



普通高等教育“十一五”国家级规划教材
PUTONG GAODENG JIAOYU SHIYIWU GUOJIAJI GUIHUA JIAOCAI

RENENG YU DONGLI GONGCHENG
ZHUANYE YINGYU

热能与动力工程 专业英语

(第三版)

阎维平 主编
柳成文 副主编



中国电力出版社

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Thermal Energy & Power



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内 容 提 要

本书为普通高等教育“十一五”国家级规划教材。

本书包含了热能与动力工程专业的基础知识和专业知识,介绍了近年来该领域的新发展和新技术。书中正文和阅读材料来源广泛,涉及的专业英语词汇量多,专业特点突出,内容丰富。全书共分九章,除第一章为热能与动力工程专业的基础知识外,其余八章分别介绍了锅炉、汽轮机、火力发电厂、热工自动化、发电厂集控运行、燃气轮机与联合循环、制冷与空调、环境工程等内容。各章后的阅读材料包括科技写作、英文专业期刊投稿、专利申请、技术转让、国际行业标准与规范介绍、签证申请、国外大学研究生入学申请、专业人员简历、单位及转换与专业常用缩写词等。

本书可作为高等学校本科热能与动力工程专业的专业英语教材,也可作为高职高专电力技术类、能源类及相关专业的专业英语教材,还可供工作在这一专业领域内的工程技术人员和管理人员阅读参考。

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前 言

本书自第一版发行以来，受到了许多高等学校同行们的青睐，被选为专业英语的教科书，为培养具有现代知识的新一代发电厂技术人员作出了贡献。同时在使用过程中，教师 and 同学们给作者提出了许多建设性的意见，在此深表感谢。

此次修订，在第二版的基础上，对全书内容进行了较大的修改，删去了原书的第七、第九和第十一章，更新了其他章节的内容，体现了电力行业的新技术和新发展。同时，根据广大读者的强烈要求，本书在网络上提供各章正文的译文，供读者参考。

本书第三版由华北电力大学阎维平担任主编，山西大学工程学院柳成文任副主编。书中第一章由叶学民编写，第二章由梁秀俊编写，第三章由崔映红编写，第四章由阎维平编写，第五章由王琦编写，第六章由张文革编写，第七章由柳成文编写，第八章由谢英柏编写，第九章由邢德山编写。阎维平负责策划并统稿。

由于编者水平所限，书中仍难免存在缺点和不足，敬请读者批评指正。

编 者

2009.08

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Chapter 1 Introduction to Thermal Sciences

1.1 Fundamental of Engineering Thermodynamics

Thermodynamics is a science in which the storage, transformation, and transfer of energy are studied. Energy is stored as internal energy (associated with temperature), kinetic energy (due to motion), potential energy (due to elevation) and chemical energy (due to chemical composition); it is transformed from one of these forms to another; and it is transferred across a boundary as either heat or work.

In thermodynamics, we will derive equations that relate the transformations and transfers of energy to properties such as temperature, pressure, and density. Substances and their properties, thus, become very important in thermodynamics. Many of our equations will be based on experimental observations that have been organized into mathematical statements or laws; the first and second laws of thermodynamics are the most widely used.

1.1.1 Thermodynamic system and control volume

A thermodynamic system is a fixed quantity of matter contained within some enclosure. The surface is usually an obvious one (like that surrounding the gas in the cylinder). However, it may be an imagined boundary (like the deforming boundary of a certain amount of mass as it flows through a pump).

All matter and space external to a system is collectively called its surroundings. Thermodynamics is concerned with the interactions of a system and its surroundings, or one system interacting with another. A system interacts with its surroundings by transferring energy across its boundary. No material crosses the boundary of a system. If the system does not exchange energy with the surroundings, it is an isolated system.

In many cases, an analysis is simplified if attention is focused on a particular volume in space into which, or from which, a substance flows. Such a volume is a control volume. A pump, a turbine, and an inflating or deflating balloon are examples of control volumes. The surface that completely surrounds the control volume is called a control surface.

Thus, we must choose, in a particular problem, whether a system is to be considered or whether a control volume is more useful. If there is mass flux across a boundary, then a control volume is required; otherwise, a system is identified.

