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Technical Notes

Design rules for small low speed wind tunnels

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1. INTRODUCTION

Even with today's computers, a wind tunnel is an essential tool in engineering, both for model tests and basic research. Since the 1930s, when the strong effect of free-stream turbulence on shear layers became apparent, emphasis has been laid on wind tunnels with low levels of turbulence and unsteadiness. Consequently most high performance wind tunnels were designed as closed-circuit types (Fig. 1(*a*)) to ensure a controlled return flow. However, as will be seen below, it is possible with care to achieve high performance from an open-circuit tunnel, thus saving space and construction cost. 'Blower' tunnels (with the fan at entry to the tunnel, Fig. 1(*b*)) facilitate large changes in working section arrangements; to cope with the resulting large changes in operating conditions, a centrifugal fan is preferable to an axial one. For ease of changing working sections the exit diffuser is often omitted from small blower tunnels, at the cost of a power factor greater than unity. This paper concentrates on the design of *small blower* tunnels but most of the information is applicable to wind tunnels in general.

A large open-circuit tunnel would be of rather inconvenient dimensions, mainly in length. Also, an open-circuit tunnel requires enough free room around it so that the quality of the return flow is not affected significantly (remember that an open-circuit tunnel in a room is really a closed-circuit tunnel with a poorly-designed return leg). The choice may also be restricted by the maximum available blower size. A working section Re per metre of more than about 3×10^6 (a speed of about 40 ms^{-1}) is rare in blower tunnels of whatever size, and commercial blowers capable of producing such a speed in a section more than about 1 m^2 in area are also rare.

The main advantage of open-circuit tunnels is in the saving of space and cost. They also suffer less from temperature changes (mainly because room volume $\gg$ tunnel volume) and the performance of a fan fitted at the upstream end is not affected by disturbed flow from the working section. One disadvantage of any open-circuit tunnel with an exit diffuser is that the pressure is always less than atmospheric and so spurious jets issue from holes left unpatched, although this can be remedied by obstructing the tunnel outlet and creating an over-pressure in the working section. The main advantage of a centrifugal blower, as distinct from an axial fan, is that it performs well over a large range of loads (the whole blade being at the same incidence and hence operating at the same lift coefficient). The only advantage of a suction tunnel, with a centrifugal or axial fan at exit, is the dubious one that air coming from the tunnel room *may* be less disturbed than that coming from a fan.

It is difficult and unwise to lay down firm design rules mainly because of the wide variety of requirements and especially the wide variety of working-section configurations. An attempt is made here to present design guidelines for the main components of a wind tunnel—the fan, wide-angle diffuser, corner vanes, settling chamber, contraction and exit diffuser (Fig. 1)—based on data from successful designs and some original experiments. For details of the data correlations see Mehta (1977) and for complete details of the experiments and design procedure see Mehta (1978).

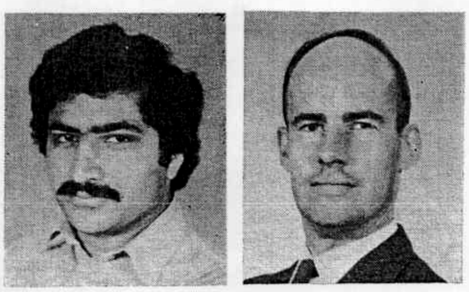
2. FANS

2.1. Axial flow fans

The usual arrangement in a closed-circuit tunnel is a stator ('pre-rotation vanes') upstream of the rotor (the fan proper), designed so that the swirl at exit is zero. In the case of an open-circuit tunnel, swirl present in the flow out of the fan *may* be dissipated before the flow reaches the intake, but a remaining advantage of pre-rotation vanes is that the flow velocity relative to the fan blades is larger than if the stator is absent or located downstream of the fan.

2.1.1. Fan solidity

The design procedure outlined by Bradshaw and Pankhurst (1964) is still an adequate guide. The only serious problem found in fan design that is not found in the design of wings for low-speed aircraft is the interference between the flow fields of the blades. This interference depends mainly on the 'solidity', the ratio of blade chord to the gap between blades (measured around the circumference). Providing that the solidity is less than unity approximately, interference is small enough to be treated as a small correction to the performance of an isolated aerofoil; for higher solidities the flow cannot be accurately related to that round an isolated aerofoil, and data for 'cascades' (rows of aerofoils arranged in the same manner as corner vanes) must be used instead. The solidity varies with radius, and in order to use the same



