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EXPERIMENTS ON A COMPRESSED AIR LOUDSPEAKER

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This work describes the development, construction, theoretical analysis and experimental evaluation of a novel type of electropneumatic sound source. The source has been specifically developed with a view to its application in active noise control systems applied in hostile environments, such as those found in the exhaust systems of gas turbines and internal combustion engines. This need arises in view of the relative fragility and large physical size of conventional loudspeakers and the high degree of non-linearity of existing electropneumatic transducers. In the new design a gas bearing is used to support the friction free motion of a sliding plate which is used to modulate the supply of compressed air. The sliding plate is driven by an electrodynamic vibrator. Experimental results demonstrate that this arrangement reduces harmonic distortion to at least 20 dB below the fundamental driving frequency for most operating conditions. In a companion paper a theoretical analysis of the transducer is presented by Chapman and Glendinning which enables predictions to be made of the acoustic volume velocity (source strength) produced by the transducer as a function of the upstream pressure and displacement of the sliding valve. The predictions of this theoretical model are found to be in good agreement with experimental results.

1. INTRODUCTION

The attenuation of the noise due to the exhaust of large ground-based gas turbines is currently accomplished by using dissipative silencers. The absorptive material incorporated in the silencer effectively attenuates high and middle frequencies while allowing free passage of exhaust gases. However, this type of silencer is not ideal for attenuation of low-frequency exhaust noise; the length and depth of the lining must be increased considerably to give adequate performance. This increases both the cost and size of the silencer. For example, the exhaust stack on the Rolls-Royce SK30 30 MW gas turbine generator set is 30·5 m high and roughly 4 m in diameter. The attenuation by using active techniques of excessive low-frequency noise radiated by a gas turbine exhaust has been demonstrated [1]. An array of 72 loudspeakers of 15 inch diameter mounted in 12 enclosures was arranged about the exhaust exit of an 11 MW gas turbine compressor installation. The system achieved an attenuation of 13 dB at the peak of the exhaust noise spectrum ($\approx$30 Hz).

The use of active control systems on internal combustion engine exhaust ducts as either an addition to or replacement for conventional reactive silencers has also been investigated [2]. In one test on a 17 litre DI engine, two loudspeakers were positioned either side of a 230 mm × 77 mm exhaust duct. At a constant engine speed of 1900 rpm the system achieved a reduction of 20 dB at the microphone position with no measurable deterioration in engine performance.

Thus, the potential for active control in both these applications has already been clearly demonstrated, but in neither instance have active techniques advanced into everyday

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engineering application. The author of the second paper [2] has cited the shortcomings of the electrodynamic loudspeakers as the main obstacle to progress towards a practicable system, and suggests the need for a light rugged transducer as a substitute. The size of the loudspeaker array used in the gas turbine application also suggests that a robust, compact transducer would make further use of active techniques more attractive.

This forms the stimulus for the work presented here. The work involves the experimental investigation of a compressed air loudspeaker (usually referred to as an electropneumatic transducer (EPT)) which generates high-intensity sound by modulating a supply of compressed air. A theoretical analysis of the device is presented in a companion paper by Chapman and Glendinning [3]. Here we review the design of existing electropneumatic transducers and present a novel design which is aimed at ensuring a high degree of linearity of the transducer operation, while being sufficiently robust to enable operation in the harsh environment of an exhaust duct. A series of experiments conducted on the transducer is presented, demonstrating the success of the design adopted and producing results which are in excellent agreement with the theory presented in the companion paper.

2. DESIGN OF THE EXPERIMENTAL TRANSDUCER

2.1. EXISTING TRANSDUCER DESIGNS

High-intensity airflow modulation sources have been commercially available since the 1950s, and there have been few changes in their design. A range of electropneumatic transducers with acoustic power outputs varying from 4 kW to 20 kW is manufactured by Ling Electronics. A similar model producing 30 kW is manufactured by Wyle Laboratories. A typical design is shown in Figure 1. Air modulation is achieved by using a sleeve valve in the form of two concentric cylinders, each with slots around its circumference. The inner cylinder, the stator, is stationary, while the outer sleeve is supported on a synthetic flexure and carries a driver coil, which lies between the poles of a large permanent magnet. In the quiescent state, the slots are aligned so that half the total port area is exposed. When the signal voltage is applied to the coil, the outer sleeve reciprocates, opening and closing the ports. The filtered air flowing inwards from the plenum chamber to the throat of the transducer is thus modulated, producing pressure perturbations upstream and downstream of the valve. These perturbations are propagated as acoustic waves. Provided that the inlet flow in the plenum chamber is symmetrical, the sleeve valve is radially balanced. The valve is mounted so that there is no contact between the stator and the outer sleeve, which almost eliminates friction and allows the

