高等学校教材

过程装备与控制工程专业英语

大学英语专业阅读教材编委会组织编写

徐鸿 董其伍 主编
吴东棣 审定

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前　　言

组织编审出版系列的专业英语教材，是许多院校多年来共同的愿望。在高等教育面向21世纪的改革中，学生基本素质和实际工作能力的培养受到了空前重视。对非英语专业的学生而言，英语水平和能力的培养不仅是文化素质的重要部分，在很大程度上也是能力的补充和延伸。在此背景下，教育部（原国家教委）几次组织会议研究加强外语教学问题，并制订有关规范，使外语教学更加受到重视。教材是教学的基本要素之一，与基础英语相比，专业英语教学的教材问题此时显得尤为突出。

国家主管部门的重视和广大院校的呼吁引起了化学工业出版社的关注，他们及时地与原化工部教育主管部门和全国化工类及相关专业教学指导委员会请求协商后，组织全国十余所院校成立了大学英语专业阅读教材编委会。在经过必要的调研后，根据学校需求，编委会优先从各校教学（交流）讲义中确定选题，同时组织力量开展编审工作。本套教材涉及的专业主要包括化学工程与工艺、石油化工、机械工程、信息工程、生产过程自动化、应用化学及精细化工、生物工程、环境工程、制药工程、材料科学与工程、化工商贸等。

根据“全国部分高校化工类及相关专业大学英语专业阅读教材编委会”的要求和安排编写的《过程装备与控制工程专业英语》教材，可供化工类及相关专业本科生使用，也可以作为同等程度（通过大学英语四级）专业技术人员的自学教材。

本教材由经过精选的、能够反映过程装备与控制工程专业基本专业内容的英语文章构成。所有文章均选自原版英文教科书、专著、科技报告或期刊。全书分成为6个部分（PART）；每个部分又包括5个教学单元（UNIT）；每单元则由1篇课文和1篇阅读材料（Reading Material）以及相关的单词及词组表（Words and Expressions）、注释（Notes）和练习题（Exercises）等组成。阅读材料提供与课文相关的背景知识，以进一步拓宽学生的视野并训练他们的阅读技能。各篇课文之间，既有一定的内在联系，又独立成章，可根据不同学时数灵活选学。在本书的最后附有本书各课的词汇总表。

本书由北京化工大学徐鸿教授和郑州工业大学董其伍教授主编，参加编写的还有北京化工大学范德顺副教授和钱才富副教授，郑州工业大学魏新利教授和吴金星讲师。全书由华东理工大学吴东棣教授审阅。在本书的编写过程中得到了化学工业出版社和大学英语专业阅读教材编委会的大力支持，在此谨致衷心的感谢。

虽然本书的全体编审人员都有多年的专业英语教学实验经验，也参阅了大量科技英语和专业方面的书刊，但由于过程装备与控制工程专业牵涉的领域宽，学科面广，限于作者水平，不妥之处在所难免，我们热诚希望使用本书的广大师生向我们提出宝贵意见。

编著

2000年1月

内 容 提 要

本教材的目的是在目前尚不具备在某些专业课中直接使用英文教材和使用英语讲授的条件下，使大学本科学生在学完基础英语之后，在高年级可通过专业英语课程做到大学四年英语不间断，熟悉科技与专业英语的特点，扩大科技与专业英语词汇量，提高科技与专业英语阅读能力。

本教材是由精通英语的原化工机械专业（现过程装备与控制工程专业）的专业教师在总结多年科技英语和专业英语教学经验的基础上编写成的。全书包括过程装备力学基础、金属材料、过程工业、过程设备、过程机械和过程装备控制等6个部分。每个部分选定5个主题，编出5个单元的教材；全书共有30个单元。每个单元则由主课文、主课文词汇表、课文注释、练习作业、阅读材料和阅读材料词汇表等组成。书后还附有词汇总表。

本教材的内容覆盖了过程装备与控制工程专业的基本专业内容。主课文和阅读材料都选自西方各著名出版社的原版英文科技书籍，并兼顾多种体裁以及英美的不同文风。全书30个单元的内容可作为60学时课程的教材。如果安排的学时较少，也可以只选用本书的部分内容而不会出现缺乏连贯性的问题。另外，本书也可以用作相关专业工程技术人员学习专业英语的自学教材。

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PART I BASIC KNOWLEDGE OF MECHANICS

Unit 1 General Equilibrium Conditions of A System

In this section, we shall consider the conditions that the forces and couples acting upon a body must satisfy in order for it to be in equilibrium①.