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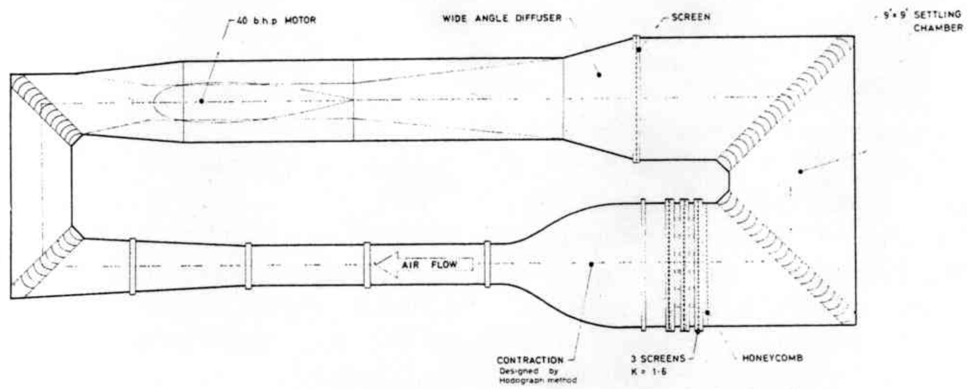

Figure 1(a). The main components of a typical closed-circuit wind tunnel.

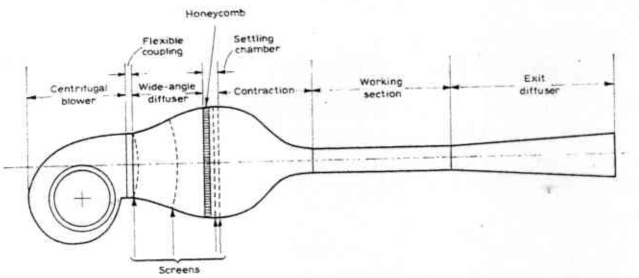

Figure 1(b). The main components of a typical blower tunnel. (Not to scale.)

design procedure for the whole length of the blade it is desirable to keep the solidity at the root below unity by mounting the fan on a central nacelle whose maximum diameter is roughly half the fan diameter.

2.1.2. *Blade design*

Axial fan efficiencies are of the order of 90% so that minimisation of losses is not usually important, and the usual procedure is to choose the blade lift coefficient to be as high as is safe, irrespective of lift/drag ratio; values of 0·7 to 0·9 are typical.

2.1.3. *Pre-rotation vanes*

Pre-rotation vanes should be run at a lift coefficient not too far above that for maximum lift/drag ratio because their wakes pass through the fan; to limit the resulting noise, the axial distance between the trailing edge of the pre-rotation vanes and the leading edge of the fan blades should be at least 20% of the vane chord and the number of fan blades should be different from the number of vanes. Pre-rotation vane solidities usually fall into the cascade range.

An alternative to pre-rotation vanes for a lightly loaded fan is a set of straightener vanes downstream of the fan.

For detailed design rules for pre-rotation vanes, fan blades and straighteners *see* Bradshaw and Pankhurst (1964).

2.2. Centrifugal blowers

Centrifugal blowers are normally used to drive open-circuit tunnels from the upstream end: a blower could be installed at the exit instead but this has no particular advantage. Single-inlet blowers could also be used to drive return-circuit tunnels by installing them in one of the corners. Single-inlet blowers are found to produce a vortex-type flow (due to the asymmetric positioning of the impeller) which would aid wall flow attachment in the wide-angle diffuser. This compensates for the non-uniformity of the flow (which is also improved by the screens in the wide-angle diffuser and the settling chamber).

2.2.1. *Advantages over other fans*

Centrifugal blowers run with reasonable steadiness and efficiency over a wide range of flow conditions (i.e. varying tunnel power factor) because the whole blade span operates at nominally the same lift coefficient. The noise and pulsations generated by a centrifugal blower are adequately low, even at off-design conditions, and the uniformity of flow varies less with advance ratio, $U/\omega r_2$ in the notation of Fig. 2. The swirl (exit vortex) produced by a single-inlet blower is also independent of advance ratio (dependent on the ratio of rotor to casing width).

2.2.2. *Types of impeller*

The most common type of blading is the backward-facing aerofoil-type (Fig. 2); forward-facing is less efficient. If the blower efficiency is not too important, these blades could be designed in the same way as corner vanes or cascades by choosing a leading edge angle of 4-5° and a zero trailing edge angle, but a more efficient blade shape is that of a cambered aerofoil with finite thickness. In the present authors' tests on blowers with aerofoil-type impellers it was found that the flow uniformity deteriorated with increasing loading. However, with backward-facing 'S' shaped blades (Fig. 2) the flow uniformity was found to *improve* with loading, presumably because these blades stall relatively early, leading to increased mixing. The cost is a higher turbulence level in the outlet flow and a reduced blower efficiency.