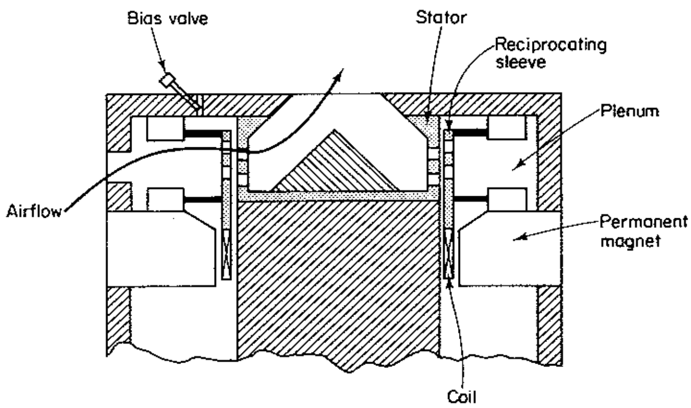

Figure 1. Schematic arrangement of a typical electropneumatic transducer design.

use of a small low-mass driver coil. The annular form of the outer sleeve means that pressure loading results solely in a circumferential compression, with a minimal change in valve clearance. The inherent strength of this design allows the mass and thickness of the outer sleeve to be kep low.

This type of transducer can suffer from scuffing of the valve surfaces as a result of mis-alignment of the outer sleeve. It is uncertain whether this is attributable to instability of the supporting flexure or servicing errors. However, it appears to be a common problem and manufacturers have attempted to reduce scuffing and the associated friction by coating the valve faces with glass-filled PTFE. Wear of the sleeve valve will mean increased clearance and hence increased leakage flow when the valve is in the closed position. This reduces the effective modulation of the flow and consequently the pneumatic–acoustic efficiency of the transducer. A similar increase in leakage flow would arise from expansion of the outer sleeve, and hence this type of valve may not work efficiently at high temperatures.

This type of transducer has been assessed by Rapier and Parkin [4] for use in acoustic fatigue testing of reactor structures. They found the valve to have an axial pressure imbalance causing it to partly close, which was brought about by the exposure of one end of the sleeve to plenum chamber pressure while the other end was exposed to the high-velocity lower-pressure gas flowing through the slot. The effect could easily have been reduced by increasing the length of the outer sleeve.

Unfortunately, all types of valves with a slot arrangement to achieve modulation will suffer from a similar imbalance. When the slots are aligned as in Figure 2, the flow through the port results in a pressure difference between either side of the slot, and the net force acts to close the valve. The magnitude of this force will vary with valve opening position, being largest when the valve is barely open, and reducing to zero as the valve becomes fully open. This inherent non-linearity will affect the acoustic output of the transducer and it should ideally be compensated for by using some electrical or mechanical means. For example, the 10 kW Ling EPT has a bias valve to offset the effect, as shown in Figure 1. However, this only allows setting of the outer sleeve mean position, and the force variation throughout the modulation cycle will still occur.

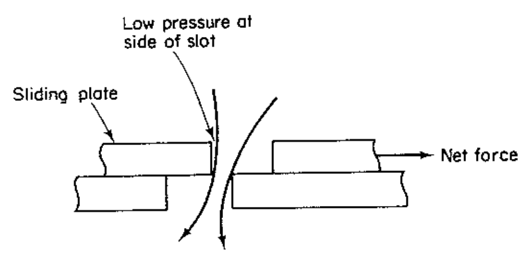

Figure 2. The airflow through a slot, illustrating the generation of an aerodynamic side force on the sliding plate.

The acoustic power output of these devices is roughly proportional to the square of the swept area of the valve. The width of the slots (d) is limited by the reciprocating mass of the valve (m), the frequency up to which full modulation is required (ω) and the modulus of the driving force (F), related by the expression $|F| = m\omega^2 d/2$. An increase in swept area will mean an increase in the reciprocating mass and therefore, to maintain the same operating frequency range, either the slot width must be reduced or the driving force must be increased. Excessive reduction of the slot width will cause increased pressure loss across the valve, which should be avoided, and thus the limit on the size of the transducer is imposed by the driver force and the cooling system.

2.2. THE EXPERIMENTAL TRANSDUCER DESIGN

The principal objective of this work was to determine whether or not an acoustic source of this type offers a suitable alternative to a conventional loudspeaker as a secondary source in an active noise control system. Hence the linearity, output volume velocity and mean airflow through the valve are all important parameters; the chosen design had to allow them to be assessed. To make a sensible study of the linearity of the valve, the valve motion had also to replicate the input signal, thus restricting non-linearity to the fluid-acoustic processes. It was also desirable that the design was sufficiently flexible to allow changes to be made in the valve geometry without complete remanufacture. Finally, while the valve was primarily intended for research, its design should be capable of being adapted to withstand the harsh conditions met within an exhaust duct. The upper frequency operating limit was chosen as 250 Hz; above this frequency absorptive attenuation techniques come into their own and also substantial valve motion is impossible without recourse to an exotic driving mechanism.

