

FLUID FORCES ON NON-STREAMLINE BODIES – BACKGROUND NOTES AND DESCRIPTION OF THE FLOW PHENOMENA

1. INTRODUCTION

The purpose of this Item is to provide background information which will assist the user of the series of Data Items on the estimation of fluid forces on non-streamline (*i.e.* bluff)* bodies. This series of Data Items gives detailed numerical data for use in a wide range of problems in aeronautical, chemical, marine, mechanical and structural engineering associated with the flow of fluid around bluff bodies. The Items present the latest available information and contain numerical data for the effect of variations in all the parameters considered of importance. They include flow charts to guide the user through procedures that will enable him to obtain numerical results without requiring him to have a specialist's understanding of the physical phenomena that take place. This Item is intended to describe the physical phenomena for those who are not specialist aerodynamicists and want to understand the rudiments of the problem. It is mainly concerned with wind flow around bluff bodies, such as buildings, but the information is also applicable to other fluid flow problems providing the Reynolds number is sufficiently high. Further, more detailed, descriptions and mathematical analyses of the problem can be found in the References quoted in Section 12. A key word index is given in Section 13.

Fluid flowing around a bluff body exerts on that body forces which fluctuate with time. These forces can be resolved into a mean (time-averaged) component on which is superimposed a fluctuating component which varies with time. The first Data Items in the series are concerned with the estimation of the mean component; a knowledge of this alone is satisfactory for many problems such as the design of most structures, particularly those having high natural frequencies of vibration and high damping. These will be followed with Data Items concerned with the estimation of the fluctuating component. It should be remarked that, although it is possible to separate the mean component from the fluctuating component, time-varying phenomena, such as turbulence, affect both the mean and the fluctuating components of the force on the body. Consequently, a discussion of such time-varying phenomena is as relevant to an understanding of the mean component as to the fluctuating component.

2. NOTATION AND NOMENCLATURE

2.1 Notation

Three coherent systems of units are given below[†].

		<i>SI</i>	<i>British</i>	
A	reference surface area	m^2	ft^2	ft^2
a	local speed of sound in fluid	m/s	ft/s	ft/s
C_D	drag coefficient, $D/(\frac{1}{2}\rho V_\infty^2 A)$			
C_F	force coefficient, $F/(\frac{1}{2}\rho V_\infty^2 A)$			
C_L	lift coefficient, $L/(\frac{1}{2}\rho V_\infty^2 A)$			

* Definitions of streamline and non-streamline (*i.e.* bluff) bodies are given in Section 2.3

[†] See Section 2.2

		SI	British	
$C_X; C_Y; C_Z$	force coefficients, $X/(\frac{1}{2}\rho V_\infty^2 A)$; $Y/(\frac{1}{2}\rho V_\infty^2 A)$, $Z/(\frac{1}{2}\rho V_\infty^2 A)$			
C_M	moment coefficient (see Section 6.1)			
C_p	pressure coefficient, $(p - p_{ref})/(\frac{1}{2}\rho V_\infty^2)$			
D	drag force (measured in free-stream direction)	N	pdl	lbf
F	force acting on body	N	pdl	lbf
f	centre frequency of narrow-band vortex shedding (see Section 11.1)	Hz	c/s	c/s
L	lift force measured normal to free-stream direction	N	pdl	lbf
$\left. \begin{matrix} L_x(u); L_y(u); L_z(u) \\ L_x(v); L_y(v); L_z(v) \\ L_x(w); L_y(w); L_z(w) \end{matrix} \right\}$	integral length scales of u , v and w components of turbulence along x , y and z axes respectively	m	ft	ft
l	representative body dimension	m	ft	ft
M	local Mach number, V/a			
p	local static pressure	N/m ²	pdl/ft ²	lbf/ft ²
p_o	total pressure	N/m ²	pdl/ft ²	lbf/ft ²
p_{ref}	reference pressure usually taken as p_∞	N/m ²	pdl/ft ²	lbf/ft ²
$\left. \begin{matrix} R_u(x, 0, 0); \\ R_u(0, y, 0); \\ R_u(0, 0, z); \end{matrix} \right\}$	correlation coefficient of u component of turbulence for two points situated on x , y or z axes respectively (see Section 10.3)			
r_u	correlation function of u component of turbulence (see Section 10.3)	m ² /s ²	ft ² /s ²	ft ² /s ²
Re	Reynolds number, $V_\infty l/\nu$			
$\left. \begin{matrix} S_u(n); \\ S_v(n); \\ S_w(n) \end{matrix} \right\}$	power spectral density functions of u , v and w components of turbulence (see Section 10.1)	m ² /s	ft ² /s	ft ² /s
St	Strouhal number, fl/V_∞			
t	time	s	s	s

		<i>SI</i>	<i>British</i>	
$V(t)$	velocity; a time-dependent quantity	m/s	ft/s	ft/s
V	mean (time-averaged) velocity	m/s	ft/s	ft/s
u, v, w	components of fluctuating velocity along x, y and z axes respectively	m/s	ft/s	ft/s
X, Y, Z	component forces acting on body in direction of x, y and z axes respectively	N	pdl	lbf
x, y, z	system of rectangular cartesian coordinates; distances along x, y and z axes respectively	m	ft	ft
α	angle of inclination of free-stream direction to body longitudinal axis		degrees	
γ	ratio of specific heat capacity of fluid at constant pressure to that at constant volume			
ϵ	equivalent height of surface roughness (see Section 9)	m	ft	ft
μ	dynamic viscosity of fluid	N s/m ²	pdl s/ft ²	lbf s/ft ²
ν	kinematic viscosity of fluid, μ/ρ	m ² /s	ft ² /s	ft ² /s
ρ	density of fluid	kg/m ³	lb/ft ³	slug/ft ³
$\sigma^2(u)$	mean square value of $u, \bar{u}^2$	m ² /s ²	ft ² /s ²	ft ² /s ²

Subscripts

bl	denotes value in boundary layer
int	denotes internal value
∞	denotes free-stream value

2.2 Units

In the system of units known as *SI* the unit of force is the newton (N). A NEWTON is defined as the force required to impart an acceleration of 1 m/s^2 to a mass of 1 kg.

In the two coherent systems of British units given in Section 2.1 the units of force are respectively the poundal (pdl) and the pound-force (lbf). A POUNDAL is defined as the force required to impart an acceleration of 1 ft/s^2 to a mass of 1 pound. A POUND-FORCE (lbf) is defined as the force required to impart an acceleration of 1 ft/s^2 to a mass of 1 slug (which is 32.17 pounds) or, alternatively, as the force required to impart an acceleration of 32.17 ft/s^2 to a mass of 1 lb. Similarly, in the metric system, a KILOGRAMME-FORCE (kgf) is defined as the force required to impart an acceleration of 1 m/s^2 to a mass of 9.807 kg.

The units of pound-force and kilogramme-force must be distinguished from the local weights of bodies having masses of 1 lb and 1 kg respectively because the gravitational force or weight of the body is dependent on the local acceleration due to gravity which may be different from the assumed standard value of 32.17 ft/s^2 or 9.807 m/s^2 .

Some conversion factors between units are given below.

$$1 \text{ newton (N)} = 1 \text{ kg} \times 1 \text{ m/s}^2 = 102.0 \times 10^{-3} \text{ kgf}$$

$$1 \text{ poundal (pdl)} = 1 \text{ lb} \times 1 \text{ ft/s}^2 = 31.08 \times 10^{-3} \text{ lbf}$$

$$1 \text{ slug} = 32.17 \text{ lb.}$$

2.3 Definition of Streamline and Non-Streamline (Bluff) Bodies

A STREAMLINE BODY is defined as a body for which the major contribution to the drag force in the free-stream direction results directly from the viscous or skin friction action of the fluid on the body. A NON-STREAMLINE or BLUFF BODY is defined as a body for which the major contribution to the drag force is due to pressure forces arising from separation of the boundary layer flow adjacent to the surface over the rearward facing part of the body. For example, a body of circular or rectangular cross section is a bluff body and so is a flat plate or aerofoil inclined at a high angle to the oncoming flow. On the other hand, a thin flat plate lying parallel and edge on to the oncoming flow is a streamline body since the flow remains attached to the surface and skin friction accounts for up to 90 per cent of the total drag.

3. GENERAL

The force exerted by a fluid on a body can be resolved along the normal and tangential directions to the surface. In the normal direction the force per unit area is called the local **PRESSURE** (see Section 6.1) while in the tangential direction the force per unit area is called the **VISCOUS** or **FRICTION STRESS** (see Section 6.2).

Both the pressure and viscous forces on the body are dependent on the local properties of the fluid at the point in question and these are affected by the history of the element of fluid at the point. Most theoretical investigations in the field of fluid dynamics are based on the concept of a perfect, frictionless and incompressible fluid. For streamline bodies this theory supplies in many cases a satisfactory description of real motions but it does fail completely to account for the force on a body in the free-stream direction (the drag force) which is, incorrectly, predicted to be zero. On a non-streamline body in particular this force depends upon so many parameters that a complete theoretical solution of the problem cannot at the moment be envisaged and most design data are of experimental origin.

