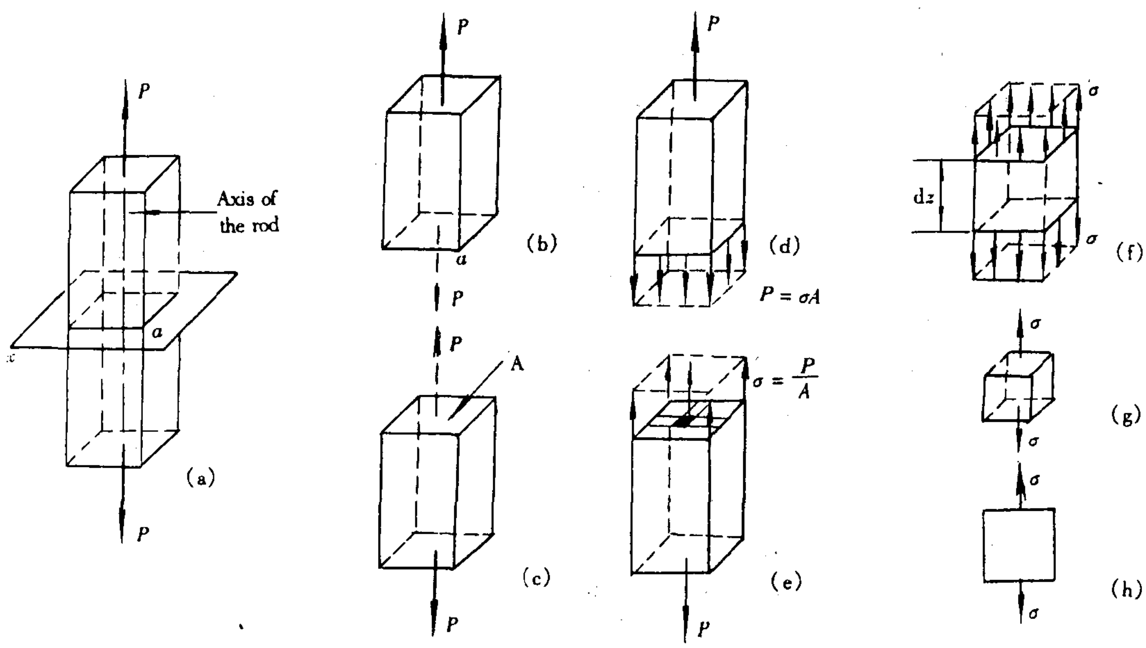

Fig. 1.4 Successive steps in the analysis of a body for stress

This *normal* stress is *uniformly* distributed over the cross-sectional area A. The nature of the quantity computed by Eq. 1-1 may be seen graphically in Figs. 1.4(d) and (e). *In general, the force P* is a resultant of a number of forces to one side of the cut or another.

If an additional cut is made parallel to the plane x-x in Fig. 1.4(a), the isolated section of the rod could be represented as in Fig. 1.4(f), and upon further "cutting," an infinitesimal cube as in Fig. 1.4(g) results. The only kind of stresses that appear here are the normal stresses on the two surfaces of the cube. Such a *state of stress* on an element is referred to as *uniaxial stress*. In practice, isometric views of a cube as shown in Fig. 1.4(g) are seldom employed the diagrams are simplified to look like those of Fig. 1.4(h). Nevertheless, the student must never lose sight of the three-dimensional aspect of the problem at hand.

At a cut, the system of tensile stresses computed by Eq. 1-1 provides an equilibrant to the externally applied force. When these normal stresses are mulitiplied by the corresponding infinitesimal areas and then summed over the whole area of a cut, the summation is equal to the applied force P. Thus the system of stresses is *statically equivalent* to the force P. Moreover, the resultant of this sum must act through the *centroid* of a section. Conversely, to have a uniform stress distribution in a rod, the applied axial force must act through the centroid of the cross-sectional area investigated. For example, in the machine part shown in Fig. 1.5(a) the stresses cannot be obtained from Eq. 1-1 alone. Here, at a cut such as A-A, a statically equivalent system of forces developed within the material must consist not only of the force P but also of a bending moment M that must maintain the externally applied force in equilibrium. This causes nonuniform stress distribution in the member. This will be treated in Chapter 7.