1.1.2 Equilibrium, process, and cycle

When the temperature of a system is referred to, it is assumed that all points of the system have the same, or essentially the same, temperature. When the properties are constant from point to point and when there is no tendency for change with time, a condition of thermodynamic equilibrium exists. If the temperature, say, is suddenly increased at some part of the system boundary, spontaneous redistribution is assumed to occur until all parts of the

system are at the same temperature.

When a system changes from one equilibrium state to another, the path of successive states through which the system passes is called a process. If, in the passing from one state to the next, the deviation from equilibrium is infinitesimal, a quasi-equilibrium process occurs, and each state in the process may be idealized as an equilibrium state. Quasi-equilibrium processes can approximate many processes, such as the compression and expansion of gases in an internal combustion engine, with no significant loss of accuracy. If the system goes from one equilibrium state to another through a series of nonequilibrium states (as in combustion), a nonequilibrium process occurs.

When a system in a given initial state experiences a series of processes and returns to the initial state, the system undergoes a cycle. At the end of the cycle, the properties of the system have the same values they had at the beginning.

The prefix *iso-* is attached to the name of any property that remains unchanged in a process. An isothermal process is one in which the temperature is held constant; in an isobaric process, the pressure remains constant; an isometric process is a constant-volume process.

1.1.3 Vapor-liquid phase equilibrium in a pure substance

Consider as a system 1 kg of water contained in the piston/cylinder arrangement shown in Fig. 1-1(a). Suppose that the piston and weight maintain a pressure of 0.1 MPa in the cylinder and that the initial temperature is 20°C. As heat is transferred to the water, the temperature increases appreciably, the specific volume increases slightly, and the pressure remains constant. When the temperature reaches 99.6°C, additional heat transfer results in a change of phase, as indicated in Fig. 1-1(b). That is, some of the liquid becomes vapor, and during this process both the temperature and pressure remain constant, but the specific volume increases considerably. When the last drop of liquid has vaporized, further transfer of heat

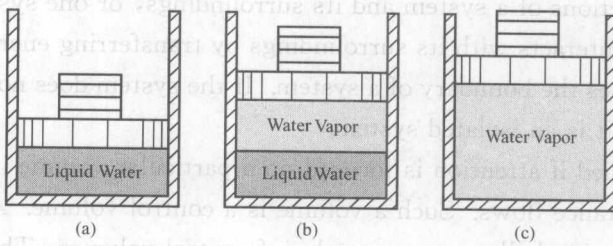


Fig. 1-1 Constant-pressure change from liquid to vapor

results in an increase in both temperature and specific volume of the vapor, as shown in Fig. 1-1(c).

The term saturation temperature designates the temperature at which vaporization takes place at a given pressure. This pressure is called the saturation pressure for the given temperature. Thus, for water at 99.6°C the saturation pressure is 0.1 MPa, and for water at 0.1 MPa the saturation temperature is 99.6°C.

If a substance exists as liquid at the saturation temperature and pressure, it is called saturated liquid. If the temperature of the liquid is lower than the saturation temperature for the existing pressure, it is called either a subcooled liquid (implying that the temperature is low-

er than the saturation temperature for the given pressure) or a compressed liquid (implying that the pressure is greater than the saturation pressure for the given temperature).

When a substance exists as part liquid and part vapor at the saturation temperature, its quality is defined as the ratio of the mass of vapor to the total mass. Thus, in Fig. 1-1(b), if the mass of the vapor is 0.2 kg and the mass of the liquid is 0.8 kg, the quality is 0.2 or 20%. Quality has meaning only when the substance is in a saturated state.

If a substance exists as vapor at the saturation temperature, it is called saturated vapor. (Sometimes the term dry saturated vapor is used to emphasize that the quality is 100%). When the vapor is at a temperature greater than the saturation temperature, it is said to exist as superheated vapor. The pressure and temperature of superheated vapor are independent properties, since the temperature may increase while the pressure remains constant.

Let us plot on the temperature-volume diagram of Fig. 1-2 the constant-pressure line that represents the states through which the water passes as it is heated from the initial state of 0.1 MPa and 20°C. Let state *A* represent the initial state, *B* the saturated-liquid state (99.6°C), and line *AB* the process in which the liquid is heated from the initial temperature to the saturation temperature. Point *C* is the saturated-vapor state, and line *BC* is the constant-temperature process in which the change of phase from liquid to vapor occurs. Line *CD* represents the process in which the steam is superheated at constant pressure. Temperature and volume both increase during this process.

