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AN INVESTIGATION INTO THE NOISE GENERATED BY VOLUME CONTROL DAMPERS IN AIR DISTRIBUTION SYSTEMS

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INTRODUCTION

Within the building services industry, comfort conditions are often maintained by the distribution of conditioned air around the building environment. These air distribution systems normally consist of a central air-handling station and a network of ducts which carries the air to its ultimate destination - the occupied space. Such ductwork, therefore, needs to be appropriately designed to be of a suitable size for the volume of air it carries in order to avoid excessive pressure drop and to provide a suitable air velocity.

Since the space available for the installation of the ductwork may be restricted due to dropped beams, small ceiling voids, etc, it is difficult to size ductwork accurately enough for the system to be self-balancing, i.e. for it to distribute air volumes to the various spaces in the building in the correct proportions simply by proportional sizing. Such balancing is generally achieved by the use of volume control dampers.

The selection of dampers, then, is clearly of importance to the overall efficiency of the system. However, in my experience supervising ductwork installation at a major prestige site (the Barbican Arts Centre), dampers are generally made to fit the ductwork with no further thought given to their characteristics. Indeed, dampers are frequently produced on site in order to compensate for omissions in the design, and are often hurriedly put together; this is unlikely to result in precision grade equipment.

Very little work has been done on damper performance generally, and still less on the noise generation characteristics. It is well known in the Building Services Industry that dampers generate noise, but this is usually accepted as being inevitable and little is done to alleviate the problem.

At present, there is neither an agreed theory nor sufficient empiric data available for accurate prediction of noise generation caused by airflow around damper blades.

Experimental Procedure

Fig. 1 shows the overall system: A centrifugal fan with self-balancing d.c. motor, capable of producing flow velocities of up to 25 m/s supplied the necessary air flow. This fan was acoustically enclosed with large plenum chambers on both inlet and outlet. The working duct was of 180mm square section downstream of a secondary plenum/ settling chamber 1m square x 1.5m deep. This arrangement allowed the incorporation of a low-loss inlet to the test duct which was calibrated against an orifice plate to provide a flow measuring device.

A Bruel and Kjaer type 4133 one inch condenser microphone, 2615 cathode follower, 2131 digital frequency analyser and 2305 sound level recorder were used for the sound power measurements.

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Parallel and opposed blade dampers were inserted into the test duct and the resulting noise generation measured as a function of pressure drop across the dampers and duct flow velocity. The sound power output from the dampers was measured according to BS 848: Part 2: 1966, that is in anechoic conditions using a subtraction method.

The free-field method used twelve measurement positions distributed over a sphere of radius 1m centred on the open end of the duct projecting into the anechoic room.

The pressure drop across the damper was measured with an Airflow Development micromanometer over a duct section of length 20 duct diameters, with the damper situated half-way along the section. Each pressure line to the micromanometer was connected to four pressure tappings in the duct. The pressure drop across the given duct section was first measured with a plain duct section and then with the damper present. This allowed the pressure drop due to the damper alone to be readily determined.

Air flow velocity measurement was by means of the static pressure at the calibrated inlet to the working duct. Such a method was found to be accurate only for static pressures of 100 Pa and above, but this was not a serious disadvantage as lower flow rates generated little sound power at the dampers.

Measurements of sound pressure level were made, at the twelve measurement positions, in the anechoic chamber, and the readings in the 30 third octave bands between 25 Hz and 20 kHz were automatically recorded on the level recorder via the digital frequency analyser.

The above procedure was repeated for four blade angles of both the opposed and parallel blade dampers at each of the chosen fan speeds. Measurements were taken twice at each fan speed to check repeatability.

Results

Figs. 2 and 3 show the one-third octave sound power level for opposed and parallel blade dampers as a function of frequency and blade angle for the maximum fan speed, duct flow rate 0.523 m^3s. There are significant differences between the sets of curves in spectral shape and levels. As the angle of attack increased the opposed blade damper produced sharp peaks in the $\frac{1}{3}$ octave band spectra, while the spectra from the parallel blade damper remained broad with no distinct tonal characteristics. Additonally, for angles of attack up to at least 45^o the opposed blade damper generates the more noise. Both these factors may be partially due to the airflow characteristics for the two dampers. The flow through the parallel blade damper is, for the range of angles considered, more streamlined than that through the opposed blade damper. In turn this meant that a given angle of attack meant a slightly higher pressure drop across the opposed blade damper.