2.2.3. *Splitter plate (tongue)*

This is an important component which affects the outlet flow uniformity and blower noise characteristics. For minimum interference with the flow uniformity, the ratio of tongue height to casing height needs to be small ($<0 \cdot 3$) and the angle and shape carefully designed. The gap between the rotor and tongue needs to be a minimum for aerodynamic reasons but optimised for minimum interaction with the outgoing flow and thus minimum noise level. The tongue design on most commercially available blowers is adequate. A badly angled tongue could be improved upon by adding a cusped fairing downstream, as shown dotted in Fig. 2.

2.2.4. *Other features and suggestions*

An inlet bellmouth helps to produce a uniform flow and reduces inlet losses, and an inlet filter (helping to reduce inlet swirl) is essential to reduce contamination of hot-wire probes. Large blowers should be mounted on antivibration mountings and connected to the tunnel with a flexible coupling to reduce vibration.

Double-inlet blowers (air entering the impeller from both sides) tend to produce a uniformly inclined flow (without a vortex) which takes a longer distance to re-attach to the bottom wall downstream of the tongue. One should therefore be more conservative in designing wide-angle diffusers for double-inlet blowers.

On the whole, commercially available single-inlet centrifugal blowers with backward-facing impellers are adequate for driving blower tunnels.

Once the maximum required fan static pressure and volume flow rate have been estimated, the makers' performance charts can be consulted. Optimisation between the efficiency, rpm and required power leads to the blower choice (*see* section 10).

3. SCREENS

Wind tunnel screens are normally made of metal wires interwoven to form square or rectangular meshes. Screens woven from nylon or polyester threads are also now being used when the wind loads are not expected to be very high (UTS of nylon ~ 70, steel ~ 1100, bronze $\sim$700–1100 MNm^{-2} and E of nylon $\not> 3$, steel$\sim$200, bronze $\sim 100\,GNm^{-2}$). The action of the gauze is described in terms of two parameters: the pressure drop coefficient, $K = f_1(\beta, R_e, \theta)$ and the deflection coefficient, $\alpha = f_2(\beta, K, \theta)$, where β is the screen open-area ratio and θ is the flow incidence angle, measured from the normal to the screen.

3.1. Main effects

(for detailed explanations *see* Mehta 1978)

Screens make the flow velocity profiles more uniform by imposing a static pressure drop proportional to (speed)^2 and thus reduce the boundary layer thickness so that the ability to withstand a given pressure gradient is increased. A screen with a pressure drop coefficient of about 2 removes nearly all variation in the longitudinal mean velocity. A screen also refracts the incident flow towards the local normal and reduces the turbulence intensity in the whole flow-field. For a given open-area ratio, it is better to have a smaller mesh for the reduction of *pre-existing* turbulence. Plastic screens tend to yield a more uniform flow beyond the boundary layer edge, mainly due to the weaving properties, and produce an 'overshoot' in the velocity profile near the edge, mainly caused by screen deflection angle which is a maximum at the wall. In terms of tackling a given pressure gradient or avoiding separation, this overshoot could be beneficial.

3.2. Open-area ratio (β)

Metal screens with very low β ($\sim 0\cdot3$) also produce an overshoot but this is caused by streamline inclination near the boundary layer edge. Low β ($< 0\cdot57$) screens also produce instabilities resulting from a random coalition of jets and presumably amalgamating to form longitudinal vortices which persist through the contraction. The coalition process is enhanced by variations in β (i.e. non-uniform weave) and by irregularities in the screen shape (i.e. wrinkles). It is therefore essential to inspect and clean wind tunnel screens regularly.

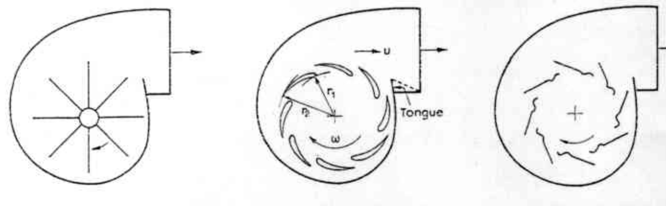

Figure 2. Different impeller types used in centrifugal blowers.

3.3. Determination of K

(ratio of pressure drop to dynamic pressure)

Although there is no *wholly satisfactory* method, Wieghardt's (1953) formula $\{K = 6\cdot5\,[1-\beta/\beta^2]\,[Ud/\beta\nu]^{-1/3}\}$, where d is wire diameter, predicts the right trend; K decreases with increasing speed up to about $Ud/\beta\nu = 600$, after which it is independent of Re. Collar's (1939) formula $\{K = 0\cdot9\,(1-\beta/\beta^2)\}$ usually over-estimates K in the high Re limit. One needs to be more careful in predicting K-values for plastic screens since,

$$K = f(\beta,\ R_e,\ \theta\ .\ .\ .\ \text{co-planarity}\ .\ .\ .).$$

where θ is angle of screen to incident flow. For $\theta \neq 0$ use

$$K_\theta = K\cos^m\theta, \text{ with } m = 1\cdot0 \text{ for screens with } \beta \sim 0\cdot6 \text{ and } m \sim 1\cdot4 \text{ for } \beta \sim 0\cdot3.$$