If the results of one experiment, or of measurements at full-scale conditions, are to be related to another situation of a different scale, it is essential that the requirements for comparability are known in detail. Application of dimensional analysis to the problem, and other considerations, shows that the force exerted on a body divided by $(1/2)\rho V_\infty^2 A$ can be expressed as a function of a series of non-dimensional parameters relating to both the characteristics of the free stream and the body surface. The force acting on a body is conveniently presented in the form of a non-dimensional force coefficient defined as

$$C_F = F / (1/2 \rho V_\infty^2 A)$$

Thus a non-dimensional general relationship for force coefficient in terms of the most important parameters can be expressed as

$$C_F = f \left\{ Re, M_\infty, \frac{\sqrt{u^2}}{V_\infty}, \frac{L_x(u)}{l} \text{ etc.}, \frac{n S_u(n)}{\sigma^2(u)}, \frac{\varepsilon}{l}, \dots \right\}. \quad (3.1)$$

For the flow pattern around two geometrically similar bodies orientated identically to two fluid streams to be similar (*i.e.* for complete **DYNAMICAL SIMILARITY**), the values of all the non-dimensional parameters must be the same in both cases. In Equation (3.1) the important non-dimensional parameters are the Reynolds number (Re), the Mach number (M_∞), the free-stream turbulence intensity $(u^2)^{1/2}/V_\infty$, the free-stream turbulence length scale ratio $(L_x(u)/l)$, the normalised power spectral density $(n S_u(n)/\sigma^2(u))$ and the surface roughness ratio (ε/l) .

It is impracticable to obtain complete identity of all parameters: however, in every instance some of the parameters have little or no effect on the flow and can be ignored. One of the most difficult tasks in the analysis of fluid flow problems around bluff bodies is to determine which are the most important parameters. The following Sections attempt to draw guidelines for this process. Each of the items on the right-hand side of Equation (3.1) is discussed in Sections 7 to 10. Finally, in Section 11 the time-dependent characteristics associated with vortex shedding are described.

Before proceeding with a description of pressure and viscous forces the concepts of ideal (inviscid) and real (viscous) fluid flows will first be discussed.

4. INVISCID FLOW

A fluid which is inviscid is sometimes called IDEAL. It produces no tangential viscous or frictional stresses or a pressure drag in the direction of the free stream. All REAL FLUIDS are viscous and may involve chemical changes. In order to develop a theory for a real fluid flow it is usually necessary to assume that it is not only inviscid, but is also non-turbulent and chemically inert. In such a theory of fluid flow around a body, layers of fluid adjacent to each other experience no viscous forces and act upon the body surface with pressure forces, normal to the body surface at each point, only. When the flow is steady a simple relation exists between the pressure, density and velocity along a STREAMLINE*. This is obtained from the equation of motion for the fluid flow and is

$$\int_1^2 \frac{dp}{\rho} + \int_1^2 V dV + \left(\int_1^2 g dz \right)^\dagger = 0 \quad (4.1)$$

where z is measured vertically from a horizontal datum plane. If further it is assumed that the flow is INCOMPRESSIBLE, that is to say that the density, ρ , does not vary with pressure (and is therefore a constant throughout the flow), this equation integrates to the well known relationship known as Bernoulli's equation

$$p + \frac{1}{2}\rho V^2 + \rho gz = \text{constant} = p_o. \quad (4.2)$$

The value, p_o , is a constant along a streamline and is known as the TOTAL HEAD or STAGNATION PRESSURE. Its value can vary from streamline to streamline if the flow field upstream of a body is a turbulent shear flow. Since the surface of a body is a stream surface it follows that in inviscid, steady incompressible flow the relation between pressure and velocity on the body surface is given by Equation (4.2).

In practice – that is in a real fluid flow – it is found that Bernoulli's equation applies to most regions of an incompressible non-turbulent flow field except in the boundary layers (see Section 5.1) immediately adjacent to solid surfaces in the flow and in the near wake of a body.

When the flow is COMPRESSIBLE (*i.e.* for gas flows at high speeds) Equation (4.2) no longer applies and Equation (4.1) has to be integrated, allowing for the variation of density with pressure. Thus for the flow of a gas along a streamline, when the gas is inviscid and non-heat conducting, and when no heat is lost or gained from the adjacent flow, the relation between density and pressure is $p/\rho^\gamma = \text{constant}$. This is the isentropic relationship which applies since the thermodynamic process is reversible. Hence, for an isentropic gas flow, Equation (4.1) when integrated becomes

$$\frac{p_1}{p_2} = \left\{ 1 - \frac{\gamma - 1}{2} \frac{V_1^2 - V_2^2}{a_2^2} \right\}^{\gamma/(\gamma - 1)}.$$

* A STREAMLINE is a curve such that its tangent at any point is in the direction of the velocity of the fluid at that point at the time considered. For a steady flow it is the path traced by an element of fluid in its motion around a body.

† The contribution of the term (gdz) in Equation (4.1) and (ρgz) in Equation (4.2) for gases can be ignored. In homogeneous liquid flows with no free surface the term ρgz does not affect the motion as it is simply the hydrostatic pressure field for the fluid at rest.

For $V_1 = 0$, $p_1 = p_o$ when the equation reduces to

$$\frac{p_o}{p} = \left\{ 1 + \frac{\gamma-1}{2} M^2 \right\}^{\gamma/(\gamma-1)} \quad (4.3)$$

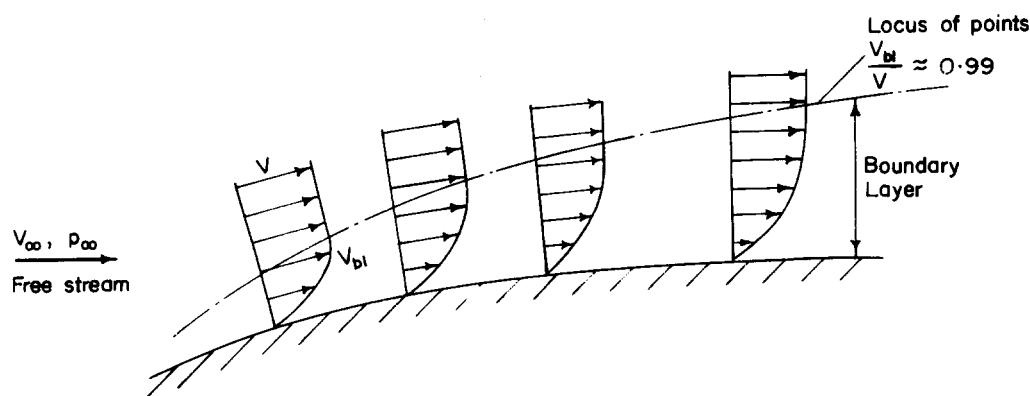
or, alternatively, for $V_2 = V_\infty$ and $p_2 = p_\infty$ it becomes

$$\frac{p}{p_\infty} = \left\{ 1 + \frac{\gamma-1}{2} M_\infty^2 \left[1 - \frac{V^2}{V_\infty^2} \right] \right\}^{\gamma/(\gamma-1)} \quad (4.4)$$

5. VISCOUS FLOW

5.1 Friction Forces, Boundary Layer, Separation

The FRICTION or VISCOUS force arises from the tangential shearing flow of a fluid along the surface of a body. The shearing forces are transmitted through the fluid shear layers adjacent to the surface as illustrated in Sketch 5.1. This layer of fluid in which a large velocity gradient exists normal to the surface is called the BOUNDARY LAYER and the flow in it is often referred to as a SHEAR FLOW. At the leading edge of a body, or more explicitly near the stagnation point (see Section 6.1), the boundary layer has only a small thickness. In general its thickness increases with distance along the surface, except in regions of high acceleration where its thickness can be reduced.



Sketch 5.1

In the boundary layer frictional forces act to slow down the fluid velocity relative to the body surface such that at the surface of the body there is no slip between the fluid and the body. When the boundary layer thickness is small compared with the body dimensions it is found that the pressure variation across the boundary layer normal to the surface can be neglected. Thus, in a real fluid, although Bernoulli's equation can only be applied outside the boundary layer, the pressure calculated at the edge of the boundary layer is also the pressure at the body surface when the boundary layer is thin.

In the flow adjacent to a surface, viscous forces are set up and their magnitude, provided the fluid is Newtonian, is equal to the product of the viscosity of the fluid and the velocity gradient normal to the flow direction, *i.e.*

$$\tau = \mu \frac{\partial V_{bl}}{\partial z}.$$

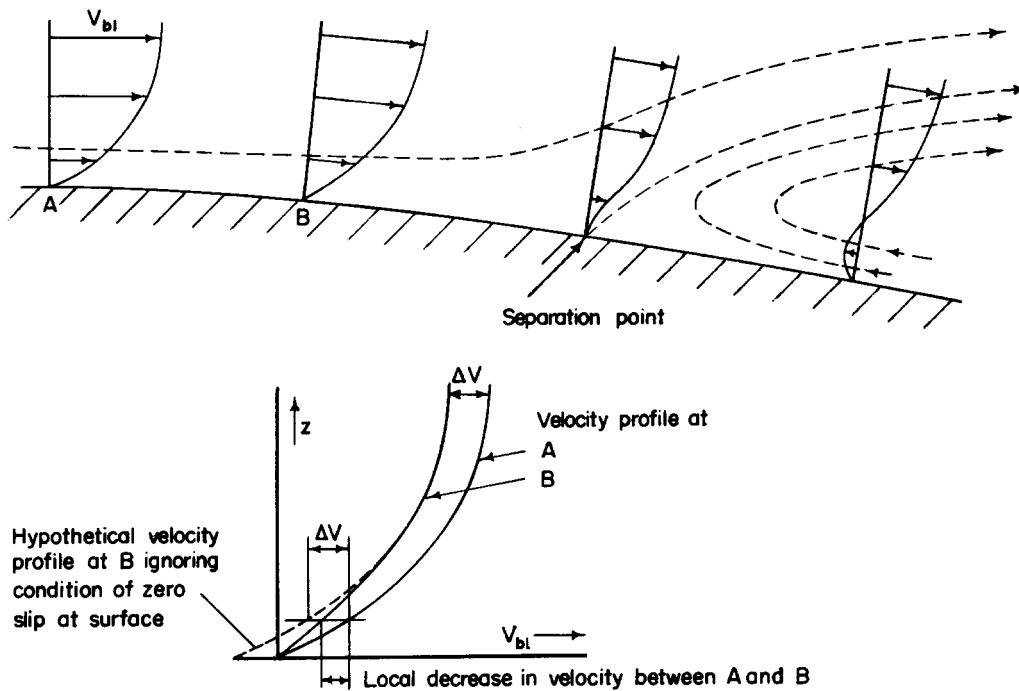
Because the viscosity (μ) of most fluids is small, shear stresses of significant magnitude only occur near the surface where a very large velocity gradient normal to the flow exists.