According to Newton's first law, the sum of the forces exerted on a body at rest must be zero. Notice, however, that this law says nothing about the moments, or rotational effects, of the forces. Clearly, the total moment must also be zero, else the body would rotate.

The fundamental problem here is that Newton's first law (and second law), as originally stated, applies only for very small bodies, or particles, with negligible dimensions and nonzero mass. However, it can be extended to bodies of finite size as follows.

Consider a system consisting of two particles, and let $\boldsymbol{f}_1$ and $\boldsymbol{f}_2$ be the forces due to the interaction between them (Fig. 1. 1). These forces are called internal forces, since they are due to interactions between bodies within the system. Assuming that the internal forces obey Newton's third law, we have $\boldsymbol{f}_1 = -\boldsymbol{f}_2$. Suppose that there are also forces, such as $\boldsymbol{F}_1$, $\boldsymbol{F}_2$, and $\boldsymbol{F}_3$, exerted on the particles due to interactions with bodies outside the system. Such forces are called external forces. Clearly, and the forces acting upon a particular particle must have the same point of application because a particle has negligible dimensions.

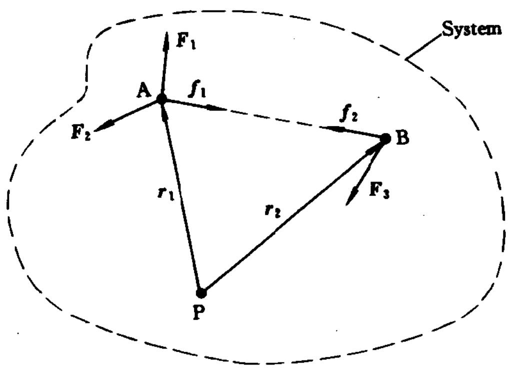


Fig. 1. 1 A system of forces

We shall say that the system is in equilibrium if each particle within it is in equilibrium. In this case, by Newton's first law, the sum of the forces acting upon each particle must be zero. For particle A we have

$$\sum \boldsymbol{F}_{\mathrm{A}} = \boldsymbol{F}_1 + \boldsymbol{F}_2 + \boldsymbol{f}_1 = \boldsymbol{0}$$

and for particle B

$$\sum \boldsymbol{F}_{\mathrm{B}} = \boldsymbol{f}_2 + \boldsymbol{F}_3 = 0$$

The total force acting upon the system is

$$\sum \boldsymbol{F} = \sum \boldsymbol{F}_{\mathrm{A}} + \sum \boldsymbol{F}_{\mathrm{B}} = \boldsymbol{F}_1 + \boldsymbol{F}_2 + \boldsymbol{F}_3 + \boldsymbol{f}_1 + \boldsymbol{f}_2 = \boldsymbol{0}$$

Now let us consider the total moment of these forces about some point P. Referring to Fig. 1. 1, we have

$$\sum \boldsymbol{M}_P=\boldsymbol{r}_1\times(\sum \boldsymbol{F}_A)+\boldsymbol{r}_2\times(\sum \boldsymbol{F}_B)$$

But $\sum \boldsymbol{F}_A=\sum \boldsymbol{F}_B=\boldsymbol{0}$; so the total moment must also be zero, as stated previously.

Since the forces $\boldsymbol{f}_1$ and $\boldsymbol{f}_2$ have the same line of action, the moment condition can be rewritten as

$$\sum \boldsymbol{M}_P=\boldsymbol{r}_1\times(\boldsymbol{F}_1+\boldsymbol{F}_2+\boldsymbol{f}_1+\boldsymbol{f}_2)+\boldsymbol{r}_2\times\boldsymbol{F}_3=\boldsymbol{0}$$

But $\boldsymbol{f}_1=-\boldsymbol{f}_2$; so the conditions on the forces and moments reduce to

$$\sum \boldsymbol{F}=\boldsymbol{F}_1+\boldsymbol{F}_2+\boldsymbol{F}_3=\boldsymbol{0}$$

and

$$\sum \boldsymbol{M}_P=(\boldsymbol{r}_1\times\boldsymbol{F}_1)+(\boldsymbol{r}_1\times\boldsymbol{F}_2)+(\boldsymbol{r}_2\times\boldsymbol{F}_3)=\boldsymbol{0}$$

In other words, if the system is in equilibrium, the sum of the external forces acting upon it is zero and so is the sum of the moments of these forces about an arbitrary point. The internal forces need not be considered because their effects cancel out.