3.4. Determination of α

(ratio of outlet angle to inlet angle)

For α the form:

$$\alpha = A + \frac{B}{\sqrt{(1+K)}}$$

where A, B are empirical constants, is a better fit than the generally accepted form:

$$\alpha = \frac{1\cdot1}{\sqrt{(1+K)}}$$

Note that the refractive index of a screen (μ) defined as in optics is equal to $1/\alpha$ for small θ. For larger θ use

$$\alpha_\theta = \frac{1}{\theta}\tan^{-1}\left\{\tan\theta - \frac{\theta}{2}\sec^2\theta\left[C - \frac{D}{\sqrt{(1+K_\theta)}}\right](E+F\theta)\right\}$$

C, D, E and F are empirical constants.

Values suggested for the empirical constants by some limited experiments (Mehta, 1978) are: $A = 0\cdot66$, $B = 0\cdot31$, $C = 0\cdot68$, $D = 0\cdot62$, $E = 1\cdot0$, $F = 1\cdot5$.

A more complete analysis of the flow through screens can be found in Mehta (1978).

4. DIFFUSERS

The flow through a diffuser depends on its geometry defined by the area ratio (A), diffuser angle (2θ), wall contour and diffuser cross-sectional shapes. Other parameters like the initial conditions, boundary layer control method and the presence of separation could also affect the flow thus making it very difficult to predict. Almost all knowledge acquired about diffusers is empirical. There are two main types:

4.1. Exit diffusers

These are fitted downstream of the working sections and have gentle expansions with a diffuser included angle usually not exceeding 5° (for best flow steadiness, although best pressure recovery is achieved at about 10°) and an area ratio not exceeding about $2\cdot5$. It is important to have a reasonable degree of flow steadiness in the exit diffuser, since otherwise the pressure recovery tends to fluctuate with time, and, therefore, so does the tunnel speed if the input power is nearly constant. The design of these diffusers is well catered for by existing methods (*see* Cockrell and Markland, 1974).

4.2. Wide-angle diffuser

This type is normally installed between the blower and settling chamber or between the fourth corner and settling chamber of a closed-circuit tunnel; the cross-sectional area increases so rapidly that separation can be avoided only by boundary layer control. A wide-angle diffuser is a means of reducing the length for a given area ratio rather than effecting a pressure recovery; generally the net pressure rise through a screened wide-angle diffuser is negative but small.

4.2.1. Boundary layer control methods

The most popular means of boundary layer control is by installing gauze screens. A screen, besides removing the direct effects of layer growth and incipient separation, gives the layer a new lease of life. In a wide-angle diffuser it is better to use several screens of relatively small K (less than about $1 \cdot 5$) because increasing K at one station has little effect on the skin friction at a station much further downstream. Other types of boundary layer control methods include splitters, suction slots, trapped vortices, vortex generators and vanes and may be preferable in diffusers with very severe geometries ($A > 5$, $2\theta > 50°$). For a review *see* Mehta (1977).

4.2.2. Design charts

The four most important parameters in a wide-angle diffuser are A, 2θ, K and n, where n is the number of screens *within* the diffuser—this includes the screens installed at the inlet and outlet. Data were collected from over a hundred wide-angle diffuser designs, mostly 'successful' (no separation, and uniform outlet flow with an acceptable turbulence level), and charts were plotted for relevant parameters, from which design rules have been derived. In Fig. 3, A is plotted against 2θ; the curves enclosing successful configurations have an approximately hyperbolic shape. As n increases, the vertex moves to a higher value of 2θ and, to a lesser extent, to a higher value of A, thus implying a stronger dependence of required n on 2θ. Figure 4 is a plot of the sum of pressure drop coefficients of all the screens, $K_{sum} \equiv \Sigma (\Delta p/q)$, vs A. The straight line EF ($A = 1 \cdot 14\, K_{sum} + 1 \cdot 0$) included the maximum number of successful configurations.

4.2.3. Overall design procedure

For a diffuser design to operate successfully it must lie to the left of the relevant curve in Fig. 3, giving the minimum number of screens required in the diffuser, and A

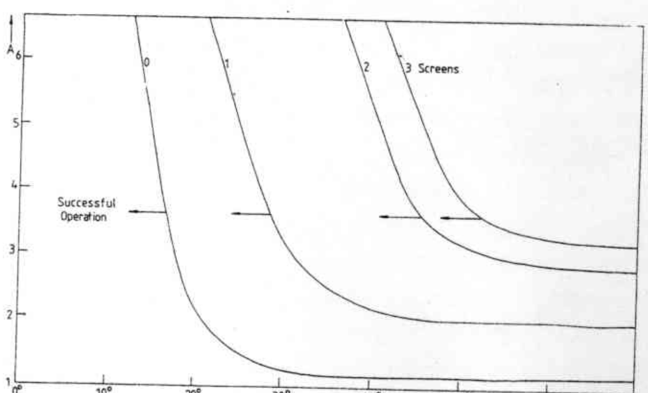

Figure 3. Design boundaries for diffusers with screens.