Several ideas for the valve configuration were considered [5]. The final choice was made between the annular sleeve design discussed above and the "sliding plate" design that was finally adopted. The annular sleeve design was rejected, since it is difficult to manufacture and assemble the coil and flexure and it is an inflexible design; to change the slot area or shapes would require complete remanufacture of the valve, and the clearance cannot be varied. In addition, turning the flow through 90 degrees causes a reduction of the pressure pulses developed at the slots.

The alternative sliding plate design does not suffer from these problems and has the additional advantage that it can be driven by a standard electrodynamic vibrator. The transducer is illustrated in the diagram of Figure 3 and is also shown photographically in Figures 4 and 5. Its main drawback is the pressure loading on the sliding plate, causing high frictional forces; this limits the throw of the valve and its linearity. These forces could be reduced by using a hydrocarbon lubricant, but this would be impracticable for high-temperature operation. Friction could also be reduced by the use of low-friction materials for the contact surfaces, e.g., PTFE. However, friction would still be appreciable and the sharp edges of the slot would be likely to wear very quickly.

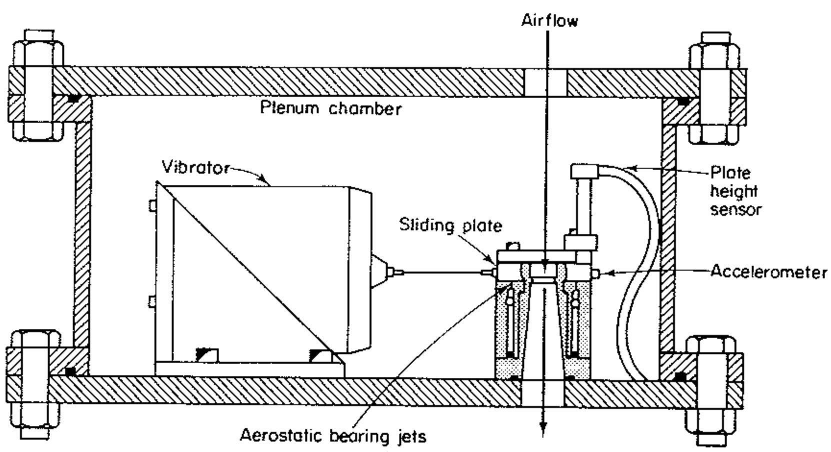

Figure 3. Diagram of the experimental electropneumatic transducer.

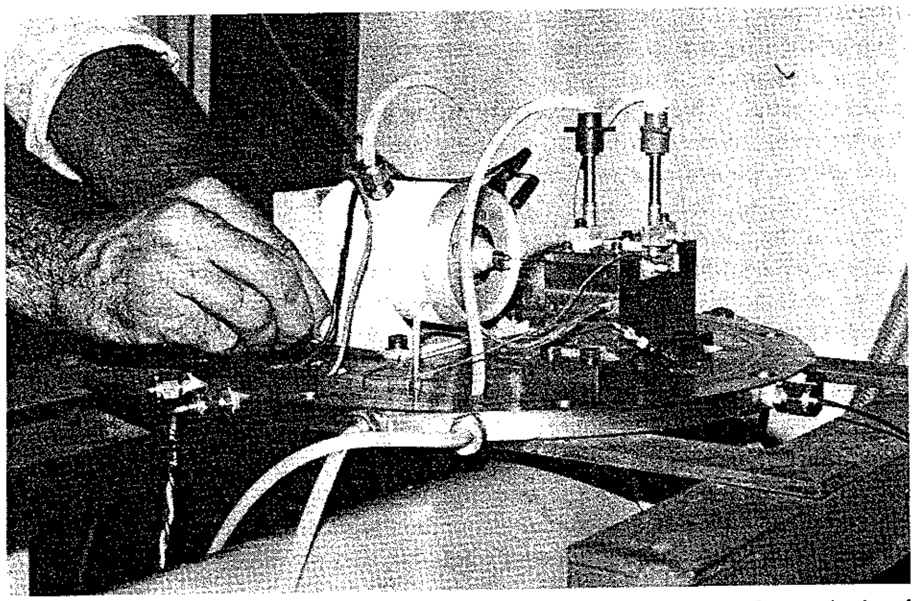

Figure 4. The vibrator and sliding plate assembly, showing the instrumentation for monitoring the plate motion.

