THE DESIGN OF LOW-SPEED WIND TUNNELS*

P. BRADSHAW and R. C. PANKHURST
National Physical Laboratory, Teddington

Summary. Aerodynamic and structural design is discussed from the viewpoint of the prospective tunnel designer, and details given of present-day practice. Comments, mostly cautionary, are made on the features of certain existing tunnels, and drawings and tabulated data are given for about thirty typical tunnels of various ages. The emphasis is on low-speed tunnels of wooden construction, but much of the discussion is applicable to the design of tunnels of larger size or higher speed.

LIST OF SYMBOLS

Notation

The fan notation used here is that of Wallis (6).

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>A</td>
<td>area (or area ratio, taking working section as unity)</td>
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<tr>
<td>c</td>
<td>chord (fan blade, corner vane: conventionally measured on a straight line joining LE and TE)</td>
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<tr>
<td>D</td>
<td>drag</td>
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<tr>
<td>d</td>
<td>diameter, especially of screen wires</td>
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<td>H</td>
<td>power input</td>
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<td>h</td>
<td>gap between adjacent vanes or blades, measured along a straight line or arc joining the leading edges</td>
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<tr>
<td>K</td>
<td>screen drag coefficient</td>
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<td>l</td>
<td>distance between screen wires</td>
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<tr>
<td>M</td>
<td>Mach number</td>
</tr>
<tr>
<td>N</td>
<td>number of blades or vanes</td>
</tr>
<tr>
<td>n</td>
<td>fan rotational speed, rev/sec</td>
</tr>
<tr>
<td>P</td>
<td>total pressure</td>
</tr>
<tr>
<td>p</td>
<td>static pressure</td>
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<tr>
<td>Δp</td>
<td>pressure difference</td>
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<tr>
<td>q</td>
<td>dynamic pressure</td>
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Wind tunnel design lies somewhere between an art and a science, with occasional excursions into propitiatory magic. Until there is a better understanding of the behaviour of turbulent shear layers in diffusers, fans and corners, it will be impossible to lay down exact principles. In the last decade or so, however, safe—or reasonably safe—rules of design have become well established, and there is now no reason why the performance of a general-purpose tunnel should fall far short of expectations, although one cannot be as confident about tunnels with special features.
In this review, we have attempted to describe the process of design of low-speed tunnels in sufficient detail for the prospective tunnel designer, and to collect the rather sparse data on existing tunnels as a demonstration of what can, and cannot, be done. The list of references is intended to be adequate but not exhaustive and is divided into sections corresponding to the sections of the report: the reader in search of more general information should consult Ref. 1.

2. HISTORY OF WIND TUNNEL DESIGN

The expense and risk of testing new ideas in full-scale flight soon became apparent to aeronautical pioneers at the end of the last century, and alternative testing methods were developed. Free-flight model testing, and the use of whirling arms and wind tunnels, became common. Except for certain stability investigations, the whirling arm quickly lost favour because of the difficulty of making reliable force and pressure measurements, quite apart from the very severe limitations on Reynolds number and Mach number, and free-flight model testing languished for the same reasons until the advent of radio control and telemetry. Powered-lift schemes, with novel stability and control problems, are well suited to free-flight model testing, but in general the accuracy of flight test data, whether model or full size, tends to be much lower than that of wind tunnel test results, and the experimental conditions are far less easily controlled, so that the wind tunnel has remained the foremost instrument of aeronautical research.

The general progress of wind tunnel design can be inferred from Table 1 and Figs. 9–34, which show low-speed tunnels designed at intervals over the last 40 years and illustrate the development of different methods of satisfying the conflicting requirements of an airstream uniform in space and time and a reasonably small dissipation of power in smoothing devices. Although the desirability of spatial uniformity and steadiness must have been realized from the first, even if means of ensuring them were not so immediately apparent, it was some time before the effect of turbulence on boundary-layer transition and separation was fully appreciated, for the true state of affairs was obscured by the observation that increasing the tunnel turbulence often had the same effect as increasing the Reynolds number (i.e. by hastening transition to a turbulent boundary layer and so forestalling laminar separation). Since the 1930’s, low-turbulence tunnel design has been improved to the point where spatial irregularities are determined by the uniformity of flow through the last screen, and temporal irregularities by the noise and vibration of the fan and the turbulence of the working section boundary layers. Further advances are likely to be in the direction of increasing compactness—by the use of boundary-layer control, for instance—and increasing accuracy of design, rather than increased airstream regularity.
The inner wall outline is known as the airline to distinguish it from the constructional outline.

Fig. 1. Proposed NPL 36 in. x 27 in. tunnel.
Descriptions of some modern wind tunnels are given in Section 15, with comments on any undesirable features. Figure 1 shows a design for a projected general-purpose tunnel at NPL, in which the various components are labelled: where several names are given, the first is the most commonly used.

3. TYPES OF TUNNEL AND THEIR USES

3.1. The use of wind tunnels for model testing

Essentially, a wind tunnel is a device for producing an airflow relative to the body under test; in nearly all cases it is immaterial whether the body, or the fluid, would normally be at rest. Usually the airflow is intended to be uniform in speed and direction, but occasionally a wind gradient, or a curved flow, has to be simulated. In addition to models of aircraft, bodies tested in wind tunnels have ranged from racing cyclists (full scale) to the Rock of Gibraltar (model), and wind tunnel experiment seeks to reproduce as closely as possible in the laboratory all the main features of the flow problem concerned. Besides possessing geometrical similarity the test body may sometimes have to reproduce the prototype distributions of mass and stiffness, as in aeroelasticity experiments. Likewise the airstream must satisfy certain requirements, notably that the conditions on its boundaries shall correspond sufficiently closely to those of the real flow, or at least that the discrepancies caused by failure to do so shall be calculable so that corrections can be applied to the test results.

The conditions of dynamical similarity are easy to state but more difficult to satisfy: the most common dimensionless groups in aerodynamics are the Reynolds number $Re = \rho U l/\mu$, the Mach number $M = U/a$, the Strouhal number $S = \omega l/U$ and the ratio of specific heats $\gamma = c_p/c_v$. This last is a property of the fluid and is very nearly 1.4 for all diatomic gases. The Reynolds number is proportional to the ratio of a typical inertia force to a typical viscous force and strongly influences the behaviour of boundary layers and wakes. The Mach number is proportional to the square root of the ratio of a typical dynamic pressure to the absolute pressure of the fluid and strongly influences density changes, especially in shock waves. In cases where the Mach number is not small enough for the fluid to be treated as incompressible it must usually be reproduced exactly, but it is often less essential to reproduce the exact full-scale Reynolds number: usual practice is to attain a Reynolds number which is a respectable fraction of full scale, to use artificial means of producing transition to turbulence in the boundary layer if the full-scale boundary layers are turbulent, and to make allowances for scale effect on the surface friction and other quantities whose variation with Reynolds number is known.

Other dimensionless groups are the Froude number $U/\sqrt{g(l)}$ which is important in free-flight manoeuvre testing and spinning tunnels, the ratio of
elastic modulus to dynamic pressure, $E/\rho U^2$, and the variety of groups involved in heat transfer, such as the Nusselt number or non-dimensional heat transfer coefficient, $Q/k\Delta T$, which is a function of Prandtl number $Pr = \mu c_p/k$; but by far the most important in low-speed experiment where the Mach number is very much less than unity is the Reynolds number, $\rho U l/\mu$. Neglecting for the moment such artifices as altering the density or viscosity of the working fluid (tunnels using water, high pressure air or other fluids, have been built with great success but the installations are necessarily complex), one sees that for exact similarity the speed of the flow should be inversely proportional to the size of the model. However, a limit to the speed of “low-speed” tunnels is set by the requirement that the Mach number shall be well below unity: in any case, the power consumption of a tunnel increases as the cube of the speed, and the loads on the tunnel structure and the model increase as the square of the speed, so that speeds of more than about 300 ft/sec are rare in low-speed tunnels. It is fortunate for aeronautical purposes that the most unpleasant manifestations of scale effect occur at Reynolds numbers nearly two orders of magnitude less than those typical of full-scale flight: at these low Reynolds numbers, extensive areas of laminar flow occur on the model so that laminar boundary-layer separation is liable to falsify the results. The assumption which underlies most wind tunnel testing is that laminar separations do not occur at full scale and therefore that full-scale conditions can be simulated by provoking transition artificially. This is possible if the Reynolds number is not so low that unduly large disturbances would be necessary to cause transition, with consequent gross distortion of the boundary layer profile. No rigid rule for the lowest acceptable Reynolds number can be given, but it is suggested that the laminar boundary-layer momentum thickness Reynolds number, $U\delta_0/\nu$, at the pressure minimum on the body, should be at least 320, which is Preston’s value for the minimum at which a fully-developed turbulent boundary-layer can exist. Thus, transition can be precipitated before the pressure minimum with a trip (wire, roughness band or similar device) of moderate size. This implies that the Reynolds number based on distance from the stagnation point to the pressure minimum should be at least $3 \times 10^6$, say a distance of 4 in. at 150 ft/sec. From this criterion one can estimate the size of model required at a given windspeed and thence the size of tunnel necessary to accommodate it. As a general rule, accurate quantitative work on three-dimensional aerofoils and other bodies is not very easy in tunnels with a product of working section diameter (feet) and speed (ft/sec) of less than 300, or $U d/\nu = 2 \times 10^8$, but smaller tunnels may be adequate for fundamental research work and teaching purposes. (In what follows we shall occasionally refer to “diameter” of parts of a tunnel when we mean “the diameter of a circle of the same area as the tunnel cross-section”.) The point of these remarks is to show the importance of trying to forecast accurately the sort of work to be done in a projected
tunnel, and of taking into account the Reynolds number required, the space and power available, and also the optimum model size for ease and accuracy of manufacture.

3.2. Types of wind tunnel and their special features

The simplest type of wind tunnel is the open-circuit or NPL arrangement (Fig. 20). Reference 56 discusses the special design problems of this layout, and most tunnels with working sections less than about 2 ft in diameter have open circuits. Nearly all large wind tunnels however, have closed circuits with cascades of corner vanes (Fig. 1). The reasons for this distinction are not entirely aerodynamical, and although the average closed-circuit tunnel has a better flow and power factor (defined as the ratio of input power to the rate of kinetic energy flux in the working section) than the average open-circuit tunnel, purely aerodynamical reasoning would lead one to expect the reverse to be the case. If identical airline arrangements of contraction, working section and diffuser were constructed (i) as an open-circuit and (ii) as a closed circuit tunnel, type (i) would be expected to have a lower power factor (providing that the diffuser exit losses were reasonably small) because of the absence of corner vane resistance, and lower working-section turbulence because of the smaller turbulence of the airstream entering the settling chamber.

A large open-circuit tunnel would be of rather inconvenient dimensions—the NPL 13 ft × 9 ft tunnel (Fig. 15) would be 285 ft × 26 ft in plan view (not counting a bellmouth) instead of the present 130 ft × 57 ft—and even small open-circuit tunnels often have their design restricted by the need to save space. However, the Indian Institute of Science at Bangalore has a 14 ft × 9 ft open-circuit open-air tunnel (68) (Fig. 31) with a maximum speed of 350 ft/sec and a power factor of about 0.15.

Open-circuit tunnels are sensitive to draughts and obstructions in the tunnel room and since the static pressure in the working section is necessarily less than atmospheric they suffer badly from the inevitable leaks, because tunnel working sections tend to become perforated with holes made during experiments for the introduction of probes, balance struts or pipelines and perfunctorily made good afterwards. If the return flow through the tunnel room is to arrive at the bellmouth with acceptably small swirl and unsteadiness, straightener vanes should be fitted after the fan just as in a closed-circuit tunnel, and a honeycomb partition across the tunnel room may be necessary. Plenty of space should be left between the tunnel and the walls, floor and ceiling of the room: Gibbings (69) has designed annular entry sections, for open-circuit tunnels installed in close proximity to a wall and to one another, which have guide vanes used as fan pre-rotation vanes in the original design but which could also be used to eliminate unwanted mean swirl in the entry stream. This design of annular entry has proved satis-
factory for two similar tunnels side by side and only about five working-section diameters apart.

Open-circuit tunnels with the fan upstream of the settling chamber and working section are called blower tunnels (Figs. 34 and 36). They are used for cascade tests, boundary-layer studies and other work in which the flow leaving the working section is greatly disturbed. Usually a diffuser is omitted: this enables liberties to be taken with the working-section arrangements at the cost of a power factor necessarily greater than unity.

Most modern tunnels have closed working sections but sometimes an "open jet" is employed (Fig. 10). Access to the model is easier but there are no other great advantages except for special purposes. The power factor of an open jet tunnel is also higher, because stagnant air is entrained into the flow from the nozzle by turbulent mixing at the perimeter of the jet and momentum is lost in the compensating outflow when the jet enters the collector. It is found from mixing layer data that the loss of momentum is equivalent to a "surface friction coefficient" of 0.020. To this must be added any loss of momentum caused by poor collector design, or by inefficiency of the first diffuser which has an entry boundary-layer momentum thickness of at least 0.01 times the working section length. Thus the power factor contribution of the NPL CAT working section jet, 6 ft in diameter and 7½ ft long, is at least 0.1 compared with about 0.015 for a similar closed working section ($C_f = 0.003$): the working section of an open jet tunnel is usually kept short, but this is inconvenient, for instance for high-lift testing, and leads to trouble with axial pressure gradients arising from the flow into the collector.

General-purpose wind tunnels are required to accommodate models of aircraft or other structures, or their components, and the working section size and shape are usually dictated by the need to minimize tunnel interference for a predetermined model size. Sometimes half-models of symmetrical structures are tested, often mounted on a false wall sufficiently far from the tunnel wall to be well clear of the thick wall boundary layer. Several tunnels with quasi-two-dimensional working sections have been built, for boundary-layer studies or smoke tests, and other tunnels with special features such as slotted walls, flexible roofs, and moving floors abound. Usually the return circuit design is not greatly affected by the working section proportions, but a short section of this review is devoted to the design problems of special-purpose tunnels.