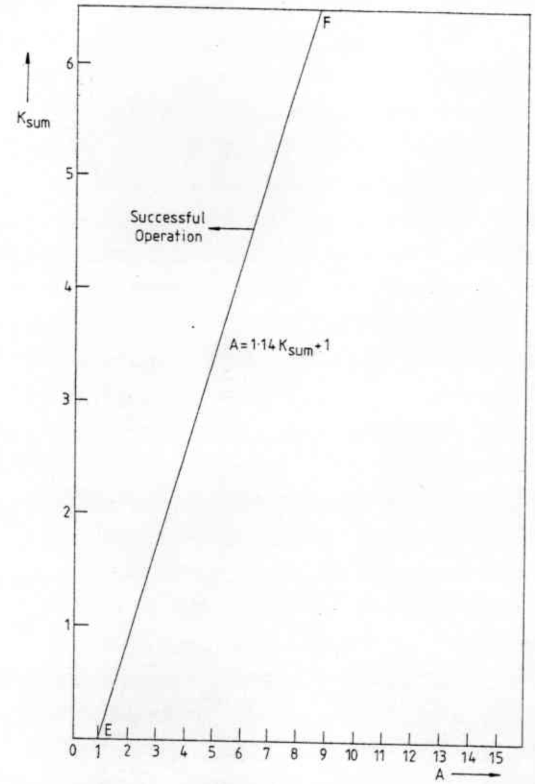

Figure 4. Overall pressure drop coefficient requirements for a diffuser with screens.

must be less than ($1 \cdot 14\, K_{sum} + 1 \cdot 0$), giving the minimum required overall pressure drop coefficient. A diffuser configuration satisfying both these curves should perform successfully *provided that* certain other design factors are kept in mind:

(*i*) *Inlet conditions*: Thin boundary layers and steady flow at the inlet are obviously beneficial.

(*ii*) *Screen Positioning*: The basic rule is to place screens where the diffuser wall angle changes suddenly, since these are the points where the flow is most likely to separate. In diffusers where no obvious location is indicated screens should be equally spaced, remembering that a screen at the diffuser entry (with a relatively high resistance) is desirable because the angle changes suddenly there.

(*iii*) *Wall shape*: The number of screens required in a diffuser could well be reduced, and the efficiency increased, by employing curved walls. Potential flow methods are sometimes used to determine wall shapes but it is often easier to design wall shapes by eye. Straight-walled diffusers (often with curved screens) are, however, often employed, because they are easier and cheaper to construct.

(*iv*) *Screen shape*: It is an advantage for the screen to intersect the diverging walls and streamlines at right angles, so that the refraction of the flow by the screens does not itself induce separation. Curved screens can be held in metal frames pressed into circular arcs and lined with wooden strips so that the gauze may be firmly embedded between two frames. It could be more difficult to dish the more flexible plastic screens (*see* section 3) which may also tend to flutter. Another alternative is to use a plane, 'variable-K' screen comprising of one screen concentrically superimposed on another.

(*v*) *Cross-sectional shape*: Most wide-angle diffusers have either rectangular or square cross-sections for ease of construction and since pressure recovery is not too important. It is advisable to fillet the corners in small tunnels, whose designs are likely to be more adventurous, to reduce the risk of large regions of flow separation.

4.2.4. *Comparison and verification of design rules*

These design rules compare well with those proposed by Kline *et al* (1957), Schubauer and Spangenberg (1948) and Squire and Hogg (1944). This is to be expected because many designers have used these rules; evidently the rules are successful, but they may be conservative. The present rules also compare well with some experiments and test cases, details of which can be found in Mehta (1977), although there is evidence that the rules are not inflexible.

5. CORNER VANES

Even some open-circuit tunnels have corners, say to deflect the efflux from a horizontal tunnel upwards to reduce draughts. Rules for the design of vanes for 90° corners are uncontroversial and probably rather conservative. Thin sheet metal vanes are used on all but the largest tunnels and, even in the latter, thick aerofoil-section vanes are used for strength rather than aerodynamic advantage. The ratio of the gap, h, between vanes (measured from leading edge to leading edge) to the chord, c, should not exceed about 0·25; the vane lift coefficient is $2h/c$. Usual practice is to make the vane as a circular arc, with short straight extensions at leading and trailing edges for ease of rolling or pressing. The trailing edge is aligned parallel with the axis of the downstream leg and the leading edge is set at a positive 'angle of incidence' of 4° to the axis of the upstream leg. This arrangement has superficial logic but differs from established cascade-design practice, and recently Ermshaus and Naudascher (1977) have successfully used a hodograph-solution design which has a *negative* angle of incidence at the leading edge and over-turns at the trailing edge so that the included angle is 105° instead of the conventional 86°. It is not clear whether a significantly higher h/c can be used with this design.