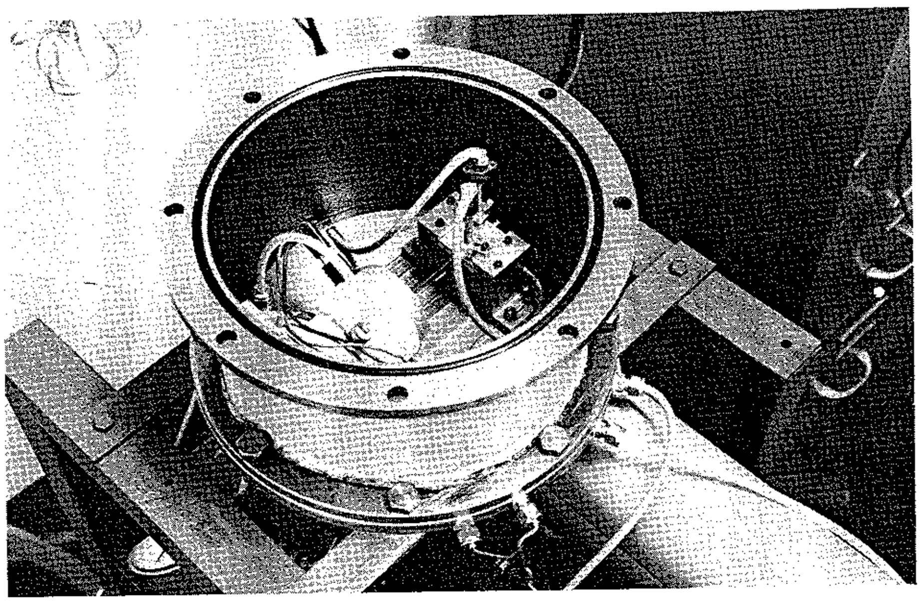

Figure 5. The installation of the vibrator and sliding plate in the cylindrical plenum chamber.

The most attractive solution to the problem is the use of an aerostatic thrust bearing to support the plate. Its advantages are that frictional forces are negligible, there is no surface contact and hence no wear of moving parts and high plenum pressures can be easily accommodated by increasing the bearing air supply pressure. In addition, very low clearances can be obtained (e.g., <12 μm), thus reducing leakage flow, and it is capable of operating at very high temperatures.

A full description of the design of the aerostatic thrust bearing is presented in reference [5]. A schematic diagram of the bearing arrangement is shown in Figure 6. The bearing consisted of an annular thrust pad which produced the support for the sliding plate in the vertical direction. This was produced by using a ring of eight jets of 152 μm diameter. The clearance on either side of the sliding plate was set at 25 μm, thus allowing for expansion effects. Lateral support for the sliding plate was then provided by four jets of 457 μm diameter. Again, full details of this design are presented in reference [5], including calculations of the natural frequencies of the bearing in both translational and rotational modes. These were found to be well above the operating range of the transducer (>1 kHz). The driving system chosen was an electrodynamic vibrator. By enclosing the vibrator in the plenum chamber, the need for any sliding seals was averted. Also it enabled the vibrator to be cooled by using plenum pressure, via a line taken from the vibrator housing.

In order to estimate the required valve swept area for a typical application, it was assumed that the transducer behaved as a constant-volume velocity source. The matching of the secondary source to the primary source can then be considered in terms of volume velocity. A relationship between the size and the position of the sources is derived in detail in reference [5]. For a secondary source placed adjacent to the primary source then the secondary source strength must be equal to that produced by the primary source.

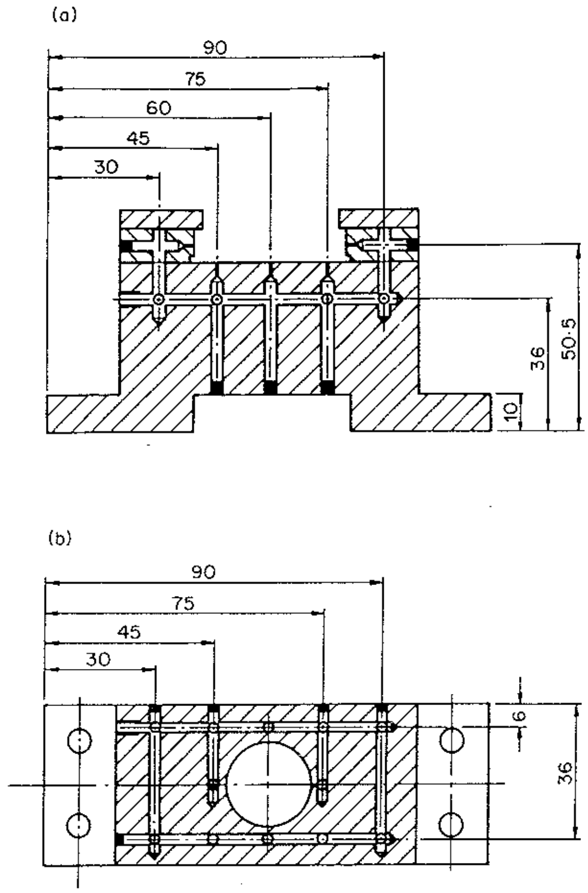

Figure 6. Details of the gas bearing used in the construction of the source, showing the location of the air jets in (a) elevation and (b) plan. The dimensions shown are in mm.

