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DETERMINING THE NOISE OF FANS IN BUILDING SERVICES

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Introduction

Much has been written on the need for sound insulation of buildings, both to attenuate the noise from outside, and also to reduce the transmission between rooms. Within any modern building, however, there are a considerable number of noise sources from building services equipment. A prime source is the Main Air Conditioning fan, which has the ability to direct its duct-borne noise to the farthest corners of the occupied space and can be a major irritant. It is remarkable, therefore, that in the original design of these systems, so little attention is paid to the correct selection of the fan from a noise viewpoint. Too often a prediction is based on some empirical "rule" which can prove far from correct when once the unit is installed. It is the intention of this paper to point out some of the pitfalls and to suggest that the requisite information be obtained from a reputable manufacturer at the earliest possible time. Unfortunately this is not always possible as the fan manufacturer will only be chosen late in the building programme when much of the design has been 'frozen'. There is every incentive, however, to conduct a feasibility study using results obtained from experiments.

Empirical Rules for Determining Fan Noise

The desire to have a simple rule by which the noise output of a fan could be deduced from its operational duty are apparent. An early attempt was by Beranek, Kamperman and Allen in their 1955 paper to the Journal of the Acoustical Society of America when the following relationship was proposed :

$$PWL = 100 + 10 \log HP \quad \text{dB re } 10^{-13} \text{ W}$$

where PWL = the overall acoustic power level of noise transmitted along a duct fitted to the inlet or outlet of a fan <u>operating at or near its peak efficiency</u>

HP = the <u>nameplate</u> horsepower of the driving motor

If updated for present-day units, the above formula becomes :

$$PWL = 91.3 + 10 \log kW \quad \text{dB re } 10^{-12} \text{ W}$$

which looks far less attractive, and could well have been a deterrent to its use!

It will be appreciated that this formula was of necessity approximate only, and was based on a series of fans tested at pressures up to about 500Pa. Subsequently, with the steady increase in system pressures up to 2500Pa in many cases, a revised formula was suggested :

$$PWL = 100 + 10 \log HP + 10 \log p \quad dB\ re 10^{-13}\ W$$

where p = pressure ins. w.g.

Again in modern units this becomes :

$$PWL = 67.3 + 10 \log kW + 10 \log p \quad dB\ re 10^{-12} W$$

where p = pressure Pa

Bearing in mind that there can be a considerable difference between absorbed and nameplate power (especially in the case of forward curved centrifugal fans) it was also suggested that the former be inserted in the formula.

A further manipulation of the power term is possible for :

$$\frac{Q \times p}{10 \times \eta\%} = kW \qquad \text{where Q is in } m^3/s, \ \text{p is in Pa}.$$

then $PWL = 57.3 + 10 \log Q + 20 \log p - 10 \log \eta\% \quad dB\ re\ 10^{-12}\ W$

This formula gives the total noise. Assuming that inlet and outlet noise are equal, then these would each, of course, be 3dB less.

And there the exercise should end, for one has to say that for very large fans and for fans at pressures above 1000Pa, the uncertainty when compared with actual noise tests can be as much as ±10dB using any of these formulae, even when the fan has been selected at its peak efficiency. This is hardly surprising for whilst some fan ranges which were current in 1955 are still available, research over the past thirty years or so has meant that we now have a very much better idea of the noise generating mechanisms within fans. Research into the cut-off and volute design of centrifugal units has, in itself, led to improvements of over 10dB whilst in axials, the importance of tip clearance, impeller-casing concentricity, rotor-stator gap, and rotor-stator vane numbers, have all been the subject of important work.

Noise Producing Mechanisms in Fans

There are three recognised ways in which acoustic energy may be derived from the kinetic energy produced by a fan. They are, in descending order of radiation efficiency, monopole, dipole and qradrupole sources (Fig. 1).

a) **Monopole source:** the most efficient generating mechanism in which the conversion from kinetic to acoustic energy is by forcing the gas within a fixed region of space to fluctuate. This may be visualized as a uniformly radially pulsating sphere surrounded by a perfectly homogeneous material of infinite extent, such that no end relections occur.