Once the working section size has been decided, and the choice made between open-circuit and return-circuit layout, and between open-jet or closed working section, the design of the rest of the tunnel depends on a compromise between the space and power available, and the quality of flow required in the working section. All tunnels of any pretensions have a contraction before the working section: the air velocity increases in passing through the contraction, but as total pressure remains constant, both mean
and fluctuating velocity variations are reduced to a smaller fraction of the average velocity. At the same time, the lower velocity of the air in the return circuit reduces the power consumption. It is usual to fit a fine-cell honeycomb grid, or wire gauze screens, or both, upstream of the contraction. The former reduces irregularities in flow direction and the latter reduce velocity variations by producing a total-pressure drop which is larger in the higher-velocity regions and smaller in the lower-velocity regions, followed by a return to constant static pressure, without appreciable total-pressure loss, further downstream. The effects of a contraction and of a screen on a flow with a small perturbation can be seen from Bernoulli’s equation. In order to accommodate a contraction it is necessary, at least in a closed-circuit tunnel, to have a corresponding expansion: the minimum loss of energy and least disturbance to the airflow are obtained when the expansion takes place through a gently-diverging passage (diffuser) which in practice is usually a cone, not necessarily of circular cross-section, of about 5° total included angle. It is conventional to locate the fan assembly in a constant-area duct at a station in this diffuser where the cross-sectional area is at least twice that of the working section. For this reason, together with the above-mentioned need to minimize disturbances in the room, and of course the desire to minimize power consumption, open-circuit tunnels usually have diffusers also. Closed-circuit tunnels need cascades of vanes, or other devices, to turn the flow smoothly through 360°, usually in four 90° corners. A good tunnel with the features mentioned above can have a power factor (ratio of motor horsepower to kinetic energy flux through the working section) of as little as 0·2, with r.m.s. spatial and fluctuating velocity variations in the working section less than 0·1 per cent of the mean velocity. It should be added however that this can only be satisfactorily achieved if there is adequate space for the tunnel structure: otherwise an increase in power consumption is inevitable.

The design of the components of the tunnel is discussed in detail in Sections 5–11, which should be applicable to the return-circuit design of supersonic tunnels as well as low-speed tunnels.

4. THE PRODUCTION OF STEADY, UNIFORM FLOW

"Unsteadiness", in wind tunnel parlance, is a velocity fluctuation of low enough frequency to be noticeable on manometers and balances. The distinction between unsteadiness and turbulence is a subjective one, but usually corresponds to a distinction between the sources of the irregularities. Unsteadiness is normally a symptom of flow separation or intermittent separation, for instance in a diffuser with too large an expansion angle, or in an excessively short contraction. An intermittent separation may be triggered by the occasional passage of an unusually intense turbulent eddy, so that very slow velocity fluctuations may occur with frequencies many
times lower than the frequencies which contain most of the turbulent energy. Because the whole flow is being deflected bodily by the displacement thickness of the separating shear layer, the fluctuations may be regarded as inviscid and irrotational. Another irrotational mode is sound, caused by

(i) mechanical vibration, generally at the fan *shaft* frequency;
(ii) dipole excitation of the moving fan blades by a spatially non-uniform flow, at the fan *blade* frequency;
(iii) dipole excitation of fixed vanes and panels by unsteadiness and turbulence, with a broad spectrum;
(iv) quadrupole emission from turbulence without excitation of any solid boundary, which is likely to be small compared with the other sound sources in a low-speed tunnel.

Allied to (iii) is the near-field pressure fluctuation, or pseudo-sound, of the turbulent boundary layers in the working section, which produces inviscid velocity fluctuations within a few boundary-layer thicknesses of the walls. It is worth remarking that tunnels with concrete or metal walls usually have an extremely long reverberation time, so that the sound intensity produced by a given source of sound power may be very high: even small areas of sound-absorbent material can greatly decrease the sound intensity. “Organ-pipe” resonance in ducts has been discussed by Whitehead\(^3\).

True turbulence, or vorticity fluctuation, is generated by the boundary layers on the return circuit walls, and the wakes of fan blades, vanes and honeycombs, but “turbulence” is frequently used as a term of abuse to cover all the above-mentioned unsteady phenomena.

Spatial non-uniformity of the mean velocity can be caused by

(i) poor corner vane design (resulting in asymmetry in the plane of the corner);
(ii) boundary-layer growth in the return circuit, especially separations caused by an excessive diffuser angle, or by poor corner design, with resulting asymmetrical flow;
(iii) poor fan or straightener vane design, causing a swirl or rotation of the whole flow about the tunnel centre line, or contributing to (ii).

Since acceptable values for the mean velocity variations across the working section of a high-performance tunnel are ±0.2 per cent and ±0.1°, i.e. about four times the r.m.s. turbulent fluctuations usually aimed at, it is found that design for low turbulence automatically ensures adequate uniformity of mean velocity. In a general-purpose tunnel, however, turbulence may be of less importance and the design of the tunnel will be determined by the required uniformity of mean velocity; but the techniques for reducing turbulence and spatial variation are very similar, and since the effects of spatial variations in wind tunnel testing are qualitatively obvious, the rest of this section will be devoted to a discussion of the effects of free-stream
turbulence in wind tunnel testing and of the general principles involved in its reduction.

The most spectacular effect of turbulence is a reduction of the boundary-layer transition Reynolds number to values which depend in an almost unknown manner on the characteristics of the free-stream turbulence. Schubauer and Skramstad⁴⁰ give a graph of flat plate transition Reynolds number against u-component r.m.s. intensity measured with various arrangements of screens in the NBS 4½ ft tunnel (Fig. 11) showing that reduction of r.m.s. intensity below 0·1 per cent had no further effect on transition. This graph has very frequently been used as an argument that 0·1 per cent turbulence is "small enough" but it is difficult to see why transition Reynolds numbers should not be increased indefinitely by reducing disturbances in the tunnel. The most likely explanation of the NBS results is that lateral vibration of the test plate relative to the airstream determined the transition position when the tunnel turbulence was very low: either oscillations in the wake behind the test plate or mechanical vibration transmitted from the nearby fan could have been responsible. Roughness of the plate is a more distant possibility.

Another effect of free-stream turbulence is in falsifying measurements of lift and drag when they vary non-linearly with incidence, as in the neighbourhood of the stall, particularly if hysteresis occurs. Here, u-component fluctuations with wavelengths greater than the chord of the model will be most important, in contrast to the fluctuations with wavelengths a few times the boundary-layer thickness which have the greatest effect on transition: these long wavelengths will comprise irrotational unsteadiness as well as turbulence.

Before the advent of hot wire anemometry a common method of specifying the turbulence in a stream was by the critical Reynolds number of spheres, the value \( Re_{crit} = 385,000 \) (obtained from flight tests) being used as a yardstick of effectively zero turbulence. The Reynolds number thus obtained was usually a function of sphere diameter and nowadays percentage turbulence seems to be preferred as a convenient number to state: this is illogical, as differences between the results from spheres of different diameters indicate the importance of the relative scales of turbulence and the variation of turbulence intensity with tunnel speed (since, for a given Reynolds number, the tunnel speed will be inversely proportional to the sphere diameter). However, quantitative measurements of turbulence intensity and scales are essential if the sources of turbulence are to be identified. The NACA⁴⁵ carried out tests with a laminar flow aerofoil in several of its wind tunnels, and compared the drag coefficients at given Reynolds numbers: since at the time of these tests laminar flow aerofoils were being extensively investigated for their own sake, no more suitable test criterion could have been asked for. It should be noted that the turbulence intensity measured with a hot wire anemometer may depend quite strongly on the lower frequency

\(^{1a*}\)
limit of the anemometer, depending on how much of the very-low-frequency unsteadiness (due to atmospheric wind gusts, power supply voltage fluctuations, servo-control hunting and so on) happens to be recorded. If possible, the effect of altering the lower cutoff frequency should be explored; otherwise, the cutoff frequency should be stated when the turbulence level is quoted.

Working section unsteadiness and turbulence can be avoided in a new tunnel, or reduced in an existing one, by two different methods, either by improving the flow in the return circuit or by installing a honeycomb, screens and a large contraction ahead of the working section. As it is difficult to suppress low-frequency unsteadiness without a very large total-pressure drop in the settling chamber, as in the Royal Melbourne Technical College (RMTC) 4½ ft x 3½ ft tunnel (see Section 15), it is necessary to ensure that severe separations do not occur in the return circuit, but high turbulence and non-uniformity of mean velocity at entry to the settling chamber can be almost entirely removed from the airstream before it reaches the working section. Tests in the RAE 4 ft x 3 ft tunnel showed that a plate obstructing about one-sixth of the cross-sectional area of the second diffuser just downstream of the fan nacelle had very little effect in the working section, but this tunnel has many screens and an exceptionally large contraction ratio, and other tunnels may not be as imperturbable. Disturbances due to the presence of a model in the working section are only likely to make themselves felt through the precipitation of separations from the diffuser or first corner, and should not be important in a well-designed tunnel unless the disturbances caused by the model are very large, in which case the working-section interference corrections will be excessive and unreliable anyway.

In general, the designer must decide, in the light of the model-testing programme planned for the tunnel, how much trouble should be taken to reduce turbulence. A useful indication of whether turbulence is low enough for a given test is to repeat it with a slightly higher turbulence level by inserting a coarse grid in the settling chamber. As a precaution, the tunnel structure should be designed to permit the installation of extra screens, and a honeycomb if not originally fitted.

The factors of reduction of mean velocity variation or turbulence intensity by screens and contractions are given below for future reference.

**Screen**

(i) *U*-component mean velocity (theoretical):

\[
\frac{2-K}{2+K} \text{ (Ref. 18)} \quad \text{or} \quad \frac{1+\alpha - \alpha K}{1+\alpha + K} \text{ (Ref. 19, p. 64)}
\]

where \( K \) is the pressure-drop coefficient and \( \alpha \) is the deflection coefficient, defined as the ratio of the exit and entry angles. Reference 21 gives \( \alpha \approx \frac{1.1}{|1+K|} \); using this value, the two expressions give
Because of this, the reduction of the lateral component by a screen or a contraction is much less than that of the longitudinal component, and so it is the desired reduction in the former which decides the screen arrangement and contraction ratio.

### 5. DRIVES AND CONTROL SYSTEMS

Wind tunnels are almost always driven by electric motors but injector pumps, aircraft and other petrol engines, water turbines and at least one gas engine have been used. Injector pumps are useful and cheap to build for
small or temporary rigs or intermittent tunnels where a laboratory compressed air supply is already available, but they are noisy and inefficient. For most tunnels the question is what type of electric motor should be used and how should it be controlled. It is usually preferable to alter the tunnel speed by altering the motor speed, so that d.c. motors are most common, shunt winding being normally preferred. Variable-pitch fans have been used, although they are better suited to the adjustment of fan efficiency than to the control of speed over a wide range. They are also complicated mechanically and the vibration at low tunnel speeds may be excessive. Motors of more than, say, 5 h.p. cannot be conveniently or economically controlled by rheostats because of the large heat dissipation involved. Ward-Leonard sets, in which a constant speed a.c. motor drives a d.c. generator whose output voltage, controlled by the field excitation current, is used to drive the fan motor, are too expensive in initial cost for motors less than about 50 h.p. though the system is very efficient and almost unrivalled for large tunnels where running costs are appreciable. In the range between 5 and 50 h.p. several types of control system compete: controllable rectifiers, saturable reactors or magnetic amplifiers, auto-transformers (for use with a.c. commutator motors or for supplying a non-controllable rectifier and a d.c. motor) movable brushgear (in the Schrage a.c. motor) and variable field excitation can be used, although the last named will not give a speed range much greater than 3:1 unless allied to a discretely-variable armature voltage. If no suitable d.c. supply is already available, one of the several commercial controllable rectifiers or magnetic amplifiers will probably be most satisfactory, but care must be taken to suppress magnetic radiation and mains interference. Induction or synchronous motors can be used to drive variable-speed fans through magnetic clutches: the torque transmitted is controlled by the clutch excitation, the fan speed increases until it accepts the given torque and the motor current increases to supply the required power. The efficiency of the system at low fan speeds is, of course, poor, but it is convenient for small tunnels whose running costs are negligible. Most electrical firms are able to offer a motor and control gear for a medium-sized wind tunnel drive from a standard range.

In all cases, the controlling action is likely to be a shaft rotation actuating a rheostat or auto-transformer, so that local manual control can be replaced by a pilot motor, remotely controlled by push-buttons or an automatic speed control system.

Types of automatic speed control are as numerous as types of motor drive, but all involve the amplification and feedback of a signal whose amplitude or sign depends on the difference between the actual and required speeds. The signal should be derived from the airspeed rather than the shaft speed, as it is convenient to maintain constant dynamic pressure during the run, and a pressure transducer or pressure balance is most suitable. If a bang-bang control (in which the error signal fed back is of constant ampli-
tude but has the same sign as the speed error) is sufficient, a pressure balance operating micro-switches is convenient, but care must be taken to avoid sluggishness. To obtain proportional or proportional-plus-derivative action a pressure transducer is essential unless a completely pneumatic servo system is to be used: photocell meniscus detectors have been used successfully. A control system should always be designed with full knowledge of the response times of the tunnel and motor to be controlled, so that there is something to be said for leaving the final arrangement of the automatic control until the tunnel has actually run: tunnel calibration and troubleshooting will in most cases take amply long for the assembly of the speed control gear. An alternative method of automatically maintaining a set speed is to vary the power factor of the tunnel continuously: the RAE 4 ft × 3 ft tunnel has a bypass round the fan, operated electrically from a pressure balance. Care must be taken to avoid upsetting the flow by dumping low velocity air into a diffuser or otherwise thickening the boundary layers. Baffles may be used to permit steady running at very low air speeds but still at a high enough fan speed and power to allow easy control of speed, as in the NPL 18 in. Low-speed Tunnel (Fig. 20). The usual source of a controlling pressure signal is the pressure difference between two holes, one in the widest part of the settling chamber and another near the narrow end of the contraction but far enough upstream to be uninfluenced by the presence of a model in the working section. In all but the smallest tunnels (i.e., whenever the contraction boundary-layer displacement thickness is reasonably small compared with the working section size) this pressure difference is a constant fraction of the empty working section dynamic pressure, in the absence of compressibility effects. It is safest to use four holes (one on each wall) manifolded together for measuring the downstream pressure, as disturbances caused by a lifting model will then tend to cancel out.

6. FANS

The design procedure given here applies to fans with pre-rotation vanes designed to produce a swirl equal and opposite to that produced by the fan, so that the swirl downstream of the fan is zero. These stator vanes are normally installed in front of the fan (Figs. 2, 3, 4) because the velocity of the fan blade relative to the airstream is thereby increased, permitting a higher fan loading for a given rotational speed. As the loading of the pre-rotation vanes and the fan blades is usually about the same, it is a matter of indifference which is disturbed by the wakes of the other. It is usual to ensure that if the number of fan blades is N the number of pre-rotation vanes is not N, 3N/2 or 2N, since the passages of the fan blades through some of the pre-rotation vane wakes would then coincide, with consequent vibration. Pre-rotation vanes can be omitted only if the fan swirl is low enough to decay completely before the stream returns to the working section. If the fan
rotational speed is high, so that the torque and swirl are low, symmetrical straightener vanes downstream of the fan may be sufficient to remove the swirl: such vanes are normally fitted as a precaution even when the fan has pre-rotation vanes. The modifications to the design procedure required if straighteners only are used and there is no pre-rotation are generally obvious.