Fig. 4 shows the overall sound power generated for both dampers as they were progressively closed to limit the airflow in the test duct. At each angle of attack the damper free area was determined and the actual air velocity through the damper calculated from the measured flow rates. While there is some spread in the data the least squares fit lines show remarkable agreement as to slope: 30.93 for parallel blades and 31.02 for opposed blades. The intercepts were respectively 25.6 and 28.3 dB.re 10^{-12}W.

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Conclusions

This work shows that for the given duct and dampers considered, the overall sound power output increased approximately as the cube of the velocity across the damper. This differs from the frequently-quoted relationship of sound power varying with the sixth power of the velocity obtained at much higher flow rates.

This variation shows that the noise production mechanism may not be independent of the size of the system and the working velocity range. It is therefore desirable that further work should investigate much wider ranges of duct size and air duct velocity.

The spread of results obtained here means that for any given combination of damper, duct size, air velocity, etc, it is not possible at this stage to predict with certainty whether an opposed or parallel blade damper would be the noisier, but it may be said that, on average, opposed blade dampers will be some 2.5 dB noisier than parallel blade dampers of the same dimensions.

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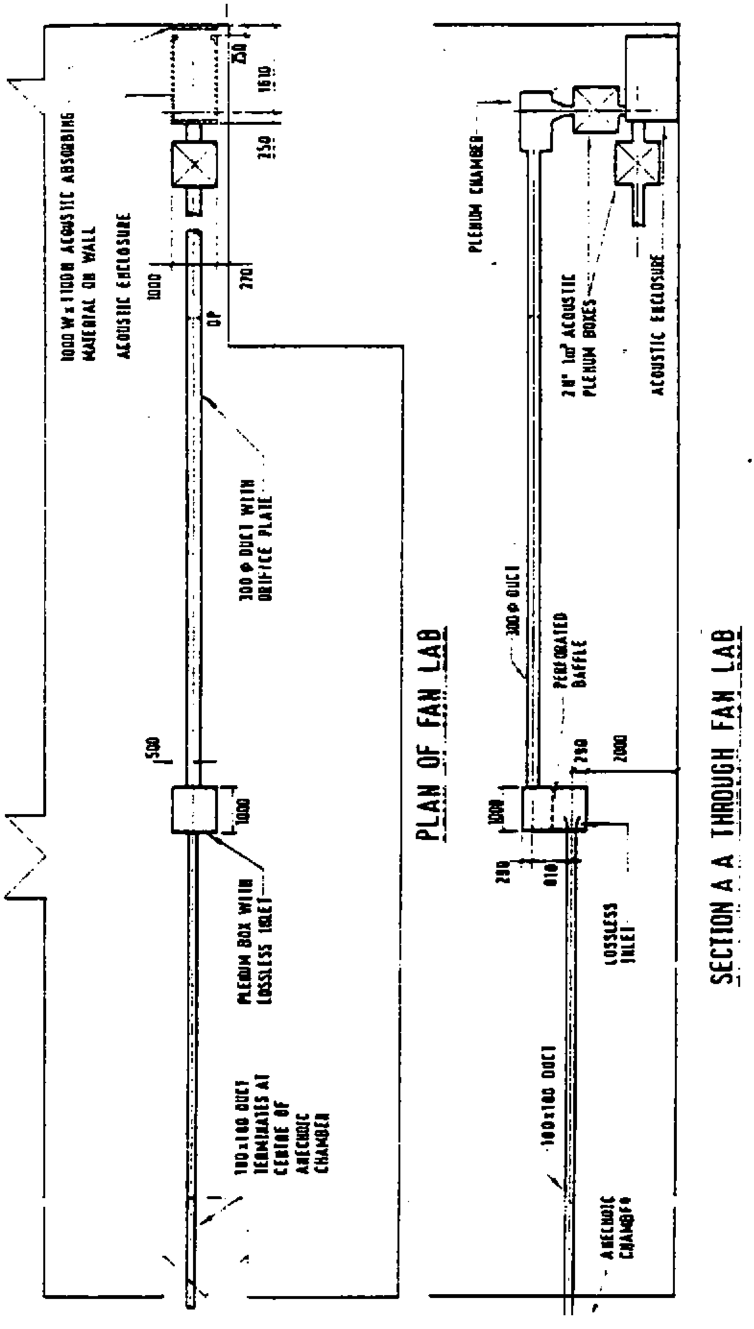