b) **Dipole source:** this is thought to be the predominent sound generating mechanism in low speed turbomachinery such as fans. Energy conversion requires the momentum within a fixed region of space to fluctuate, the process being equivalent to a uniformly pulsating sphere oscillating in the x-direction as a rigid body. Alternatively, it may be thought of as two adjacent monopoles where one is at its maximum dimension, when the other is at a minimum. Thus the dipole is vibrating along one axis. This accounts for the directional nature of the sound generated, the normal particle velocity on the sphere surface being a function of its polar location.

c) **Quadrupole source:** the least efficient energy conversion mechanism in which sound is generated aerodynamically, with no motion of solid boundaries, as in the mixing region of a jet exhaust. Within a fixed region of space there is no change of either mass or momentum. Energy conversion is by forcing the rates of momentum flux across fixed surfaces to vary. Momentum flux is the rate at which momentum in the x_i direction is being transported in the x_j direction, with corresponding velocities v_i v_j. A quadrupole source may be modelled as a double dipole, both oscillating along the same axis. It exhibits complex directionality.

Equations of Acoustic Pressure

The acoustic pressure generated by these different sources may be deduced as follows :

$$\bar{\Delta}p_{\text{monopole}} \propto \frac{1}{r} \frac{\delta \dot{M}(t)}{\delta t}$$

$$\tilde{\Delta}p_{\text{dipole}} \propto \frac{\delta}{\delta x} \left[\frac{r_{sp}}{r}, \frac{\delta M(t)}{\delta t} \right]$$

$$\bar{\Delta}p_{\text{quadrupole}} \propto \frac{\delta}{\delta x_i} \frac{\delta}{\delta x_j} \left[\frac{1}{r}, v_i, v_j, \rho, D^3 \right], F(\nu, \Delta t)$$

where $M(t)$ = rate of addition of mass from the neighbourhood of the source to its surroundings.
r = polar distance to the observer.
r_{sp} = radius of the sphere.
x = direction of oscillation.
v = momentum flux velocity.
D = characteristic dimension
ρ = ambient density of the air or gas.
ν = air or gas viscosity.
Δt = temperature change across the region.

Generally the dissipation of acoustic energy into heat by viscosity and heat conduction, is negligible over distances of less than say 100m, in which case the viscosity and temperature defect terms in the quadrupole equation may be neglected.

Development of Flow-Acoustic Theory

The equations detailed above may be applied to single sources, but within the acoustic field of a fan, the degree of radiation will depend also on the level of phase cancellation between adjacent sources. Indeed, this whole question of phase difference is seen as the way forward in the reduction of fan noise. It is leading to the introduction of scimitar shaped blades, angular cut-off pieces and other devices.

It was Lighthill who first applied dimensional analysis to the acoustic power radiated by the different sources of sound pressure, and showed that :

Sound Power $W_n \propto \frac{\rho D^2 v^4}{C}$ for a Fluctuating Mass or Monopole

$\propto \frac{\rho D^2 v^6}{C^3}$ for a Fluctuating Force or Dipole

$\propto \frac{\rho D^2 v^8}{C^5}$ for Turbulent Mixing or Quadrupole

Equally, the Ventilation Engineer can show that in a geometrically similar range of fans

Air Power $W_A \propto \rho D^2 v^3$

We may therefore state that

Sound Power $\propto W_A M_A$ for a Monopole

$\propto W_A M_A^3$ for a Dipole

$\propto W_A M_A^5$ for a Quadrupole

where M_A is the ratio of some velocity to the speed of sound, i.e. a Mach No. In high speed fans this can approach 0.3.

Overall sound power radiation for any fan type or homologous series of fans will have a sound power/rotational velocity relationship which depends on the relative contributions of the three sources. However, it is not simply a matter of how an acoustic mechanism varies with a typical speed, but rather how the flow conditions related to that acoustic mechanism vary with speed. Whilst a considerable amount of work has been done attempting to define a consistent relationship between fan rotational speed and the generated sound power, unless strict similarity is ensured, or design variations accounted for, the empirically derived equations may give rise to considerable error, as previously stated. Consequently, results from various researchers differ and the power-velocity exponent, a, is variously quoted between 4 and 6 where $W_n = kv^a$. It should here be noted that the Beranek formula and its extrapolations assume $a = 5$ as power absorbed $\propto Qp$ and $Q \propto v$ $p \propto v^2$ and the pressure term has a coefficient of 20.