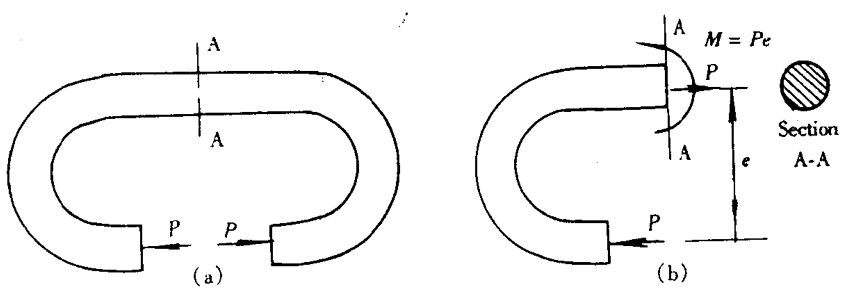

Fig. 1.5 A member with a nonuiform stress distribution at Section A-A

In accepting Eq. 1-1, it must be kept in mind that the material's behavior is *idealized*. Each and every particle of a body is assumed to contribute equally to the resistance of the force. A perfect homogeneity of the material is implied by such an assumption. Real materials, such as metals, consist of a great many grains, while wood is fibrous. In real materials, some particles will contribute more to the resistance of a force than others. Stresses are shown in Figs. 1.4(d) and (e) actually do not exist. The diagram of true stress distribution varies in each particular case and is a highly irregular, jagged affair. However, on the average, or statistically speaking, computations based on Eq. 1-1 are correct, and hence the computed stress represents a highly significant quantity.

Similar reasoning applies to compression members. The maximum normal or compressive stress can also be obtained by passing a section perpendicular to the axis of a member and applying Eq. 1-1. The stress so obtained will be of uniform intensity as long as the resultant of the applied forces coincides with the *centroid* of the area at the cut. However, one must exercise additional care when compression members are investigated. These may be so slender that they may not behave in the fashion considered. For example, an ordinary meter under a rather small axial compression force has a tendency to buckle sideways and collapse. The consideration of such *instability* of compression members is deferred until Chapter 13. *Equation 1-1 is applicable only for axiaily loaded compression members that are rather chunky*, i.e., to short blocks. As will be shown in Chapter 13, a block whose Least dimension is approximately one-tenth of its length may usually be considered a short block. For example, a 50 mm by 100 mm wooden piece may be 500 mm long and still be considered a short block.

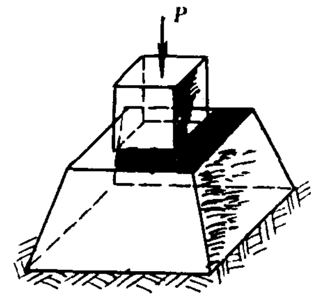

Fig.1.6 Bearing stresses occur between the block and pier

Situations often arise where one body is supported by another. If the resultant of the applied forces coincides with the centroid of the contact area between the two bodies, the intensity of force, or stress, between the two bodies can again be determined from Eq. 1-1. It is customary to refer to this normal stress as a <u>*bearing stress*</u>⑥. Figure 1.6, where a short block bears on a concrete pier and the latter bears on the soil, illustrates such a stress. The bearing stresses are obtained by dividing the applied force P by the corresponding area of contact.

5. Average Shearing Stress

Another situation that frequently arises in practice is shown in Figs. 1.7(a), (c), and (e). In all of these cases the forces are transmitted from one part of a body to the other by causing stresses in the plane parallel to the applied force. To obtain stresses in such instances, cutting planes as A-A are selected and free-body diagrams as shown in Figs. 1.7(b), (d), and (f) are used. The forces are transmitted through the respective cut areas. Hence, *assuming* that the stresses that *act in the plane of these cuts are uniformly distributed*, one obtains a relation for stress

$$\tau = \frac{P}{A} \quad \text{or} \quad \frac{force}{area} \quad \left[\frac{\mathrm{N}}{\mathrm{m}^2}\right] \tag{1-2}$$

where τ by definition is the *shearing* stress, P is the total force acting across and parallel to the cut, often called *shear*, and A is the cross-sectional area of the cut member. For reasons to be discussed later, unlike normal stress, the shearing stress given by Eq. 1.2 is only *approximately* true. For the cases shown, the shearing stresses actually are distributed in a nonuniform fashion across the area of the cut. The quantity given by Eq. 1-2 represents an *average* shearing stress.

