inter•noise 83

THE PERFORMANCE OF LOW NOISE LAMINAR FLOW FANS

S.L. Merry (1), S.A.L. Glegg (2) and A.G. Herbert (3)

(1) and (2) Institute of Sound and Vibration Research, Southampton University.
(3) Wolfson Unit for Noise and Vibration Control, Southampton University.

INTRODUCTION

The laminar flow fan differs from more conventional bladed fans, in that the rotor consists of a stack of closely-spaced discs, with a central hole for the air inlet (see Fig. 1). Air enters the interdisc spacing and acquires a tengential acceleration through friction on the discs, which provides a radial pressure rise and air through-flow. As the air exhausts at the periphery of the discs, it is collected in a standard scroll-shaped volute. The absence of blades and the preservation of laminar flow conditions in the narrow interdisc spacing ensure that the operation of this type of fan is inherently quiet.

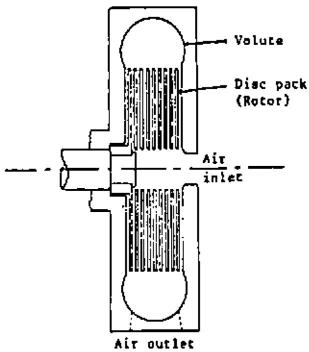

Fig. 1
Diagram of laminar flow fan.

MATHEMATICAL MODEL OF FAN PERFORMANCE

Experimental results have shown that laminar flow fans do not obey the conventional laws of aerodynamic performance scaling for changes of geometry and rotational speed. A mathematical model for predicting the head/flow characteristics has therefore been developed.

The fan head rise and flow rate are expressed non-dimensionally as:
Head coefficient, $\psi = g\Delta H/\omega^2 D^2$; Flow coefficient, $\phi = Q/\omega D^2 bN$

where: ω = rotational speed; Q = air flowrate; D = rotor diameter; b = interdisc spacing; ΔH = head rise across fan; N = number of spaces.

The relationship between fan generated head and air flow rate will take the general form:

$$\psi = \tfrac{1}{2}\mu_s - \psi_f(\phi) - \beta_i(\phi) - \beta_{vol}\mu_s^2 \qquad (1)$$

The first term on the right hand side of equation (1) represents the ideal fan head, as given by the Euler equation. The remaining terms describe the frictional, inlet and volute losses, respectively; negligible leakage is assumed.

Tangential and radial air flow in a laminar fan occurs mainly in the boundary layer next to the discs. Hence the ratio of the interdisc spacing to the boundary layer thickness, b/δ, influences the fan performance. For a rotating disc, the boundary layer thickness is of the order $\sqrt{\nu/\omega}$ (ν = kinematic viscosity), so that $(b/\delta)^2 = \omega b^2/\nu$. This parameter is called the "disc-spacing" Reynolds number, Re_b.

The tangential velocity profile of air between two discs consists of two high velocity boundary layers surrounding a central core which rotates at lower velocity, as a solid body. From this concept, the following expression for the slip factor (μ_s) has been derived:

$$\mu_s = \frac{2}{h}(1 - e^{-h}(1 + h))Re_b^{-\frac{1}{2}}\left(\frac{r_2}{r_1}\right)^{\frac{1}{2}} + e^{-h} \qquad (2)^{**}$$

where h = constant, r = radius and subscripts 1 and 2 refer to inlet and outlet conditions on the rotor, respectively.

By consideration of the radial shear forces on an element of fluid between two discs, the frictional through-flow losses are found to be:

$$\psi_f = k\left(1 - \frac{r_1}{r_2}\right)\phi \qquad (3)^{**}$$

where k, which is constant for a given disc spacing, may be determined experimentally. Empirical formulae for volute losses give $\beta_{vol} \simeq 0.035$, whilst the fan inlet losses prove negligible when related to ψ_f. The above formulae for μ_s, ψ_f and β_{vol} are inserted into equation (1) to give the theoretical head/flow relationship for a laminar flow fan.

**Further work is in hand to confirm the assumed radial scaling in equations (2) and (3)

COMPARISON OF THEORY AND EXPERIMENTAL RESULTS

The head flow characteristics of a small laminar flow fan have been investigated. Using a fixed disc geometry, with $r_1 = 40$ mm and $r_2 = 130$ mm, the effect of speed variation, as well as different disc spacing and number of discs, was examined. The fan was tested in three configurations: (a) 22 discs, 0.77 mm spacing; (b) 14 discs, 1.54 mm spacing; (c) 11 discs, 2.31 mm spacing. Test speeds were 50-125 rps, corresponding to $Re_b = 12.4 - 280$.