It is important to note that wind tunnel fans must frequently be designed to give the maximum practicable pressure rise for a given diameter and rotational speed rather than to operate at maximum efficiency. Typically, fan efficiencies are of the order of 90 per cent, and as it is by no means certain that the tunnel power factor can be estimated to 10 per cent accuracy, engineering convenience and cheapness are more important than the utmost efficiency. The design procedure has been reduced by Wallis to the consultation of a series of charts. Since some at least of these charts are based on empirical rules which are not definitive, it is as well to recapitulate the analysis (see also Mair) and to mention the assumptions and empiricisms.
The steps in design are outlined in (i) to (vi) below: the difficulties in each are discussed in (a) to (f).

(i) Estimate the power factor of the tunnel, allowing a factor 1.1 for fan inefficiency.

(ii) Choose the fan area (or the mean speed through the fan), the ratio of nacelle diameter to fan diameter, and the rotational speed.

(iii) Hence calculate the ideal total-pressure rise through the fan disc. It is usual to make the total-pressure rise independent of radius unless the entry velocity profile is unusual. Deduce the swirl at each radius. For each annulus of radius $r$,

$$\text{power input} = (\text{tangential force component}) \times (\text{blade rotational velocity})$$

$$= (\text{rate of change of tangential momentum}) \times (\text{rotational velocity})$$

$$= (\text{mass flow}) \times (\text{swirl velocity}) \times (\text{rotational velocity})$$

$$= (2\pi r \, dr \cdot \rho \, U) \cdot (\omega_r \, r) \cdot (\Omega \, r)$$
and power output
\[ = (\text{total pressure rise}) \times (\text{volume flow}) \]
\[ = (\Delta P) \cdot (2\pi r \, dr \cdot U) \]
so that total power output
\[ = 2\pi \int U \Delta P \cdot r \, dr = \frac{1}{2} \rho \, U_0^2 \lambda \, A_0 \cdot \Delta P \] at radius \( r \) follows from this and the chosen radial distribution.

Equating input and output, we find (since \( \lambda \) includes the fan inefficiency),
\[ \omega_p = \frac{\Delta P}{\rho \, \Omega \, r^2}. \]

(iv) Calculate the product of blade section lift coefficient and solidity at each radius, both for the fan blades and the pre-rotation vanes. Assuming, with some theoretical justification, that the effective swirl at the blade position is the mean of the values far upstream (\( \omega_p \, r \)) and far downstream (zero), the effective velocity relative to the fan blade is
\[ U \csc \varphi \]
at an angle $\varphi = \tan^{-1} \frac{U}{\Omega r + \frac{1}{2} \omega_p r}$ to the fan disc (since the pre-rotation opposes the fan rotation).

By equating the tangential force component on each blade element to the rate of change of tangential momentum produced by that blade element,

$$C_L \frac{1}{2} \theta (U \csc \varphi)^3 \ c \ \sin \varphi \ \varphi \ dr = \frac{\partial U}{\partial \omega_p} rh \ dr,$$

$C_L$ being defined as usual in a direction perpendicular to the effective velocity vector.

Therefore

$$C_L \frac{c}{h} = 2 \frac{\omega_p r}{U} \sin \varphi.$$

For the pre-rotation vanes, we put $\Omega = 0$ and apply the same analysis so that

$$C_{L_p} \frac{c_p}{h_p} = -2 \frac{\omega_p r}{U} \sin \varphi_p$$

and

$$\tan \varphi_p = -\frac{2U}{\omega_p r}.$$

Note that $C_{L_p} \frac{c_p}{h_p}/C_L \frac{c}{h} = -\frac{\sin \varphi_p}{\sin \varphi} \approx 1.5$, typically.

(v) Decide the maximum safe lift coefficient at each radius, and deduce the solidity $c/h$, which must be about 1.5 times as large for the pre-rotation vanes as for the fan. Choose the number of blades, $N = 2\pi r/h$ so that the blade aspect ratio is greater than about unity to minimise secondary flow effects.

(vi) Choose the blade section and use its performance data to calculate the blade incidence and thus the angle of inclination of the blades to the plane of the fan disc.

(a) It is very difficult to make an accurate estimate of the power factor: for convenience, the power factor contributions of each section of the tunnel circuit are discussed separately, but on reference to these sections it will be clear that the estimate is at best an inspired guess. A 10 per cent underestimate of power factor implies a drop of only 3 per cent in tunnel maximum speed, providing the fan efficiency is not greatly affected: the danger is that the fan blades may tend to stall, causing vibration and poor flow in the return circuit. On the other hand, an over-generous estimate means that the full power of the motor cannot be absorbed at the design speed, but this is clearly the better side on which to err, and it is usually possible to increase the speed of a shunt motor by reducing the field excitation.
(b) It is usual to have an area ratio of about 2 or 3 to 1 between the working section and the fan. A very much larger value would result in a poor velocity profile and high turbulence at inlet to the fan, and also necessitate a larger and more expensive fan assembly, whereas a much smaller value would imply a high fan rotational speed and more vibration. To absorb a given amount of power, the fan must produce a given thrust per unit area. If we imagine a series of fans of different duct areas $A$, say, but the same solidity and lift coefficient, we see that the effective velocity over the fan blades must be constant:

$$U^2 + r^2\Omega^2 = \text{const},$$

so that the tip speed increases to a limit as the area increases, though the rotational speed $\Omega$ finally decreases towards zero, since $Ur^2 = \text{constant}$. An area ratio of 2 fits in well with the usual closed-circuit tunnel arrangement of a diffuser immediately after the working section and a fan immediately after the second corner.

The fan rotational speed and nacelle diameter are chosen to give reasonable blade angles, particularly near the blade root. It is usual in large tunnels to choose a nacelle diameter of about 0.5-0.6 fan diameters and a rotational speed to give blade angles of not more than about 45° near the root. High rotational speeds, giving low values of $U/\Omega r$, reduce the blade area needed, and reduce sensitivity to velocity profile, since

$$U \frac{\partial \varphi}{\partial U} \approx \frac{U}{\Omega r} \frac{1}{1 + \left(\frac{U}{\Omega r}\right)^2}.$$  

However, high rotational speeds lead to unnecessary vibration, and the tip speed should always be kept below about half the speed of sound, so that one usually chooses the lowest speed that will give safe values for the blade lift coefficient: this implies that the blade solidity should be high, but an excessively high solidity will lead to severe blockage and induced-camber interference between the blades, whose behaviour can no longer be predicted by small corrections to isolated-aerofoil theory, and will also cause constructional problems if the blades overlap in plan view.

(c) Although the mean speed through the fan is known, the velocity profile is most unlikely to be uniform: it is usual to choose the total pressure rise through the fan to be independent of radius, which will partially restore uniformity, rather than to design for a uniform total pressure at outlet, which would require an unacceptably high lift coefficient near the tips where the axial velocity is low. It is, however, necessary to estimate the inlet profile to calculate the torque at each radius and, of course, to
calculate the velocity of the airflow relative to the blades. Uncertainty about the inlet profile is the biggest single source of error in wind tunnel fan design, and renders any very refined analysis of the other errors somewhat futile. Wallis says that in tests on a series of axial compressors designed on the basis of various assumptions about the inlet velocity profile, the best design was found to be that based on a uniform profile rather than on actual velocity profile, and comments that from a practical point of view the constant axial velocity design assumption appears to be as good as any other. Wallis points out that the overall performance of a fan is not too sensitive to velocity profile shape although he mentions the increased likelihood of tip stalling: perhaps the right view to take is that a conservatively-designed fan will still work efficiently away from its design conditions, but that this is no reason for failing to make as good an estimate as possible of what the design conditions actually are. The profile is best estimated by reference to tests on tunnels of similar layout: velocity distributions are given for the RAE 4 ft × 3 ft tunnel in Ref. 50, for the RAE 10 ft × 7 ft tunnel in Ref. 7 and for the NPL boundary-layer tunnel in Fig. 5: the flow in tunnels with more conventional diffusers should be no worse than that shown in Fig. 5. In the case of a large tunnel it is safest to make a scale model of the first diffuser and corners.

Fig. 5. Velocity contours upstream of NPL boundary-layer tunnel fan before final adjustment of diffuser blowing. Figures are values of \( u/u_{\text{Ref}} \).
and measure the profile, but even this precaution has been known to fail. The effect of scale operates in two opposing ways: in a model, the pressure coefficient at separation is less, but the displacement thickness of the boundary-layer is greater, thus reducing the pressure rise in the diffuser.

(d) By taking the swirl at the "blade position" to be the mean of the values "far upstream" and "far downstream" we are assuming that the pre-rotation vanes are many vane chords upstream of the fan blades: this is not usually the case. The distance between the pre-rotation vane trailing edge and the fan blade leading edge should be at least half a vane chord so that vane wakes shall not be too strong.

(e) Because of the influence of the boundary-layers on the nacelle and duct, the maximum lift coefficient of the blade section near root and tip will almost certainly be less than in two-dimensional flow, and the maximum lift/drag ratio will occur at a smaller lift coefficient. There is no general information to enable one to correct the two-dimensional characteristics, even if the inlet boundary-layer profile were known accurately. Mair suggests that the fan blade lift coefficients should not exceed 0.9 or 1.0 at the root and 0.7 at the tip in order to avoid stalling, due to boundary-layer influence at the tip and a combination of this and blade interference at the root. Wallis gives details of Howell's relations for optimum performance of high-solidity cascades, which support Mair's figure of $C_L \leq 0.9$ for a typical root solidity of about 1.0. The pre-rotation vane $C_L$ should be kept well below these values, implying high solidity. The resulting values of $Nc$ at root and tip may prove to be noticeably different, in which case a redistribution of $C_L$, or recourse to tapered blades will be necessary. There is no aerodynamic reason for tapering the fan blades, and constant chord is to be preferred for constructional reasons, so that if possible the mid-span value of $NcC_L$ should be modified to give the same total thrust but keeping a ratio of $(NcC_L)_{root}/(NcC_L)_{tip}$ of about 9/7. Pre-rotation vanes are more conveniently made untwisted, with a moderate taper if necessary. The maximum permissible solidity of the fan or pre-rotation vanes depends on the allowances the designer is prepared to make for blade interference. Solidities of up to 2 are in common use in multi-stage axial compressors, which achieve efficiencies of up to 90 per cent, and a reliable set of empirical rules for high-solidity cascade design has been developed. Wallis gives details of these rules, which should be adequate for pre-rotation vane design; unfortunately the extrapolation of cascade design rules to the low solidities (0.5 or less) found near the tips of conventional wind tunnel fans is very risky, so that if cascade design rules are used near the root it will be necessary to change to isolated-aerofoil theory at part-span. The two ways of avoiding this are to use a very large nacelle, so that the
solidity is nearly constant over the span and can be chosen in the cascade-design range, or to restrict the root solidity to a value at which corrections to isolated-aerofoil theory can be confidently made. It seems best in the latter case to keep the fan root solidity below unity, and there should be no great difficulty in doing this unless the tunnel power factor is unusually high or the fan speed necessarily low. The NPL Boundary-layer Tunnel design power factor was 0.4 and the ratio of axial speed to fan blade speed at the root was 0.95, both high values, and the root solidity came to 1.0 exactly. The blade chord should be kept low enough to give a reasonable blade aspect ratio, preferably not much less than unity, in order that any shortcomings of flow near the root or tip do not affect a disproportionately large part of the blade area. At the same time it is well to keep the blade Reynolds number above the range in which laminar separations are liable to occur: the presence of turbulence in the inlet flow is probably an advantage in this respect but a chord Reynolds number of $2 \times 10^6$ is a fairly safe lower limit, corresponding to a blade chord of 7 in. at 50 ft/sec. Trip wires or roughness on the blades might cure a laminar separation.

(f) There seems to be no point in departing from the usual flat-bottomed Clark Y type of section for fan use. High-lift sections with concave lower surfaces are usually predicted by high-solidity cascade design techniques and could usefully be employed near the root, but a flat-bottomed section is easier to make, particularly as regards setting the blade angle. Pre-rotation vanes, as mentioned above, are best designed as camber lines by cascade rules and then faired with an RAF 30 or NACA 4-digit symmetrical section. Mair suggests the use of NACA 6712 as a pre-rotation vane section, and gives its ordinates: it resembles a Clark Y section with a 20 per cent chord flap at 20°. If the inlet velocity profile is more than usually uncertain, the provision of adjustable trailing edges on the pre-rotation vanes may be wise, not, of course, to improve the axial velocity distribution but to prevent fluctuations in blade lift and consequent vibration. NACA 6712 could well be approximated by a Clark Y section with adjustable trailing edges. Lift and drag data for sections in the low Reynolds number range are scarce ($Uc/\nu = 10^8$ if $U = 150$ ft/sec and $c = 1$ ft) so that some difficulties may arise in the design of small fans but the zero-lift angle $\alpha_0$ ought not to depend greatly on Reynolds number and the drag coefficient is not very important, so that only the variation of $dC_L/d\alpha$ with Reynolds number is critical. Mair gives details of $\alpha_0$ and $dC_L/d\alpha$ in terms of thickness for Clark Y at $Re = 3 \times 10^6$. Wallis gives data for RAF 6E (a flat-bottomed 10 per cent thick aerofoil) showing little variations of $dC_L/d\alpha$ down to $Re = 3 \times 10^6$ although the blunt trailing edge of this section may minimise scale effect.
As suggested above, it seems best to limit the root solidity of the fan to about 1.0 and treat blade interference as a small correction to the performance of an isolated aerofoil. The main effects on a two-dimensional cascade are (i) a change of lift curve slope, (ii) a change of zero-lift angle due to camber and (iii) a change of zero-lift angle due to thickness. Charts for all these have been given by Mair: more details of Weinig's theoretical results for the lift slope of flat plates in cascade are given by Wislicenus and Horlock. Approximate values of the corrections for flat-bottomed aerofoils are

\[
\left( \frac{dC_L}{d\alpha} \right)_{\text{isolated}} \approx 1 + \frac{\pi^2}{12} \left( \frac{c}{h} \right)^2 \cos^2 \beta \quad \text{for} \quad \frac{c}{h} < 0.8
\]

\[\Delta \alpha_0 \approx 48 \left( \frac{c}{h} \right)^2 \left( \frac{l}{c} \right) \cos^2 \beta \quad \text{degrees}\]

\[\Delta \alpha_0 \approx 18 \left( \frac{c}{h} \right)^2 \left( \frac{l}{c} \right) \sin^2 \beta \quad \text{degrees}\]

(iii) agrees with the detailed calculations for a 10 per cent aerofoil reported by Schlichting. The change of zero-lift angle due to thickness is +2° for a 10 per cent aerofoil with \(h/c = 1\) near \(\beta = 45^\circ\), and this is equivalent to a \(C_L\) error of about 0.2. To put this in perspective it may be remarked that a change in incidence of 2° would also result from a change of 4 per cent in axial velocity for \(\beta = 45^\circ\). The net effect of lift-slope change and camber is small near \(\beta = 45^\circ\); the two corrections are of opposite sign, and they could probably be neglected for most wind tunnel fans, because the root blade angle is usually quite near 45° (RAE 4 ft × 3 ft 48.7°, RAE 11½ ft × 8½ ft No. 2 45°, NPL BL tunnel 47°) and even if the angle is much less at the tips the gap/chord ratio is correspondingly greater.