The shearing stress, as computed by Eq. 1-2, is shown diagrammatically in Fig. 1.7(g). Note that for the case shown in Fig. 1.7(e) there are *two planes* of the rivet that resist the force. Such a rivet is referred to as being in *double* shear.

In cases such as those in Fig. 1.7(c) and (e), as the force P is applied, a highly irregular pressure develops between a rivet or a bolt and the plates. The *average* nominal intensity of this pressure is obtained by dividing the force transmitted by the projected area of the rivet onto the plate. This is re-

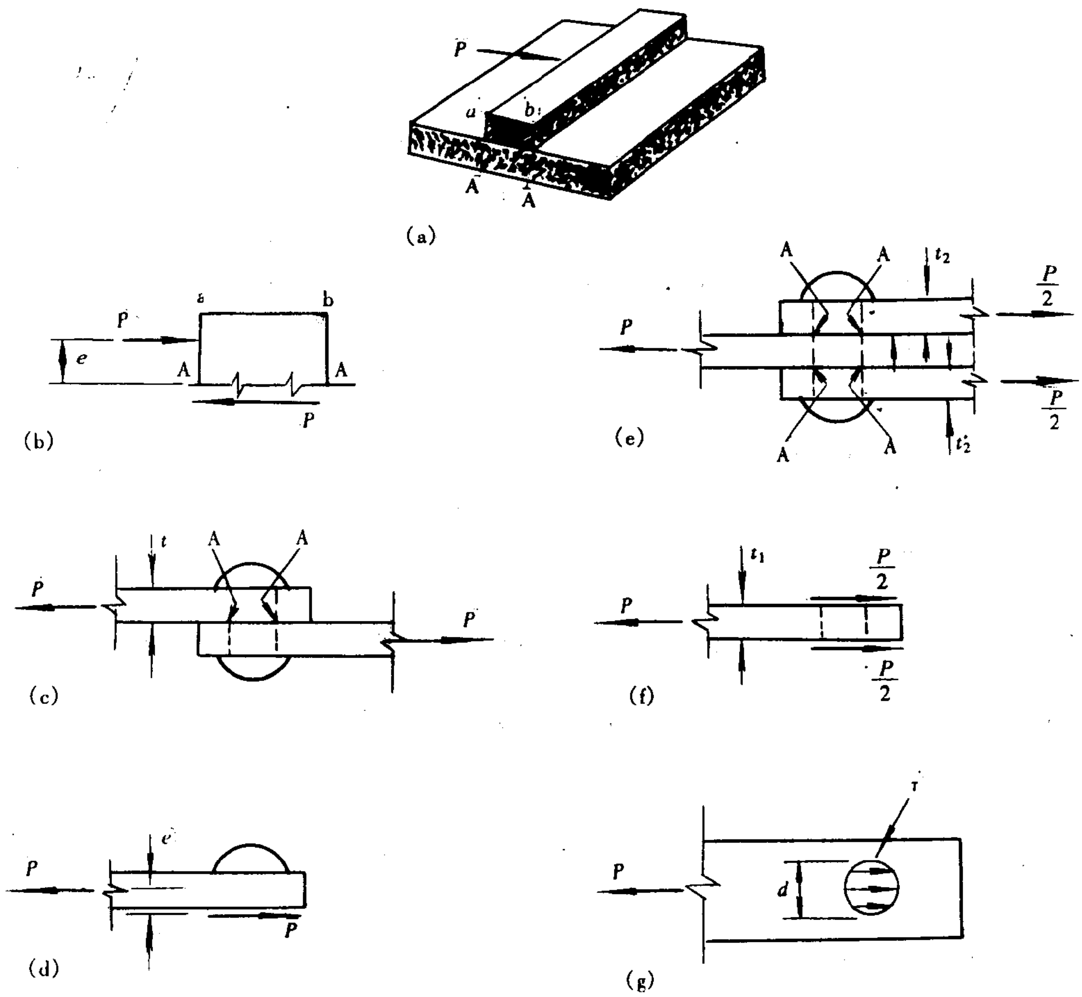

Fig.1.7 Loading conditions causing shearing stresses

ferred to as the *bearing stress*. The bearing stress in Fig. 1.7(c) is $\sigma_b = P/(td)$, where t is the thickness of the plate and d is the diameter of the rivet. For the case in Fig. 1.7(e) the bearing stresses for the middle plate and the outer plates are $\sigma_1 = P/(t_1 d)$ and $\sigma_2 = P/(2t_2 d)$, respectively.

6. Problems in Normal and Shearing Stress

Once P and A are determined in a given problem. Eqs. 1-1 and 1-2 are easy to apply. These equations have a clear physical meaning. Moreover, it seems reasonably clear that the desired magnitudes of stresses are the *maximum stresses*, as they are the greatest imposition on the strength of a material. The greatest stresses occur at a cut or section of *minimum* cross-sectional area and the greatest axial force. Such sections are called *critical sections*. The critical section for the particular arrangement being analyzed can usually be found by inspection. However, to determine the force P that acts through a member is usually a more difficult task. In the majority of problems treated in this text the latter information is obtained from statics.

For the equilibrium of a body in space, the equations of statics require the fulfillment of the following conditions:

$$\left.\begin{array}{ll} \Sigma F_x = 0 & \Sigma M_x = 0 \\ \Sigma F_y = 0 & \Sigma M_y = 0 \\ \Sigma F_z = 0 & \Sigma M_z = 0 \end{array}\right\} \quad (1\text{-}3)$$