In addition to the viscous force, the mixing process of eddying fluid in a turbulent boundary layer flow (see Section 5.2) causes a greater interchange of momentum between fluid layers and produces an effective shear stress which is additive to the shear stress produced by the true viscosity of the fluid. The effective shear stress is called the REYNOLDS STRESS and in certain cases can be represented by an EDDY VISCOSITY multiplied by the local velocity gradient normal to the surface (see References 3 to 5).

For a rough surface, the roughness elements themselves act as small bluff bodies and eddies are cast off them which cause an increase in the shear forces. The earth's boundary layer is thick (about 200-600 m) compared to building heights and so buildings in the atmosphere must be treated as surface roughness elements in a boundary layer of shear flow.

The complete equations for the flow of a viscous fluid do not have solutions in closed form, except for certain elementary flows. However, it is possible to simplify these equations to describe the flow in a boundary layer. The latter equations can be solved if the flow outside the boundary layer is known to an adequate approximation. The methods, however, in general only have application when the boundary layer remains attached to the body surface. They do not apply in those regions, such as the downstream faces of bluff bodies, where boundary layer separation has occurred and the wake is both thick and unsteady. In this case very large discrepancies between the calculated pressure forces and measured values are found to occur.

A fluid forced to flow around a body attempts to resume its original undisturbed conditions of flow. In a real flow this is not achieved until the flow has progressed some way downstream of the body because of viscous effects in the boundary layer. Broadly speaking, over the forward facing part of the body the flow is accelerated and the local pressure decreases, and over the rearward facing part of the body the flow is retarded and the pressure increases again (see, for example, Sketch 5.5). A pressure increasing with distance along the surface (*i.e.* a positive pressure gradient) is compatible with the velocity at the edge of the boundary layer decreasing. On the other hand, at the surface itself a necessary condition is that the flow velocity shall be zero relative to the surface. In order that this condition of zero slip be maintained the velocity profile in the boundary layer must change as the flow moves downstream along the body. Considering a decelerating flow as in Sketch 5.2, at each value of z there will be a reduction in velocity in passing downstream from A to B and this reduction will vary from zero at the wall to ΔV at the edge of the boundary layer. If the pressure gradient is large enough, or is maintained sufficiently far along the surface, then there often comes a point at which the velocity gradient normal to the surface, at the surface, becomes zero. At this point the viscous shear force must also be zero which means that the boundary layer can no longer progress along the surface and thus separates. Downstream of this point there is a region of reversed flow close to the surface as illustrated in Sketch 5.2. A positive pressure gradient acting along a surface is thus called an ADVERSE PRESSURE GRADIENT; a negative pressure gradient is conversely called a FAVOURABLE PRESSURE GRADIENT because a boundary layer is stabilised in these conditions.



Sketch 5.2

It should also be noted that discontinuities in surface slope, if sufficiently large (*e.g.* the sharp edges of many buildings and structures), will also cause the boundary layer flow to separate at the discontinuity.

Downstream of the separation of a boundary layer, the flow outside the separated regions does not follow the contours of the body surface; the region between the separated boundary layer and the surface is filled with an eddying flow in which the velocity and direction vary with time in an almost random manner and bear little or no relation to that of the free stream. In addition, the pressure along a normal to the surface no longer remains independent of distance from the surface over the thickness of the boundary layer.

One of the important adverse effects of separation, when it extends over the rearward facing part of the body, is that the expected pressure rise towards the rear of the body referred to earlier is prevented. A consequent increase in pressure drag results because the area of relatively low pressure on the rearward facing area of the body in the separated flow region acts to produce an increase in drag force.

For streamlined bodies at small angles of incidence (α) in a low speed flow the boundary layer usually leaves the trailing edge smoothly. As the angle of incidence is increased a progressive separation of the boundary layer develops, usually on the upper surface, as the adverse pressure gradient is increased by the increasing incidence. When the streamlined body is an aerofoil the lift suddenly falls beyond a certain angle of incidence and the drag rapidly increases. This is the result of separation of the boundary layer over most of the upper surface and is referred to as "STALLING". For those bluff bodies covered by the Data Items separation, for all practical considerations, always takes place. The exception to this is for a very low Reynolds number flow (which usually implies a highly viscous liquid flow of low velocity). For example, separation in the flow around a circular cylinder does not occur for values of Reynolds number less than about 5.

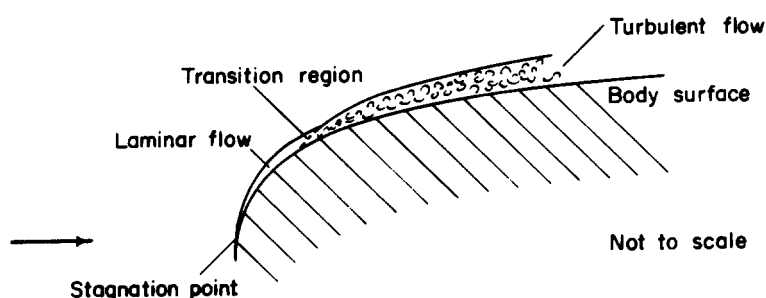
In the study of the flow over buildings it must be noted that the atmospheric wind upstream of the building is itself a boundary layer. The variation with height of its mean velocity and turbulence intensity and scale has several particular effects on the flow. For instance, the flow can never be considered two-dimensional in a vertical plane because significant transverse flows develop around the sides of the building resulting in vertical components of flow. In addition, separation of the “ground” boundary layer occurs just upstream of the forward face of the building with the result that the lower portion of the building close to the ground is engulfed in a separated flow region (see Sketch 5.5). A vortex in front of the building is formed in this separated region and its ends are swept downstream with the result that significant three-dimensional effects are produced.

5.2 Laminar and Turbulent Flow, Transition, Effect on Separation

In the flow over a smooth surface at low Reynolds numbers (which usually imply low velocity) every fluid particle moves with uniform velocity along a uniform path. Adjacent fluid layers slide over each other and only friction forces act between them. There is no macroscopic mixing of fluid elements between layers as in the case of turbulent flow. Viscous forces slow down the particles near the surface in relation to those in the external stream but the flow is well-ordered and is said to be a LAMINAR FLOW. In flows at low to moderate Reynolds numbers the boundary layer at its point of origin is normally laminar. Laminar boundary layers can only exist when disturbances such as turbulence, noise, etc. outside the boundary layer are of low amplitude and do not excite resonances within the layer, when the external pressure gradient is favourable and the surface of the body is sufficiently smooth.

The orderly pattern of laminar flow ceases to exist at higher Reynolds numbers (which usually imply higher velocities) and strong mixing of all the particles occurs. In this case (TURBULENT FLOW) there is superimposed on the main motion a subsidiary eddying motion (turbulence) which causes mixing.

These two flow regimes, laminar and turbulent, and the TRANSITION from laminar to turbulent, can be observed in the boundary layer. Transition in a boundary layer takes place over a range of critical Reynolds number where the characteristic length in the definition of Reynolds number (see Section 7) is distance along the surface from the stagnation point. The range of Reynolds numbers over which transition takes place is itself affected by many parameters, the most important ones being the pressure distribution in the external flow, the roughness of the body surface and the intensity of turbulence in the external flow.



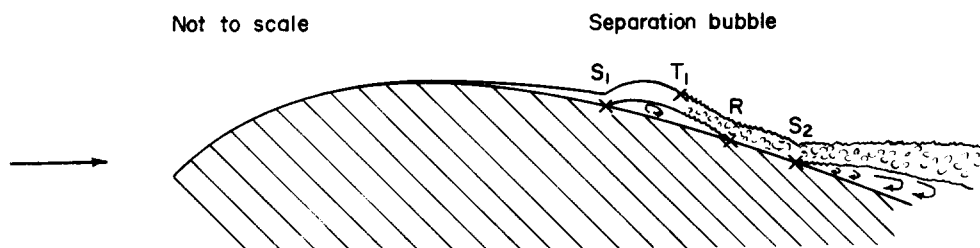
Sketch 5.3

The major effect of the mixing of fluid elements in a turbulent boundary layer, and the consequent inter-change of fluid momentum between layers is that the thickness of the layer increases (because eddy motions redistribute the momentum in the fluid flow between the surface and the edge of the boundary layer). Furthermore, the mixing process causes the addition of an effective shear stress, represented by eddy viscosity (see Section 5.1), to the true viscous shear force, and as a consequence the retarded fluid layers adjacent to the surface can be pulled further along the surface into regions of higher pressure. Thus a turbulent boundary layer is thicker, is able to progress further against an unfavourable pressure gradient and thus first separates at a point further along a surface than would a laminar one under the same conditions.