Although we shall not go through the details, it should not be too difficult to see that the preceding results hold for a system consisting of any number of particles acted upon by any number of external forces, provided the internal forces obey Newton's third law. In particular, these results apply to bodies of finite extent, since such bodies can be thought of as consisting of a large number of very small pieces, or particles②. Thus, we have the following general equilibrium conditions:

If a system is in equilibrium, then

$$\sum \boldsymbol{F}=\boldsymbol{0} \quad \text{and} \quad \sum \boldsymbol{M}_P=\boldsymbol{0} \tag{1.1}$$

where $\sum \boldsymbol{F}$ is the sum of the external forces acting upon the system and $\sum \boldsymbol{M}_P$ is the total moment of these forces about an arbitrary point, including the moments of any couples which may be acting.

Equations (1.1) are *necessary* conditions for equilibrium; i. e., if the system is in equilibrium, these equations must be satisfied. They are not, in general, *sufficient* conditions for equilibrium; satisfaction of these equations does not necessarily guarantee that the system will be in equilibrium. This presents no difficulties, however, for we shall be dealing only with systems known to be in equilibrium. Equations (1.1) are both necessary and sufficient conditions for equilibrium of a rigid body. Proof that they are sufficient requires use of Newton's second law and other knowledge beyond the level of this text.

It is important to note that Eqs. (1.1) hold for any system in equilibrium, regardless of the material of which it is comprised③. For example, they hold for a mass of fluid at rest, as well as for solid bodies. They also apply to moving systems under certain conditions, since Newton's first law, upon which they are based, applies to particles moving with constant velocity as well as to particles at rest④. For instance, Eqs. (1.1) hold for bodies that move in a straight line at constant speed without rotation and for bodies that rotate at a constant rate about a fixed axis through their mass center. Typical examples are an airplane in straight, level flight at constant speed and the pulley on an electric motor rotating at constant speed. However, problems involving motion of any kind are usually relegated to texts on dynamics.

When expressed in component form, Eqs. (1.1) yield the six scalar equations:

$$\sum F_x=0 \qquad \sum F_y=0 \qquad \sum F_z=0$$
$$\sum M_{px}=0 \qquad \sum M_{py}=0 \qquad \sum M_{pz}=0 \tag{1.2}$$

These equations can be used in a force analysis of a system to solve for unknown information concerning the external forces and couples acting. Since there are six equations, we can generally solve for six unknowns. If all of the unknowns concerning the external forces and couples can be determined from the equilibrium equations, the problem is said to be *statically determinate*. If not, it is said to be *statically indeterminate*.

When there are more unknowns than equations of equilibrium in a problem, it is tempting to try to obtain additional equations by considering moments about more than one point. Unfortunately, this procedure does not work.

(Selected from: Karl K. Stevens, *Statics & Strength of Materials*, 2nd Edition, Prentice-Hall Inc. 1987.)

Words and Expressions

1. couple [ˈkʌpl] *n.* 力偶，电偶
2. exert [igˈzəːt] *v.* 施加（压力），用（力）
3. fundamental [fʌndəˈmentl] *a.* 基本的，基础的，主要的
4. negligible [ˈneglidʒəbl] *a.* 可以忽略的，微不足道的
5. moment [ˈməumənt] *n.* 力矩，弯矩，转矩
6. equilibrium [ˌiːkwiˈlibriəm] *n.* 平衡（状态，性）
7. cancel out 相约，相消
8. preceding [priˈsiːdiŋ] *a.* 以前的，上述的
9. pulley [ˈpuli] *n.* 滑轮，皮带轮
10. relegate [ˈreligeit] *vt.* 归类，委托
11. component [kəmˈpəunənt] *n.* 分力，分量，构件，成分
12. scalar [ˈskeilə] *n.*; *a.* 纯量（的），标量（的）
13. statically determinate 静定的
14. statically indeterminate 静不定的，超静定的