The first theoretical study of noise from rotating machinery was probably that of Gutin in 1948. His basic equation assumed a steady state where the blade loading distribution was independent of time. Here an element of gas within the area swept by the rotor was considered to receive an impulse periodically with the passing of a blade. The impulses were treated as a series of dipole sources distributed throughout the swept area, and of constant strength at any radius. The dipole source amplitudes were obtained from the thrust and torque loading conditions, the fundamental frequency of the noise generated being $z\Omega$ where z is the blade number and Ω is the rotational frequency (revs/sec). The resultant sound field can be analysed into a series containing the fundamental frequency and its integer harmonics. It is assumed that the acoustic pressure satisfies the homogeneous wave equation

$$\frac{\delta^2 p}{\delta t^2} - C^2 \frac{\delta^2 p}{\delta x^2} = 0$$

The fluid surrounding the blade surfaces must, therefore, have velocities which are low compared to the speed of sound, such that acoustic waves can travel radially from their source. For fans operating at high pressures or unstable parts of their characteristic curves, this may not be the case and it is then necessary to consider the fluid as a perfect acoustic medium containing quadrupole sound sources of $T_{ij} = \rho v_i v_j + p_{ij} - C^2\delta_{ij}$. As previously stated, the last two terms in this stress tensor may usually be ignored as the quadrupole strength density becomes equal to the 'fluctuating Reynolds Stress' of the gas around the blades.

It is, therefore, possible to itemise the source components of the whole radiation field such that sound produced by a fan may be regarded as generated by monopole sources related to volume displacement, dipoles distributed over the machine surfaces and quadrupoles of strength density T_{ij} distributed throughout the surrounding gas.

Lighthill's acoustic analogy was to regard density variations within the gas as being driven by a source distribution $K = \frac{\delta^2 \rho}{\delta t^2} - C^2\nabla^2\rho$ for the general case of an unbounded fluid, but in the real world, solid boundaries are present. Modifications to the theory are, therefore, necessary to take account of reflections at these surfaces and also for an uneven quadrupole distribution as these may only exist external to the blades. Those still interested are referred to the works of Curle and ffowcs-Williams who have enabled surface force distributions and moving boundaries to be considered.

Practical Sources of Fan Noise

These may be grouped under the following headings :

a) thickness noise due to the passage of blades through the air - a quadrupole source

b) torque and thrust noise - quadrupole sources

c) rotation noise due to the blades passing a fixed point e.g. cut-off - a dipole source

d) vortex shedding due to flow separation from the blades - a dipole source with some Reynolds No. dependence.

e) air turbulence noise due to shear forces when the blades are stalled - a quadrupole source.

f) interference noise due to contact between turbulent wakes and obstructions

g) pulsation noise - where at high system pressures the flowrate regularly varies and a pitched tone is produced at the frequency of the pulses - a monopole source.

Of all these, the most important in a well designed fan will probably be due to vortex shedding and are, therefore, dipoles where noise $\propto v^6$.

Spectral Distribution

The spectral shape of the noise from a fan varies according to its design. In very general terms, an axial flow fan will have its highest noise in the octave band containing the blade passing frequency, $z\Omega$ (Blade No. x rev/sec) with a declination of 2 dB per octave on either side. The peak at blade passing frequency can exceed the general spectral level by 4 to 10 dB, being especially severe where the impeller is eccentric in its casing. There may also be additional tones generated at interactive frequencies determined by (blades + vanes), (blades-vanes) etc, the strength of these being dependent on the gap between them, and the ratio $\frac{\text{Blade No.}}{\text{Vane No.}}$

A centrifugal fan will have a spectrum with its peak towards the lower frequencies. The declination is of the order of 3 to 7 dB per octave band dependent on blade shape, but this general statement requires a host of provisos. In backward bladed fans, the blade passing tone and its harmonics may be of especial importance. With the flat inclined type, they are easily identified above the general broad band background. With backward curved blades, they are not so pronounced, and are lowest with backward aerofoil designs.

Effects of the Casing and External Ductwork on the Sound Produced

Sound waves produced by a source within a duct will undergo reflection, interference and decay according to the frequency of the emitted wave.