The pressure drop through thin vanes of standard (86°) design is estimated by Bradshaw and Pankhurst (1964) to be about $1\cdot2\ (Uc/\nu)^{-1/4}$ times the dynamic pressure.

6. HONEYCOMBS

Honeycombs are effective for removing swirl and lateral mean velocity variations, as long as the flow yaw angles are not greater than about 10°. Large yaw angles cause the honeycomb cells to 'stall' which reduces their effectiveness besides increasing the pressure loss.

6.1. Reduction of turbulence

An incidental effect of honeycombs is to reduce the turbulence level in the flow. Essentially, the lateral components of turbulence, like those of mean velocity, are inhibited by the honeycomb cells and almost complete annihilation is achieved in a length equivalent to about 5–10 cell diameters. Honeycombs themselves shed small scale turbulence, the level of which is found to be higher when the cell flow is laminar than when it is turbulent: this is attributed to a basic instability of the laminar near wakes. Note that the cell flow in most wind tunnel honeycombs is *laminar* and so Lumley and McMahon's (1967) analysis, which assumes turbulent flow, will *not* apply. With a laminar cell flow the *net* reduction is greatest for the shortest honeycomb (Loehrke and Nagib, 1976). It turns out that the shear layer instability in the near wake has a strength proportional to the shear layer thickness and so for the longest honeycomb, the ratio of turbulence generated to that suppressed is greatest.

6.2. Optimum cell size

For maximum overall benefit the cell length should be about 6–8 times its diameter. The cell size should be smaller than the smallest lateral wavelength of the velocity variation (roughly 150 cells per settling chamber 'diameter', i.e. 25 000 total, are adequate). The cross-sectional shape of the honeycomb cells is usually hexagonal, but sometimes square or triangular, the shape being chosen mainly for ease in construction. Impregnated paper honeycombs are adequate for small tunnels. Aluminium honeycombs made for aircraft sandwich construction have more precise dimensions than paper honeycombs and are to be preferred for high performance tunnels and large tunnels where the wind loads may be expected to be high. The cells of all honeycombs are often partly obstructed by burrs which can be fatigued off with an air hose.

7. SETTLING CHAMBERS

7.1. General arrangement

The usual arangement consists of a honeycomb (with about 25 000 cells) followed by screens, the number and K-value depending on the turbulence level requirements. If severe yaw or swirl is expected in the flow from the wide-angle diffuser, it is advisable to install one screen upstream of the honeycomb, so that the flow angles are reduced. A screen with $K=1\cdot5$ reduces yaw and swirl angles by a factor of about 0·7 for swirl angles of about 40°. The honeycomb should be installed some way downstream of the wide-angle diffuser exit, so that the flow static pressures and angles have had a chance to become more uniform. Since screens with small β (less than about 0·57) tend to produce instabilities, presumably in the form of longitudinal vortices, at least one screen with a larger β ($>0\cdot57$) should be used (in the most downstream position) if a truly two-dimensional boundary layer is required in the working section. Another alternative is to place the honeycomb downstream of the screens but this at best results in a rise in the turbulence level and is not recommended in general.

7.2. Spacing between screens

There are two important properties to consider:

(*i*) For the pressure drops through the screens to be completely independent, the spacing should be such that the static pressure has fully recovered from the perturbation before reaching the next screen (i.e. $dp/dy=0$).

(*ii*) For full benefits from the turbulence-reduction point of view, the minimum spacing should be of the order of the large energy containing eddies.

It has been found that a screen combination with a spacing equivalent to about 0·2 settling chamber diameters performs successfully. The optimum distance between the last screen and the contraction entry has also been found to be about 0·2 cross-section diameters. If this distance is much shorter significant distortion of the flow through the last screen may be expected. On the other hand, if this distance, or for that matter the overall length of the settling chamber, is too long then unnecessary boundary layer growth occurs.

7.3. Installation of components

Screens are normally tacked onto wooden frames. More care is necessary when tacking on plastic screens since these, being more flexible, tend to wrinkle along the lines of tension. The honeycomb is usually just push-fitted into its own frame. A useful arrangement for small tunnels is to rest the wide-angle diffuser, screen frames and contraction on a table and clamp them by drawbolts, so that frames can be withdrawn easily. On larger tunnels, it is advisable to equip the settling chamber (and wide-angle diffuser) components with castors for ease of removal. Even in tunnels made of metal or concrete, the screens are normally installed in separate frames which can be withdrawn from the tunnel for cleaning or replacement.