The change in \(C_{L_{\text{max}}}\) is just as important as the changes in zero-lift angle and lift-curve slope, but it is not predictable by the above means since it depends on the effect of the induced velocity on the aerofoil boundary layer. In general, \(C_{L_{\text{max}}}\) is less for aerofoils in cascade, but as most fan blade sections are of the rear-stalling type and unlikely to suffer a rapid decrease in \(C_L\) after the stall, the lift loss penalty is likely to be small. The drag increase is not likely to be very important because we are of course considering only the root section of the blade so that the contribution to the shaft torque will be small. Even if the total fan blade drag were double the estimate, \(1 - \eta\) would be rather less than doubled (say \(\eta = 0.87\) instead of 0.93) and the tunnel speed at given power would therefore be only 2 per cent below the design value. In fact the only important consideration is deterioration of flow downstream of the fan, leading to buffeting of the straightener vanes and possibly to separation from the fan nacelle. To this extent, however, the effect of stall in a single-stage fan is less than in a multi-stage compressor.
The behaviour of the blades near the root is influenced as much by the nacelle boundary-layer as by blade interference. If the fan is driven by an extension shaft, the fluid passing over the blade roots has had to negotiate the shaft fairing at the second corner and the length of the rotating shaft, with a region of sharply retarded flow near the nacelle nose, as well as the nacelle itself: the boundary layer is therefore likely to be somewhat disturbed although not as thick as the tunnel wall boundary-layer.

Providing that the flow at entry to the fan is not in an excessively disturbed state because of separations in the first diffuser or in the fan duct itself, and that the blade lift coefficients are chosen within the range suggested above, it is unlikely that grave shortcomings in performance will appear and even less likely that they will be irremediable by adjusting the pre-rotation vane trailing edges or the motor speed (always assuming, of course, that the power factor has not been badly underestimated). But one can hardly claim any greater reliability than this for a design process which depends so much on empirical limits.

The fan duct and nacelle design problems are more structural than aerodynamic but excessive liberties should not be taken merely to simplify construction. Unless the nacelle diameter is very large, the expansion in duct diameter needed to maintain constant area up to the fan can be drawn in by eye. It is not a good idea to maintain a constant diameter of duct and use the nacelle to produce a slight contraction before the fan: it would be better to reduce the diffuser area ratio pro rata and have a constant-area duct for some distance before the fan. A section change from octagonal to circular is usually necessary but it should not be too abrupt. The best method seems to be the use of flat facets between the octagon and an intermediate hexadecagon, and between this hexadecagon and another circumscribing the circle: the final fairing can be done with filling compound, either in the mould or on the final structure. Some expansion can be permitted over the fan nacelle tail if it is really necessary to keep the overall length of the tunnel to a minimum, but the equivalent cone angle should not exceed 5°: an excessive adverse pressure gradient over the rear may lead to boundary-layer separation and persistence of the wake as in the RAE 4 ft × 3 ft tunnel. The nacelle itself should have a length-to-diameter ratio of at least 3, with a constant-diameter section from 30 per cent to 40 per cent of the length for mounting the fan. The best design method is probably to take a well-known fairing such as the NACA 4-digit series and modify by eye where necessary. The nose shape is usually almost ellipsoidal though a smooth fairing into the fan extension shaft, if any, would be better.

The power factor contribution of the fan duct walls is adequately estimated by assuming a zero-pressure-gradient local surface friction coefficient, say 0.0025 for tunnels of 4 ft² working section area and over: alternatively the fan duct may be lumped together with the diffuser and the nacelle treated as an isolated body of revolution.
Preston\textsuperscript{(11)} has shown that a fan, like a freely-rotating windmill, will tend to remove circumferential variations in mean velocity and also tend to modify the radial velocity distribution to correspond more closely to that for which it was designed.

7. DIFFUSERS

Townsend\textsuperscript{(12)} has shown that a turbulent boundary-layer can survive retardation to zero free stream velocity without separating only if the velocity varies more slowly than a certain power of the distance from the boundary-layer origin: this power depends on the Reynolds number but is always more positive than $-\frac{1}{2}$. Almost invariably, the diffuser of a wind tunnel is conical so that the velocity in a very long diffuser would eventually become proportional to the inverse square of the distance from the entry. The position at which the power law index falls below $-\frac{1}{2}$ depends on the diffuser angle and the thickness of the boundary-layer at entry, and in the present state of boundary-layer theory precise design rules cannot be given. In any case, some safety margin is necessary so that separations shall not be caused by unsteadiness of the flow into the diffuser. The usual rule for diffuser design is to use an equivalent cone of $5^\circ$ total included angle and to avoid thick entry boundary layers, but this is not infallible: the cone angle should clearly depend on the area ratio.\textsuperscript{(13)}

If the absence of any asymmetrical disturbances (such as may be caused by models at incidence or poor corner vanes) could be guaranteed, $7^\circ$ diffusers, as used in many tunnels of 1940 vintage, would probably be satisfactory at least for the small area ratios for which they were then commonly used. As is demonstrated in the discussion of the RMTC $4\frac{1}{2} \text{ ft} \times 3\frac{1}{2} \text{ ft}$ tunnel (Section 15) such asymmetrical disturbances tend to be fed back round the tunnel circuit and produce serious disruption of the flow.

The diffuser of a return-circuit tunnel is normally divided into two sections by the fan, which in practice usually gives an almost uniform total pressure profile at exit: the first and second diffusers can therefore be considered separately, albeit with caution. The area ratio of a diffuser leg rarely exceeds 5, and trouble has been experienced with separation in very long $5^\circ$ conical diffusers. The modern trend is towards a moderate expansion of about 2.5 to the first corner, whose power factor contribution is then only about 0.02, and a similar expansion ratio in the second diffuser. If a large area ratio is needed in order to provide a large contraction ratio, a wide-angle diffuser is installed before the settling chamber. A $45^\circ$ equivalent cone with about three screens fitted at intervals has been found\textsuperscript{(50)} to maintain satisfactory flow with an area ratio of 4 : 1. The purpose of the screens is to smooth out velocity variations, in particular regions of incipient separation.\textsuperscript{(14)} It would be an advantage for the screens to intersect the walls at right angles so that the refraction of the flow by the screens
does not itself tend to produce separations, but this presents constructional difficulties. Gibson\textsuperscript{(15)} has described a wide-angle diffuser designed mathematically to have constant velocity on the wall except for a jump at one point, where a screen is fitted to prevent boundary-layer separation (Fig. 34). There was actually a small net static-pressure rise through this diffuser, whereas wide-angle diffusers designed by eye generally produce a fall in static pressure. As losses at this position in a conventional tunnel will in most cases be negligible because the dynamic pressure is so low, it is doubtful whether mathematical design is warranted, particularly in view of the probable non-uniformity of the velocity at entry. Small regions of separated flow probably occur in most empirically-designed wide-angle diffusers but do not seem to produce unsteadiness in the working section. Adequate space should be allowed between the wide-angle diffuser and the honeycomb so that the outer cells of the latter run full.

It is occasionally suggested that the effective angle of a diffuser may be reduced by inserting longitudinal splitter vanes. Splitters are indeed beneficial to a diffuser with incipient separation, but it is clear that the only reduction in diffuser effective angle is caused by the increase in boundary-layer displacement thickness on the side of the splitters: obstacles introduced along streamlines of a steady inviscid flow have no effect, and in either case, the same pressure rise has to be surmounted in the same distance. Splitters will of course reduce any flow asymmetry or unsteadiness, and their drag helps to re-establish a uniform velocity profile.(44)

The cross-section shape of diffusers is not particularly critical. Many tunnels have octagonal working sections and it is usual for the first diffuser at least to be octagonal also. The second diffuser, in which energy losses and low velocity regions produced by secondary flow are less important, can be square or rectangular in cross section for ease of construction and installation of corner vanes. It is good practice to allow a parallel portion before the fan, possibly from the first corner onwards, to permit the flow to recover from the diffuser pressure gradient and establish a low form-factor velocity profile before entering the fan.

Diffusers for open-circuit tunnels are similar in design except that the cone angle employed is often greater, on the argument that the state of the flow after the fan is unimportant. However, the flow through the fan has a great upstream influence on the steadiness of the working section flow. There are arguments for employing a smaller cone angle than 5°, since the ratio of diffuser entry boundary-layer thickness to equivalent diameter is usually greater for an open-circuit than a closed-circuit tunnel because open-circuit tunnels tend to be smaller and to have a larger working section length-to-diameter ratio than closed-circuit tunnels, and also suffer more from leaks. Any leaks in a tunnel diffusing to atmosphere result in jets of air entering perpendicular to the walls at speeds up to that of the flow in the working section, and the effect on the tunnel wall boundary-layers
may be considerable. Outward leaks in the return circuit of a tunnel whose working section is vented to atmosphere must clearly be balanced by a flow through the breathers, but the breathers can be designed as flush injector-type slots, as in the RAE 13 ft × 9 ft tunnel, so that the air enters nearly tangentially to the walls.

If a diffuser fails to perform properly and large modifications to the structure are impracticable then, once the source of the trouble has been located, a number of remedies can be applied. Apart from the aforementioned splitters, vortex generators of the aerofoil or air jet type can be fitted. (See the discussion of the RMTC and RAE 13 ft × 9 ft tunnels, Section 15). If a compressed air supply is available, tangential air jets may be of assistance in cleaning up incipient separations: as these will occur first near the corners of the cross section, circular jets, rather than two-dimensional slots, may be adequate. The NPL 7 ft × 7 ft, 9 ft × 7 ft and 13 ft × 9 ft tunnels have freely-rotating windmills in the second diffuser to transfer kinetic energy from fast-moving to slowly-moving regions of the flow: the transfer is chiefly spatial but there may be some effect on the larger turbulent eddies. Unfortunately the passage of the windmill blades through regions of varying velocity induces vibration and airflow pulsations at the blade frequency, so this sort of palliative is ill suited to a low-turbulence tunnel.

Screens in small-angle diffusers are not to be recommended except as an extreme remedy: the best a screen can do is to reduce the total pressure of high velocity regions and leave the low velocity regions unaffected, thus defeating the object of having an efficient diffuser. It is also doubtful whether tangential blowing in an open-circuit tunnel using air at atmospheric total pressure can ever be very effective, since the whole purpose of diffusion is to increase the static pressure to that of the atmosphere. Suction applied to a diffuser can be aerodynamically effective but is not usually preferable to blowing except for constructional reasons. It should be avoided in closed-circuit tunnels because it would be necessary to let atmospheric air into the tunnel to compensate for the quantity extracted, and this would disturb the flow near the breathers and introduce dirt.

The efficiency of a diffuser is usually defined, for incompressible flow, as

\[ \eta = \frac{\Delta p}{\frac{1}{2} \rho U_1^2 \left(1 - \frac{1}{A^2}\right)} \]

where \( U_1 \) is the entry velocity, \( A \) the area ratio and \( \Delta p \) the pressure rise: the efficiency must clearly decrease as \( A \) increases, and reach a maximum for some cone angle between 0 and 180°. In fact the efficiency seems to be nearly constant at 0.9 for the range of area ratios usual in tunnel practice and for angles of 5–10°, providing that the entry boundary-layers are not so thick as to produce separation. An alternative definition is that of total-
pressure loss coefficient

\[ C_P = \frac{\Delta P \Delta \lambda}{\frac{1}{2} \rho U_1^2} = \left(1 - \frac{1}{A^2}\right)(1 - \eta), \]

where \( \Delta P \) is the total-pressure loss averaged over the mass flow (not simply averaged over the cross section). \( C_P \) is the quantity actually required for power factor calculation, since \( \Delta \lambda = \Delta P \frac{1}{2} \rho U_0^2 \) where \( U_0 \) is the velocity in the working section. The figures for efficiency are based on the results of Patterson (see Ref. 1) but we cannot pretend that the data are really adequate for all tunnel design purposes especially as the first diffuser may absorb a quarter of the tunnel power: the diffuser is the component most sensitive to design faults and so even accurate data on efficiency of ostensibly similar diffusers could not be used with complete confidence. A figure of \( \eta = 0.9 \) or \( C_P = 0.08 \) should be applicable to diffusers of conventional cross section.