In flows, such as past streamline shapes, where boundary layer separation does not occur the velocity gradient immediately adjacent to the surface when the boundary layer is laminar is less than when the boundary layer is turbulent: the drag force composed of viscous and pressure forces in the direction of the flow is less in the former case. However, if the boundary layer is laminar and separation occurs, then, for a given free-stream velocity, the drag force is usually greater than when the boundary layer is turbulent, even if the latter also separates. The reason for this is that, on rounded bodies, transition to turbulent boundary layer flow causes the separation point to move downstream, more to the rear of the body, which considerably decreases the width of the wake*. Thus, in this case, the adverse effects on the expected pressure recovery towards the rear of the body referred to in Section 5.1 are confined to a smaller area and hence the drag force is less. On sharp-edged bodies separation is fixed at the forward sharp edge and the drag force is less affected by the state of the boundary layer.

Drag forces on bodies can, as described in the preceding paragraph, depend considerably upon the type of flow. Therefore the results of two experiments, one conducted in laminar flow and the other in turbulent flow, can be considerably different. It is essential to ensure that similar flow regimes occur for comparisons to be meaningful.

In some instances it is possible for the boundary layer flow to be laminar at separation (S_1 in Sketch 5.4) and for transition (T_1 in Sketch 5.4) to occur in the separated boundary layer. The properties of the then turbulent layer may be such that the boundary layer REATTACHES (R in Sketch 5.4) to the surface. Conditions may also be such that this reattached turbulent layer separates again (S_2 in Sketch 5.4). For a direct comparison of experiments, all these phenomena must occur at corresponding positions relative to the model.



Sketch 5.4

It is also possible for a boundary layer which has become turbulent while still attached to separate and then REATTACH at a point downstream: reattachment is not necessarily only associated with transition. When reattachment occurs, the separated region is usually called a SEPARATION BUBBLE and often further designated a laminar separation bubble or a turbulent separation bubble.

* The WAKE of a body is defined as the region downstream of a body where the flow velocity is less than the free-stream value and where there is a loss in momentum corresponding to the drag or resistance of the body to the fluid motion. This is also a region of reduced total pressure and thus measurements of total pressure downstream of a body can be used to define the extent and growth of the wake.

6. THE DETERMINATION OF PRESSURE AND VISCOUS FORCES

6.1 Pressure Forces

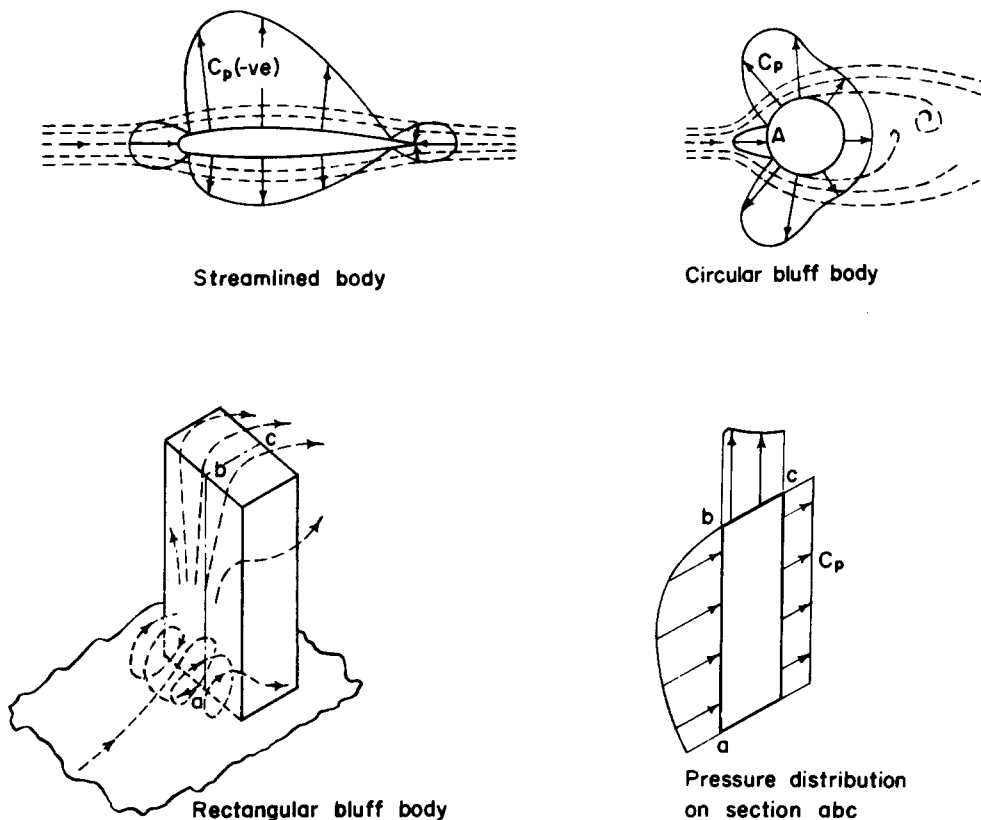
For a steady flow past a body, on which the boundary layer does not separate, the local pressures and velocities are related by Bernoulli's equation (Equation (4.2) for incompressible flow). When the flow far upstream is everywhere uniform with pressure p_∞ and velocity V_∞ Equation (4.2) can be written (for gases)

$$p + \frac{1}{2}\rho V^2 = p_o = p_\infty + \frac{1}{2}\rho V_\infty^2 \quad (6.1)$$

The free-stream TOTAL PRESSURE or TOTAL HEAD, p_o , is also the pressure at the STAGNATION POINT of the body (near the most forward part of the body) where the flow is brought to rest. The difference between the total pressure, p_o , and the static pressure, p , for incompressible flow, is equal to the KINETIC PRESSURE, $\frac{1}{2}\rho V^2$. In compressible flow this difference is no longer $\frac{1}{2}\rho V^2$ and is known as the DYNAMIC PRESSURE.

In most flows at moderate to high Reynolds numbers pressures, forces and moments, made non-dimensional with respect to $\frac{1}{2}\rho V_\infty^2$, $\frac{1}{2}\rho V_\infty^2 A$ and $\frac{1}{2}\rho V_\infty^2 Al$ respectively, vary little with change in velocity for a given body. Thus pressure is expressed as a pressure coefficient, C_p , usually defined as

$$C_p = (p - p_\infty) / \frac{1}{2}\rho V_\infty^2 \quad (6.2)$$



Sketch 6.1 Illustrations of flow patterns and pressure distributions for streamline and bluff bodies

In some cases (e.g. measurements at full-scale conditions) C_p is defined relative to a reference pressure

which is not p_∞ . However, when evaluating pressure forces from an integration of pressures over a surface as in Equations (6.3) and (6.4), C_p must be defined in the form of Equation (6.2). Typical surface pressure distributions are illustrated in Sketch 6.1 for three particular bodies.

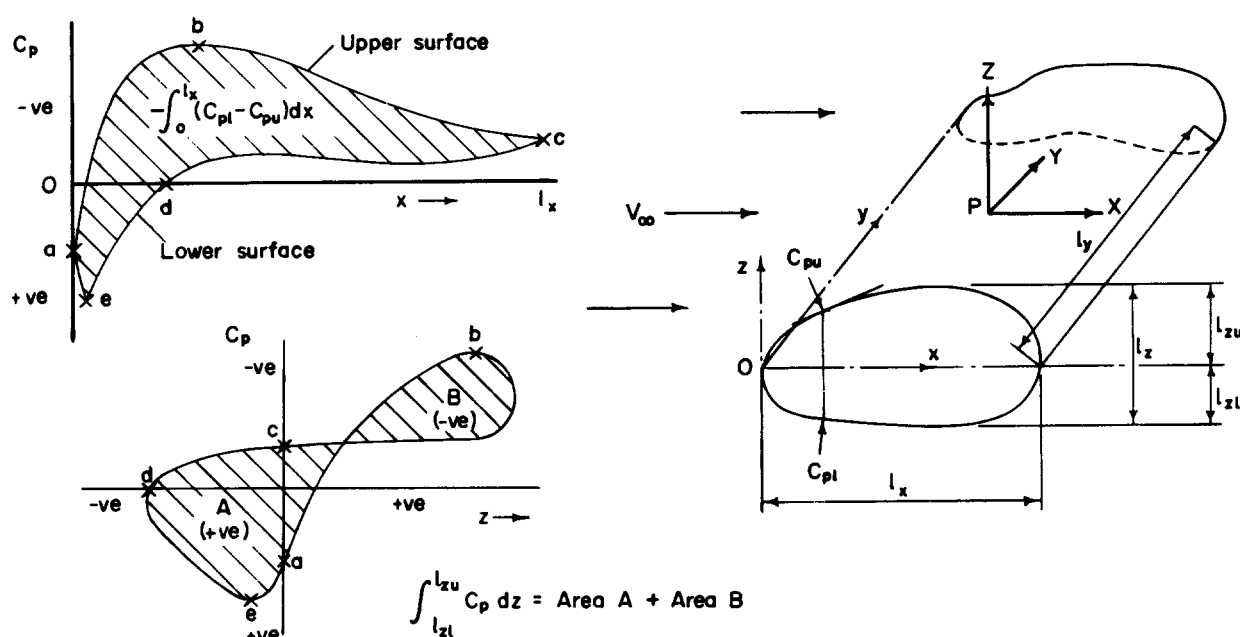
It follows from Equation (6.1) that the maximum positive value of C_p in incompressible flow is 1.0 which is achieved by bringing the flow to rest. The point where this happens is called the STAGNATION POINT (point A in Sketch 6.1). It is easily possible to achieve negative values of C_p greater than 1.0 by accelerating the flow; in fact, on bluff bodies values as large as -2.5 are commonplace and on aerofoils at high incidence even higher values are achieved.