Figure 1 : Diagram of finalised test rig

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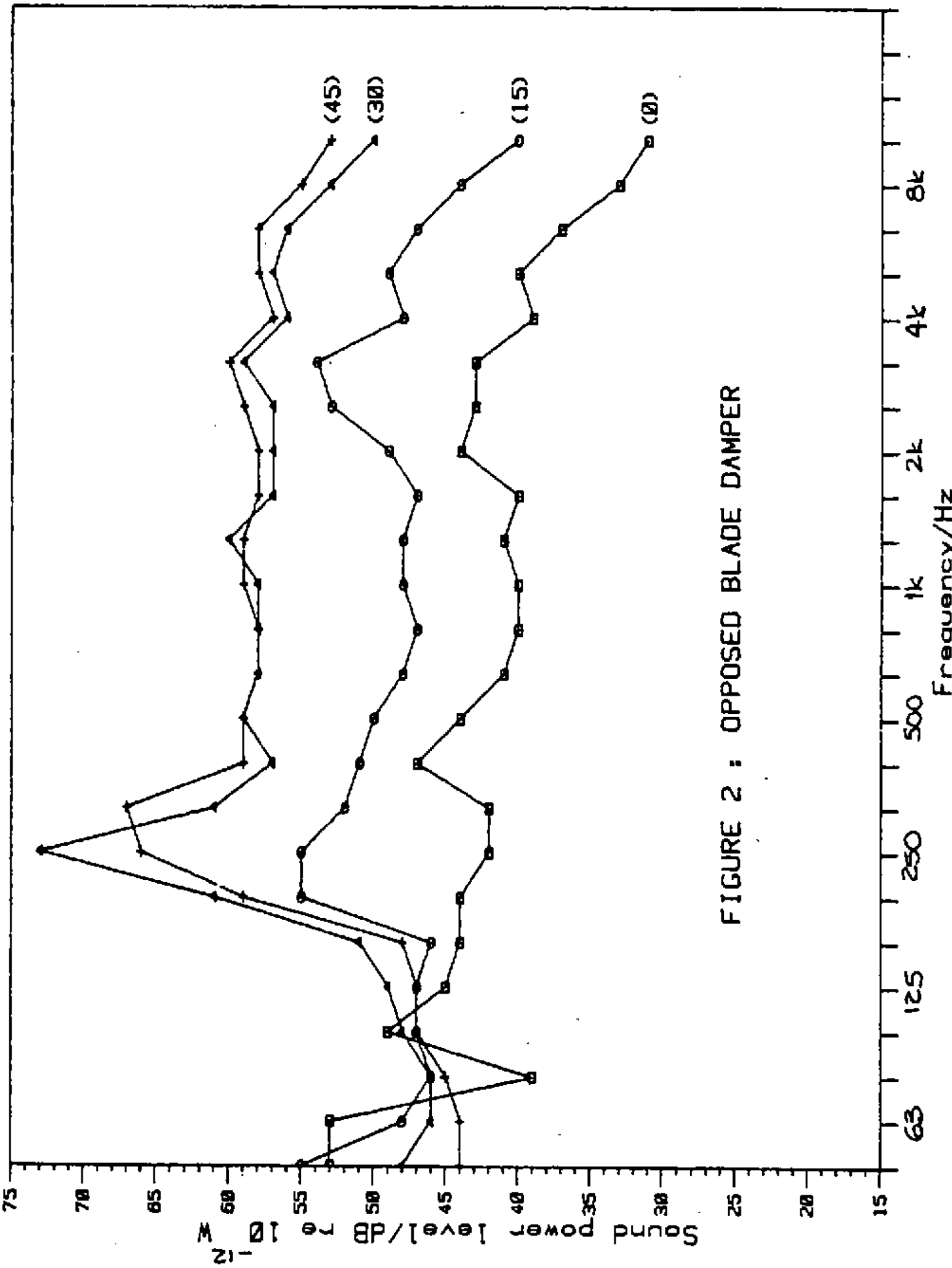