Notes

① 本句可译为：“在这一部分，我们将研究为了使一个物体保持平衡，作用在其上的力和力偶所必须满足的条件”。

② 本句可译为：“特别是，这些结果也适用于有限尺寸的物体，因为这样的物体可认为是由大量微体或质点组成的。”句中“in particular”意思是“特别是，尤其是”。

③ 本句可译为：“重要的是要注意到，方程式（1.1）适用于任何平衡系统，而不管组成该系统的物质是什么”。句中 hold for 意思是：“适用于”。

④ 本句可译为：“在某种条件下，它们（指两方程式）也适用于运动系统，因为它们是建立在牛顿第一定律的基础上，而牛顿第一定律既适用于匀速运动的质点，也适用于静止的质点。”句中 since 引导一个原因状语从句。

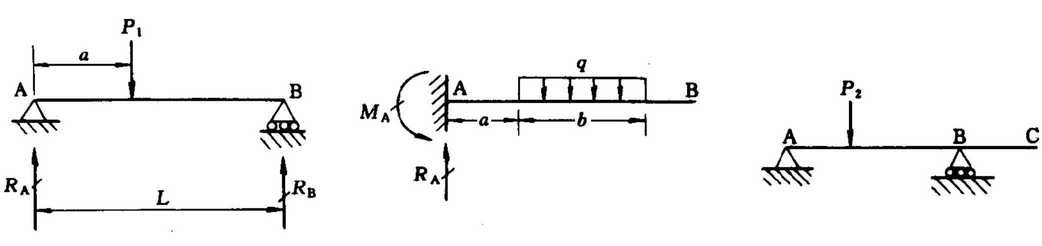
Exercises

1. After reading the text above, summarize the main idea of it in oral English.
2. Answer the following questions, according to the text:
   (1) Can you describe Newton's first law? what's it?
   (2) What forces are called internal forces?
   (3) What are the general equilibrium conditions of a system?
   (4) Can Eqs. (1.1) apply to moving systems under certain conditions? why?
3. Translate paragraph 3 into Chinese.
4. Put the following into Chinese, by reference to the text:
   at rest rotational effects interaction negligible arbitrary fundamental
5. Put the following into English:
   内力 外力 平衡 必要条件 充分条件 静定的 超静定的
6. Translate the following sentences into English:
   (1) 如果作用在物体上的总力矩不为零，则它就会发生转动。
   (2) 如果系统内的各个质点都处于平衡状态，那么该系统也是平衡的。

Reading Material 1

Static Analysis of Beams

A bar that is subjected to forces acting transverse to its axis is called a beam. In this section we will consider only a few of the simplest types of beams, such as those shown in Fig. 1.2. In every instance it is assumed that the beam has a plane of symmetry that is parallel to the plane of the figure itself. Thus, the cross section of the beam has a vertical axis of symmetry. Also, it is assumed that the applied loads act in the plane of symmetry, and hence bending of the beam occurs in that plane. Later we will consider a more general kind of bending in which the beam may have an unsymmetrical cross section.



(a) A simple supported beam (b) A cantilever beam (c) A beam with an overhang

Fig. 1.2 Types of beams

The beam in Fig. 1.2(a), with a pin support at one end and a roller support at the other, is called a *simply supported beam*, or a *simple beam*. The essential feature of a simple beam is that both ends of the beam may rotate freely during bending, but they cannot translate in the lateral direction. Also, one end of the beam can move freely in the axial direction (that is, horizontally). The supports of a simple beam may sustain vertical reactions acting either upward or downward.

The beam in Fig. 1.2(b) which is built-in or fixed at one end and free at the other end, is called a *cantilever beam*. At the fixed support the beam can neither rotate nor translate, while at the free end it may do both. The third example in the figure shows a beam with an overhang. This beam is simply supported at A and B and has a free end at C.

Loads on a beam may be *concentrated forces*, such as P_1 and P_2 in Fig. 1.2(a) and (c), or distributed loads, such as the load q in Fig. 1.2(b). Distributed loads are characterized by their intensity, which is expressed in units of force per unit distance along the axis of the beam. For a uniformly distributed load, illustrated in Fig. 1.2(b), the intensity is constant; a varying load, on the other hand, is one in which the intensity varies as a function of distance along the axis of the beam.