The component of the force on a body in a given direction is found by resolving the local normal force on the body surface in that direction, and integrating over the body surface. It can easily be seen that this is equivalent to an integration of the unresolved pressure over the surface area projected normal to the given direction. Thus, for example, referring to Sketch 6.2,

$$C_X = \frac{1}{l_x l_y} \int_0^{l_y} \int_{l_{zl}}^{l_{zu}} C_p dz dy = \frac{X}{\frac{1}{2} \rho V_\infty^2 l_x l_y}, \quad (6.3)$$

$$C_Z = \frac{1}{l_x l_y} \int_0^{l_y} \int_0^{l_x} C_p dx dy = \frac{Z}{\frac{1}{2} \rho V_\infty^2 l_x l_y}, \quad (6.4)$$

and a similar expression can be developed for C_Y .



Sketch 6.2

In Equations (6.3) and (6.4) l_x , l_y and l_z are the lengths of the body in the x , y and z directions respectively, and X and Z are the forces in the x and z directions respectively. When a body surface can be subdivided into two separate surfaces (e.g. the upper and lower surface of an aerofoil) then the total component force on the body is obtained by subtracting the integrated pressure force in the negative axis direction from that

in the positive axis direction. Thus, in the case illustrated

$$C_Z = \frac{1}{l_x l_y} \int_0^{l_y} \int_0^{l_x} (C_{pl} - C_{pu}) dx dy \quad (6.5)$$

where C_{pl} and C_{pu} are the pressure coefficients on the lower and upper surfaces respectively.

In the case of hollow bodies the net pressure force acting on an element of body surface must be estimated taking into account the fact that the internal pressure may be different from the free-stream static pressure, p_∞ . Thus the mean force *per unit area* acting normally to the face of an element of surface is

$$(\bar{C}_p - C_{p(int)}) \frac{1}{2} \rho V_\infty^2$$

where $\bar{C}_p$ is the integrated mean value of the external pressure coefficient on the surface element and $C_{p(int)}$ is the internal pressure coefficient.

Components of force along the free-stream direction are called DRAG forces and the components of force perpendicular to the free stream, usually in the vertical sense, are called LIFT forces. This follows aeronautical conventions. The force normal to both the free-stream direction and the lift force is called SIDE or LATERAL force. The point within a body through which the total resultant force can be considered to act is called the CENTRE OF PRESSURE (e.g. point P in Sketch, 6.2).

Pressure forces can, as their name suggests, always be found by integrating the measured pressure distributions over the surface. The force component obtained by integrating in the direction of the fluid stream is called the FORM DRAG or the BOUNDARY LAYER NORMAL PRESSURE DRAG. It is necessary to differentiate this from the TOTAL DRAG which includes the VISCOUS FORCE or FRICTION DRAG.

Moment data due to fluid forces on a body are also usually expressed in the form of a non-dimensional coefficient. The MOMENT COEFFICIENT is defined as

$$C_M = \frac{M}{\frac{1}{2} \rho V_\infty^2 A l}$$

where l is a representative body length. If the body centre of pressure position is known then the moment, M , about a specified body axis is given by the sum of the product of all the total component forces along body axes normal to the specified axis and their respective moment arms between the centre of pressure point and the specified axis.

6.2 Viscous Forces

The net VISCOUS FORCE or FRICTION DRAG is obtained by resolving the local viscous stress, which acts in a direction tangential to the surface, in the direction of the free stream and integrating it around the surface of the body.

It is difficult to measure viscous drag separately; it is usually obtained by measuring the total drag of a model in a balance (or by calculating the loss in the momentum in the wake) and then subtracting the form drag, previously derived by integration of the measured pressure distribution around the body, from this total. This can produce poor accuracy when it is a case of subtracting two quantities of almost equal size.

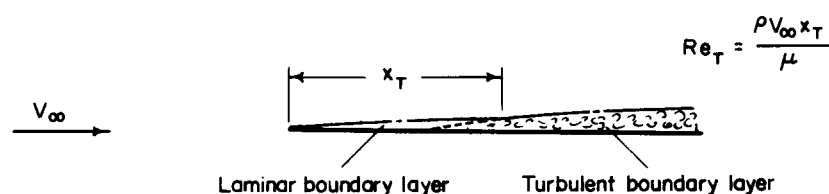
Although a separated boundary layer can and does play an extremely important part in determining the forces on a bluff body, it is usually the case that the actual friction drag is negligible for this class of bodies. The exception to this arises for flows for which the Reynolds number is very small (which usually implies a highly viscous liquid flow) when the friction force forms the major part of the drag.

7. EFFECT OF REYNOLDS NUMBER

The total force coefficient on a body was stated in Equation (3.1) to be a function of several parameters, including the Reynolds number and Mach number. Reynolds number is defined as $Re = \rho V_{\infty} l / \mu$. Now the inertia force acting on a body is of the order $\rho V_{\infty}^2 l^2$ while the viscous force is of the order $\mu (V_{\infty} / l) l^2$. Hence the Reynolds number can also be represented by

$$Re = \frac{\rho V_{\infty} l}{\mu} = \frac{\rho V_{\infty}^2 l^2}{\mu \frac{V_{\infty}}{l} l^2} = \frac{\text{Inertia Force}}{\text{Viscous Force}} \quad (7.1)$$

Reynolds number, together with pressure gradient, surface roughness and free-stream turbulence, prescribes transition. Its value at transition is called the TRANSITION REYNOLDS NUMBER (Re_T). For a smooth two-dimensional flat plate at zero incidence in a fluid stream of negligible turbulence its value is not less than about 5×10^5 based on x_T (see Sketch 7.1).



Sketch 7.1

For bodies without sharp edges, where there is a laminar boundary layer separation and the velocity is increased sufficiently to promote transition before that separation point, the position of separation will be changed. The boundary layer becomes turbulent at the transition point and the flow separation point is transferred downstream, relative to its position had the boundary layer remained laminar, for the reasons given in Section 5.2. This rearward movement of the separation point, as well as its character, has a marked influence on the pressure distribution over the rearward surface of the body with the result that a marked drop in the drag coefficient occurs as illustrated, for example, in Sketch 9.1. The Reynolds number based on the characteristic length of the body (such as the diameter for a circular cylinder) at which this sudden drop in the drag coefficient occurs is called the CRITICAL REYNOLDS NUMBER. It should not be confused with the transition Reynolds number, even though both are associated with the result of transition from laminar to turbulent flow in the boundary layer.

Separation is strongly dependent upon whether the boundary layer is laminar or turbulent and consequently is affected by Reynolds number. Fortunately, in the case of flow around bluff bodies, if either laminar flow predominates or turbulent flow is well established in the boundary layer, the exact value of Reynolds number tends to become unimportant. The reason for this is that if either transition to turbulent boundary layer flow does not occur, or the Reynolds number is sufficiently large that further increases in Reynolds number do not significantly alter the transition point position, then the separation point is essentially fixed and the drag coefficient varies only slowly with change in Reynolds number. In other words, providing the Reynolds number, coupled with surface roughness and free-stream turbulence, is sufficient to produce the same type of boundary layer flow, there are ranges of Reynolds number where the effect of Reynolds number is small. This result is not true, of course, for streamlined bodies on which the boundary layer remains attached.

If the body is sharp edged, separation will occur at the edge and Reynolds number becomes fairly unimportant whatever the state of the boundary layer, but especially in a turbulent stream.

If the body has no sharp edges, recent work would appear to suggest that by the use of surface roughness not to scale, the effects of high Reynolds numbers can be reproduced experimentally at considerably lower values (see, for example, Reference 6).

8. EFFECT OF MACH NUMBER

Mach number is defined as the ratio of the local speed of flow to the corresponding local speed of sound, *i.e.*

$$M = V/a \quad (8.1)$$

where $a^2 = (\gamma p)/\rho$. It is a measure of the effect of compressibility in the flow. When M is small compared with unity the fluid may be regarded as incompressible.