8. CONTRACTIONS

A contraction:

(*i*) Increases the mean velocity which allows the honeycomb and screens to be placed in a low speed region, thus reducing pressure losses.

(*ii*) Reduces both mean and fluctuating velocity variations to a smaller fraction of the average velocity.

The most important single parameter in determining these effects is c, the contraction ratio. The factors of reduction, as derived by Batchelor (1970) for $c \gg 1$, are:

U-component mean velocity: $1/c$
V or W-component mean velocity: $\sqrt{c}$
u-component rms intensity: $1/2c\ \{3\,(\ln 4c^3 - 1)\}^{1/2}$
v or w component rms intensity: $(3c)^{1/2}/2$.

(The factor of reduction of percentage velocity variation is given by multiplying the above expressions by $100/c$).

The design of a contraction centres on the production of a uniform and steady stream at its outlet, and requires the avoidance of flow separation. Two more desirable criteria include minimum exit boundary layer thickness and minimum contraction length. A design satisfying all criteria will be such that separation is just avoided and the exit non-uniformity is equal to the maximum tolerable level for a given application (typically $\pm\frac{1}{2}\%$ velocity variation outside the boundary layers).

8.1. Contraction lengths

It is always possible to avoid separation in the contraction by making it very long, but this results in an increase of tunnel length, cost and exit boundary layer thickness.

8.2. Contraction ratio

Since the power factor contribution of screens in the settling chamber varies as $1/c^2$, large contraction ratios are advantageous. But large contraction ratios mean higher construction and running costs besides possible problems of noise and separation near the ends. Therefore, contraction ratios between about 6 and 9 are normally used, at least for the smaller tunnels.

8.3. Cross-sectional shape

In any contraction with a non-circular cross-section, the flow near the walls tends to migrate laterally, especially near corners of a polygonal section. In any case the boundary layers near the corners will be more liable to separate. However, recent investigations (Mehta, 1978) show that this does not cause a problem in a well-designed *square* contraction; the effect of boundary layer migration in a contraction whose cross-sectional aspect ratio changes along its length can be reduced by adding small 45° corner fillets, but rapid termination of these in the working section must be avoided.

Two-dimensional contractions are sometimes preferred on tunnels used for boundary layer studies, where the working section is wide but shallow. However, if the boundary layers are thick, the plane walls tend to develop strong secondary flows. Also, 2-D contractions require about 25% more length to attain the same uniformity of pressure distribution as axisymmetric ones.

8.4. Wall shape

8.4.1. Theoretical design

The solution of the Laplace equation or the Stokes-Beltrami equation is relatively easy for simple geometries and many analytical solutions have been derived. With the onset of computers many numerical schemes have also been proposed. For a review *see* Mehta (1978).

There is no wholly satisfactory method of theoretical wall shape design, as distinct from analysis. The application of all these methods requires the establishment of some criteria and then the application of trial-and-error techniques for which limited guidance is given.

8.4.2. Design by eye

Designers have often used the rather unscientific method of design by eye. Note that the actual form of a contraction contour is not very important except near the ends, and that smoothness in contour shape is much more important than exact dimensions. In general the wall radii of curvature should be less at the narrow end and each end must join the parallel sections so smoothly that at least the first and second derivatives of the curve are zero (or very small) at the ends.

9. WORKING SECTIONS

Working section design is totally dependent on the requirements of the individual experimenter. Blower tunnels are more flexible in accepting a variety of working sections (with and without exit diffusers).

The flow out of a contraction often takes a distance equivalent to about 0·5 diameters before the non-uniformities are reduced below an acceptable level. Also, if a turbulence grid is installed, it may take up to 10–15 mesh lengths before a homogeneous flow is obtained. These requirements often fix the minimum length of the working section. The streamwise pressure gradient is best controlled by installing tapered fillets.

It is advisable to mount removable side panels on pinned hinges on large working sections which makes their 'single-handed' removal easier and safer.

Drag forces, being proportional to (velocity)^2, change by twice as large a fraction as the mean velocity; lift forces change because of the change in mean velocity and because of the influence of tunnel walls on the effective angle of incidence. Lift interference on a complete aircraft model in a rectangular-section tunnel is minimised if the ratio of working section breadth to height is $\sqrt{2}$ (with model span less than three-quarters of the breadth) so most general purpose aerospace tunnels are made with approximately this aspect ratio. However tunnels for measurements in boundary layers, growing on the tunnel floor say, have an optimum breadth-to-height ratio of about five since all that is necessary is that a reasonable thickness of irrotational flow shall remain between the roof and floor boundary layers at the end of the test section (a diffuser with such a non-uniform entry flow would not be very efficient). Tunnels for testing building complexes or natural terrain at model scale can also have a large breadth/height ratio; conversely, tunnels for testing isolated tower buildings or smokestacks can have a breadth-to-height ratio *less* than unity, although the ratio of model breadth to tunnel breadth must still be kept small to minimise interference.