FIGURE 2 : OPPOSED BLADE DAMPER

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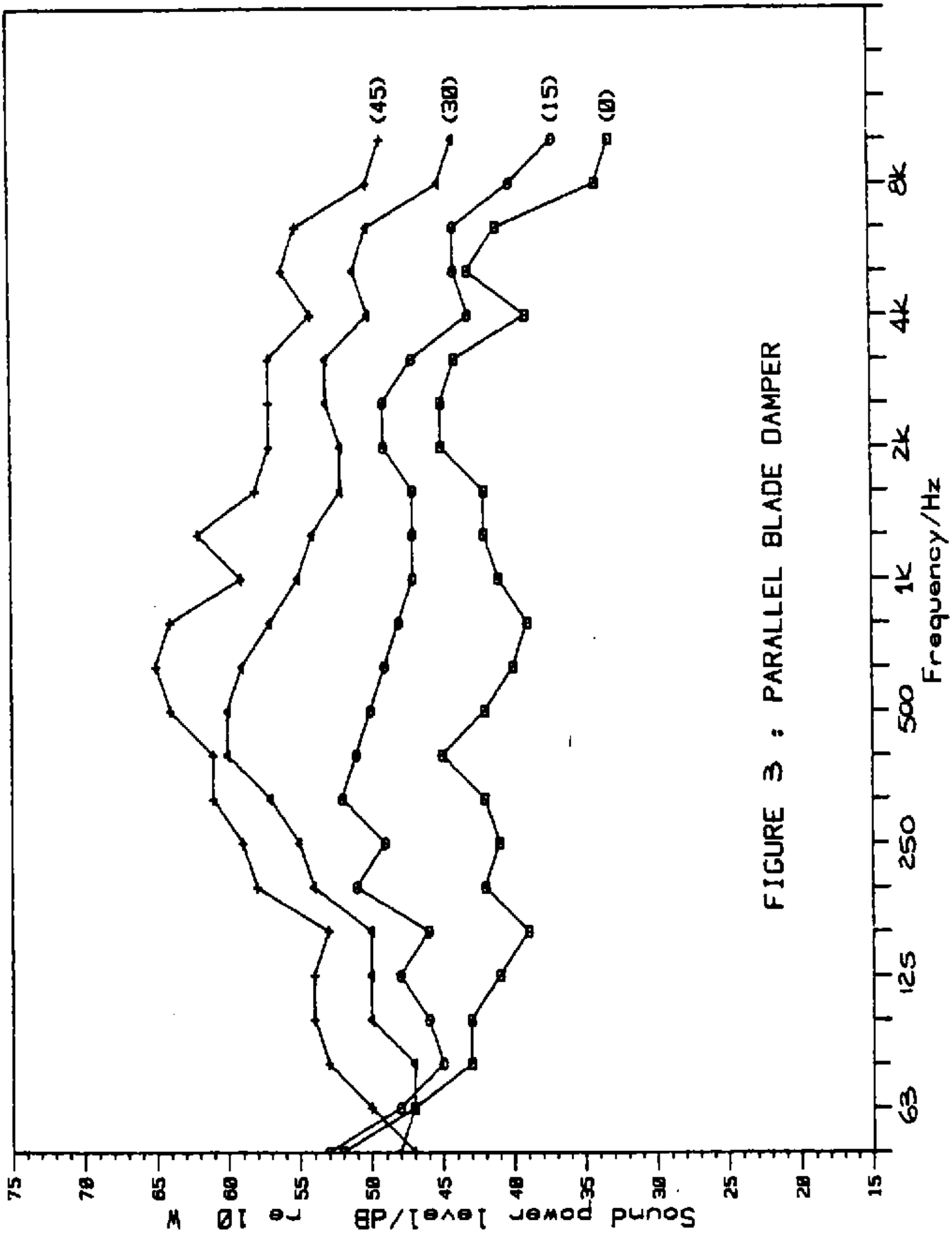