To allow for compressibility, Equations (4.3) and (4.4) must be used instead of Equation (4.2). The expression for pressure coefficient then becomes

$$\begin{aligned} C_p &= \frac{p - p_\infty}{\frac{1}{2}\rho_\infty V_\infty^2} = \frac{p - p_\infty}{\frac{\gamma}{2}p_\infty M_\infty^2} \\ &= \frac{2}{\gamma M_\infty^2} \left\{ \left[1 + \frac{\gamma-1}{2} M_\infty^2 \left(1 - \frac{V^2}{V_\infty^2} \right) \right]^{\gamma/(\gamma-1)} - 1 \right\}. \end{aligned} \quad (8.2)$$

Expanding Equation (8.2) by the binomial theorem and substituting $\gamma = 1.4$ for air it becomes

$$C_p = \left[1 - \left(\frac{V}{V_\infty} \right)^2 \right] + \frac{1}{4} M_\infty^2 \left[1 - \left(\frac{V}{V_\infty} \right)^2 \right]^2 + \frac{1}{40} M_\infty^4 \left[1 - \left(\frac{V}{V_\infty} \right)^2 \right]^3 + \dots \quad (8.3)$$

Clearly, for M_∞ less than about 0.2, it is reasonable to ignore the effects of compressibility, as, for example, in the calculation of wind loads on buildings and structures.

On slender wings and bodies at small incidences, an approximate inviscid flow theory due to Prandtl and Glauert defines the relation between the pressure coefficient in compressible (C_{pc}), and incompressible flow (C_{pi}), at corresponding points on the same body, in the form

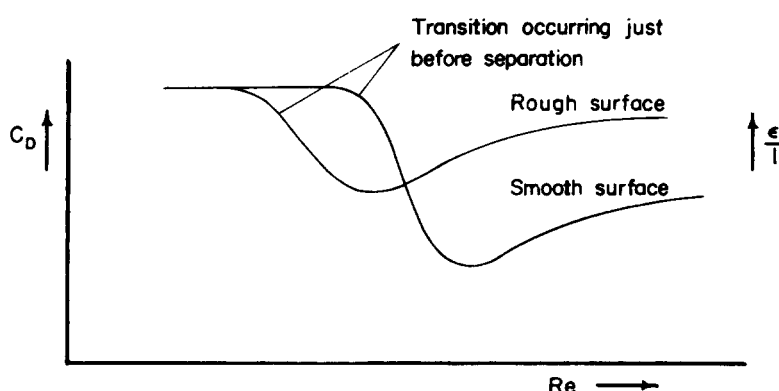
$$C_{pc} = C_{pi} \times \frac{1}{\sqrt{1 - M_\infty^2}}. \quad (8.4)$$

Equation (8.4) can be used in practice provided the maximum local Mach number in the flow is less than unity and no separation of the boundary layer occurs.

For bluff bodies a simple relation does not in general exist between C_{pc} and C_{pi} . In addition, for values of M_∞ of the order 0.45 and greater, the flow becomes supersonic over part of the body surface and shock waves develop. The actual flow and pressure distribution must be obtained from experiment.

9. EFFECT OF SURFACE ROUGHNESS

The effect of surface roughness is to cause transition from laminar to turbulent boundary layer flow to occur at a lower Reynolds number than if the surface is smooth. The consequent effect of this is usually to increase the drag coefficient, compared with the smooth surface value, but in some circumstances, over certain ranges of Reynolds number, the reverse happens as illustrated in Sketch 9.1. In this example of a non sharp-edged bluff body, at sufficiently low Reynolds numbers the laminar boundary layer separates early in its development, the wake is wide and the pressure drag is large. However, if, for the same Reynolds number, transition to turbulent boundary layer flow is provoked by an increased surface roughness (or the addition of a transition wire to the model) before separation occurs then the boundary layer is turbulent and therefore remains attached further round the body surface than the laminar boundary layer. Consequently, the wake width and the pressure drag are reduced and a net reduction in the drag coefficient, over the smooth surface value, occurs.



Typical variation of drag coefficient with Reynolds number and surface roughness for a rounded bluff body

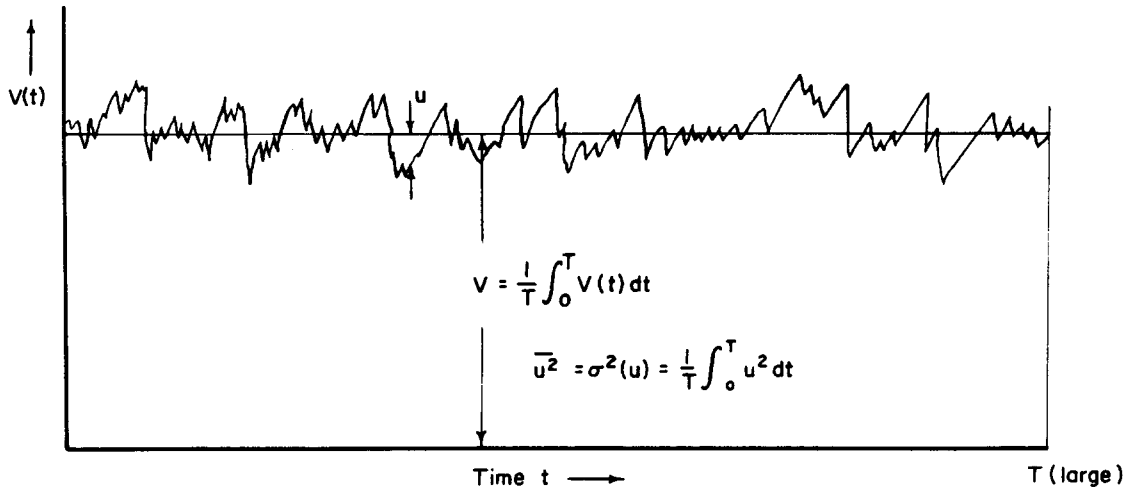
Sketch 9.1

In practice, surface roughness varies from one body to another and depends on material texture, surface finish, extent of corrosion and the build up of deposits (e.g. scale, rust, ice, etc.). For the purposes of estimating drag forces it is convenient to define an equivalent surface roughness height, ϵ . The equivalent roughness height of a rough body refers to the size of uniform particles evenly distributed over the smooth surface of a geometrically identical body which gives the same resistance to motion under identical flow conditions as the naturally rough body. It is usually assumed that the equivalent roughness height is independent of Reynolds number so that the ratio ϵ/l is a non-dimensional parameter influencing the value of the drag coefficient.

10. FREE-STREAM TURBULENCE

The flow mechanisms through which varying degrees of free-stream turbulence can affect the mean forces (and the fluctuating forces due to vortex shedding) acting on a body are mentioned in Sections 5.2 and 11.1. The fluctuating velocity component in a turbulent free stream (as illustrated in Sketch 10.1) will also produce on a body fluctuating forces which vary with time about a mean value. A knowledge of the structure of the turbulence in terms of its energy distribution (power spectrum) is important in determining the nature of both the mean and the fluctuating forces. If the body is large compared with the scale of turbulence then gust velocities produced by individual turbulent eddies will not occur simultaneously over the body. The gust velocities are then not fully correlated over the body. Thus a knowledge of both the power spectrum, and the correlation functions with respect to time and space, is necessary in order to describe the nature and spatial characteristics of turbulence. These properties can be defined in statistical terms and in the

following Sections a description of the relevant parameters is given as an aid to the understanding of the concepts and terminology which are used to describe the properties of turbulence.



Sketch 10.1

10.1 Power Spectrum

The fluctuating velocity component in free-stream turbulence is random in character but can be regarded as being compounded of oscillations of cosine form of varying amplitude (b) and frequency (n), i.e. it can be represented by a Fourier cosine series of the form

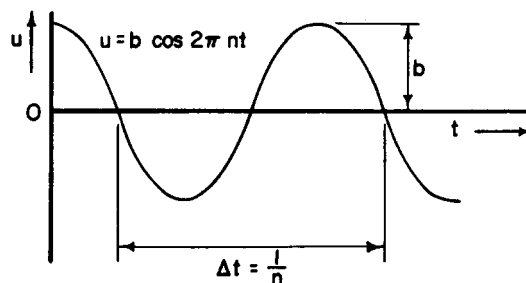
$$u = \sum_{n=1}^{\infty} b_n \cos(2\pi n t) \quad n = 1, 2, 3, \dots \quad (10.1)$$

where

$$b_n = 2 \int_0^{\infty} u \cos(2\pi n t) dt.$$

For a simple fluctuating velocity of the cosine form

$$u = b \cos(2\pi n t) \quad (10.2)$$



Sketch 10.2

the mean square of the fluctuating velocity, $\overline{u^2}$, sometimes called the VARIANCE, is given by

$$\overline{u^2} = \sigma^2(u) = \frac{1}{\Delta t} \int_0^{\Delta t} u^2 dt = \frac{b^2}{2}.$$

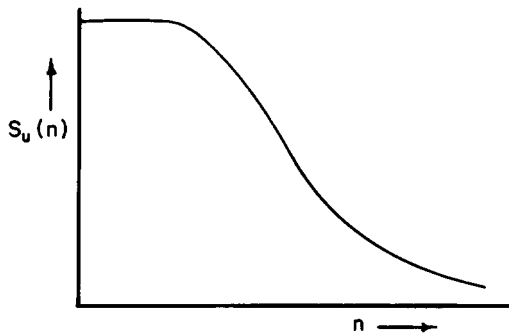
This is also a measure of the kinetic energy or average power contained in the fluctuations. For a series of compounded fluctuations of cosine form the variance must be given by

$$\sigma^2(u) = \sum_{n=1}^{\infty} \frac{b_n^2}{2}.$$

In practice, in random turbulence there will be so many individual frequencies that they can be considered to exist as a continuous range of frequencies. The power spectrum can then be defined as

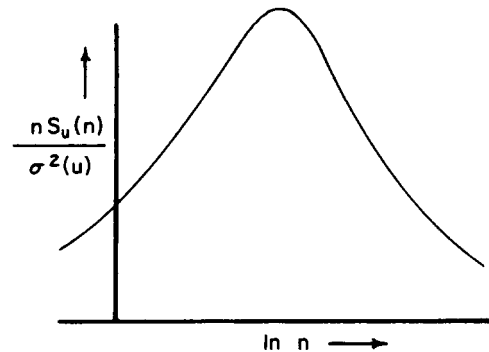
$$\int_0^{\infty} S_u(n) dn = \sigma^2(u) \quad (10.3)$$

and this is also a measure of the total energy present in the fluctuations. The quantity $S_u(n) \cdot \delta n$ is a measure of the energy associated with that component within the narrow frequency range between n and $n + \delta n$ and $S_u(n)$ is known as the POWER SPECTRAL DENSITY of fluctuating velocities, u , at frequencies n . A typical distribution of the power spectral density function is shown in Sketch 10.3.