At zero flow rate, equation (1) reduces to:

$$\psi_o = \tfrac{1}{2}\mu_s - \beta_{vol}\mu_s^2 \qquad (1a)$$

Experimental values of ψ_o were used to calculate the corresponding slip factor μ_s, which is plotted as a function of $Re_b^{-\frac{1}{2}}$ in Fig. 2.

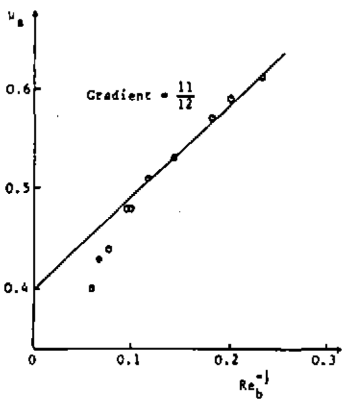

Fig. 2 Slip factor (μ_s) as a function of Re_b.

According to equation (2), $\mu_s \propto Re_b^{-\frac{1}{2}}$ and therefore the data points in Fig. 2 should fall on a straight line which intercepts the ordinate at $\mu_s = e^{-h}$. Clearly, the points fall on a curve, but to a close approximation, a linear relationship exists in the interval $0.1 > Re_b^{-\frac{1}{2}}$ ($Re_b < 100$), which is the range of rotational speed and disc spacing of greatest practical interest. The corresponding value of e^{-h} is 0.4, giving h=0.916. This cross-checks with the value of h obtained from the gradient of the graph.

When the appropriate value for μ_s (calculated from equation (2) with h = 0.92) is inserted into equation (1a), we obtain:

$$\psi_o = 0.2017\, Re_b^{-\frac{1}{2}} - 0.02893\, Re_b^{-1} + 0.0944.$$

The experimental head/flow data for the laminar fan operating in the range $Re_b < 100$ can now be collapsed onto a single curve, by plotting a modified head coefficient ψ' (where $\psi' = \psi - 0.2017Re_b^{-\frac{1}{2}} + 0.02893Re_b^{-1}$) versus Φ/k as shown in Fig. 3.

For wide disc spacings ($Re_b > 100$) the tangential velocity of the central core (which is expressed as the constant e^{-h} in equation (2)) may be a decreasing function of Re_b. Therefore the present model does not apply to this area of fan operation, which requires further investigation.

NOISE MEASUREMENTS

The noise from laminar flow fans is generally difficult to distinguish above the noise from its motor and bearings or from flow in associated ducts. To overcome this problem, a quiet drive unit has been developed for accurate noise measurements, which will be reported in the future.

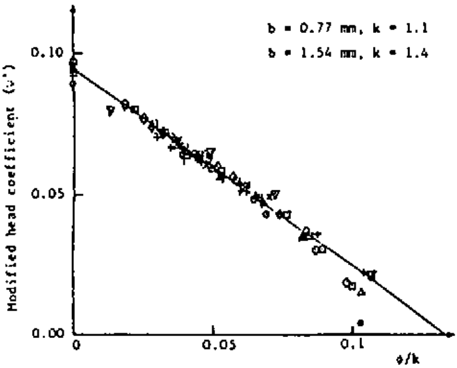

Fig. 3 Empirical collapse of head/flow data for $Re_b < 100$.

However, significant noise reductions have been obtained when standard centrifugal fans have been replaced by laminar fans of equivalent aerodynamic performance. The elimination of blade passage frequency is of particular importance and noise reductions of up to 10 dB(A) have been observed. For instance, Fig. 4 shows the noise reduction when a laminar fan was used to replace a centrifugal fan in an air distribution system, whose outlet was close to a voice communication microphone. In this case the major problem was caused by the blade passage frequency of the centrifugal fan coinciding with the peak response of the communication microphone at 2500 Hz. The noise reduction obtained by using the laminar fan is more than 10 dB, except at 500 Hz, which corresponds to the passing frequency of the disc fixing bolts.

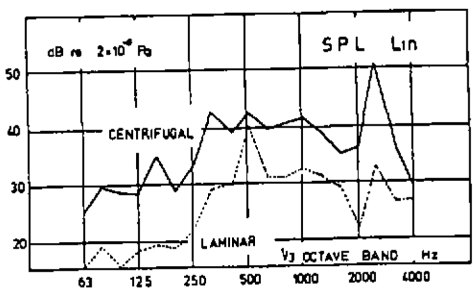

Fig. 4 Comparison of noise levels from a laminar flow and centrifugal fan.