If the secondary source is placed at any other position between the primary source and the open end of the duct, its required strength may need to be very much greater than that of the primary source, depending on the frequency and the exact location. A calculation of the swept area and mean air flow rate required to attenuate the exhaust noise of a 1300 cm^3 SI engine was mady by using the results of Meyer [6]. Full details are given in reference [5]. It was estimated that a valve swept area of 69 mm^2 and an air flow rate of 0·016 m^3 would be required to attenuate the tone at the firing frequency.

The vibrator selected was a Ling Dynamics Model 201, developing 27 N thrust (when cooled), giving an estimated peak-to-peak displacement of 0·4 mm at 250 Hz and of 1 mm at 100 Hz. In order to explore the full potential of the transducer at low frequency, while minimizing the pressure loss through the slots, a slot width of 1 mm was chosen. Although at 250 Hz there was insufficient peak-to-peak displacement to fully open and close a slot of this width, full flow modulation could still be achieved by offsetting the mean position. The slot width of 1 mm was able to be fully swept at frequencies up to 160 Hz.

3. EXPERIMENTATION

3.1. INSTRUMENTATION

The EPT was mounted at the closed end of a duct 110 mm in diameter and 2 m long (see Figure 7 and Figures 4 and 5). The plenum pressure was monitored with a gauge and adjusted by using a diaphragm valve in the main air supply line. Following the valve the air passed through an orifice plate, a filter and then via a flexible hose to the plenum chamber. The pressure difference across the orifice plate was measured with a precision gauge and the total flow rate calculated according to reference [7]. The net flow through the valve was deduced by subtracting the vibrator air cooling flow rate, which was found by using a rotameter. A separate supply was used to feed the aerostatic thrust bearing, again through a filter, the pressure being regulated with a globe valve. The input signal to the vibrator was supplied by a Brüel and Kjaer (B&K) type 1023 sine generator, driving a Quad 50E 50 W amplifier, the output current of which was monitored with an ammeter to avoid overheating the driver coil. A B&K 12 mm pressure microphone was used to

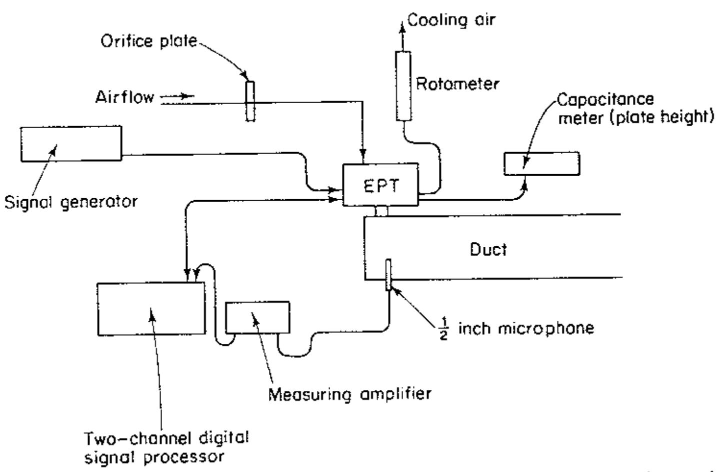

Figure 7. The installation of the electropneumatic transducer in the experimental duct.

measure the acoustic pressure in the duct. This was mounted with its diaphragm 10 mm proud of the duct wall, perpendicular to the exit of the transducer.

Vertical displacement of the slides was gauged with a pair of 6 mm Wayne Kerr capacitance probes, mounted above diagonally opposite corners of the slider. Horizontal displacement was initially determined with a miniature accelerometer, with its output connected via a charge amplifier to an oscilloscope and measuring amplifier. This last technique proved to be unsatisfactory as it gave only the value of the displacement and not the absolute position of the valve. This proved to be a very important factor in the satisfactory operation of the valve. Under steady state conditions, the absolute location of the valve was found to be very sensitive to the aerodynamic side forces discussed above and illustrated in Figure 2. Evidence of this was found when the main air supply was turned on with no input signal to the vibrator. The flow rate increased gradually with plenum pressure and then suddenly decreased, indicating that the valve was shutting off. A capacitance probe was thus used in order that the absolute location of the sliding plate could be determined for each set of modulation experiments. It was found that the offset force on the slider was very sensitive to both peak-to-peak displacement and modulation frequency so the centre position of the valve was re-adjusted for each reading. The flow over the vibrator coil produced a steady drag force which enabled this adjustment to be made.

3.2. EXPERIMENTAL PROCEDURE

For a series of sinuosidal inputs in the range 0–200 Hz, simultaneous measurements were made of valve displacement and acoustic pressure; these were analyzed by using a

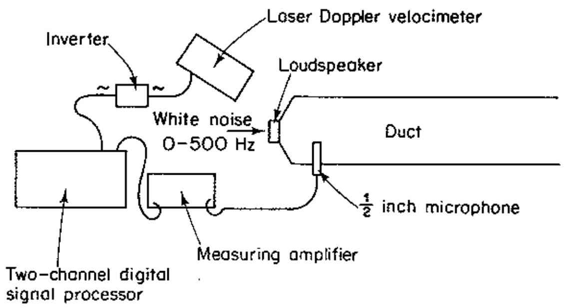

Figure 8. Experimental arrangement for the determination of the input impedance of the experimental duct.