Centrifugal fans usually run at lower Mach Nos. than Axial fans and the predominant tones have wavelengths larger than characteristic impeller or duct dimensions. The overall radiated sound power may be greatly affected by reflection properties of the casing and ductwork. This can lead to some distortion of the sound power and directivity pattern, especially at low frequencies. Whilst an uncased centrifugal impeller usually gives a flat frequency spectrum, the addition of a case leads to enhancement of the noise at well defined frequencies, related to the casing geometry. Flowrate variations do not significantly affect the overall shape of the cased spectra, although the magnitude, in particular frequency bands, can vary.

It is clear, therefore, that the overall radiated sound power can be quite different from the generated power. The casing may act as a Helmholtz resonator and a major casing dimension may relate to the wavelength of some important frequency.

The acoustic power radiated by a sound source is also dependent upon the impedance against which it operates. It is of importance to ensure that no standing waves are formed in the duct as these would greatly increase the noise output. Noise produced by sudden expansions or by inlet flow distortions are all of importance.

Typical Sound Ratings

From all the above it will be seen that it is virtually impossible to determine the sound power of a fan for a specific duty without knowing the characteristics of the particular design to be used. Nevertheless, it is appreciated that a demand will still exist for some predictive measurement.

In an attempt to meet this demand, Figure 2 has therefore been produced. Again, it is assumed that the fan has been selected to operate at its best efficiency point and is handling air of standard density. PWL is the level of sound power transmitted along a duct attached to the fan inlet or outlet (this in itself may be ±3 dB). L_p is derived from the fan total pressure and L_q from the volumetric flowrate.

$$PWL = L_p + L_q \text{ dBW re } 10^{-12} \text{ W}$$

The air duty has been used rather than the size, speed or mechanical power input so that fans of differing type or efficiency may be compared. On the diagram, straight lines have been drawn through 84 dBW, 250 Pa at slopes corresponding to $PWL \propto N^{3.5}$ to N^6 where N is the fan speed. The area bounded by the dashed lines covers the range within which L_p may be expected to lie. Axial and forward curved centrifugal fans will be located around the middle of the band, whilst backward curved and mixed flow designs will be in the lower half. The lowest values will be found from aerofoil bladed centrifugal fans. At very high pressures radial tipped blades often have to be used for strength considerations. These are not so quiet and hence the lower limit line has been curved upwards.

Variation in Sound Power with Flowrate

At a constant fan speed, the sound power generated will be dependent on the system resistance against which the fan has to operate. It is, therefore, of importance to ensure that this has been correctly calculated. The change in noise at constant fan speed for some typical designs is shown in Fig 3. Differences up to 10dB are common and will occur quite sharply if the characteristic contains a marked stall point.

Of recent years, Variable Air Volume systems have become of great importance and are recognised as an Energy Efficient solution to the whole question of Air Conditioning. It is rare for a building to continuously require the design flowrate determined by temperature, occupancy, solar heat gain, relative humidity or other criteria. Some percentage of the maximum flowrate must, therefore, be delivered by the fan. A distribution curve (Fig 4) can be constructed and this indicates the percentage running time against percentage flow. How this affects the noise produced, depends on the method used.

a) **Simple damper control:** in this case the fan simply works along its characteristic. Noise will generally increase according to fan design as previously stated.

b) **Speed control:** noise may be expected to reduce with decreasing fan speed according to the relationship $PWL_2 - PWL_1 = 10a \frac{\log N_2}{\log N_1}$ where a is some exponent between 4 and 6.

Note that, because of resonances and phenomena still the subject of analysis, the variation may not be continuous. There can be 'peaks' on the graph (Fig 5). It should also be remembered that this curve does not take account of motor noise. Where the motor is contained within the fan duct, as with a typical direct driven axial flow fan, the reduction in noise may be less. With certain types of inverter control the electrical waveform may be sufficiently distorted to <u>increase</u> the motor noise at reduced speed. Fig 6 shows the overall effect.

c) **Inlet vane control:** this type of control may be used with Mixed Flow fans, with a noise penalty of up to 10 dB at small opening angles (Fig 7). It should not be used with Axial Fans where the noise penalties are severe (Fig 8). With centrifugal fans, the effect on noise down to about 50% design flow is minimal, but below this figure instability can be a problem with the wider high flow designs, such that noise will increase.

d) **Disc throttle control:** this patented control for centrifugal fans (UK No. 2,119,440B) varies the flow by narrowing the effective blade width and a monotonic reduction in noise with decreasing flowrate is achieved. The reductions are especially noteworthy at low frequencies where other controls are ineffective. A comparison with inlet vane control is shown in Fig 9.

e) **Variable pitch in motion axial fans:** noise reduces with decreasing flow throughout the whole range of performance and no discontinuities or distortions are apparent (Fig 10).