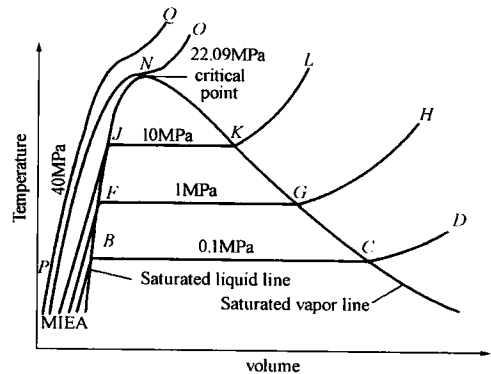


Fig. 1-2 Temperature-volume diagram

Table 1-1 Some Critical-Point Data

	Critical temperature	Critical pressure	Critical volume
	°C	MPa	m ³ /kg
Water	374.14	22.09	0.003155
Carbon dioxide	31.05	7.39	0.002143
Oxygen	-118.35	5.08	0.002438
Hydrogen	-239.85	1.30	0.032192

is called the critical point. At the critical point the saturated-liquid and saturated-vapor states are identical. The temperature, pressure, and specific volume at the critical point are called the critical temperature, critical pressure, and critical volume. The critical-point data for some substances are given in Table 1-1.

1.1.4 The first law of thermodynamics

The first law of thermodynamics is commonly called the law of conservation of energy.

In a similar manner, a constant pressure of 10 MPa is represented by line *IJKL*, for which the saturation temperature is 311.1°C. At a pressure of 22.09 MPa, represented by line *MNO*, we find, however, that there is no constant-temperature vaporization process. Instead, point *N* is a point of inflection with a zero slope. This point

In elementary physics courses, the study of conservation of energy emphasizes changes in kinetic and potential energy and their relationship to work. A more general form of conservation of energy includes the effects of heat transfer and internal energy changes. Other forms of energy could also be included, such as electrostatic, magnetic, strain, and surface energy.

Historically, the first law of thermodynamics was stated for a cycle: the net heat transfer is equal to the net work done for a system undergoing a cycle.

1.1.5 The second law of thermodynamics

The second law of thermodynamics can be stated in a variety of ways. Here we present two: the Clausius statement and the Kelvin-Planck statement.

Clausius Statement

It is impossible to construct a device that operates in a cycle and whose sole effect is the transfer of heat from a cooler body to a hotter body.

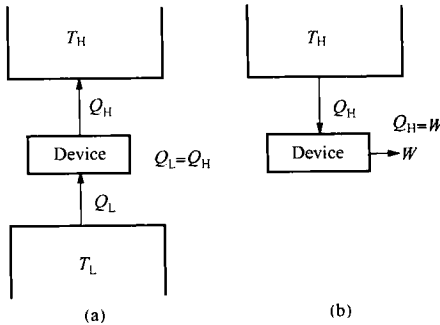


Fig. 1-3 Violations of the second law

This statement relates to a refrigerator (or a heat pump). It states that it is impossible to construct a refrigerator that transfers energy from a cooler body to a hotter body without the input of work; this violation is shown in Fig. 1-3(a).

Kelvin-Planck Statement

It is impossible to construct a device that operates in a cycle and produces no other effect than the production of work and the transfer of heat from a single body.

In other words, it is impossible to construct a heat engine that extracts energy from a reservoir, does work, and does not transfer heat to a low-temperature reservoir. This rules out any heat engine that is 100 percent efficient, like the one shown in Fig. 1-3(b).

1.1.6 The Carnot cycle

The heat engine that operates the most efficiently between a high-temperature reservoir and a low-temperature reservoir is the Carnot engine. This is an ideal engine that uses reversible processes to form its cycle of operation; such a cycle is Carnot cycle. The Carnot engine is very useful, since its efficiency establishes the maximum possible efficiency of any real engine. If the efficiency of a real engine is significantly lower than the efficiency of a Carnot engine between the same limits, then additional improvements may be possible.