Power spectral density function

Sketch 10.3



Normalised power spectral density function

Sketch 10.4

If Sketch 10.3 is replotted as $n S_u(n) / \sigma^2(u)$ against $\ln n$ as in Sketch 10.4 then the area under this curve is

$$\frac{1}{\sigma^2(u)} \int_{-\infty}^{\infty} n S_u(n) d(\ln n) = \frac{1}{\sigma^2(u)} \int_0^{\infty} S_u(n) dn = 1 \quad (10.4)$$

and the curve is called the NORMALISED POWER SPECTRAL DENSITY FUNCTION.

Now the local turbulent velocity is a vector and has components in the three directions x , y and z , and so the foregoing applies to the three velocity components u , v and w in turn and there exist power spectral density functions $S_u(n)$, $S_v(n)$ and $S_w(n)$. When the statistical properties of turbulence become the same in the three directions, so that $\overline{u^2} = \overline{v^2} = \overline{w^2}$, the turbulence is described as ISOTROPIC. Atmospheric turbulence is normally anisotropic but in certain cases (away from the immediate vicinity of the ground) an isotropic model of turbulence can be employed.

10.2 Time Correlations

Time correlation functions, such as the autocovariance and autocorrelation functions, describe the time scale of the random fluctuating component of turbulence and are a correlation of pairs of fluctuating quantities measured at a point in space at times t and Δt .

The AUTOCOVARIANCE is obtained by measuring instantaneous fluctuating components at a given point in space over a period of time and then taking the mean value of the product of pairs of fluctuating values measured at times t and $t + \Delta t$. Thus the autocovariance is defined as

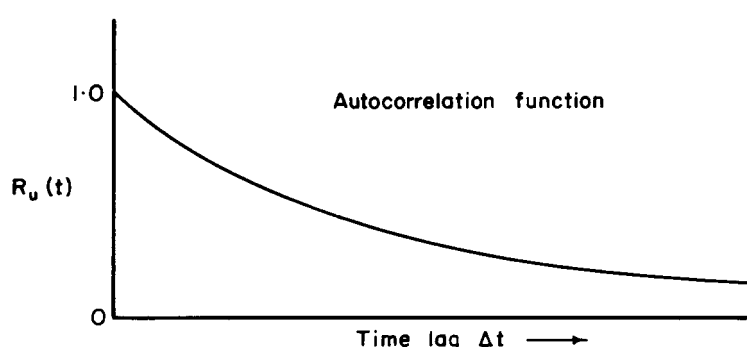
$$\overline{u_t \cdot u_{t + \Delta t}}.$$

When the time lag is zero the autocovariance is equal to the variance, $\sigma^2(u)$.

When the autocovariance is normalised by dividing by the variance, the resulting quantity is called the AUTOCORRELATION coefficient, $R_u(t)$, where

$$R_u(t) = \frac{\overline{u_t \cdot u_{t + \Delta t}}}{\sigma^2(u)}. \quad (10.5)$$

The variation of $R_u(t)$ with Δt is called the autocorrelation function.



Sketch 10.5

Thus the autocorrelation function, or more precisely the integral

$$\int_0^{\infty} R_u(t) d(\Delta t)$$

is a measure of the time interval over which there exists a dependency between the mean values of the fluctuating component at a point in space. When $R_u(t)$ is close to unity then the measured pairs of values of the fluctuating component can be considered to occur in the same average eddy. When Δt is large and $R_u(t) \approx 0$ then the behaviour patterns of each of the paired values are independent of each other and there is no correlation between values.

It can be shown (References 1, 2) that the autocorrelation function and the power spectral density function are related by a pair of simple Fourier transformations, *i.e.*

$$S_u(n) = 4\sigma^2(u) \int_0^{\infty} R_u(t) \cos(2\pi nt) dt$$

and

$$R_u(t) = \sigma^2(u) \int_0^{\infty} S_u(n) \cos(2\pi nt) dn.$$

10.3 Space Correlations

The autocorrelation function, which describes the time-sequential properties of turbulence at a point in space, is satisfactory for bodies which are small in relation to the spatial scale of turbulence. However, this reveals nothing about the random behaviour of turbulence in space and this characteristic is particularly important for bodies which are large compared with the spatial scale of turbulence.

If the fluctuating component, u , is measured simultaneously at two points in the flow and these instantaneous fluctuating values u_1 and u_2 are multiplied together, the time averaged value of the product is called a CORRELATION (r): e.g.

$$r_u = \overline{u_1 \cdot u_2}. \quad (10.6)$$

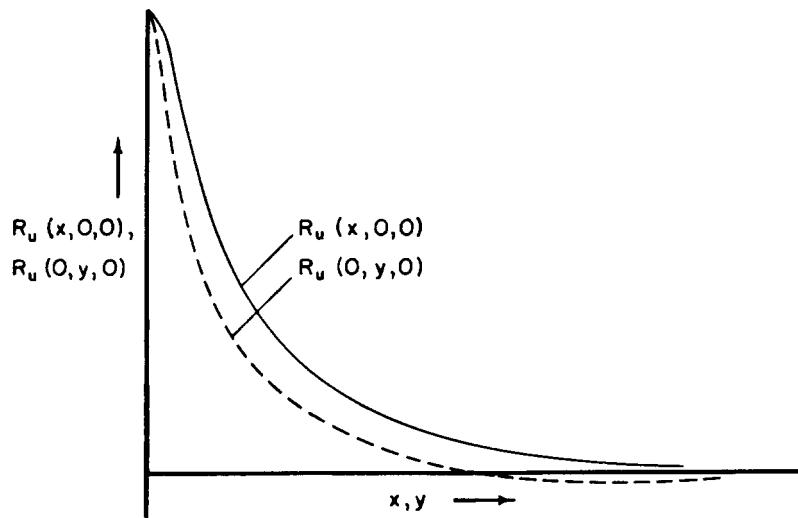
It is advantageous to have this quantity in non-dimensional form, so it is divided by the product of the root mean square values at the points to become a CORRELATION COEFFICIENT,

$$R_u = \frac{\overline{u_1 \cdot u_2}}{\sqrt{\overline{u_1^2}} \sqrt{\overline{u_2^2}}}. \quad (10.7)$$

However, the relative location of the two points must be known. It is therefore better to write, for example,

$$R_u(x, 0, 0) = \frac{\overline{u_1 \cdot u_2}}{\sqrt{\overline{u_1^2}} \sqrt{\overline{u_2^2}}} \quad (10.8)$$

so that it is obvious that the two points are located x apart in the x direction with $y = z = 0$. If the two points are close together and the value of R_u is close to unity, then this implies that the variations, with time, of the fluctuating components u_1 and u_2 at the two points are related in some way and that the two points can be considered to occur in the same average eddy. On the other hand, if the two points are far apart compared with the scale of turbulence then the time-dependent behaviour pattern of u_1 and u_2 at the two points will not be linked and the value of R_u is close to zero.



Sketch 10.6

If $R_u(x, 0, 0)$ is measured for a large number of values of spacings x and plotted as in Sketch 10.6, the area under the curve has units of length and is called the INTEGRAL LENGTH SCALE, $L_x(u)$ where

$$L_x(u) = \int_0^\infty R_u(x, 0, 0) dx. \quad (10.9)$$

This gives some idea of the size of eddies of u in the x direction. There will be nine scales in all, permutating x, y, z and u, v and w . For similar types of turbulence, a knowledge of the value of length scale is useful, but it must be appreciated that different power spectral density functions can be consistent with the same integral length scale.

When the turbulence is isotropic then the relationship between the various correlation coefficients is such that

$$\text{and} \quad \left. \begin{aligned} L_x(u) &= 2L_y(u) = 2L_z(u) \\ L_x(u) &= L_y(v) = L_z(w) \end{aligned} \right\} \quad (10.10)$$

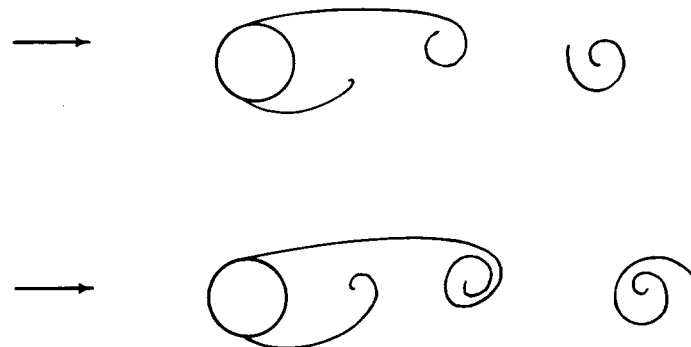
For the purpose of estimating the effect of free-stream turbulence on the magnitude of the mean forces acting on a body, the turbulence is often defined simply by the INTENSITY $((u^2)^{1/2}/V_\infty, \text{ etc.})$ and the integral length scale.