CONCLUSIONS

The use of empirical 'laws' to determine fan noise can be frought with danger. In all possible cases, reference should be made to actual tests. If the flowrate varies, care should be taken in selecting an appropriate method. The sound output may increase if the ducting resistance has been incorrectly assessed and the fan does not operate at the correct point on its characteristic. Ductwork impedance can determine the fan noise, particularly at low frequencies. The need for good inlet and outlet connections cannot be understated.

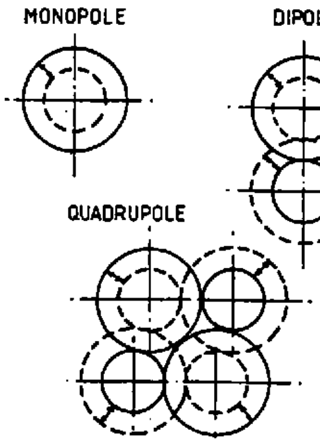

Fig 1 Noise Sources

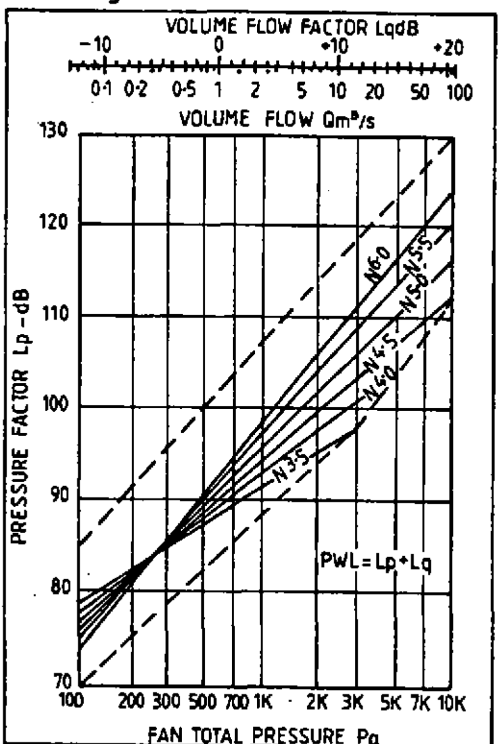

Fig 2 Sound power level and fan duty

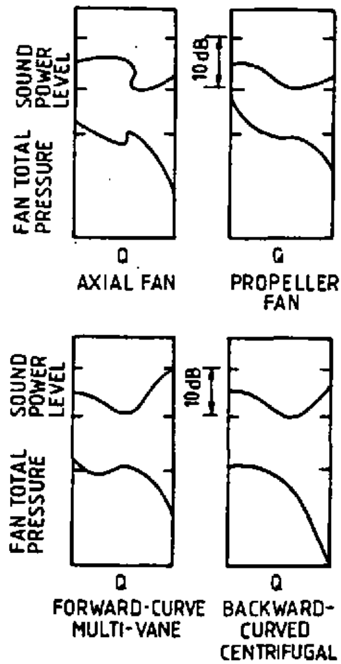

Fig 3 Typical shapes of sound power level characteristics

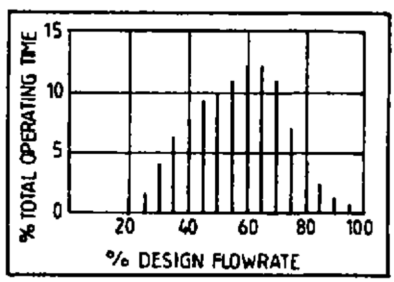