The ideal Carnot cycle in Fig. 1-4 is composed of four reversible processes: 1→2: Isothermal expansion; 2→3: Adiabatic reversible expansion; 3→4: Isothermal compression; 4→1: Adiabatic reversible compression. The efficiency of a Carnot cycle is

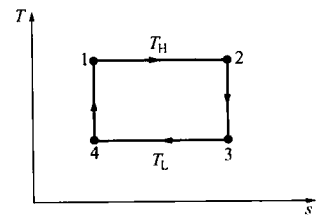


Fig. 1-4 The Carnot cycle

$$\eta = 1 - \frac{T_L}{T_H} \quad (1 - 1)$$

Note that the efficiency is increased by raising the temperature T_H at which heat is added or by lowering the temperature T_L at which heat is rejected.

1.1.7 The Rankine cycle

The first class of power cycles that we consider are those utilized by the electric power generating industry, namely, power cycles that operate in such a way that the working fluid changes phase from a liquid to a vapor. The simplest vapor-power cycle is called the Rankine cycle, shown schematically in Fig. 1-5(a). A major feature of such a cycle is that the pump requires very little work to deliver high-pressure water to the boiler. A possible disadvantage is that the expansion process in the turbine usually enters the quality region, resulting in the formation of liquid droplets that may damage the turbine blades.

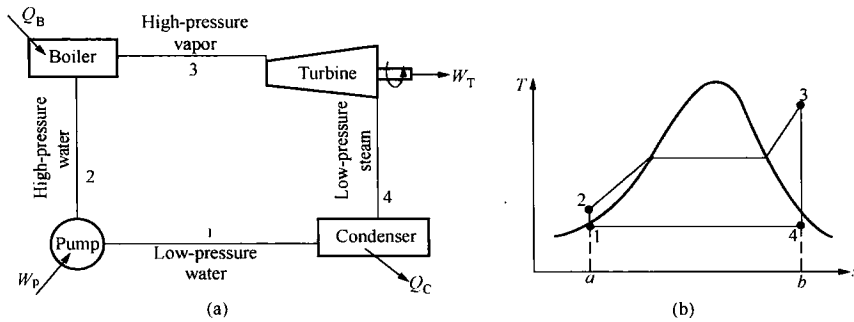


Fig. 1-5 The Rankine cycle

(a) The major components; (b) The T - s diagram

The Rankine cycle is an idealized cycle in which friction losses in each of the four components are neglected. The losses usually are quite small and will be neglected completely in initial analysis. The Rankine cycle is composed of the four ideal processes shown on the T - s diagram in Fig. 1-5(b): 1→2: Isentropic compression in a pump; 2→3: Constant-pressure heat addition in a boiler; 3→4: Isentropic expansion in a turbine; 4→1: Constant-pressure heat rejection in a condenser.

The pump is used to increase the pressure of the saturated liquid. Actually, states 1 and 2 are essentially the same, since the high-pressure lines are extremely close to the saturation curve; they are shown separated for illustration only. The boiler (also called a steam generator) and the condenser are heat exchangers that neither require nor produce any work.

If we neglect kinetic energy and potential energy changes, the net work output is the area under the T - s diagram, represented by area 1-2-3-4-1 of Fig. 1-5(b); this is true since the first law requires that $W_{\text{net}} = Q_{\text{net}}$. The heat transfer to the working substance is represented by area a -2-3- b - a . Thus, the thermal efficiency η of the Rankine cycle is

$$\eta = \frac{\text{area } 1-2-3-4-1}{\text{area } a-2-3-b-a} \quad (1 - 2)$$

that is, the desired output divided by the energy input (the purchased energy). Obviously,

the thermal efficiency can be improved by increasing the numerator or by decreasing the denominator. This can be done by increasing the pump outlet pressure p_2 , increasing the boiler outlet temperature T_3 , or decreasing the turbine outlet pressure p_4 .

1.1.8 The Reheat cycle

It is apparent that when operating in a Rankine cycle with a high boiler pressure or a low condenser pressure it is difficult to prevent liquid droplets from forming in the low-pressure portion of the turbine. Since most metal cannot withstand temperatures above about 600°C , the reheat cycle is often used to prevent liquid-droplet formation: the steam passing through the turbine is reheated at some intermediate pressure, thereby raising the temperature to state 5 in the T - s diagram of Fig. 1-6. The steam then passes through the low-pressure section of the turbine and enters the condenser at state 6. This controls or completely eliminates the moisture problem in the turbine. Often, the turbine is separated into a high-pressure turbine and a low-pressure turbine. The reheat cycle does not significantly influence the thermal efficiency of the cycle, but it does result in a significant additional work output, represented in the figure by area $4-5-6-4'-4$ of Fig. 1-6. The reheat cycle demands a significant investment in additional equipment, and the use of such equipment must be economically justified by the increased work output. If reheat is not used to avoid droplet formation, the condenser pressure must be quite high, resulting in relatively low cycle efficiency. In that sense, reheat significantly increases cycle efficiency when compared to the cycle with no reheat but with the higher condenser pressure.