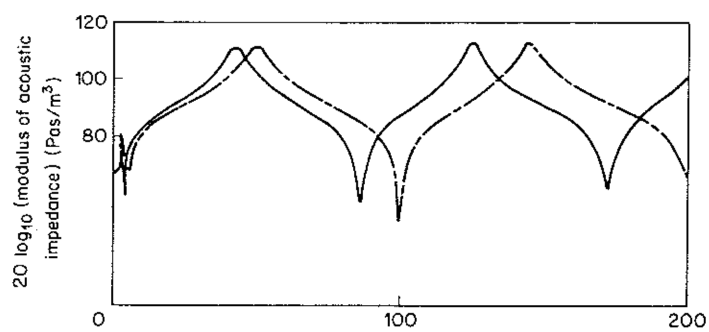

Figure 9. Measured values of the acoustic input impedance of the experimental ducts. ——, 2 m duct; - - -, 1·72 m duct.

two-channel digital signal processor. Measurements were made for peak-to-peak valve displacements of 50–100% of the slot width. The plenum pressure was maintained at 246 kPa. In order to obtain data for input frequencies avoiding duct resonance or anti-resonance, two duct lengths were used (2.0 m and 1·71 m) and measurements made in the frequency ranges between resonance and anti-resonance. The duct input impedances were determined experimentally (see Figure 8) by driving the ducts with a band-limited white noise (0–500 Hz) via a loudspeaker, and dividing the instantaneous acoustic pressure by the product of the velocity and area of the speaker cone. The impedances are given in Figure 9. Dividing the modulus of the duct acoustic pressure obtained with the EPT by the modulus of the duct impedance at the same frequency gives the modulus of the equivalent volume velocity of the transducer.

4. DISCUSSION OF RESULTS

A typical narrow band spectrum of the acoustic pressure measured in the duct is shown in Figure 10. This corresponds to a vibrator input signal at a frequency of 55 Hz. Some of the series of higher harmonics of this frequency are visible above the broadband noise generated in the duct by the turbulent flow exhausting from the EPT. The 2nd, 4th, 5th, 6th and 8th harmonics can be clearly identified. They are, however, at least 35 dB lower in magnitude than the fundamental. The spectrum of the slider displacement at this condition also showed that the harmonic distortion of the slider displacement signal was similarly small. The results of a systematic investigation of the harmonic content of the slider displacement signal as a function of the slider mean position are shown in Figure 11. This was carried out at a frequency of 90·4 Hz and the results are expressed relative to the level of the signal at the level of the fundamental. Again, the level of the second harmonic is shown to be between 20 and 30 dB below that of the fundamental and only weakly dependent on the mean position of the slider. The corresponding results for the harmonic content of the acoustic volume velocity signal are shown in Figure 12. These results demonstrate that the design adopted was successful in minimizing harmonic distortion. Further results on the level of distortion at other frequencies have been presented in reference [3].

The results of a series of experiments to determine the variation of acoustic volume velocity with frequency and peak-to-peak displacement are shown in Figure 13. The volume velocities were determined as described above by dividing the moduli of the

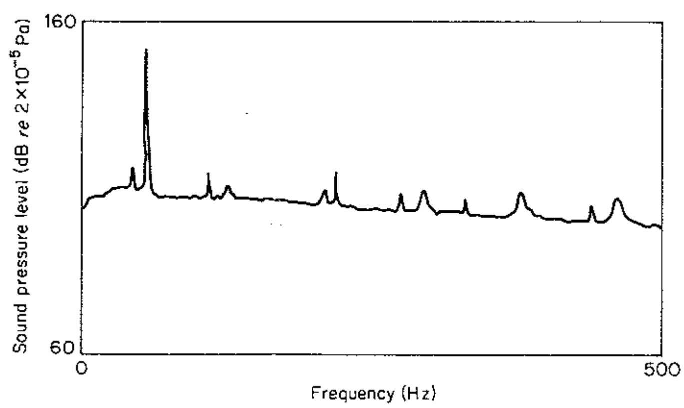

Figure 10. A typical narrow-band spectrum of the pressure fluctuations measured in the duct. The duct length was 2 m excited by an input frequency of 55 Hz, with a slider peak-to-peak displacement of 0·8 mm.