11. VORTEX SHEDDING

11.1 Two-Dimensional Flow

The boundary layer that forms on a body immersed in a fluid continues downstream of the body in the form of a wake – a region of strongly retarded flow. On streamlined bodies, for which the flow remains attached along the whole body length, the wake is of narrow width. On bluff bodies, from which the boundary layer separates prematurely, the phenomenon of separation is associated with the formation of vortices and a large energy loss in a wide wake. At moderate Reynolds numbers the flow in the wake of a bluff body is dominated by a periodic train of alternating vortices (see Sketch 11.1) known as the KÁRMÁN VORTEX STREET. The boundary layers that separate from the two sides of the body are separated by a region of the order of the thickness of the body. The boundary layers tend to roll up in this region producing large vortices which, when they have achieved a certain size, separate from the body and move downstream. These vortices are much larger than the boundary layer immediately ahead of the separation point, approaching the size of the body generating them, and producing fluctuations of large scale (and low frequency) in the fluid flow. In certain regions of Reynolds number, this shedding process is inhibited and a coherent set of vortices is not produced although a random shedding process of smaller scale still occurs.

The mechanism of shedding has been established and is found to be the interplay between the boundary layers from either side of the body. If this interaction is prevented by any means, physical or fluid, then the “clock” mechanism described below stops and the regular vortex shedding breaks down into general turbulence.



Sketch 11.1

The boundary layer from one side grows in the wake behind the body and eventually starts to entrain fluid from the boundary layer from the opposite side. This slows down the growth of the first vortex and drags the second boundary layer across behind the body. The fully grown vortex then separates from the boundary layer and moves downstream, and the second boundary layer, which has moved across behind the body, begins to grow into a vortex. It is essential for this development for there to be free access across the wake at the rear of the body. Should a restriction be placed here, the shedding phenomenon cannot occur.

The alternating frequency of vortex shedding is not a discrete value but a narrow band of values. The central or predominant frequency of the narrow band of values is usually fairly easy to define and this frequency, presented in a non-dimensional form, is called the STROUHAL NUMBER,

$$St = \frac{fl}{V_{\infty}}.$$

For a circular cylinder normal to the flow, or a rectangular plate with its surface normal to the flow, the body dimension, l , in the definition of Strouhal number is taken as the cylinder diameter and plate width respectively. For a very long (two-dimensional) circular cylinder in incompressible flow the Strouhal number is approximately 0.2 for subcritical Reynolds numbers (see Section 7) and for a surface-mounted flat plate the Strouhal number has a normal value of about 0.14.

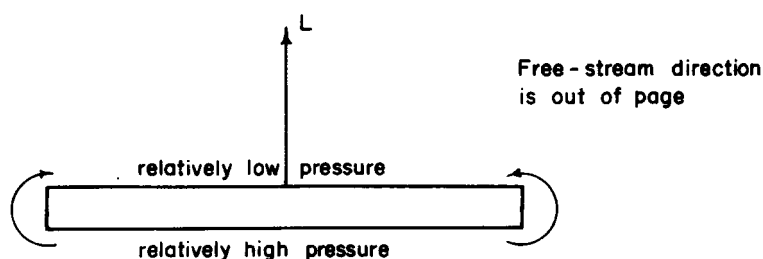
The process of shedding vortices alternately from each side of a bluff body causes a fluctuating lateral force (normal to the drag force) to be exerted on the body through changes in the pressure distribution on the body itself. The alternating frequency range of the fluctuating force is the narrow band of values centred around the Strouhal number frequency, f . The magnitude of the time-dependent alternating lateral force can be very considerable (of the same order as the mean drag force) and the dynamic effect on a structure which is free to move or is lightly damped can be very significant.

The effect of free-stream turbulence on vortex shedding has not been systematically investigated but increasing degrees of free-stream turbulence do not change the predominant frequency of vortex shedding but do increase the bandwidth of frequencies. It also decreases the peak value of the fluctuating force found at the Strouhal number frequency.

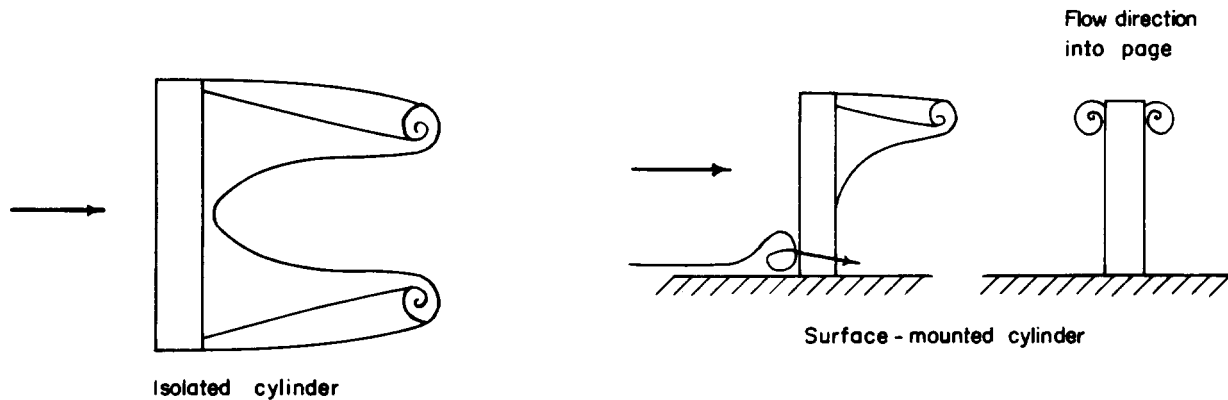
11.2 Three-Dimensional Flow, Trailing Vortices

TRAILING VORTICES (see Sketch 11.3) are generated at the free ends of a finite-length cylinder which is producing a lift (or side) force normal to both the free-stream direction and the cylinder major axis. They are due essentially to the three-dimensional nature of the flow.

A simple physical explanation of how the trailing vortex system is generated can be obtained by considering the flow development near the ends of a cylinder. Assuming that the cylinder is producing a lifting force in the conventional sense (Sketch, 11.2) then this implies that, on average, pressures on the upper surface are less than those on the lower surface. Thus, because there can be no discontinuity in pressure between upper and lower surfaces adjacent to the ends of the cylinder, there will be a flow of fluid around the ends of the cylinder from the relatively high pressure region on the lower surface to the lower pressure region on the upper surface as illustrated in Sketch 11.2. This lateral motion of fluid, compounded with the forward motion of the free stream, induces a spiral or vortex motion in the flow over the cylinder with the intensity of vorticity (rotational motion) being strongest at the cylinder ends. A result of this is that, as the vortex sheet (formed by the separated boundary layers shed from the sides of the body) moves downstream, it tends to curl up at its free ends into discrete vortices as illustrated in Sketch 11.3.



Sketch 11.2



Sketch 11.3

The energy expended to generate the trailing vortex system manifests itself as a drag force, called the INDUCED or VORTEX DRAG. This drag force appears by way of a modification to the surface pressure distribution appropriate to two-dimensional flow past an equivalent infinitely long cylinder at the same inclination to the free stream.

Clearly, for the flow around an infinitely long cylinder (two-dimensional flow) the trailing vortex system is not generated because the lateral movement of fluid due to the end effects is eliminated (although vortex shedding of the kind described in Section 11.1 may occur) and the induced drag is zero. For streamlined cylinders (such as aircraft wings) the magnitude of the induced drag can be theoretically estimated in many cases (see, for example, References 3 to 5). For bluff cylinders, on which substantial regions of separated flow occur, successful theoretical treatment is not possible and the induced drag is included as part of the total pressure drag obtained by integrating the measured pressure distribution over the whole cylinder as in Equation (6.3).

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13. KEY WORD INDEX

For convenience, a short key word index is given to assist the user of this Item, and other Data Items, to locate particular passages which are of background interest. It is not intended to be a complete index and the Engineering Sciences Data Index should also be consulted for reference to related Data Items in which specific data can be found.

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THE PREPARATION OF THIS DATA ITEM

The work on this particular Item was monitored and guided by the Fluid Mechanics Steering Group which has the following constitution:

Chairman	
Mr W.F. Wiles	– Rolls Royce, 1971, Ltd, Hucknall
Members	
Mr E.C. Firman	– Central Electricity Research Laboratories
Mr B.H. Fisher	– Consulting Structural Engineer
Dr G. Hobson	– English Electric – AEI Turbine Generators Ltd
Mr T.V. Lawson	– University of Bristol
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Mr C. Scruton	– National Physical Laboratory.

The Item was accepted for inclusion in the Aerodynamics Sub-series by the Aerodynamics Committee which has the following constitution:

Chairman	
Prof. G.M. Lilley	– University of Southampton
Vice-Chairmen	
Prof. D.W. Holder	– University of Oxford
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The technical work involved in the assessment of the available information and the construction and subsequent development of the Data Item was undertaken by

Mr N. Thompson	– Head of Wind Engineering Group.
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