8. CORNERS

Provided that great liberties are not taken in design, no trouble should be experienced with the corner vanes of a closed-circuit tunnel, but some care is needed to keep down the resistance of the first and second corners, where the speed is high, as it may make up 10 or 15 per cent of the total power factor. The airspeed at the third and fourth corners is usually so low that losses do not matter, and it would be an ill-designed corner indeed that added appreciably to flow unsteadiness in this region. Corner vane camber lines can be designed by cascade theory to approximate to any given camber line in an infinite stream, but in practice a circular-arc camber line, subtending an angle slightly less than 90°, is adequate. The thickness of section is best decided in the light of the construction method adopted: it has been found that sheet metal vanes are adequate, and indeed may be more efficient than streamlined section vanes since excessively thick vanes in a cascade cause blockage, and the resulting adverse pressure gradients over the rear of the vanes will cause rapid boundary-layer growth and loss of efficiency. It is found that a gap/chord ratio of \( \frac{1}{4} \), a L.E. angle of 4–5° and a T.E. angle of zero give the best performance. Note that in many drawings of wind tunnels, possibly including some of the figures reproduced from other reports in this paper, the shapes and even the numbers of corner vanes may be drawn conventionally. In tests at a Reynolds number of about \( 2 \times 10^5 \) a pressure drop coefficient of 0.06 was obtained neglecting secondary flow losses at the roots. Tests in the RAE 4 ft \( \times \) 3 ft tunnel at \( Re = 1.9 \times 10^8 \) gave \( C_P = 0.033 \) so that

\[ C_P \approx 1.2 \left( \frac{U_c}{v} \right)^{-1/4} \]
may be assumed. It is unwise to permit any expansion round the corner
though this has often been done in the past (see NPL 13 ft x 9 ft tunnel
and Cavendish 15½ in. tunnel): if this is tried, the corner vane spacing
should be reduced below the value of 0.25 chord normally adopted, in order
to decrease the vane lift coefficient ($C_L = 2(h/c)$ for a 90° corner), and it
is on the whole doubtful whether the total resistance would be any lower
than that of a conventional corner plus diffuser. The overall length of the
tunnel is unlikely to be worth consideration, as it is determined by the settling
chamber and working section length together with the length of the first
diffuser required to produce a sufficient expansion before the first corner to
keep down the resistance; the only way of reducing tunnel length appreci-
ably would be by shortening the first diffuser. 180° corners with long splitters
or vanes have been used (see NACA 12 ft tunnel, Section 15), but for small
wooden tunnels the multiplicity of frames required make them unattractive.
On the whole it seems unlikely that any great improvement on the 86° sheet
metal vane is possible without more research effort than most tunnel de-
signers are prepared to make. The gap/chord ratio of 0.25 was suggested,
on the basis of tests, as giving the lowest resistance coefficient (i.e. $L/D$ max)
but somewhat larger ratios might be used in the third and fourth corners
without actually stalling the vanes, providing that the entry flow were
sufficiently uniform.

9. SCREENS AND HONEYCOMBS

The reduction of turbulent fluctuations by screens is shown by Batche-
lor(19) to increase monotonically with $K$, the resistance coefficient, for the
$u$-component and to have a very flat maximum at $K \approx 4$ for the $v$-compo-
nent. It appears from very recent work at NPL that screens of open-area
ratio, $\beta$, less than about 0.57 suffer from an instability of the flow through
the pores which causes the emerging jets to coalesce in random (steady)
patterns, and that the resulting small variations of flow direction in the
working section produce spanwise variations of boundary-layer thickness
and surface shear stress, of the order of 10 per cent of the mean, in nomin-
ally two-dimensional boundary layers. Further work is needed to establish
the limiting open-area ratio more accurately for different conditions. A dis-
cussion of the instability per se is given by Morgan(20) and an indication of
peculiar flow behind some fine screens is given in Ref. 21. If the local Rey-
nolds number, $U_\infty d/\beta v$, of the wires exceeds about 80, compared with about
150 for an isolated cylinder, the wakes of the wires will become turbulent;(21)
the turbulence is of very small spatial scale, and is likely to decay to an un-
noticeable level in the working section. The pressure drop coefficient $K$
of a screen is a function of the open-area ratio, $\beta$, and the wire Reynolds
number $U d/\beta v$. An adequate collapse of most experimental data in the
range of these two variables representative of tunnel use is obtained by
plotting $\beta^2 K/(1 - \beta)$ against $Ud/\beta v$. $\beta^2 K/(1 - \beta)$ is the drag coefficient of the wires considered as isolated cylinders in a uniform stream of speed $U/\beta$ and appears to vary as about $9 Re^{-0.4}$ for $50 < Re < 200$, decreasing more slowly above $Re = 200$. Wieghardt\(^{(22)}\) gives $\beta^2 K/(1 - \beta) = 6.5 Re^{-1/3}$ for $60 > Re > 600$. A suitable gauze is 16 mesh, 28 s.w.g. wire (0.0148 in. diameter). The open-area ratio $\beta = (1 - d/l)^2$ is 0.58 and $K$ is about 1.25 at 30 ft/sec rising to 1.6 at 12 ft/sec.

Experiments quoted in Ref. 19 show that the reduction factors for the energy of the $u$ and $v$ components (i.e. of $u^3$ and $v^3$) are respectively 0.15 and 0.3 for $K = 2$ so that if enough screens are used to reduce the $v$-component to an acceptable level the $u$-component level will be acceptable also. It is difficult to estimate the turbulence level upstream of the first screen but in the RAE 4 ft $\times$ 3 ft tunnel,\(^{(52)}\) which has as good a return circuit flow as any, each component has an intensity on the centre line of 12 per cent just upstream of the third corner and 5 per cent just downstream of the fourth corner and upstream of the screens in the rapid expansion: the decrease is partly due to the honeycomb effect of the corner vanes but part is due to natural decay, which would be absent if the diffusion continued right up to the fourth corner. If we take a $v$-component r.m.s. intensity of 5 per cent as typical, and require a value of 0.2 per cent in the settling chamber (leading to 0.05 per cent in the working section with a contraction ratio of 12) we find that about six ($K = 1.6$) screens are needed. Bearing in mind that at least two screens are required for a wide-angle diffuser, and that a honeycomb, if fitted, will help to reduce large-scale turbulence as well as mean velocity variations, we conclude that about four or five ($K = 1.6$) screens are needed in the settling chamber itself to produce a working section turbulence level low enough for nearly all purposes.

The full effect of multiple screens will only be felt if they are mounted far enough apart for the turbulence in the wire wakes of one screen to decay before the next screen is reached. 500 wire diameters should be sufficient in most cases, implying 7 in. for 0.0148 in. wires. There should be no difficulty in accommodating four screens at this spacing, and indeed a much smaller spacing would cause difficulties in construction and cleaning.

Screens are not very effective for removing swirl and lateral mean velocity variations: for this purpose a honeycomb, with cell length at least six or eight times the cell diameter, is preferable providing that the flow incidence does not exceed say 10°. The cell size and cross-section are not very important and will probably be determined by availability: impregnated paper honeycomb with hexagonal cells of widths down to $\frac{1}{4}$ in. is suitable for small sizes though care is necessary to find a piece with straight uniform cells. Since any deflection of the honeycomb under its own weight or under air loads would be most undesirable it is necessary for sizes more than about 3 ft $\times$ 3 ft to incorporate a supporting grid or to use metal honeycomb. This is normally made by spot-welding cor-
rugated strip between flat strips, giving a roughly triangular shape, but stacks of tubing with hexagonal section ends, sweated together as in car radiator manufacture, have been used. Aluminium honeycomb, made for aircraft sandwich construction, is also available. The only restriction on the cell width is that it shall be smaller than the smallest lateral wavelength of the velocity variation: 50 cells per settling chamber diameter should be adequate for closed-circuit tunnels fitted with screens. If a reasonably low turbulence level is required in an open-circuit tunnel without screens, where the honeycomb is the only source of small-scale turbulence, it is necessary to use a much smaller cell size so that the honeycomb turbulence decays more quickly.

The resistance coefficient of a honeycomb is usually about 0.5, which is small compared with that of screens and does not contribute appreciably to the tunnel power factor.

10. CONTRACTIONS

There is no wholly satisfactory method of contraction design either in two or in three dimensions, and many existing contractions have been designed by eye, with or without the precaution of testing a model before construction of the full-size tunnel. The difficulty in design is that any contraction of finite length must have regions of adverse velocity gradient on the walls at each end (a proof of this for a two-dimensional contraction is given below) so that there is a danger that the boundary-layer will separate. In addition, the surface streamlines in a contraction of polygonal (e.g. rectangular) cross-section cross the intersections of the sides, leading to severe secondary flow in the boundary-layer near the corners: even if such crossflows are avoided, as in a “two-dimensional” contraction, the boundary-layer near the corners will be more liable to separate than the boundary-layers near lines of symmetry.

To compromise between difficulties of construction and undesirable boundary-layer effects in the corners, contraction cross-sections are usually octagonal, and it is assumed for design purposes that both the potential flow and the boundary-layer behave as in a contraction of circular cross-section. Some contractions are “two-dimensional”.

Ideally, one would specify the velocity on the contraction wall as a function of distance along the wall, make a boundary-layer calculation to check that separation did not occur, and then deduce the required wall shape. Lighthill's\(^{(23)}\) two-dimensional method permits exact design of contractions of either finite or infinite length, but requires the wall velocity to be specified as a function of azimuth angle on the circle from which the contraction shape is conformally transformed, and the relation between this angle and distance along the contraction wall can only be obtained after transformation. Several methods for two-dimensional and axisymmetric flow start from the specification of the velocity on the centre line \(r = 0\) in terms
of axial distance $x$ and find a solution for the stream function $\psi$ in series form, with terms $X_p(x)R_p(r)$: pressure gradients on the walls are invariably steeper than those on the centre line, and the choice of a suitable velocity variation is nearly as difficult as an outright choice of wall shape. The equation for $\psi$ is $\psi_{xx} + \psi_{rr} - \psi/r = 0$, and the axial component of velocity is $\psi_x/r$. Lilley (unpublished) sets $\psi = \frac{1}{2}x r^2 + R(r) X(x)$, and finds $r^2 R_{rr} - r R_r + R r^2/c^2 = 0$ and $X_{xx} = X/c^2$. Putting $X = A \exp(x/c) + B \exp(-x/c)$, there follows $R = C(r/c) J_1(r/c)$: the contraction is designed in two parts, for $x > 0$ and $x < 0$, and the two parts matched at $x = 0$. Thwaites (4) puts $X(x) = \sin px$, so that $1/c = ip$, $R = C_p(1/p) I_0(pr)$, and sums over $p = 1$ to $N$ to enable a more general expression for centre line velocity to be used: unfortunately the flow is periodic in $x$ and errors are introduced when the contraction is terminated in parallel ends. Cohen and Ritchie (5) put $R = r^{2p}$, sum over $p = 1$ to $\infty$, and derive a recurrence relation for $X_p$ in terms of $\frac{1}{2}X_1$ which is the centre line velocity: this method seems to be the most general but the amount of computation required is correspondingly high. The usual procedure is to examine the streamlines of the flow calculated from the centre-line velocity distribution and to choose as the wall shape one which gives a reasonably short contraction without too sharp an adverse pressure gradient (the centre line velocity distribution is chosen monotonic, or the streamlines would all be onion-shaped). Another group of methods for two-dimensional flow is based on the hodograph technique, in which one specifies the relation between the wall velocity and the wall angle (that is, between the $U$ and $V$ velocity components) as a curve in the $(U, V)$, or hodograph, plane and then transforms to the $x, y$ plane, in which the potential function $F = \phi + i \psi$ is determined by the relation $dF/dz = U - i V$ where $z = x + iy$. Jordonson's (26) hodograph method is interesting because it implies that the effect of the narrow end of a contraction on the flow in the wide end can be approximately represented by a point sink so that the shape of the wide end becomes independent of the shape of the narrow end (and vice versa) for large contraction ratios where the central section can be approximated by a frustum of a cone or, in two dimensions, by a trapezium. The RAE 4 ft x 3 ft tunnel contraction has a conical central section: a (diverging) conical entry flow is also assumed in the Foelsch method for supersonic nozzle design. It would appear possible to design a family of contractions of various area ratios by scaling one wide end and one narrow end shape.

Derivation of an axisymmetric or three-dimensional shape from a two-dimensional one is unlikely to rise above the level of guesswork, since most contractions have a large wall angle, making any simple-minded scaling of cross-sectional area inaccurate. An exception is the approximate method of Whitehead et al. (7) for calculating the flow in an axisymmetric contraction once the flow in the two-dimensional contraction with the same
wall shape is known: here it is assumed that the streamlines are the same in both cases, but the velocity along the streamlines is made to satisfy the axisymmetric continuity equation. We may expect that the growing availability of high-speed computers will encourage the development of numerical methods for two-dimensional and axisymmetric flow. For instance, Lighthill's method has been adapted to calculate the wall shape corresponding to a given distribution of velocity on the wall itself, rather than on the transform circle, by successive approximation. Smith and Pierce (28) give a numerical method of calculating the two-dimensional or axisymmetric flow about or within a given boundary using a distribution of sources on the boundary, and state that computing times for 40-point solutions on typical aerofoils were about 15 min on an IBM 701. The state of the art should therefore improve in the next few years, but there is less likelihood of progress in methods for calculating the development of boundary layers, especially when secondary flow occurs.

The electric tank, resistance network or conducting paper analogues can be used to solve the inverse problem of finding the wall velocity for a given wall shape in two-dimensional flow. The tank can be used to simulate a sector of axisymmetric flow by tilting its bottom so that the depth of liquid decreases to zero on the centre line, but there is no straightforward way of using conducting paper for axisymmetric flow. The use of a deep electric tank for fully three-dimensional flow is likely to be rather more difficult and certainly less informative than testing a model in air.

If a contraction is to be designed entirely by eye, it should be noted that wall radii of curvature should be less at the narrow end of the contraction than at the wide end: the temptation to draw an antisymmetrical S-shaped wall should be resisted, unless particularly mild velocity gradients are required at the high-speed end. The radii of curvature at the wide end should be large enough to ensure that the flow through the last screen is nearly axial. We may note here that an "infinite-length" contraction can be conveniently terminated at the last screen and the walls made parallel upstream of this: since the wall velocity falls to a minimum at the discontinuity in wall slope, any extra adverse gradient introduced by the truncation will occur upstream of the screen.

Occasionally it is desired to keep the contraction exit boundary-layer laminar, for instance for jet test rigs, but in all but the smallest tunnels the boundary-layer is likely to become turbulent near the wide end of the contraction, where the velocity gradient is adverse and the wall concave (leading to Taylor–Görtler instability). A sufficiently rapid contraction will reduce the thickness of the boundary-layer enough for it to become laminar again. Preston's (3) condition for the existence of a fully-developed turbulent boundary-layer is \( U \delta / \nu > 320 \) but the thickness of the layer would have to remain well below this limit for some distance before a steady laminar layer was established.
If a model contraction is to be tested, care may be needed to simulate the transition behaviour of the full-size version. The most convenient arrangement for a model test is a suck-down model fitted with a bellmouth, possibly including the wide-angle diffuser (if any), screens and honeycomb, as well as the settling chamber and contraction. It would be worthwhile to test the model with a non-uniform or asymmetric inlet velocity profile, by obstructing part of the bellmouth, to ensure that any non-uniformities in the full-size tunnel can be accepted.

The inevitability of an adverse velocity gradient at each end of a finite-length contraction can be seen by considering the locus of the point \((U, V)\), representing the velocity components, in the hodograph plane. The point on the wall at \(x = -\infty\) corresponds to \((U, 0)\) and \(x = +\infty\) corresponds to \((nU, 0)\) where \(n\) is the contraction ratio. If the point, \(P\) say, at which the wall begins to curve (i.e. where \(V\) begins to increase) coincides with the point at infinity \((U, 0)\), then the wide end of the contraction must be of infinite length. If \(P\) lies to the left of \((U, 0)\) there is an adverse velocity gradient: if \(P\) lay to the right of \((U, 0)\) then the velocity on the parallel walls would be greater than \(U\), the velocity nearer the centre line would be less than \(U\) by continuity, so that the velocity would have a minimum within the fluid, which is impossible in general.