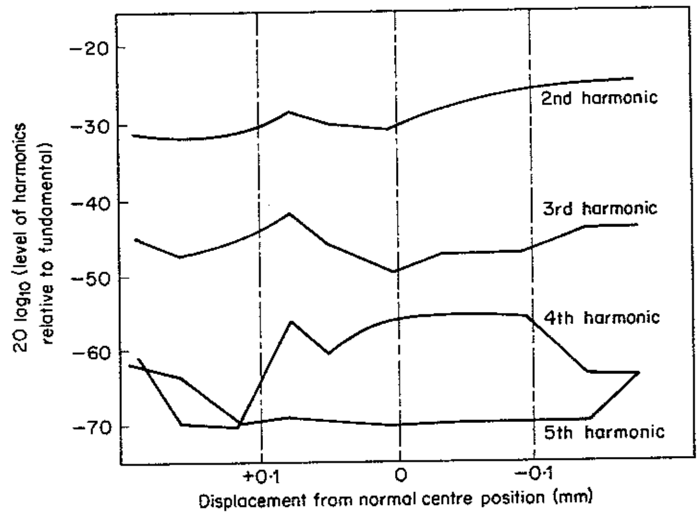

Figure 11. Variation of the distortion of the slider displacement signal with the mean position of the slider. Results shown are for an input frequency of 90·4 Hz, a duct length of 1·72 m and a slider peak-to-peak displacement of 0·8 mm.

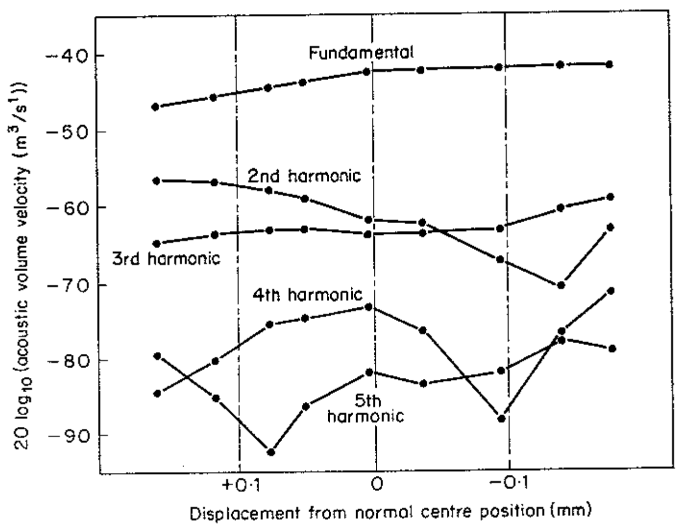

Figure 12. Variation of the distortion of the acoustic volume velocity with the mean position of the slider. Results shown are for an input frequency of 90·4 Hz, a duct length of 1·72 m and a slider peak-to-peak displacement of 0·8 mm.

measured acoustic pressures by the moduli of the measured acoustic impedances of the duct. Also shown in Figure 13 is a plot corresponding to the theoretical predictions of Chapman and Glendinning [3]. This theory gives a relationship between the acoustic particle velocity in the duct δu_3 and the fluctuations in the valve area δA_1 (see equation (29) of reference [3]) such that

$$\delta u_3 = \left(\frac{\alpha}{z_3 + z_i}\right) \delta A_1, \tag{1}$$

where the notation of reference [3] is adopted. Thus z_3 is the specific acoustic impedance of the duct into which sound is radiated, z_i is the ratio of mean pressure $\bar{p}_3$ to mean

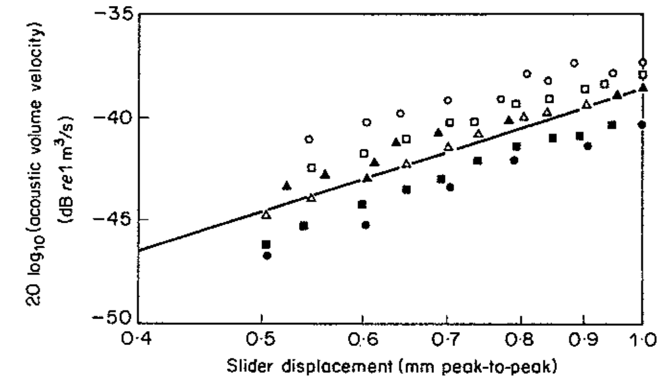

Figure 13. Measured values of acoustic volume velocity as a function of the amplitude of the slider displacement for a range of excitation frequencies. The solid line is the theoretical prediction. □, 54·9 Hz; ▲, 138·5 Hz; △, 78·2 Hz; ●, 90·4 Hz; ○, 41·1 Hz; ■, 75 Hz.