The first column[7] of Eq. 1-3 states that the sum of *all* forces acting on a body in any (x, y, z) direction must be zero. The second column notes that the summation of moments of *all* forces around *any* axis parallel to any(x, y, z) direction must also be zero for equilibrium. In a *planar* problem. i.e., all members and forces lie in a single plane such as the x-y plane, relations $\Sigma F_z = 0, \Sigma M_x = 0$, and $\Sigma M_y = 0$, while still valid, are trivial.

These equations of statics are directly applicable to deformable solid bodies. The deformations tolerated in engineering structures are usually negligible in comparison with the over-all dimensions of structures. Therefore, *for the purposes of obtaining the forces in members, the initial undeformed dimensions of members are used in computations.*

There are problems where equations of statics are not sufficient to determine the forces in, or those acting *on*, the member. For example, the reactions for a straight beam, shown in Fig. 1.8, supported vertically at three points, cannot be determined from statics alone. In this planar problem there are four unknown reaction components, while only three *independent* equations of statics are available. Such problems are termed statically indeterminate[8]. The consideration of statically indeterminate problems is postponed until Chapter 11. For the present, and in the succeeding nine chapters of this text, *all structures and members considered will be statically determinate*, i.e., all of the external forces acting on such bodies can be determined by Eqs. 1-3. There is no dearth of statically determinate problems that are practically significant.

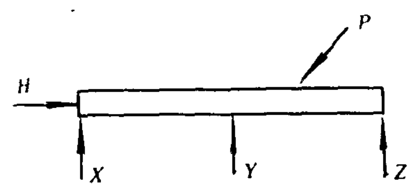

Fig. 1.8 A statically indeterminate beam

Equations 1-3 should already be familiar to the student. However, several examples where they are applied will now be given, the professional techniques for their use being stressed[9]. These examples will serve as an informal review of some of the principles of statics and will show applications of Eqs. 1-1 and 1-2.