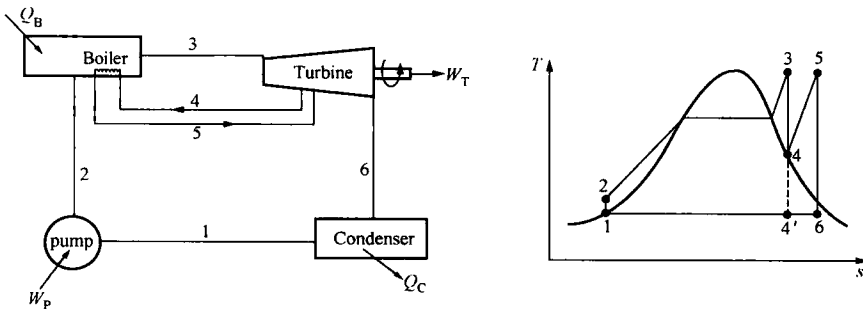


Fig. 1-6 The Reheat cycle

1.2 Fundamental of Fluid Mechanics

Fluid motions manifest themselves in many different ways. Some can be described very easily, while others require a thorough understanding of physical laws. In engineering applications, it is important to describe the fluid motions as simply as can be justified. This usually depends on the required accuracy. Often, accuracies of $\pm 10\%$ are acceptable, although in some applications higher accuracies have to be achieved. The general equations of motion are very difficult to solve; consequently, it is the engineer's responsibility to know which simplifying assumptions can be made. This, of course, requires experience and, more important,

an understanding of the physics involved.

Some common assumptions used to simplify a flow situation are related to fluid properties. For example, under certain conditions, the viscosity can affect the flow significantly; in others, viscous effects can be neglected greatly simplifying the equations without significantly altering the predictions. It is well known that the compressibility of a gas in motion should be taken into account if the velocities are very high. But compressibility effects do not have to be taken into account to predict wind forces on buildings or to predict any other physical quantity that is a direct effect of wind. After our study of fluid motions, the appropriate assumptions used should become more obvious. Here we introduce some important general approaches used to analyze fluid mechanics problems and give a brief overview of different types of flow.

1.2.1 Lagrangian and Eulerian descriptions of motion

In the description of a flow field, it is convenient to think of individual particles each of which is considered to be a small mass of fluid, consisting of a large number of molecules, that occupies a small volume that moves with the flow. If the fluid is incompressible, the volume does not change in magnitude but may deform. If the fluid is compressible, as the volume deforms, it also changes its magnitude. In both cases the particles are considered to move through a flow field as an entity.

In the study of particle mechanics, where attention is focused on individual particles, motion is observed as a function of time. The position, velocity, and acceleration of each particle are listed as $s(x_0, y_0, z_0, t)$, $\mathbf{V}(x_0, y_0, z_0, t)$, and $\mathbf{a}(x_0, y_0, z_0, t)$ and quantities of interest can be calculated. The point (x_0, y_0, z_0) locates the starting point—the name of each particle. This is the Lagrangian description, named after Joseph L. Lagrange, of motion that is used in a course on dynamics. In the Lagrangian description many particles can be followed and their influence on one another noted. This becomes, however, a difficult task as the number of particles becomes extremely large, as in a fluid flow.

An alternative to following each fluid particle separately is to identify points in space and then observe the velocity of particles passing each point; we can observe the rate of change of velocity as the particles pass each point, that is, $\partial \mathbf{V}/\partial x$, $\partial \mathbf{V}/\partial y$ and $\partial \mathbf{V}/\partial z$ and we can observe if the velocity is changing with time at each particular point, that is, $\partial \mathbf{V}/\partial t$. In this Eulerian description, named after Leonhard Euler, of motion, the flow properties, such as velocity, are functions of both space and time. In rectangular, Cartesian coordinates the velocity expressed as $\mathbf{V} = \mathbf{V}(x, y, z, t)$. The region of flow that is being considered is called a flow field.

