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FAN NOISE

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INTRODUCTION. In any air distribution system, the major source of noise is almost invariably the fan. It is the fan which determines the eventual noise levels in the ventilated spaces, or, for a given level to be achieved, how much additional attenuation will be necessary in the system.

In fact, the designer of the system can rarely do anything to reduce the basic noise produced by a given fan. The amount of noise to be expected under normal operating conditions is a function of its design, and will have been assessed by the manufacturer, along with the aerodynamic performance. It may appear then that the system designer has little scope for influencing the noise at source.

There are, however, two things he can do. First, having decided upon the duty required, he may be able to select a type of fan which produces not necessarily less noise than other types giving the same duty, but noise of a different frequency content that may be attenuated in the system more economically. The second possible course of action is to increase the rated noise of the fan through being unaware of potentially noise generating installations.

Fan Selection. Perhaps the best way to illustrate how a judicious choice of fan can save on cost of additional attenuation is to use a specific example. In many cases the choice is between centrifugal and axial. In a worked example Novick(1) has quoted the sound power levels to be expected from a fan of each type, giving a duty of 6000 cfm at 1½" SWG. The figures, which are based on ASHRAE data(2), are reproduced here on figure 1, together with a typical criterion for the sound power levels at the system outlet. It can be seen that this axial fan will require considerably less low frequency attenuation in the system between fan and outlet, than will the centrifugal. This is important because some form of packaged attenuator will almost certainly be required. Since the physical size (and cost) of attenuators is nearly always fixed by the amount of low frequency attenuation, it follows that a smaller unit can be used with the axial.

Similar considerations apply with some types of propeller fan. While they are not normally used in a system as such, their location in, for example, outside walls of plant room, can give rise to problems of noise affecting nearby residences or offices. The usually low frequency character of the noise then makes the solution difficult - a packaged attenuator for example having to be disproportionately large because of the large free area required to avoid excessive additional system resistance. In such a case, it may be better to install a different type of fan requiring a smaller silencer.

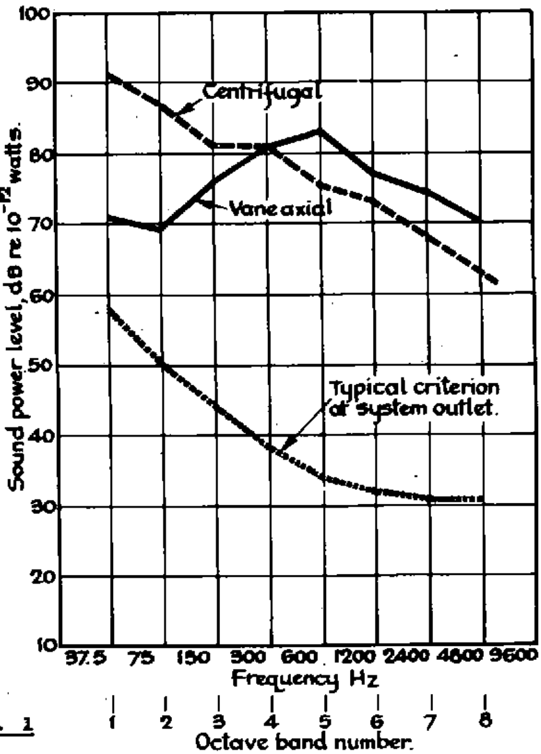

FIG. 1

In general, whatever the type of fan, it is best selected to work as near as possible to its maximum efficiency point. If the installation design is such that the predominant source of noise is "vortex-shedding" (see below), then the amount of acoustic energy produced will normally be a minimum when the blade is working at its highest aerodynamic efficiency. If the flow is significantly reduced, the blade works at a higher relative incidence, the pressure fluctuations on its surface due to boundary layer separation become stronger, and more acoustic energy is radiated. In the extreme case, if the incidence is high enough to cause complete break-away over the upper blade surface (stall), increases of up to 10 dB may result at low and mid-frequencies. Not all fans exhibit these characteristics of course. Indeed, many show little or no change of sound level over the whole of their operating range. Where there is doubt, however, the manufacturer should be consulted. When comparing then, the noise characteristics of different fan types available for a given duty, the comparison should always be on the basis of at least octave band sound power level produced at that duty. It should also be ascertained from the manufacturer, by how much the noise will vary due to the variation of duty likely to be encountered when the system is finally commissioned.

Installation design. The noise rating of a fan which has been provided by the manufacturer, should have been obtained using one of the current standard test procedures, e.g. BS 848[3]. The noise levels will therefore apply for normal installations where the flow into the impeller is reasonably clean. Then, the predominant source of noise will be the so called "vortex-shedding". That is to say that the randomly fluctuating pressures on the blade surface which cause the acoustic radiation, arise from the action of its turbulent boundary layer, rather than from any "external" disturbance of the inlet flow. This is also the situation for many practical inlet configurations generating a flow which does contain disturbances in the form of turbulence, but only of low intensity (ratio of perturbation velocity of the turbulence to the mean flow velocity).

Occasionally, however, the design of the system immediately upstream of the impeller is such that the flow onto the blades be seriously disturbed. Then, the interaction of the generated turbulence and the blade can cause significant increases in noise - as much as 10 dB has been observed during tests in which turbulence was artificially introduced into a fan intake.[4] The mechanism by which this additional noise is generated is illustrated in figure 2. When the fluctuating velocity in the turbulent flow is superimposed on the mean flow, the direction of the flow relative to the blade (i.e. the angle of incidence) fluctuates in a similar manner. There is therefore a fluctuating lift set up on the blade, and this acts as a source of noise. The frequency content of the noise is related to the spectrum of turbulent velocity fluctuations and will therefore appear over a fairly wide range of frequencies. The level of additional noise depends on the amplitude of the lift fluctuations, which in turn are proportional to the amplitude of the velocity perturbation ("W" in figure 2) and roughly inversely proportional to the ratio of the spatial scale of the turbulence (or "eddy" size) to the blade chord. The object should therefore be to avoid designs of the intake system, which produce large shear gradients giving rise to high turbulence intensities, and those producing separated flow containing eddys of a size comparable to the blade chord. In practical terms, this means the avoidance of sharp bends close to the impeller, (or at least the provision of generous turning vanes), upstream dampers or louvres which have to work at high angles of attack being placed as far from the fan as possible, and the provision of bell

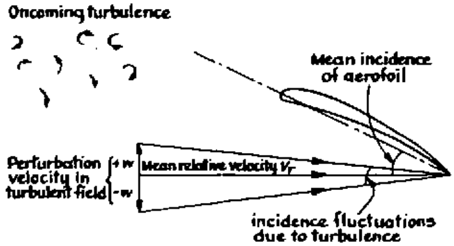

FIG. 2

mouths or coned inlets on all short intakes from atmosphere or plenums.

As a final point, although the foregoing has been in the context of broad band noise arising from a generally turbulent inlet flow, it should be remembered that the wakes from "discrete" obstacles like instrument probes, or internal duct supports, can produce additional pure tone noise by a similar mechanism. As a design rule, the distance between such obstacles and the impeller should never be less than about twice the cross stream dimension of the obstacle.

REFERENCES

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