The beams shown in Fig. 1.2 are statically determinate because all their reactions can be determined from equations of static equilibrium. For instance, in the case of the simple beam supporting the load P_1[Fig. 1.2(a)], both reactions are vertical, and their magnitudes can be found by summing moments about the ends; thus, we find

$$R_A=\frac{P_1(L-a)}{L} \qquad R_B=\frac{P_1a}{L}$$

The reactions for the beam with an overhang [Fig. 1.2(c)] can be found in the same manner.

For the cantilever beam [Fig. 1.2(b)], the action of the applied load q is equilibrated by a vertical force R_A and a couple M_A acting at the fixed support, as shown in the figure. From a summation of forces in the vertical direction, we conclude that

$$R_A=qb$$

and, from a summation of moments about point A, we find

$$M_A=qb\left(a+\frac{b}{2}\right)$$

The reactive moment M_A acts counterclockwise as shown in the figure.

The preceding examples illustrate how the reactions (forces and moments) of statically determinate beams may be calculated by statics. The determination of the reactions for statically indeterminate beams requires a consideration of the bending of the beams, and hence this subject will be postponed.

The idealized support conditions shown in Fig. 1.2 are encountered only occasionally in practice. As an example, long-span beams in bridges sometimes are constructed with pin and roller supports at the ends. However, in beams of shorter span, there is usually some restraint against horizontal movement of the supports. Under most conditions this restraint has little effect on the action of the beam and can be neglected. However, if the beam is very flexible, and if the horizontal restraints at the ends are very rigid, it may be necessary to consider their effects.

Example *

Find the reactions at the supports for a simple beam loaded as shown in Fig. 1. 3(a). Neglect the weight of the beam.

Solution

The loading of the beam is already given in diagrammatic form. The nature of the supports is examined next and the unknown components of these reactions are boldly indicated on the diagram. The beam, with the unknown reaction components and all the applied forces, is redrawn in Fig. 1. 3(b) to deliberately emphasize this important step in constructing a free-body diagram. At A, two unknown reaction components may exist, since the end is pinned. The reaction at B can only act in a vertical direction since the end is on a roller. The points of application of all forces are carefully noted. After a free-body diagram of the beam is made, the equations of statics are applied to obtain the solution.

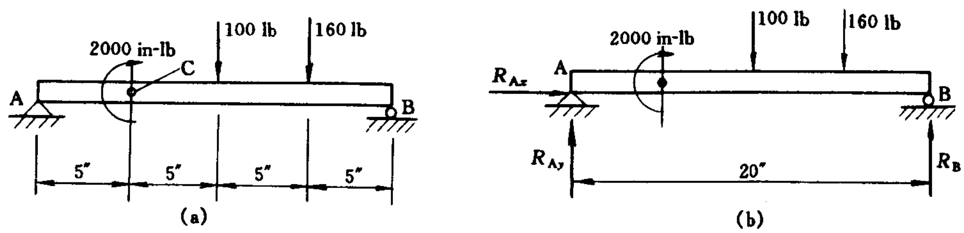


Fig. 1. 3 A simple beam

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

$\sum M_A = 0+, 2000 + 100(10) + 160(15) - R_B(20) = 0, R_B = +2700\ \text{lb} \uparrow$

$\sum M_B = 0+, R_{Ay}(20) + 2000 - 100(10) - 160(5) = 0, R_{Ay} = -10\text{lb} \downarrow$

Check: $\sum F_y = 0 \uparrow +, -10 - 100 - 160 + 270 = 0$

Note that $\sum F_x = 0$ uses up one of the three independent equations of statics, thus only two additional reaction components may be determined from statics. If more unknown reaction components or moments exist at the support, the problem becomes statically indeterminate.

Note that the concentrated moment applied at C enters only into the expressions for the summation of moments. The positive sign of R_B indicates that the direction of R_B has been correctly assumed in Fig. 1. 3(b). The inverse is the case of R_{Ay}, and the vertical reaction at A is downward. Note that a check on the arithmetical work is available if the calculations are made as shown.