1.2.2 Pathlines and streamlines

Two different lines help us in describing a flow field. A pathline is the locus of points traversed by a given particle as it travels in a field of flow; the pathline provides us with a "history" of the particle's locations. A photograph of a pathline would require a time exposure of an illuminated particle.

A streamline is a line in the flow possessing the following property: the velocity vector of each particle occupying a point on the streamline is tangent to the streamline, that is, $\mathbf{V} \times d\mathbf{r} = 0$. Since \mathbf{V} and $d\mathbf{r}$ are in the same direction; recall that the cross product of two vectors in the same direction is zero. A photograph of a streamline cannot be made directly. For a general unsteady flow the streamlines can be inferred from photographs of short pathlines of a large number of particles.

1.2.3 One-, two-, and three-dimensional flows

In the Eulerian description of motion the velocity vector, in general, depends on three space variables and time, that is, $\mathbf{V} = \mathbf{V}(x, y, z, t)$. Such a flow is a three-dimensional flow, because the velocity vector depends on three space coordinates. The solutions to problems in such a flow are very difficult and are beyond the scope of an introductory course. Even if the flow could be assumed to be steady [i. e., $\mathbf{V} = \mathbf{V}(x, y, z)$], it would remain a three-dimensional flow.

Often a three-dimensional flow can be approximated as a two-dimensional flow. For example, the flow over a wide dam is three-dimensional because of the end conditions, but the flow in the central portion away from the ends can be treated as two-dimensional. In general, a two-dimensional flow is a flow in which the velocity vector depends on only two space variables. An example is a plane flow, in which the velocity vector depends on two spatial coordinates, x and y , but not z [i. e., $\mathbf{V} = \mathbf{V}(x, y)$].

A one-dimensional flow is a flow in which the velocity vector depends on only one space variable. Such flows occur in long, straight pipes or between parallel plates. The velocity in the pipe varies only with r i. e., $u = u(r)$. The velocity between parallel plates varies only with the coordinate y i. e., $u = u(y)$. Even if the flow is unsteady so that $u = u(y, t)$, as would be the situation during startup, the flow is one-dimensional.

As for developed flows, the velocity profiles do not vary with respect to the space coordinate in the direction of flow. This demands that the region of interest be a substantial distance from an entrance or a sudden change in geometry. There are many engineering problems in fluid mechanics in which a flow field is simplified to a uniform flow: the velocity, and other fluid properties, are constant over the area. This simplification is made when the velocity is essentially constant over the area, a rather common occurrence. Examples of such flows are relatively high speed flow in a pipe section, and flow in a stream. The average velocity may change from one section to another; the flow conditions depend only on the space variable in the flow direction.

1.2.4 Newtonian fluid and non-Newtonian fluid

A Newtonian fluid is a fluid whose stress versus rate of strain curve is linear and passes through the origin. The constant of proportionality is known as the viscosity. A simple relation to describe Newtonian fluid behavior is $\tau = \mu du/dy$. τ is the shear stress exerted by the fluid, μ is the fluid viscosity, du/dy is the velocity gradient perpendicular to the direction of shear.

If a fluid does not obey this relation, it is termed a non-Newtonian fluid, of which there are several types, including polymer solutions, molten polymers, many solid suspensions and most highly viscous fluids. In a non-Newtonian fluid, the relation between the shear stress and the strain rate is nonlinear, and can even be time-dependent. Therefore a constant coefficient of viscosity can not be defined. A ratio between shear stress and rate of strain (or shear-dependent viscosity) can be defined, this concept being more useful for fluids without time-dependent behavior.

1.2.5 Viscous and inviscid flows

A fluid flow may be broadly classified as either a viscous flow or an inviscid flow. An inviscid flow is one in which viscous effects do not significantly influence the flow and are thus neglected. In a viscous flow the effects of viscosity are important and cannot be ignored.

To model an inviscid flow analytically, we can simply let the viscosity be zero; this will obviously make all viscous effects zero. It is more difficult to create an inviscid flow experimentally, because all fluids of interest (such as water and air) have viscosity. The question then becomes: Are there flows of interest in which the viscous effects are negligibly small? The answer is "Yes, if the shear stresses in the flow are small and act over such small areas that they do not significantly affect the flow field." This statement is very general, of course, and it will take considerable analysis to justify the inviscid flow assumption.