10. ESTIMATION OF TUNNEL POWER FACTOR

Having decided the size and configuration of a wind tunnel, the next design step is to estimate the tunnel power factor, λ (equal to $H/\frac{1}{2}\rho_o U_o^3 A_o$, where H is the shaft input power and subscript o refers to working section conditions) so that the fan and drive unit can be selected. It is difficult, but in fact not essential, to estimate the power factor very accurately; adequate extra power must be installed to cope with a variety of model or working section configurations, not known in advance.

The pressure losses in a wind tunnel are due to diffuser losses, resistive components such as screens, and friction on the tunnel walls. The total pressure loss due to each component can be estimated separately, and then summed and divided by the blower efficiency η, typically 0·8, to give the tunnel power factor. Typical values for a tunnel similar to that shown in Fig. 1(*b*) are given below.

(*i*) Losses due to skin friction.

$$\eta\Delta\lambda_1 = \frac{\Delta P}{\frac{1}{2}\rho_o U_o^2} = \left(\frac{A_o}{A}\right)^2 \int C_f \frac{S}{A} \mathrm{d}x,$$

where S is the duct local perimeter and remembering that the area ratio is the reciprocal of the velocity ratio. It is normally only necessary to estimate skin friction losses in the working section ($A/A_o=1$). Those in the diffusers are normally accounted for in the efficiency and the other components do not contribute significantly.

Therefore, $\eta\Delta\lambda_1 \simeq C_f SL/A$, where L is the working section length. Typical value: $\eta\Delta\lambda_1 \sim 0{\cdot}07$, assuming $C_f \sim 0{\cdot}003$.

(*ii*) Losses due to screens, honeycomb and corner-vanes.

$$\eta\Delta\lambda_2 = K\left[\frac{A_o}{A}\right]^2.$$

So for a tunnel with four screens (two in the wide-angle diffuser with $A/A_o=4$ and 6 respectively) each with $K=1{\cdot}5$ (for $U=5$–10 m/s), and a honeycomb with $K=0{\cdot}5$ we have, taking $c=9$, typical value: $\eta\Delta\lambda_2=0{\cdot}18$ (the screen at $A/A_o=4$ contributes 0·094).

(*iii*) Loss of total head in the exit diffuser.

$$\eta\Delta\lambda_3 = (1-\eta_D)\left[1-\left(\frac{A_o}{A_{\mathrm{out}}}\right)\right],$$

where η_D is the diffuser efficiency. This is a loss due to the inefficiency of the diffuser in transforming kinetic energy into 'pressure energy' and is caused by boundary layer growth and non-uniformity of the flow. The efficiency of a wide-angle diffuser with screens is generally negative but Δp is small.

For a conical diffuser with $A{\sim}2{\cdot}5$ and $2\theta{\sim}5°$, Cockrell and Markland (1974) suggest $\eta_D=0{\cdot}8$, but this may be lower for diffusers with rectangular cross-sections, typical value: $\eta\Delta\lambda_3=0{\cdot}25$ for $\eta_D=0{\cdot}7$ and $A=2{\cdot}5$.

(*iv*) Loss of total head at the exit of an open-circuit tunnel.

In an open-circuit tunnel, the amount of kinetic energy lost at the exit and dissipated into heat adds to the total losses.

$$\eta\Delta\lambda_4 = \left(\frac{A_o}{A_{\mathrm{out}}}\right)^2$$

[$=1$ for blower tunnels with no exit diffuser], typical value: $\eta\Delta\lambda_4=0{\cdot}16$ for $A=2{\cdot}5$.
Therefore the estimated overall tunnel power factor, $\lambda \equiv \sum_{n=1}^{4}\Delta\lambda_n/\eta \simeq 0{\cdot}825$ for the tunnel considered (with an exit diffuser), taking $\eta=0{\cdot}8$.

Once the tunnel power factor has been estimated and the required fan static pressure rise determined, one can set about the selection of the optimum fan size. The *dynamic* pressure rise through a blower is usually ignored and can be thought of as a safety factor in the calculations.

The fan outlet flow will be least turbulent when the fan is operating near maximum efficiency. Fan efficiency is a function of the dimensionless flow rate; the pressure rise coefficient is a (weak) function of the dimensionless flow rate also, so that requiring maximum efficiency specifies both dimensionless flow rate and pressure rise coefficient. So for a given required flow rate and pressure rise, two equations are obtained which can be solved to give the fan size and optimum operating rpm. In practice the manufacturer's performance charts should be searched for a fan size (and rpm) giving near maximum efficiency for the required flow rate and pressure rise.

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