velocity $\bar{u}_3$ in the duct, and $\alpha = (5/6)^3 p_0 c_0 / A_3 \bar{u}_3$, where p_0 is the plenum pressure, c_0 the sound speed in the air in the plenum and A_3 the duct cross-sectional area. For the experimental work undertaken, the value of $|z_3|$ was considerably smaller than $|z_i|$. Thus, for example, for the 2·0 m long duct at 55 Hz then the results of Figure 9 show that $|z_3| \approx 6{\cdot}7 \times 10^2$ Pas/m, whereas with a mean flow of $\bar{u}_3$ of typically 1 m/s then $|z_i|$ is of the order of 10^5. Since all the experiments conducted were at frequencies between resonances and anti-resonances of the two ducts used, it is reasonable to neglect the term z_3 in the denominator of equation (1), since it is always considerably smaller than z_i. (Note that even the largest value of $|z_3|$ measured was a factor of 30 smaller than z_i.) Under these circumstances equation (1) can be written as

$$\delta q_3 = (5/6)^3 (p_0 c_0 / \bar{p}_3) \delta A_1, \tag{2}$$

where $\delta q_3 = \delta u_3 A_3$ is the acoustic volume velocity produced at the duct entrance by the electropneumatic transducer. This is therefore an extremely simple and useful relationship, which shows the clear dependence of the transducer output on the pressure ratio across the valve and the valve area modulation. The experimental results are clearly in good general agreement with this prediction, where values of $p_0 = 2{\cdot}46 \times 10^5$ Pa, $c_0 = 343$ m/s and $\bar{p}_3 = 1{\cdot}02 \times 10^5$ Pa have been used in drawing the theoretical plot. The results do appear to exhibit some dependence on frequency, although there is no systematic variation that could be detected. The deviation from theoretical predictions could easily be due to the accumulation of errors associated with the measurements of acoustic pressure, acoustic impedance and slider displacement.

Finally, it may be useful to give a preliminary assessment of the potential advantages to be had from employing an electropneumatic transducer of the type investigated. Although the particular transducer used had only a modest capability in terms of its ability to produce volume velocity (at 50 Hz its output is of the order of that which could be produced by a conventional 0·15 m diameter loudspeaker) then such a design could offer advantages, provided that it could be built with a larger valve cross-sectional area. In order to obtain a direct comparison with conventional loudspeakers, one can assume that a valve of diameter d could be engineered and compare its volume velocity output with that of a loudspeaker of the same diameter. One can also assume that the electromagnetic device driving both the valve and the loudspeaker is capable of producing a displacement $\delta\xi$. Thus the change in valve area produced by such a displacement can be written as $\delta A_1 = \delta\xi (1/2)(\pi d^2/4)^{1/2}$, where the factor of (1/2) appears since only half the area of the valve transmits flow. Equation (2) can now be used to calculate the volume velocity of the transducer in terms of d and $\delta\xi$. Similarly, the volume velocity δq produced by a loudspeaker of diameter d and displacement $\delta\xi$ at a frequency ω is given simply by $\delta q = \omega\delta\xi(\pi d^2/4)$. It therefore follows that the ratio of the volume velocity of the transducer to that of a loudspeaker of the same diameter and displacement is given by

$$\delta q_3/\delta q = (5/6)^3(1/\sqrt{2\pi^3})(p_0/p_3)(\lambda/d), \tag{3}$$

where λ is the wavelength of the sound in air (with sound speed c_0) at the frequency ω of interest. This expression reduces to

$$\delta q_3/\delta q = 0{\cdot}073(p_0/\bar{p}_3)(\lambda/d). \tag{4}$$

Thus if the valve could be operated at a pressure ratio $(p_0/\bar{p}_3)$ having a value in the region of 3 (which is common for conventional EPTs), then the ratio $\delta q_3/\delta q$ would be around $0{\cdot}2\ \lambda/d$. Thus if $d = 0{\cdot}2$m and $\lambda = 6{\cdot}86$ m (corresponding to 50 Hz) then $\delta q_3/\delta q = 6{\cdot}86$. Thus a 0·2 m diameter EPT could produce around seven times the volume velocity of a conventional loudspeaker of the same diameter and displacement. In terms of acoustic power output, for the same radiation impedance and no loading effects, then the power output of the EPT would be $20 \log_{10}(6{\cdot}86) \approx 17$ dB greater than that of the equivalent loudspeaker. This advantage clearly becomes greater as the frequency becomes lower and λ increases. However, there are clearly issues of efficiency of energy conversion to be considered, especially with respect to the power necessary to provide the compressed air supply. These factors are application-specific and will not be considered here. It also remains to be established whether such a valve could indeed be built to operate as successfully with a larger cross-sectional area and with an electromagnetic actuator that is truly comparable with that of a loudspeaker.

5. CONCLUSIONS

The mechanical design of the transducer was successful. Distortion of the valve motion was low (less than 25 dB) and was found to vary with frequency and peak-to-peak displacement. It was thought to be a result of varying aerodynamic rather than mechanical forces. Harmonic distortion of the acoustic volume velocity output of the valve was generally of the order of 20 dB below the fundamental. This too was found to vary with the valve centre position and frequency. A linear relationship was found between the acoustic volume velocity and the peak-to-peak valve displacement. The theoretical model of the transducer due to Chapman and Glendinning [3] gave a good prediction of the experimental volume velocities.

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