(Selected from Stephen P. Timoshenko and James M. Gere, *Mechanics of Materials*, Van Nostrand Reinhold Company Ltd. , 1978.

* Selected from Egor P. Popov, *Introduction to Mechanics of Solids*, Prentice-Hall Inc. , 1968.)

Words and Expressions

1. transverse ['trænzvəːs] *a.* 横向的，横切的，横截的
2. symmetry ['simitri] *n.* 对称性，对称现象，均匀
3. pin support 铰支座
4. roller support 滚轴支座
5. translate [træns'leit] *v.* 平移，移动
6. lateral ['lætərəl] *a.* 横向的，水平的
7. sustain [səs'tein] *vt.* 支撑，承受住，维持
8. cantilever ['kæːntiliːvə] *n.* 悬臂（梁），支撑木
9. overhang ['əuvə'hæŋ] *n.* 突出物，外伸，悬空
10. intensity [in'tensiti] *n.* 强度，密度
11. reaction [ri'ækʃən] *n.* 反作用（力），反力
12. magnitude ['mægnitjuːd] *n.* 大小，量级，幅度
13. equilibrate [iːkwi'laibreit] *v.* （使）平衡，（使）相称
14. inverse ['in'vəːs] *a.* （相）反的，逆的；*n.* 倒数
15. counterclockwise ['kautə'klɔkwaiz] *a.*；*ad.* 逆时针方向（的）
16. deliberately [di'libəritli] *ad.* 审慎的，故意的

Unit 2 Stress and Strain

1. Introduction to Mechanics of Materials

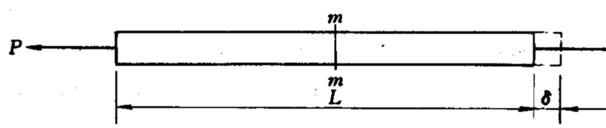
Mechanics of materials is a branch of applied mechanics that deals with the behaviour of solid bodies subjected to various types of loading. It is a field of study that is known by a variety of names, including "strength of materials" and "mechanics of deformable bodies." The solid bodies considered in this book include axially-loaded bars, shafts, beams, and columns, as well as structures that are assemblies of these components. Usually the objective of our analysis will be the determination of the stresses, strains, and deformations produced by the loads; if these quantities can be found for all values of load up to the failure load, then we will have obtained a complete picture of the mechanical behaviour of the body.

Theoretical analyses and experimental results have equally important roles in the study of mechanics of materials. On many occasions we will make logical derivations to obtain formulas and equations for predicting mechanical behaviour, but at the same time we must recognize that these formulas cannot be used in a realistic way unless certain properties of the material are known. These properties are available to us only after suitable experiments have been made in the laboratory. Also, many problems of importance in engineering cannot be handled efficiently by theoretical means, and experimental measurements become a practical necessity. The historical development of mechanics of materials is a fascinating blend of both theory and experiment, with experiments pointing the way to useful results in some instances and with theory doing so in others[①]. Such famous men as Leonardo da Vinci (1452-1519) and Galileo Galilei (1564-1642) made experiments to determine the strength of wires, bars, and beams, although they did not develop any adequate theories (by today's standards) to explain their test results. By contrast, the famous mathematician Leonhard Euler (1707-1783) developed the mathematical theory of columns and calculated the critical load of a column in 1744, long before any experimental evidence existed to show the significance of his results[②]. Thus, Euler's theoretical results remained unused for many years, although today they form the basis of column theory.

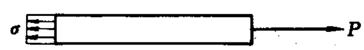
The importance of combining theoretical derivations with experimentally determined properties of materials will be evident as we proceed with our study of the subject[③]. In this section we will begin by discussing some fundamental concepts, such as stress and strain, and then we will investigate the behaviour of simple structural elements subjected to tension, compression, and shear.