Fig 4 Typical fan operating load profile

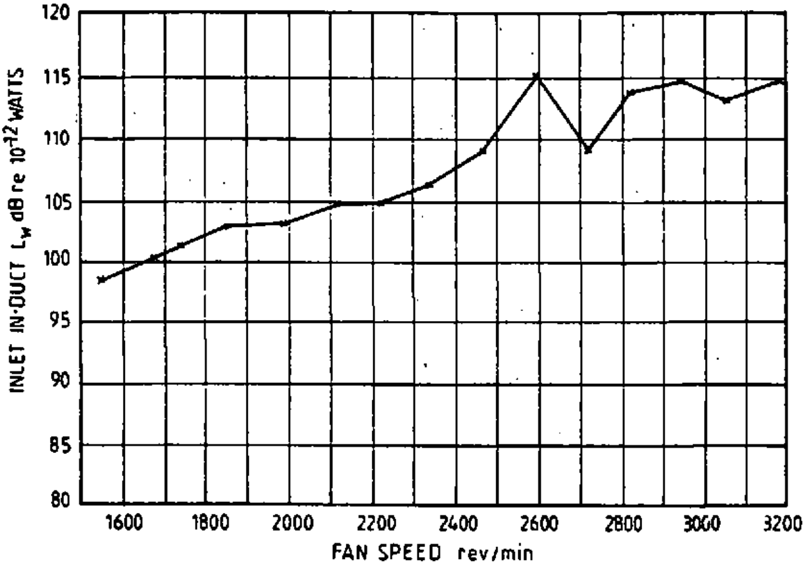

Fig 5 Variation in noise level with speed of 560mm Axcent 1 Mixed Flow Fan

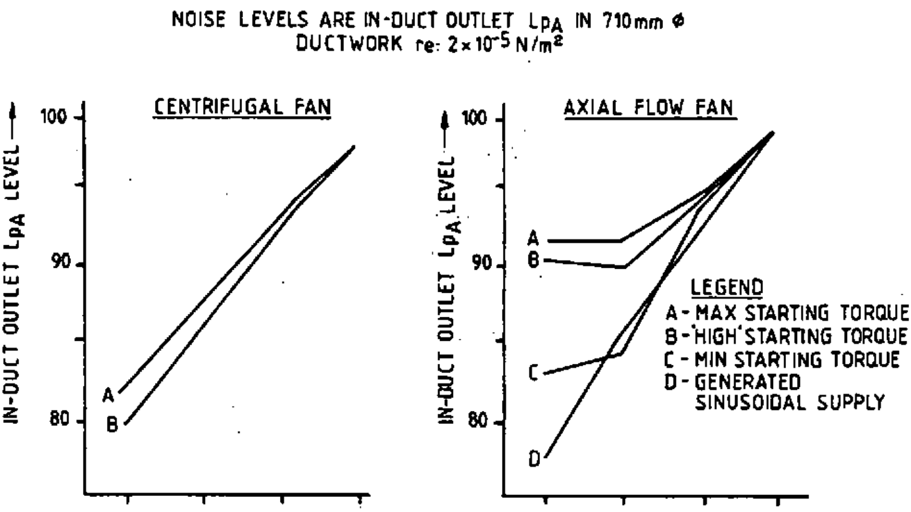

Fig 6 Variation in noise level with speed of Fans according to motor type and control

630 MF @ 1500 rev/min

Vane Angle	Flowrate m^3/s	Fan Pressure Pa	In duct PWL dB re 10^{-12} W								
			Tot	63	125	250	500	1k	2k	4k	8k
Full open	2.4	410	91	84	79	83	86	83	80	75	67
80°	2.37	405	92	85	80	84	87	84	80	75	67
70°	2.3	382	94	88	82	86	88	85	80	75	67
60°	2.17	356	96	91	85	87	89	86	81	76	67
50°	2.05	320	97	94	87	89	90	87	81	76	67
40°	1.89	277	99	96	89	90	90	87	81	75	67
30°	1.67	221	100	98	91	90	90	86	80	75	67
20°	1.39	154	98	96	90	89	88	84	79	74	67
10°	0.76	56	100	98	92	89	87	85	80	76	69
Closed	0	6	101	98	96	92	91	88	84	81	74