FIGURE 3 : PARALLEL BLADE DAMPER

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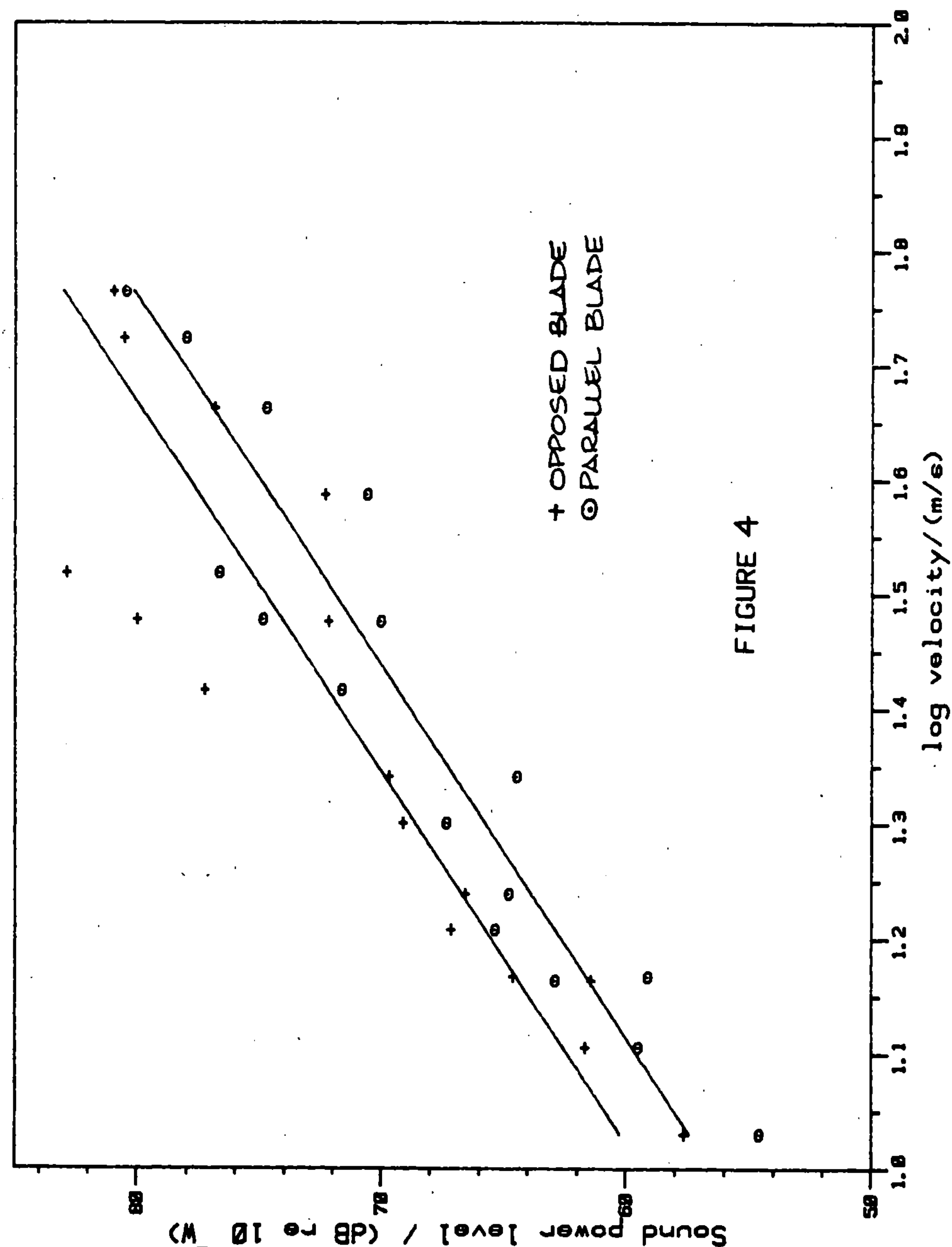

FIGURE 4