2. Stress

The concepts of stress and strain can be illustrated in an elementary way by considering the extension of a *prismatic bar* [see Fig. 1. 4(a)]. A prismatic bar is one that has constant cross section throughout its length and a straight axis. In this illustration the bar is assumed to be loaded at its ends by axial forces P that produce a uniform stretching, or *tension*, of the bar. By making an artificial cut (section *mm*) through the bar at right angles to its axis, we can isolate part of the bar as a free body [Fig. 1. 4(b)]. At the right-hand end the tensile force P is applied, and at the other end there are forces representing the action of the removed portion of the bar upon the part that remains. These forces will be continuously distributed over the cross section, analogous to the continuous distribution of hydrostatic pressure over a submerged surface. The intensity of force, that is, the per unit area, is called the stress and is commonly denoted by the Greek letter σ. Assuming that the *stress* has a uniform distribution over the cross section [see Fig. 1. 4(b)], we can readily see that its resultant is equal to the intensity σ times the cross-sectional area A of the bar. Furthermore, from the



(a)



(b)

Fig. 1. 4 Prismatic bar in tension

equilibrium of the body shown in Fig. 1. 4(b), we can also see that this resultant must be equal in magnitude and opposite in direction to the force P. Hence, we obtain

$$\sigma=\frac{P}{A} \tag{1. 3}$$

as the equation for the uniform stress in a prismatic bar. This equation shows that stress has units of force divided by area——for example, Newtons per square millimeter (N/mm^2) or pounds per square inch (psi). When the bar is being stretched by the forces P, as shown in the figure, the resulting stress is a *tensile stress*; if the forces are reversed in direction, causing the bar to be compressed, they are called *compressive stresses*.

A necessary condition for Eq. (1. 3) to be valid is that the stress σ must be uniform over the cross section of the bar. This condition will be realized if the axial force P acts through the centroid of the cross section, as can be demonstrated by statics. When the load P does not act at the centroid, bending of the bar will result, and a more complicated analysis is necessary. Throughout this book, however, it is assumed that all axial forces are applied at the centroid of the cross section unless specifically stated to the contrary④. Also, unless stated otherwise, it is generally assumed that the weight of the object itself is neglected, as was done when discussing the bar in Fig. 1. 4.

3. Strain

The total elongation of a bar carrying an axial force will be denoted by the Greek letter δ[see Fig. 1.4(a)], and the elongation per unit length, or *strain*, is then determined by the equation

$$\varepsilon=\frac{\delta}{L} \tag{1.4}$$

where L is the total length of the bar. Note that the strain ε is a nondimensional quantity. It can be obtained accurately from Eq. (1.4) as long as the strain is uniform throughout the length of the bar. If the bar is in tension, the strain is a *tensile strain*, representing an elongation or a stretching of the material; if the bar is in compression, the strain is a *compressive strain*, which means that adjacent cross sections of the bar move closer to one another.

(Selected from Stephen P. Timoshenko and James M. Gere, *Mechanics of Materials*, Van Nostrand Reinhold Company Ltd., 1978.)

Words and Expressions

1. stress [stres] *n.* 应力，受力（状态，作用）
2. strain [strein] *n.* 应变；*v.* 使变形
3. deformable [diˈfɔːməbl] *a.* 可［易］变形的
4. shaft [ʃɑːft] *n.*（传动，旋转）轴；柱身
5. derivation [deriˈveiʃən] *n.* 推导，导出，推理
6. axially-loaded *a.* 受轴向载荷的
7. blend [blend] *n.* 混合（物）；*v.* 混合，掺和
8. tension [ˈtenʃən] *n.* 拉伸，张力，拉力
9. shear [ʃiə] *n.* 剪切，剪力；*v.* 剪切，剪断
10. prismatic [prizˈmætik] *a.* 等截面的
11. at right angles to 与…垂直，与…成直角
12. analogous [əˈnæləgəs] *a.* 类［相］似的，模拟的
13. hydrostatic [ˌhaidrəuˈstætik] *a.* 静水（力）学的，流体静力学的
14. submerge [səbˈməːdʒ] *v.* 浸没，沉没，淹没
15. denote [diˈnəut] *v.* 表示，指示
16. resultant [riˈzʌltənt] *n.* 合力；*a.* 合成的，总的
17. centroid [ˈsentrɔid] *n.* 矩心，质心，重心，形心
18. elongation [ilɔːŋˈgeiʃən] *n.* 拉张，伸长，延伸率
19. adjacent [əˈdʒeisənt] *a.* 相邻的，临近的
20. free-body 自由体，隔离体

Notes

① 本句可译为："材料力学的发展历史是理论与实验极有趣的结合，在一些情况下是实验指明