11. WORKING SECTIONS AND INSTRUMENTATION

The most popular cross-sectional shape for a general purpose tunnel is a rectangle of about \(1/2\) to 1 ratio, with corner fillets. It is usual for the larger dimension to be horizontal to accommodate aircraft models mounted on a balance above the roof or below the floor. The size of the fillets varies: usually they are merely large enough to prevent excessive boundary-layer growth in the corners and to accommodate the lights, but sometimes the tunnel cross-section is arranged to be a good approximation to an ellipse. A worthwhile reduction in cross-sectional area and power consumption may be obtained without much increasing the interference on a wing mounted on the major axis, but the interference on off-centre and floor-mounted models will be increased. Any increase in cross-sectional area to compensate for boundary-layer growth is usually accomplished by tapering the fillets.

Wind tunnel balances are discussed in detail in Ref. 1. The conventional type of balance is of the null-displacement type, in which forces (weight or electromagnetic) are applied, one for each component of force or moment on the model, to different parts of a frame to which the model is attached by rigid supports or by wires. Although the most accurate method in principle is to use weights, solenoids used on the Kelvin current balance principle are much more satisfactory for automatic balancing and data recording: electrical signals are almost invariably to be preferred if any further pro-
cessing is required, but an alternative is the use of a jockey weight on a
leadscrew with, for instance, a shaft digitizer. Strain gauge “balances” are
commonly used in high-speed tunnels in order to avoid the loss of time
involved in manual or automatic balancing systems, and might well be
used more frequently in low-speed tunnels although their accuracy is
smaller than that of a weighbeam-type balance, which may be accurate
to one part in 10,000 of full load, and resistance strain gauges are notoriously
sensitive to temperature.

Manometer systems, like balances, can be divided into two classes,
automatic-reading (and expensive), and hand-operated (and cheaper).
Automatic systems for high-speed tunnels have been described by Frederick
et al.\(^{30}\)

Flow visualization methods have been described by Maltby and Keating\(^{31}\)
who add a warning\(^{32}\) about the carbon monoxide content of products of
combustion.

More refined testing techniques, such as hot wire anemometry and un-
steady flow measurement, are outside the scope of this review.

12. TUNNELS FOR SPECIAL PURPOSES

12.1. Smoke tunnels

The chief difficulty in smoke tunnel design is the introduction of smooth
laminar smoke filaments into the airstream, and the need to avoid turbu-
lence in the filaments or in the wake of the smoke strut limits the maximum
speed rather severely: speeds of more than 50 ft/sec are rarely achieved
although smoke introduced in the settling chamber has been used success-
fully in a supersonic tunnel. Fortunately, the types of investigation for
which smoke techniques are most useful do not usually require a parti-
cularly high Reynolds number, and a low speed facilitates observation of
unsteady flow.

A 20 per cent thick smoke strut designed by M. T. Gee is used in the
NPL smoke tunnel. The trailing-edge radius is 2 per cent of the chord, with
smoke tubes 3 per cent chord in diameter and 14 per cent chord long. Suction
is applied over the rear part of the surface to suppress the wake. The trailing-
edge flap sections, fitted between the smoke tubes and 7 per cent chord in
length, have to be carefully adjusted to eliminate trailing vorticity and ensure
equal spacing of the filaments throughout the working section. For the same
reason the tubes must be accurately parallel to the free stream direction. An
improvement in maximum speed can be achieved by putting the smoke strut
in a (two-dimensional) contraction, where the favourable pressure gradient
helps to eliminate mean velocity variations. The best type of aerofoil section
is one with a linear adverse velocity gradient downstream of the suction peak:
this gradient can be made numerically less than the velocity gradient in the
contraction so that the velocity over the smoke strut increases monotonically.
If this contraction is followed by a contraction in the other dimension, then
the Reynolds number of a smoke tube producing a given size of filament in
the working section is reduced by a factor equal to this latter contraction
ratio, but clearly a wholly two-dimensional contraction is useless in this
latter respect. A double contraction of this sort is used in the Princeton
University 12 in. × 16 in. smoke tunnel. Another approach is to place
the smoke strut just upstream of a screen in the settling chamber, usually
with the smoke tubes projecting through the screen to prevent condensation
troubles: the screen must be of very uniform weave and flatness to avoid
deflection of the filaments.

Apart from the smoke strut, the chief requirement of a smoke tunnel is low
turbulence to prevent diffusion of the filaments: the total angle of spread of
a passive contaminant is about 2.5 °/U radians or 0.03 in./ft for °/U = 0.1
per cent r.m.s. Some care is needed with the room ventilating arrangements:
even if the comparatively inoffensive vaporized paraffin “smoke” is used,
forced ventilation is advisable. Natural ventilation, if sufficient, tends to
produce unsteadiness of the tunnel flow.

12.2. Boundary-layer tunnels

Tunnels intended for boundary-layer studies can have shallow working
sections to economize on power, and to facilitate control of the pressure
gradient. Blower tunnels are very suitable for this sort of work. If the tunnel
is fitted with a diffuser, some steps are necessary to prevent separation of the
boundary layers, which will be proportionately very thick. The NASA Lewis
boundary-layer channel which is sucked down by an exhauster pump
has a flexible roof with auxiliary suction. Clauser's boundary-layer tunnel had
screens at the downstream end of the working section to provide back
pressure which discharged air from the working section roof for pressure
gradient control. A brief description of the Liverpool University blower
tunnel (Fig. 34) is given by Gibson. The NPL boundary-layer tunnel
(BLT) is shown in Fig. 32.

12.3. Slotted working sections

The Admiralty Research Laboratory water tunnel has a slotted
working section intended to reduce wall interference on large cylindrical
bodies, and this principle could be applied more generally to low-speed
tunnels, though trouble was experienced with severe pressure fluctuations
due to jet instability in the water tunnel, and the corrections to be made
for wall interference are uncertain. High lift models may require large cor-
rections for tunnel interference, and when it is desired to simulate the motion
of an aircraft near the ground it is strictly necessary to provide a moving
floor to eliminate the effect of the tunnel wall boundary-layer: some tests in
the NPL smoke tunnel have shown, however, that application of suction to the tunnel floor will usually give a sufficient approximation to the flow with a moving floor.

12.4. *Tests of floor-mounted models*

Tunnels for industrial or non-aeronautical use may also require attention to the tunnel floor boundary-layer if the flow round a model mounted on the ground is to be investigated, but in many cases the provision of a false floor, extending a little upstream of the likely flow field of the model, should be sufficient. The use of an inclined screen to simulate a natural wind gradient has been described by Elder\(^{(38)}\): other methods are also used.

12.5. *Water tunnels*\(^{(55, 64)}\)

These are generally similar in layout to wind tunnels, except that the circuit is usually vertical, with the fan at the bottom to minimize cavitation. Tunnels deliberately used for cavitation studies require some method of ensuring re-absorption of air bubbles to keep the concentration of dissolved air constant: a device first used in the United States is an extra, low-velocity, duct in the return circuit located at the lowest possible level, since the re-absorption time is long and decreases with increasing pressure. Water takes 73 sec to pass through the re-absorber of the ARL 30 in. water tunnel (Fig. 29) at 60 ft/sec in the working section. Care has to be taken to prevent contamination of the water and corrosion of the structure. Because of the large temperature coefficient of viscosity, careful control of temperature is important, and continuous refrigeration is necessary in larger tunnels.

12.6. *Cascade tunnels*

Tunnels for general cascade investigations are usually of the blower type (Fig. 36): the flow leaving the cascades may pass straight to atmosphere or, if it is desired to constrain the outlet flow direction, into a short parallel duct. Boundary-layer suction is usually applied to the side walls of the tunnel to reduce secondary-flow effects and permit the simulation of two-dimensional flow.

Other special types, such as icing or spinning tunnels,\(^{(39)}\) are generally major undertakings requiring more detailed study than that provided by the present paper.

13. CONSTRUCTION

The aerodynamic design of a tunnel is usually influenced by structural considerations, particularly in point of cross-section shape. Even if a tunnel is made from concrete or plastic, moulds have still to be constructed. Sheet metal can be spun into cylindrical or conical shape but this is inconvenient
for the larger sections of a tunnel, and metal tunnels often need the application of a damping material such as “aquaplas” to prevent structural vibration. In small tunnels, fibreglass is the most convenient material for sections with compound curvature: plaster-of-paris moulds, shaped by a plane template, can be used for surfaces of revolution. Most small low-speed tunnels are chiefly made of plywood in a timber frame, but this method has been used for quite large and fast tunnels, such as the RAE 4 ft x 3 ft tunnel (290 ft/sec). The Bristol Aeroplane Company 12 ft x 10 ft tunnel is of wooden sandwich construction.

Wooden tunnels differ from most other wooden structures in having a framework on the outside and panels on the inside, which leads to minor difficulties. Sketches of suitable layouts for octagonal and square sections are shown in Figs. 6 and 7. Drumming of plywood panels must be kept to a minimum, preferably by keeping the panel resonant frequencies above the fan blade frequency at maximum speed. In most cases deflection rather than ultimate strength will decide the material size: large deflections cause leaks to appear on repeated loading. The elastic modulus of wood can be taken as $5 \times 10^6$ lb/in.$^9$ It is safest to design the whole of a return-circuit tunnel to withstand the full stagnation pressure difference, as at least three-quarters of this pressure will have been reached by the end of the first diffuser. The settling chamber of an open-circuit tunnel needs only to support its own weight, but the rest of the tunnel should be able to withstand the working-section static pressure difference. The RAE 4 ft x 3 ft tunnel was designed with a plenum chamber round the working section and pressure-equalizing vents in the return circuit: the settling chamber of this tunnel is a 22 ft octagon. The tunnel seems, however, to have been run at maximum speed with the working section at atmospheric pressure. Transient pressures on
starting and stopping will not exceed the stagnation pressure: only one instance of the failure of a tunnel working section during stopping has ever come to light, and the chief considerations in working section design are ease of access, model mounting and modification.

Metal corner vanes are best mounted between carefully shaped distance pieces in a separate frame. Details of a press tool used at NPL are shown in Fig. 8. The vanes must be prevented from twisting, and on the larger corners some cross-bracing may be needed.

Except in very large tunnels the motor will normally be mounted outside the tunnel rather than in the fan nacelle. The mass and vibration of rotating parts could be reduced by the latter arrangement, but standard electric motors do not seem to be small enough to fit easily in a nacelle of 0·6 times the fan diameter unless the tunnel is fairly large: for given speed and power factor, the motor diameter will vary approximately as the ½ power of the tunnel diameter. The extension shaft bearing must be very firmly supported, and it is convenient to attach it via the pre-rotation vanes to a frame encircling the tunnel structure and let the fan overhang. Multiple bearings on rigid shafts are to be avoided: motor vehicle propeller shafts with universal joints at each end are suitable for small tunnels. It should not be necessary for mechanical purposes to fit a bearing where the shaft passes through the second corner vanes but the shaft should be faired where it is not parallel to the airstream. The problem is less severe for an open-circuit tunnel which usually has fans mounted directly on the motor shaft at the diffuser exit.

The larger wooden sections of the tunnel may need a metal supporting structure, and the fan bearing will almost certainly need to be connected to

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**Fig. 7. Stages in construction of square sections.**

NPL BLT sizes: main frames 3 in. × 1½ in., plywood ¼ in., longerons 3 in. × 2 in. (max. cross-section 5 ft square, section length 5 ft, internal pressure 30 lb/ft².)
the motor by a rigid metal or concrete structure, probably including the floor of the room.

Wind tunnel fans are almost invariably made of wood but moulded fibreglass construction might be attractive for a large many-bladed fan,

![Diagram of press tool](image)

**Fig. 8.** Press tool for 12 in. arc length, 86° corner vanes: material — laminated mahogany.

**Tool dimensions**

Vane dimensions: Arc length 12 in. (10 in. circular arc with 1 in. straight extensions at leading and trailing edges). Radius 6.75 in., chord 11.16 in.

Material — 99% aluminium, 16 s.w.g.

Press load — 1 ton

The tool dimensions to allow for springing back were found by a trial with a sample of the material used, but should be fairly accurate for other samples of "pure" aluminium.

although the difficulty of making sharp trailing edges should not be underestimated. It should not normally be necessary to fit adjustable trailing edges to the fan blades but as stated in Section 6 it may be wise to fit them to the pre-rotation vanes, which, if they are to support the fan bearing, can conveniently be light alloy castings. The straightener vanes, which normally support the fan nacelle tail section, are simply made from solid wood, or ply
on a wooden frame. The fan duct should have some sort of metal reinforcement near the fan position in order that the gap between the duct and the blades can be kept small. $\frac{1}{30}$ in. per foot blade radius is a figure to aim at: Wallis\(^6\) quotes the percentage loss in efficiency for axial compressors as about $(300 \times \text{clearance})/(\text{blade lengths})$.

Contractions are normally built from plywood on a wooden frame: frequent cross-members are needed to support the thin plywood necessary. Extra strength may be obtained by bending the ply over a suitable pattern, gluing on a continuous layer of square strips laid along the generators, and completing the sandwich with another sheet of ply to form a rigid structure which can then be screwed into the support frame with other similar slabs to form a polygonal duct. An extremely strong structure for large panels can be built in this way by using sheet metal instead of ply: aluminium-faced end-grain blockboard is commercially available, at least in plane sheets.

Screens have been supported in many different ways: the simplest attachment for small wooden tunnels is to clamp the gauze between two wooden frames which can then form part of the pressure-bearing structure. Small screens can be pre-tensioned and installed flat, but larger screens or those bearing high air loads should be allowed to sag slightly, though they should still become taut enough to eliminate wrinkles when the tunnel is run, at least at the higher speeds. The RAE 11 ft \( \times \) 8$\frac{1}{2}$ ft No. 2 tunnel\(^44\) screens have a sag of $1\frac{1}{2}$ ft in 24 ft but there seems to be no effect on the mean velocity distribution. As an example of the stresses involved, we may note that the radial force per unit length of perimeter of a circular screen of diameter $d$ and sag $\delta$, with a pressure drop $p$, is $p \frac{d^2}{16} \delta$, so that a 7 ft diameter screen of pressure drop coefficient 2, passing air at 12 ft/sec with a sag of 3 in. would have a tension of 17 lb/ft. If the screen had 20 0.015 in. diameter wires per inch the average stress in each wire would be 400 lb/in.\(^2\) Considerable stress concentrations are likely to occur and the nominal wire stress should always be kept an order of magnitude below the ultimate, especially if there are joins in the screen. Wire screens are made principally for the paper-making trade and widths of more than 6 ft are not easily obtained: joining by brazing needs some care, but almost invisible seams can be produced.