New Words

1. axial [ˈæksiəl] *a*. 轴向的
2. component [kəmˈpəunənt] *n*. 分量,构件
3. vessel [ˈvesl] *n*. 容器
4. adequate [ˈædikwit] *a*. 充分的
5. shaft [ʃɑːft] *n*. 杆,轴

6. torque [tɔːk] *n*.扭矩
7. composite [ˈkɔmpəzit] *a*.合成的
8. rigid [ˈridʒid] *a*.刚性的
9. deflect [diˈflekt] *v*.(使)弯曲,挠曲
10. sag [sæg] *vi*.下垂
11. misalignment [ˈmisəˈlainmənt] *n*.未对准,安装误差
12. crack [kræk] *v*.断裂
13. plaster [ˈplɑːstə] *n*.灰泥
14. underneath [ʌndəˈniːθ] *n*.下面
15. slender [ˈslendə] *a*.细长的
16. compressive [kəmˈpresiv] *a*.压缩的
17. collapse [kəˈlæps] *v*.倒塌
18. buckling [ˈbʌkliŋ] *n*.弯折,弯曲
19. initial [iˈniʃəl] *a*.最初的
20. configuration [kənˌfigjuˈreiʃən] *n*.形状
21. stability [stəˈbiliti] *n*.稳定
22. maximum [ˈmæksiməm]([复]-s 或 maxima) *n*.极大值,最大量
23. vacuum [ˈvækjuəm]([复]-s 或 vacua) *n*.真空
24. minimum [ˈminiməm]([复]-s 或 minima) *n*.极小值,最小量
25. feasibility [ˌfiːzəˈbiliti] *n*.可能性,现实性
26. mission [ˈmiʃən] *n*.任务,使命
27. mechanics [miˈkæniks] *n*.[用作单]力学
28. stiffness [ˈstifnis] *n*.刚度
29. deformation [ˌdiːfɔːˈmeiʃən] *n*.形变
30. characteristic [ˌkæriktəˈristik] *n*.特性,特征
31. various [ˈvɛəriəs] *a*.多种的
32. prior [ˈpraiə] *a*.先前的
33. precedent [ˈpresidənt] *n*.先例
34. empirical [emˈpirikəl] *a*.经验的
35. rule [ruːl] *n*.规则,法则
36. rational [ˈræʃənl] *a*.理性的,有理的
37. tension [ˈtenʃən] *n*.拉,拉力
38. notable [ˈnəutəbl] *a*.值得注意的
39. indelible [inˈdelibl] *a*.不能消除的
40. offshore [ˈɔ(ː)fˈʃɔː] *a*.近海的
41. civil [ˈsivl] *a*.民用的
42. architectural [ˌɑːkiˈtektʃərəl] *a*.建筑(学)上的
43. metallurgist [meˈtælədʒist] *n*.冶金学家
44. fundamental [ˌfʌndəˈmentl] *a*.基本的,主要的