Based on experience, it has been found that the primary class of flows, which can be modeled as inviscid flows, is external flows, that is, flows which exist exterior to a body. Inviscid flows are of primary importance in flows around streamlined bodies, such as flow around an airfoil or a hydrofoil. Any viscous effects that may exist are confined to a thin layer, called a boundary layer, that is attached to the boundary, such

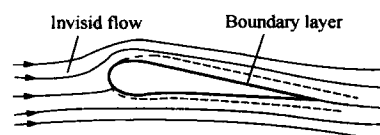


Fig. 1 - 7 Flow around an airfoil

as that shown in Fig. 1 - 7; the velocity in a boundary layer is always zero at a fixed wall, a result of viscosity. For many flow situations, boundary layers are so thin that they can simply be ignored when studying the gross features of a flow around a streamlined body. For example, the inviscid flow solution provides an excellent prediction to the flow around the airfoil, except inside the boundary layer and possibly near the trailing edge. Inviscid flow is also encountered in contractions inside piping systems and in short regions of internal flows where viscous effects are negligible.

Viscous flows include the broad class of internal flows, such as flows in pipes and conduits and in open channels. In such flows viscous effects cause substantial "losses" and account for the huge amounts of energy that must be used to transport oil and gas in pipelines. The no-slip condition resulting in zero velocity at the wall, and the resulting shear stresses, lead directly to these losses.

1.2.6 Laminar and turbulent Flows

A viscous flow can be classified as either a laminar flow or a turbulent flow. In a laminar flow the fluid flows with no significant mixing of neighboring fluid particles. If dye were injected into the flow, it would not mix with the neighboring fluid except by molecular activity; it would retain its identity for a relatively long period of time. Viscous shear stresses always influence a laminar flow. The flow may be highly time dependent or be steady.

In a turbulent flow fluid motions vary irregularly so that quantities such as velocity and pressure show a random variation with time and space coordinates. The physical quantities are often described by statistical averages. In this sense we can define a "steady" turbulent flow; a flow in which the time-average physical quantities do not change in time. A dye injected into a turbulent flow would mix immediately by the action of the randomly moving fluid particles; it would quickly lose its identity in this diffusion process. A laminar flow and a turbulent flow can be observed by performing a simple experiment with a water faucet. Turn the faucet on so the water flows out very slowly as a silent stream. This is laminar flow. Open the faucet slowly and observe the flow becoming turbulent. Note that a turbulent flow develops with a relatively small flow rate.

The flow regime depends on three physical parameters describing the flow conditions. The first parameter is a length scale of the flow field, such as the thickness of a boundary layer or the diameter of a pipe. If this length scale is sufficiently large, a flow disturbance may increase and the flow may be turbulent. The second parameter is a velocity scale such as a spatial average of the velocity; for a large enough velocity the flow may be turbulent. The third parameter is the kinematic viscosity; for a small enough viscosity the flow may be turbulent.

The three parameters can be combined into a single parameter that can serve as a tool to predict the flow regime. This quantity is the Reynolds number, named after Osborne Reynolds, a dimensionless parameter, defined as $Re = VL/\nu$, where L and V are a characteristic length and velocity, respectively, and ν is the kinematic viscosity; for example, in a pipe flow L could be the pipe diameter and V could be the average velocity. If the Reynolds number is relatively small, the flow is laminar; if it is large, the flow is turbulent. This is more precisely stated by defining a critical Reynolds number, Re_{crit} , so that the flow is laminar if $Re < Re_{crit}$. For example, in a flow inside a rough-walled pipe it is found that $Re_{crit} \approx 2000$. This is the minimum critical Reynolds number and is used for most engineering applications. If the pipe wall is extremely smooth and free of vibration, the critical Reynolds number can be increased as the fluctuation level in the flow is decreased; values in excess of 40 000 have been measured. The critical Reynolds number is different for every geometry, e. g. , it is 1500 for flow between parallel plates using the average velocity and the distance between the plates.

In a boundary layer that exists on a flat plate, due to a constant-velocity fluid stream, the length scale changes with distance from the upstream edge. A Reynolds number is calculated using the length x as the characteristic length. For a certain x_T , Re becomes Re_{crit} and