The honeycomb can also be mounted in a wooden frame. In the NPL boundary-layer tunnel the honeycomb and six gauze frames are clamped together by long drawbolts secured to the settling chamber frame.

The sections of the tunnel must have leak-proof joints. Rubber gaskets, or, for sections which are unlikely to be dismantled, sealing tape, can be used, but leak testing after assembly may be advisable in any case. The leaks from the NPL boundary-layer tunnel return circuit totalled 150 ft\(^3\)/min when first tested: this was only $\frac{1}{3}$ per cent of the working section flow rate but was rather too high for the special purposes of the tunnel. The leaks were reduced to 100 ft\(^3\)/min by the use of adhesive cloth tape.
Closed-circuit tunnels built entirely of wood may suffer from a large temperature rise which, for tunnels of different size, but similar shape, is proportional to $U^3t$ where $t$ is a typical panel thickness: if the panel deflections are kept at a constant percentage of diameter $d$ of the tunnel, $t \propto U^{2/3}d$, so that $\Delta T \propto U^{11/3}d$. A tunnel of $4 \text{ ft}^2$ working section area and $2000 \text{ ft}^2$ surface area, built of $\frac{1}{2} \text{ in.}$ thick wood and absorbing 12 h.p. would have a final temperature rise of about $5^\circ\text{C}$: this would produce a change of 2 per cent in Reynolds number if the tunnel were run at constant dynamic pressure, and would seriously interfere with the use of hot wires. However the insertion of a few metal panels, preferably near the end of the second diffuser, would greatly reduce the temperature rise. Tunnels with high maximum speeds may need refrigeration arrangements.

14. TUNNEL TESTING

The immediate sign of any misbehaviour is the failure of the tunnel to reach its design speed: even small shortcomings may indicate malfunction, almost certainly in the first diffuser or fan. If the deficit is serious the first step is to find the fan power by measuring the electrical power input and allowing for the motor efficiency and bearing friction, preferably by running the motor with its extension shaft but without the fan. If the fan is absorbing the design power at the design speed (the speed of a shunt-wound d.c. motor varies very little unless it is grossly overloaded) the velocity profile at entry to the fan section should be measured to see if it corresponds to that chosen for the fan design calculations. If the fan is absorbing too much power for a given shaft speed, blade stall is indicated: this may also be caused by a poor entry velocity profile. If the entry velocity profile is fairly close to that assumed, then errors must have occurred in the fan design calculations or power factor estimation. A useful way of checking blade stall is to traverse a hot wire radially behind the fan and calculate the wake width from oscilloscope traces. More detailed investigation of the fan behaviour should obviously start with the quantities in whose calculation the designer has least confidence, but a check on the pre-rotation vane swirl and wakes might usefully precede any observations on the fan itself.

Deficiencies in the fan entry velocity profile can be traced to malfunction of the first diffuser or, less probably, of the corner vanes in a closed-circuit tunnel. Separation in the first diffuser may be an unsteady phenomenon caused by, or interacting with, a separation (and resulting unsteady flow) elsewhere, but in any case it indicates an excessive cone angle or a thick boundary-layer at entry. Vortex generators may be sufficient to clear up a slight separation.

It is pertinent, if not comforting, to remark that malfunction of one component of a wind tunnel designed too near the limit may precipitate trouble in the other sections which will disappear when the origin of the trouble is
removed, as in the case of the RMTC 4½ ft x 3½ ft tunnel. The best way of
detecting separations is probably to use a Preston surface pitot tube to
measure the surface friction coefficient, as flow visualization methods are
difficult to use inside a tunnel: tufts may be adequate if they can be seen.
Windows in the tunnel wall rarely provide a good enough view: an axial
view is to be preferred and a periscope might be useful if the walls can be cut
out for its insertion. It is worth installing a few flush-fitting electrical sockets
for inspection lamps at different parts of the circuit. Any tendency to asym-
metric flow can be observed by reading the difference in total pressure
recorded by two pitot tubes on opposite sides of the tunnel, but the origin of
asymmetry must be sought at the corners.

If the turbulence in the working section is higher than anticipated or
desired, a frequency spectrum should be measured immediately. Peaks at
the fan shaft frequency indicate unbalance and peaks at the fan blade fre-
quency imply an asymmetrical velocity profile at entry to the fan, whose
origin is to be sought at the corner vanes. Low frequency contributions are
the result of unsteadiness in the return circuit, as discussed above, or possibly
in the contraction. Higher frequency turbulence indicates that the screens
and honeycomb are ineffective: little can be done about the intensity of
turbulence—as distinct from unsteadiness—in a diffuser, as the mean
velocity and turbulence will in any case approximate to fully-developed pipe
flow.

An interesting and well-documented search for the origin of tunnel
turbulence is that carried out on the RAE 4 ft x 3 ft tunnel\(^{(50-53)}\): most of
the trouble was traced to separations in the wide-angle diffuser and contract-
ion, and it was acknowledged that these two components were too close to-
gether.

The contraction ratios of 6 to 12 commonly used in modern tunnels should
be sufficient to provide a very uniform mean flow in the working section
unless the last screen is of poor quality. Total and static pressure differences
from the reading of a pitot-static tube at, say, the centre of the working
section can be read on a Chattcock gauge or other sensitive manometer.
Measurement of small flow angles is very difficult and is probably not worth
while: static pressure variations give the same information, as

\[
\alpha = \frac{1}{\rho U^2} \int \frac{\partial p}{\partial y} \cdot dx. 
\]

A useful method of finding the mean incidence over the area of a wing is to
measure its lift when set at approximately the nominal zero-lift incidence to
the horizontal, both upright and inverted. The difference between the two
values of \( C_L \) is \( 2\varepsilon \partial C_L/\partial \alpha \), where \( \varepsilon \) is the average deviation of the tunnel
stream from the horizontal in the vicinity of the wing. If the tunnel has no
balance, the difference between surface static pressures near the leading
edge can be used instead, employing the wing as an extra-large yawmometer.
15. ANALYSIS OF THE FEATURES OF SOME MODERN TUNNELS

In this section we descend from the general to the particular and comment on the good and bad features of some tunnels built since about 1937, when the importance of low turbulence was realized. The tunnels have been chosen on the basis of availability of information; other tunnels might have provided better examples had details been readily available, but designers seem reluctant to investigate the flow in their tunnels properly unless something is wrong, and reluctant to publish the results in either case. We hope that those whose results and descriptions we have used will forgive us the clarity of our hindsight; by way of demonstrating impartiality we include an analysis of the faults of the NPL boundary-layer tunnel. A table of dimensions and performance of tunnels of various ages is appended.

15.1. National Bureau of Standards (U.S.A.) 4½ ft tunnel (1938) (Fig. 11)

This was one of the first tunnels to be built after the realization that low turbulence was desirable, and it is still classed as a very good tunnel, especially in view of its simple layout. The position of the fan in front of the first corner does not seem to have affected the flow through the corner vanes sufficiently to cause unsteadiness, even though no straightener vanes are fitted, and the noise level in the working section is clearly not excessive: it may be presumed that the improved symmetry of flow into the fan compensates for its nearness to the working section. No honeycomb is fitted but the uniformity of flow in the working section is good enough to depend only on the last screen: the moderate contraction ratio of 7:1 is too small to make the effect of screen inhomogeneity negligible and axial vorticity shed by the last screen was found to produce spanwise variation in thickness of the laminar boundary-layer on a flat plate. The corner guide vanes are of very small chord (6½ in. in the first to third corners and 2½ in. in the fourth) which may help to reduce large-scale turbulence in the return circuit.
15.2. RAE Farnborough 11½ ft × 8½ ft No. 2 tunnel (1942) (Fig. 13)

The chief defect of this tunnel is the employment of 7° diffusers, with the result that a local separation caused by too rapid a change from circular to square section downstream of the fan, made asymmetrical by underturning at the second corner, ruined the flow in the second diffuser. As in the case of the RMTC tunnel it was only possible to find the source of the trouble after curing the flow in the working section because of the positive feedback of disturbances. The 11½ ft tunnel was calibrated before the invention of vortex generators, which would probably have suppressed the separation, and the course finally adopted was to insert an "eggbox" of horizontal and vertical
Fig. 13. RAE 11½ ft x 8½ ft No. 2 tunnel (1942).
splitters in the second diffuser, together with a honeycomb and three screens in the settling chamber. The chief function of the eggbox is probably to provide resistance (made asymmetrical by the addition of more slats on the outside of the circuit) rather than to prevent lateral motion. The power factor increment due to the eggbox was 0.06, and the screens increased the power factor by 0.1, both at 400 ft/sec. The air is cooled by brine circulated through the first, third and fourth corners.

15.3. *NPL 13 ft × 9 ft tunnel*(46, 70) (1942) (Fig. 14)

The performance of this tunnel is better than might be expected from its undesirable features. There is an expansion round each of the four corners, whose vanes are very widely spaced: the flow separates from the first corner and the outer part of the second corner, and the third and fourth corners under-turn slightly. As the first corner is immediately downstream of the working section with no diffuser in between, its contribution to the power factor is immense. A freely-rotating windmill in an *irregular octagonal* section just before the third corner helps to reduce mean velocity variations, but introduces pulsations which contribute greatly to the turbulence in the working section: windmill blade resonances occur at several speeds. The NPL *9 ft × 7 ft* tunnels also have windmills, in regular octagonal sections, but the pulsations are much less severe: the mean flow distribution at the windmill is probably better than in the *13 ft × 9 ft* tunnel. The vorticity fluctuations in the *13 ft × 9 ft* working section are fairly small, thanks to the long settling chamber which helps to make up for the absence of screens. The contraction shape is of the type favoured just before the war, with a gradual sweep into the working section after a fairly sharp initial curvature: this arrangement produces severe static pressure gradients in the first few feet of the working section. The total-pressure variation across the width of the working section is about 2 per cent. (The corners have since been modified.)

15.4. *RAE Farnborough 10 ft × 7 ft high-speed tunnel*(47) (1942) (Fig. 15)

Like the CAT (Fig. 10) this variable-density tunnel has an annular return circuit with a very poor shape at entry to the settling chamber. Guide vanes had to be fitted to the inner wall, and mixing vanes were installed in the return circuit. The longitudinal component of turbulence at low speeds was 0.4 per cent. The tunnel has now been modified to have an *8 ft × 6 ft* working section, and the low-frequency lateral component, measured with a *wing-mounted* microphone, is now about 0.7 per cent(66). The annular return circuit is the most economical layout for a pressure tunnel but none has yet been found to give really good flow. The tunnel fan is very heavily loaded and has 13 blades. There is a gradual entry to the
diffuser and a gradual change to a cylindrical duct round the fan, instead of the usual conical diffuser: the former refinement is advisable to retard the formation of shock waves in a highly subsonic tunnel.

![Diagram of tunnel](image)

**FIG. 15.** RAE 10 ft × 7 ft high-speed tunnel (1942).

![Diagram of tunnel](image)

**FIG. 16.** Cavendish Laboratory tunnel (1940).
15.5. RAE Farnborough 4 ft × 3 ft tunnel (1946) (Fig. 18)

This is one of the best tunnels ever built, and its only bad features are a rather ill-shaped fan nacelle, whose wake persists as far as the screens, and an insufficient distance between the rapid expansion and the contraction resulting in a tendency to separation downstream of the screens and consequent regions of "high" turbulence in the working section. It is now realized that a contraction ratio of over 30 is not really necessary for most purposes: the variation of velocity over the central portion of the working section was less than ±0.1 per cent with 20 mesh 0.017 in. diameter screens, and
uniformity of this order is only likely to be needed for fundamental laminar-flow stability experiments. However a study of this tunnel is of great use to designers, not least because the flow has been painstakingly explored and reported.

15.6. **NACA Ames 12 ft pressure tunnel (1946)** (Fig. 19)

This is one of a number of pressure tunnels built with 180° “racecourse” bends, which have structural advantages over sharp corners in a pressure shell. This tunnel, for instance, has nine cascades of vanes in the first bend,

![Diagram of NACA Ames 12-ft low-turbulence pressure wind tunnel (1946).](image)

and so the lift of each cascade is only \( \sin 10^\circ / \sin 45^\circ = 0.25 \) of a 90° cascade: thus the gap/chord ratio can be larger, although the total area of all cascades is \( \frac{\pi}{2 \sqrt{2}} \sin \left( \theta / 2 \right) \) = 1.1 times that of two 90° corners where \( \theta \), the deflection angle, is 20°. Scale effect will give a higher drag coefficient: clearly there is no advantage in using this sort of corner in an ordinary tunnel, although a racecourse bend was at first proposed for the RAE 4 ft x 3 ft tunnel.

The other features of the tunnel include a rapid expansion of area ratio 2 without any screens, and turbulence-reducing screens of only 12 and 16 mesh. The first diffuser has an area ratio of 4, but the working section is only 2.2 diameters long, giving a reasonably thin inlet boundary layer.

FIG. 22. English Electric Co. 9 ft × 7 ft tunnel (1949).

FIG. 23. Wichita University 10 ft × 7 ft tunnel (1950).
Fig. 24. Vickers Armstrongs (Weybridge) 13 ft × 9 ft tunnel (1951).

Fig. 25. Imperial College 5 ft × 4 ft tunnel (1951).

Fig. 26. Manchester University low-turbulence tunnel (1949).

$k$ denotes a pressure drop equal to the dynamic head.
15.7. Royal Melbourne Technical College (Australia) 4½ ft × 3½ ft tunnel \(^{(61)}\) (1953) (Fig. 27)

This tunnel has a number of cumulative bad features: the diffuser cone angles are rather large, and the corner vanes are widely spaced. Because of the low contraction ratio (4 : 1) and the low resistance in the return circuit (power factor 0.18), disturbances arising from underturning at the corners are fed back round the circuit, causing an asymmetric separation in the first diffuser. The return circuit resistance was increased by installing a 60 mesh screen ahead of the honeycomb (the position was doubtless dictated by structural considerations). The fan then stalled, and the working section velocity distribution, though much improved, developed a minimum at a radius of about nine inches each side of the centre line. Finally, the screen was removed, and vortex generators were installed at entry to the first diffuser, at a rather sharp expansion just before the fan, and at the expansion caused by the tapering tail cone. The working section velocity distribution was then slightly inferior to that obtained with a screen, but the power factor was the same as in the original tunnel whereas the screen increased it threefold.