45. concept [ˈkɔnsept] *n*. 概念
46. portion [ˈpɔːʃən] *n*. 部分
47. discipline [ˈdisiplin] *n*. 训练
48. curriculum [kəˈrikjuləm]([复]-s 或 curricula) *n*. (一门)课程
49. equilibrium [ˌiːkwiˈlibriəm]([复]-s 或 equilibria) *n*. 平衡
50. fabricate [ˈfæbrikeit] *vt*. 建造
51. specimen [ˈspesimin] *n*. 样品,试样
52. phenomenon [fiˈnɔminən]([复]phenomena) *n*. 现象
53. occurrence [əˈkʌrəns] *n*. 发生,出现
54. vital [ˈvaitl] *a*. 不可缺少的
55. contradistinction [ˌkɔntrədisˈtiŋkʃən] *n*. 对比
56. blend [blend](-ed 或 blent) *vt*. 混合
57. postulate [ˈpɔstjuleit] *n*. 假定,公设
58. statics [ˈstætiks] *n*. 静力学
59. primarily [ˈpraimərili] *ad*. 第一,原来,主要地
60. presume [priˈzjuːm] *vt*. 推断,假定
61. familiar [fəˈmiljə] *a*. 熟悉的
62. limit [ˈlimit] *n*. 极限(值)
63. topic [ˈtɔpik] *n*. 题目,课题
64. simplicity [simˈplisiti] *n*. 简单
65. numerous [ˈnjuːmərəs] *a*. 许多的
66. formula [ˈfɔːmjulə]([复]-s 或 formulae) *n*. 公式
67. visualize, visualise [ˈvizjuəlaiz] *vt*. 设想
68. quantity [ˈkwɔntiti] *n*. (数)量,值
69. diagrammatic [ˌdaiəgrəˈmætik] *a*. 图解的
70. sketch [sketʃ] *n*. 草图
71. dividend [ˈdividend] *n*. 被除数
72. section [ˈsekʃən] *n*. 切割,截面,段落
73. resistance [riˈzistəns] *n*. 抵抗,阻力
74. balance [ˈbæləns] *n*. 平衡
75. uniform [ˈjuːnifɔːm] *a*. 均匀的
76. approach [əˈprəutʃ] *n*. 接近,近似法
77. employ [imˈplɔi] *vt*. 使用
78. respective [risˈpektiv] *a*. 各自的
79. support [səˈpɔːt] *n*. 支承,支柱
80. equation [iˈkweiʃən] *n*. 方程(式),等式
81. principal [ˈprinsəpəl] *a*. 主要的
82. arbitrary [ˈɑːbitrəri] *a*. 任意的
83. separate [ˈsepərit] *a*. 分离的, *vt*. 使分离

84. original [əˈridʒənəl] *a*.最初的
85. conclusion [kənˈkluːʒən] *n*.结论
86. briefly [briːfli] *ad*.短暂地
87. orient [ˈɔːrit] *vt*.使定向
88. significant [sigˈnifikənt] *a*.有意义的,有效的
89. reduce [riˈdjuːs] *vt*.减少,把…分解,简化
90. acceleration [əkˌseləˈreiʃən] *n*.加速度
91. multiply [ˈmʌltiplai] *v*.& *n*.乘,增加
92. opposite [ˈɔpəzit] *a*.相对的,相反的
93. view [vjuː] *n*.观察,观点
94. instantaneously [ˌinstənˈteinjəsli] *ad*.瞬时的
95. infinitesimal [ˌinfiniˈtesiməl] *a*.无穷小的
96. magnitude [ˈmægnitjuːd] *n*.大小,量值
97. intensity [inˈtensiti] *n*.强度
98. incline [inˈklain] *v*.(使)倾斜
99. customary [ˈkʌstəməri] *a*.习惯的,通常的
100. perpendicular [ˌpəːpənˈdikjulə] *a*.垂直的,正交的
101. parallel [ˈpærələl] *a*.平行的
102. resolution [ˌrezəˈljuːʃən] *n*.解决,分解,解析
103. normal [ˈnɔːməl] *a*.正交的,法线的
104. elementary [ˌeliˈmentəri] *a*.单元的
105. homogeneous [ˌhɔməˈdʒiːnjəs] *a*.均匀的,齐次的
106. macroscopic [ˌmækrəuˈskɔpik] *a*.宏观的
107. mental [ˈmentl] *a*.内心的,脑力的
108. vector [ˈvektə] *n*.矢量,向量
109. scalar [ˈskeilə] *n*.标量
110. sum [sʌm] *n*.总和
111. metric [ˈmetrik] *a*.公制的
112. system [ˈsistim;ˈsistəm] *n*.系统
113. abbreviate [əˈbriːvieit] *vt*.缩写
114. major [ˈmeidʒə] *a*.较大的,主要的
115. announce [əˈnauns] *vt*.宣布
116. convert [kənˈvəːt] *vt*.变换
117. inevitable [inˈevitəbl] *a*.不可避免的
118. predominant [priˈdɔminənt] *a*.主要的,占优势的
119. derive [diˈraiv] *vt*.导出
120. Pascal [ˈpæskəl] *n*.帕(斯卡)
121. prefix [ˈpriːfiks] *n*.前缀
122. numerical [nju(:)ˈmerikəl] *a*.数值的