Fig 7. Noise Levels of Mixed Flow Fan with Inlet Vane Control

48 in. JSR 16° P.A @ 1480 rev/min

Vane Angle	Flowrate m^3/s	Fan Pressure Pa	In duct PWL dB re 10^{-12} W								
			Tot	63	125	250	500	1k	2k	4k	8k
Fan only	24.1	Free Inlet and Delivery	105	93	90	96	94	94	92	98	95
Full open	23.5		122	98	100	122	110	108	101	97	96
79°	22.8		122	98	100	122	115	111	103	96	94
67°	21.8		123	97	101	119	117	111	101	96	92
56°	20.6		122	98	102	118	116	109	100	95	90
45°	19.2		120	100	103	118	112	106	100	94	87
34°	17.4		118	101	104	118	109	106	100	93	86
23°	14.9		117	102	104	116	107	105	99	93	85
11°	13.1		117	102	102	116	107	105	98	91	84
Closed	0		116	101	101	115	106	104	97	90	83

Fig 8. Noise Levels of Axial Flow Fan with Inlet Vane Control

Disc Throttle noise levels have been determined for a 400/28X backward inclined bladed fan, operating a 4 pole speed. The noise reduction factors, L_{DT} indicate a substantial and consistent noise saving at reduced effective widths. One other feature to note is the very considerable noise reduction in the 63 Hz octave band. L_{DT} for the Disc Throttle is always negative or zero. For the IVC however K_{IVC} can be positive especially at the higher frequencies. No dependency between L_{DT} and blade passing frequency exists. The Disc Throttle only reduces flow/acoustic noise and has no effect on tongue/interaction mechanisms.

$\frac{b_E\%}{b}$ \ Hz	63	125	250	500	1k	2k	4k	8k
100	0	0	0	0	0	0	0	0
80	-1	-1	-1	0	0	0	0	0
60	-4	-4	-4	-2	-2	-2	-2	-2
40	-9	-5	-4	-4	-4	-4	-4	-4
20	-13	-3	-4	-6	-5	-5	-5	-5
0	-13	-3	-5	-5	-5	-5	-5	-5

Fig 9. Disc Throttle v. Inlet Vane Control

To estimate the noise level reductions in eight frequency bands achievable by reducing the fan effective width:

$PWL_{DTi} = PWL + L_{DTi}$

where

- PWL = Sound power level at full effective fan width (with or without the Disc Throttle) dB_W
- PWL_{DT} = Sound power level at reduced effective fan width dB_W
- L_{DT} = Noise reduction factors dB
- $\frac{b_E\%}{b}$ = Percentage full effective fan width
- i = 63, 125, 250, 500, 1k, 2k, 4k and 8k Hz octave bands

Example illustrating typical noise level reductions using a Disc Throttle device, compared with those using an IVC.

dB \ Hz	63	125	250	500	1k	2k	4k	8k
PWL	104	100	96	92	93	87	82	74
L_{DT}	-9	-5	-4	-4	-4	-4	-4	-4
K_{IVC}	-2	0	+1	+2	+2	+2	+2	+2
PWL_{DT}	95	95	92	88	89	83	78	70
PWL_{IVC}	102	100	97	94	95	89	84	76
ΔPWL	7	5	5	6	6	6	6	6

$PWL_{IVCi} = PWL + K_{IVCi}$

K_{IVC} = IVC noise factor typical dB

PWL_{IVC} = level at reduced flow using IVC dB_W

$\Delta PWL = PWL_{IVC} - PWL_{DT}$ (positive) dB

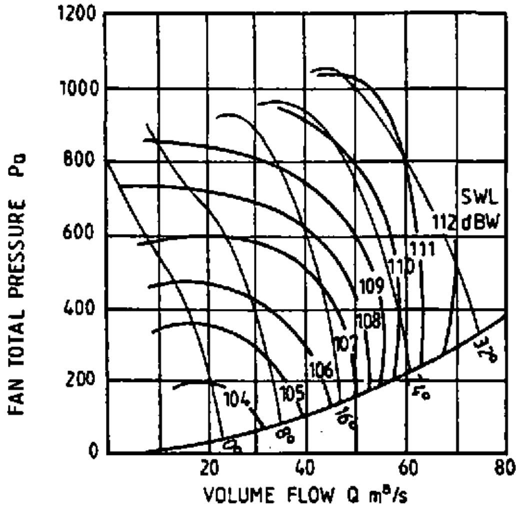

Fig 10 Contours of equal sound power level for 1600 mm 975 rev/min variable pitch Axial Flow Fan