A further useful modification would be the replacement of the hemispherical fan nacelle nose by a more conventional shape, thus reducing the expansion just before the fan and probably reducing the fan root boundary-layer thickness.

This demonstration of "instability" of tunnel flow to small disturbances gives some clue to the large differences in performance which exist between almost identical tunnels. The RMTC tunnel is, except for the fan section and corner vanes, a model of the Australian ARL 9 ft × 7 ft tunnel, but the flow in the latter appeared to be fairly satisfactory.\(^{(62)}\)
15.8. RAE Bedford 13 ft × 9 ft tunnel\(^{(21)}\) (1954) (Fig. 28)

Unlike the 4 ft × 3 ft tunnel, which was intended to explore design features for the Bedford tunnels, the 13 ft × 9 ft has no bulge. The area ratio is 4·0 to the first corner and 6·5 to the fan. The working section is 3·3 diameters long and vortex generators were fitted in the first diffuser, whose entry boundary-layers were rather thick, to prevent separation. The flow in the second diffuser was then adversely affected by the modification of the fan entry velocity profile but was cured by introducing a grid to increase resistance over the inner portion of the fan disc. The tunnel power factor is about 0·22, but would scarcely have been increased by reducing the diffuser length and fitting a bulge: it seems likely that contractions of the order of 16 will be more than sufficient for future low-speed tunnels and that designers will generally prefer to employ a rapid expansion with screens at the end of a diffuser of moderate area ratio.

15.9. ARL 30 in. water tunnel\(^{(64)}\) (1954) (Fig. 29)

This tunnel has a slotted-wall working section 5·5 diameters long, but it has been necessary to modify the diffuser entry to reduce resonances caused by the streamwise amplification of fluctuations in the flow near the walls, and the usable length is only 4·5 diameters. These resonances were investigated theoretically in Ref. 65 and it was shown that the use of a constant pressure plenum chamber round the working section would virtually eliminate the trouble: this is easy to arrange in a wind tunnel, but the plenum chamber of a water tunnel is necessarily of constant volume. The slotted wall arrangement was chosen to allow the testing of relatively large bodies of revolution, with diameters as large as one-third of the working section diameter, without excessive tunnel interference. In practice it has been found that the boundary conditions appear to vary with Reynolds
number. The first diffuser has an entry diameter 1.1 times the working section diameter to collect the whole of the flow: the diffuser angle is 6.2° and the area ratio is 3.76 to the first corner. A separation occurs in the diffuser and appears to be the cause of the low-frequency unsteadiness which makes it necessary to average readings over periods of at least a minute: trials with one form of vortex generator produced no improvement. Experience with this tunnel has shown that while a slotted wall does provide an economical tunnel design capable of accommodating large, long models, there are disadvantages: not enough is known yet about how to minimize these.

15.10. *AVA Göttingen 3 m tunnel* (1957) (Fig. 30)

The working section length of this open-jet tunnel is about twice the working section height, so that a power factor contribution of at least 0.16 is to be expected. The flow is turned through two 180° bends with staggered guide vanes over the inner half only: the first bend has an area expansion of 1.5 : 1 and the second bend an expansion of 3 : 1. There is a rapid change of section from circular to 3 : 1 rectangular just before the second bend. It was originally intended to fit rotating circular cylinders at the inner walls of the bends, to prevent boundary-layer separation, but model tests showed that the power required would be excessive. Since the tunnel power factor is 0.78 (say 0.6 if a closed working section were fitted) of which only 0.06 is due to the screens the bends cannot be said to diffuse very efficiently, and their failure to turn the flow adequately is demonstrated by a 2 per cent variation in total pressure across the working
section, although the longitudinal component of turbulence is only 0.05 per cent and the sphere critical Reynolds number is 390,000 (free air $Re_{crit} = 410,000$), defined here as the Reynolds number at which the pressure at the rear of the sphere becomes equal to ambient pressure.

Fig. 30. AVA 3 m open jet tunnel (1959).

Fig. 31. Indian Institute of Science 14 ft × 9 ft tunnel (1960).
15.11. *NPL Boundary-layer tunnel (1961)* (Fig. 32)

The first diffuser combines a $5^\circ$ expansion with a change of shape from a 59 in. $\times$ 9 in. rectangle to a regular octagon in a length of 7$\frac{1}{2}$ ft. Although tangential blowing is used to clean up the boundary layers on the long sides of the rectangles, the boundary layers on the corresponding sides of the octagon are much thicker than on the other sides, and since diffusion continues right up to the fan the entry velocity profile has minima near roof and floor, resulting in a component of vibration at twice the fan blade frequency. The only solution apart from extensive boundary-layer control would be a more gradual section change and a smaller diffuser angle, but the "turbulence" level in the working section is quite satisfactory. Originally, the 3 in. diameter fan shaft had no fairing at the second corner: the power factor increment was about 0.03.

Table 1 gives details of all the recent general purpose tunnels on which reports are accessible to us, together with some information supplied by private communication, but it does not pretend to include all the interesting tunnels designed in the last few years in universities, firms and elsewhere. Some of the area ratios have been found by scaling rather small published drawings and may not be absolutely accurate. The dates quoted are usually the commissioning dates, but in some cases only the date of the published report is immediately available. We would welcome corrections or additions to the list.
Table 1. Some tabulated features of low-speed tunnels

<table>
<thead>
<tr>
<th>Name</th>
<th>Date</th>
<th>max speed ft/sec</th>
<th>h.p.</th>
<th>λ</th>
<th>A₁</th>
<th>A₂</th>
<th>A₃</th>
<th>Fig.</th>
<th>Ref.</th>
</tr>
</thead>
<tbody>
<tr>
<td>NPL Duplex 14 ft × 7 ft open circuit</td>
<td>1919</td>
<td>100</td>
<td>400</td>
<td>1.9</td>
<td></td>
<td>2.1</td>
<td></td>
<td>9</td>
<td>40</td>
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<tr>
<td>NPL Compressed Air Tunnel 6 ft dia. open jet 25 atm</td>
<td>1930</td>
<td>90</td>
<td>500</td>
<td>0.44</td>
<td>1.64</td>
<td>1.64*</td>
<td>3.6</td>
<td>10</td>
<td>41</td>
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<tr>
<td>NBS 4½ ft octagonal</td>
<td>1938</td>
<td>75</td>
<td></td>
<td>2.33</td>
<td></td>
<td>2.33*</td>
<td>7.1</td>
<td>11</td>
<td>42</td>
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<td>Cavendish laboratory 15½ in. × 15½ in.</td>
<td>1940</td>
<td>75</td>
<td></td>
<td>1</td>
<td>1.87</td>
<td>11.4</td>
<td></td>
<td>16</td>
<td>48</td>
</tr>
<tr>
<td>Langley 7½ ft × 3 ft pressure tunnel, 10 atm</td>
<td>1941</td>
<td>470</td>
<td>2,000</td>
<td>0.33</td>
<td>5.1</td>
<td>5.3</td>
<td>17.8</td>
<td>12a</td>
<td>43</td>
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<tr>
<td>RAE 11½ ft × 8½ ft No. 2</td>
<td>1942</td>
<td>415</td>
<td>4,800</td>
<td>0.32</td>
<td>2.2</td>
<td>2.22</td>
<td>6.18</td>
<td>13</td>
<td>44</td>
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<td>NPL 13 ft × 9 ft</td>
<td>1942</td>
<td>240</td>
<td>750</td>
<td>0.32</td>
<td>1</td>
<td>1.3</td>
<td>4</td>
<td>14</td>
<td>46</td>
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<td>RAE 10 ft × 7 ft high-speed tunnel §</td>
<td>1942</td>
<td>M = 0.85</td>
<td>4,000</td>
<td>0.18</td>
<td>2</td>
<td>2*</td>
<td></td>
<td>7</td>
<td>47</td>
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<td>McKinnon Wood design 13 ft × 9 ft</td>
<td>1944</td>
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<tr>
<td>RAE 4 ft × 3 ft</td>
<td>1946</td>
<td>290</td>
<td>200</td>
<td>0.32</td>
<td>2.4</td>
<td>2.4</td>
<td>31.2</td>
<td>18</td>
<td>50</td>
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<td></td>
<td>11,000</td>
<td>4</td>
<td>4</td>
<td>25</td>
<td>19</td>
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<td>Penn. State 48 in. water tunnel</td>
<td>1948</td>
<td>83</td>
<td>2,000</td>
<td>0.2</td>
<td>3.5</td>
<td>9</td>
<td>21</td>
<td>55</td>
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<td>English Electric 9 ft × 7 ft</td>
<td>1949</td>
<td>187</td>
<td>250</td>
<td>0.32</td>
<td>1.24</td>
<td>1.77</td>
<td>5</td>
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<td>Manchester Univ. 20 in. × 20 in.</td>
<td>1949</td>
<td>120</td>
<td>6</td>
<td>0.53</td>
<td>2.4</td>
<td>3.5</td>
<td>20</td>
<td>26</td>
<td>60</td>
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<td>Wichita Univ. 10 ft × 7 ft</td>
<td>1950</td>
<td>265</td>
<td></td>
<td>1.5</td>
<td>1.5*</td>
<td>6</td>
<td>23</td>
<td>57</td>
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<tr>
<td>Vickers 13 ft × 9 ft</td>
<td>1951</td>
<td>350</td>
<td>2,000</td>
<td>0.2</td>
<td>2.52</td>
<td>3.61</td>
<td>10.7</td>
<td>24</td>
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<td>Imperial College 5 ft × 4 ft</td>
<td>1951</td>
<td>165</td>
<td>50</td>
<td>0.28</td>
<td>2.71</td>
<td>1.71</td>
<td>4.93</td>
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<td>Royal Melbourne Tech. College 4½ ft × 3½ ft</td>
<td>1953</td>
<td>235</td>
<td>130</td>
<td>0.32</td>
<td>2.54</td>
<td>2.5</td>
<td>4</td>
<td>27</td>
<td>61</td>
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<td>Bristol Aeroplane Company 12 ft × 10 ft</td>
<td>1953</td>
<td>300</td>
<td>1,950</td>
<td>0.28</td>
<td>2.14</td>
<td>2.81</td>
<td>6.65</td>
<td></td>
<td>63</td>
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<tr>
<td>RAE 13 ft × 9 ft</td>
<td>1954</td>
<td>300</td>
<td>1,500</td>
<td>0.22</td>
<td>3.7</td>
<td>6.5</td>
<td>16</td>
<td>28</td>
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<td>ARL 30 in. water tunnel</td>
<td>1954</td>
<td>60</td>
<td>850</td>
<td>0.36</td>
<td>4*</td>
<td>2.3*</td>
<td>9</td>
<td>29</td>
<td>64</td>
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<tr>
<td>AVA Göttingen 3 m. open jet</td>
<td>1959</td>
<td>180</td>
<td>1,000</td>
<td>0.78</td>
<td>1</td>
<td>1.5</td>
<td>5.45</td>
<td>30</td>
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<tr>
<td>Liverpool Univ. 4 ft × 2 ft blower tunnel</td>
<td>1959</td>
<td>120</td>
<td>50</td>
<td>1.6</td>
<td>2.65</td>
<td>9</td>
<td>34</td>
<td>15</td>
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<td>Indian Inst. of Science 14 ft × 9 ft</td>
<td>1960</td>
<td>350</td>
<td>1,610</td>
<td>0.15</td>
<td></td>
<td>3.6</td>
<td>14</td>
<td>31</td>
<td>68</td>
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<td>NPL 59 in. × 9 in. boundary-layer tunnel</td>
<td>1961</td>
<td>160</td>
<td>12</td>
<td>0.4</td>
<td>1.7</td>
<td>2.2</td>
<td>12</td>
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<td>Southampton Univ. 7 ft × 5 ft</td>
<td>1961</td>
<td>200</td>
<td>200</td>
<td>0.55</td>
<td>1.3</td>
<td>1.22</td>
<td>4.82</td>
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<td>NPL 36 in. × 27 in. project</td>
<td>1962</td>
<td>200</td>
<td>35</td>
<td>0.3</td>
<td>1.8</td>
<td>2.2</td>
<td>9</td>
<td>1</td>
<td></td>
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<tr>
<td>Cambridge Univ. Eng. Lab. 4 ft × 5½ ft</td>
<td>1963</td>
<td>180</td>
<td>100</td>
<td>0.36</td>
<td>2.8</td>
<td>2.8</td>
<td>7.1</td>
<td>35</td>
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</table>

* Fan before first corner. § All these figures refer to the tunnel before modification to 8 ft × 6 ft (see p. 50)  † sic
Fig. 33. University of Southampton 7 ft × 5 ft low-speed wind tunnel (1961).
(The settling chamber is used as an alternative working section.)

Fig. 34. Liverpool University Engineering Department Blower Tunnel (1959).
Fig. 35. Cambridge University Engineering Dept. 4 ft × 5 ft 6 in. tunnel (1963).

Fig. 36. Cambridge University Engineering Laboratory Cascade Tunnel (1951).
16. CONCLUSIONS

A tunnel with the following features

(a) a 1.5:1 rectangular working section not more than three equivalent diameters long (to prevent excessive boundary-layer growth),
(b) a 5° first diffuser of 2 or 2.5 to 1 area ratio, with a gradual section change to a regular octagon,
(c) corners with thin circular arc vanes subtending an angle of 85-86°, spaced at 0.25 chord,
(d) a constant area duct between the first and second corners,
(e) a fan boss of about 0.6 times the fan diameter, with a length/diameter ratio of 3,
(f) a fan duct giving constant area up to the fan and a subsequent expansion not exceeding the equivalent of a 5° cone downstream,
(g) a 5° second diffuser with a gradual change of section from circular to square and an area ratio of about 3:1,
(h) a rapid expansion of about 2:1 with at least two screens,
(i) a honeycomb with cell length/width ratio of at least 6 or 8, mounted well downstream of the rapid expansion,
(j) about four screens each with a pressure drop coefficient of 1.6, say 16 mesh 28 s.w.g. wire,
(k) a 12:1 contraction which has been proved satisfactory in model tests

should have adequate performance for nearly all purposes. The fan should be designed according to the principles of Section 6 but is not included in the above assurance.

We hope that tunnel designers will either try the experiment of departing from this stereotype or, at least, calibrate their tunnels properly and publish the results, good or bad, so that wind tunnel design may come to involve more science, less art, and no magic at all.

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Cascade tunnels


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