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ELECTROPNEUMATIC TRANSDUCERS AS SECONDARY ACTUATORS FOR ACTIVE NOISE CONTROL, PART I: THEORETICAL ANALYSIS

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A theoretical analysis is presented of an acoustic source which operates by the release of compressed air through an aperture, the area of which is made to vary with time. A distinction is made between sonic and subsonic compressed air sources (depending on the Mach number at the throat), and in both cases simple equations are derived that describe the acoustical characteristics of the device. The theory is also developed with a view to using the source as a secondary actuator in an active noise control system, and the pneumatic efficiency and linearity of compressed air sources are discussed. Although the sonic source has a high internal acoustic impedance and its output is almost linearly dependant on the aperture opening, it is shown to be very inefficient. The subsonic source can be much more efficient than the sonic one but the output is generally no longer a linear function of aperture opening. A simple method of linearizing such a source is discussed. A numerical comparison between compressed air sources and electrodynamic loudspeakers shows that the former offer a useful alternative for active noise control, especially when the secondary source has to act in an extreme environment. The subsonic compressed air source is particularly useful when efficiency is a major issue. Details and results of the experimental evaluation of a subsonic compressed air source built at the Laboratory of Acoustics of the Faculté Polytechnique de Mons (Belgium) are presented in companion paper (Part II).

INTRODUCTION

The principle of the control of sound by active techniques was described for the first time in 1936, in a patent published in the United States by Lueg [1]. Lueg's idea was to create a destructive interference between the sound field generated by the unwanted noise source (often called the primary source) and that emitted by

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(a) secondary source(s). Since the mid 1970s, the interest of scientists in active control has been continuously growing. Research of the past 25 years of Active Noise Control (ANC) is documented in several thousand publications and has resulted in a reasonable understanding of the underlying physics principles involved, as well as in the development of efficient control algorithms. One of the merits of the research was to demonstrate that active methods were particularly useful for the attenuation of low frequency sound fields. Industrial applications of active noise control started to appear in the mid 1980s, to solve noise problems for which the advantages of active techniques were readily apparent. These industrial developments were, however, impeded by the lack of powerful secondary sound sources. For example, the active control of automobile exhaust noise was proven theoretically feasible but very difficult to implement in practice: the diaphragm of electrodynamic loudspeakers placed in the exhaust pipe is likely to be quickly damaged by the high temperature (about 300°C in normal conditions) exhaust gas flow. This difficulty was recognized by Roure [2], for example, who quoted today's impossibility of designing loudspeakers able to resist extreme environments.

Secondary sources for active noise control must ideally be lightweight, compact, robust (i.e., able to work for many years without showing significant changes in their characteristics), cheap, linear, efficient, easily and accurately controllable. Moreover, depending on the application considered, secondary sources can be requested to supply large volume velocities (order of magnitude: $0{\cdot}05\ m^3/s$) at very low frequencies (10 to 20 Hz, say), to be able to resist extreme (hot, humid, corrosive) environments without being damaged, or to be highly directive. Some solutions to this design problem have already been proposed, for example, Inoue *et al*. [3] suggested protecting the diaphragm of electrodynamic loudspeakers acting in a high temperature (250°C) gas duct using a thin film of inert material. The loudspeaker was cooled and isolated from corrosive combustion gas by pouring air into a duct connected between the main smoke duct and the loudspeaker. The production of large volume velocities at very low frequencies was considered by Blondel [4] and by Sjösten *et al*. [5], who showed that an efficient secondary source could be implemented by the use of vented-box loudspeakers, especially when the primary noise is tonal.

The solutions mentioned above are based on the modification of electrodynamic loudspeakers. Alternative possibilities also exist. A secondary actuator proposed by Renault *et al*. [6] controlled the periodic pulsed flow of fluid in the exhaust pipe of a combustion engine by using a flap placed in the duct. By making the flap oscillate, it was possible to control the pressure fluctuations dowstream in the duct. The main drawback of the oscillating flap is that its use is restricted to the control of acoustical flows in ducts. Another idea is to replace the electrodynamic loudspeaker by an electropneumatic trandsucer (EPT). Broadly speaking, electropneumatic transducers (sometimes called pneumatic sound generators or compressed-air loudspeakers) are devices in which an airflow is modulated by a valve controlled by an electrodynamic driver or by an electrohydraulic driver. Electropneumatic transducers can produce extremely high acoustic powers (100–200 kW), are controllable and their architecture is such that they are rugged and able to resist extreme environments. They are thus good candidates for active noise control but have not been widely investigated. One of the first papers on electropneumatic sources is due to Webster *et al*. [7], who measured the performance of a sonic electropneumatic transducer used as a speech and alarm transducer on the flight deck of an aircraft carrier (sonic sources refer to sources in which the Mach number is unity at the throat of the device). Fiala *et al*. [8] designed a sonic electropneumatic transducer capable of producing 6 kW of acoustic power, and their paper contains both the description of the architecture of the transducer and measurements of the efficiency, of the frequency response and of the linearity of the device.

Meyer [9] appears to have been the first to propose a comprehensive theory of the electropneumatic transducer. In Meyer's analysis the flow is divided into three regions: the nozzle, a mixing zone and an acoustic zone, in which acoustic plane waves are propagating. The analysis is separated into two cases: sonic and subsonic nozzle velocitites (subsonic sources refer to sources in which the Mach number in the throat is significantly less than unity). The result of Meyer's analysis is the establishment of a relationship between instantaneous values of the pressure at the end of the mixing zone and of the area of the nozzle throat. Unfortunately, this relationship was proposed only in a graphical form. Moreover, Meyer recognized that his theory was not very accurate for predicting the performance of the electropneumatic transducer at modulation frequencies below 100 Hz and that an extension of the theory into the lower subsonic flow region was necessary. The theory of sonic electropneumatic transducers was also considered by Chapman and Glendinning [10], who reworked the problem using a similar approach to that of Meyer, but was able to derive an explicit relationship between the pressure and velocity at the end of the mixing region. This enabled the determination of the amplitude and phase of the acoustic pressure at the source output as a function of the area of the nozzle throat. An experimental analysis of a sonic source [11] proved that the results of the theoretical study were fairly accurate, but unfortunately this reference provided no data on the efficiency of the source.

A recent contribution to the development of the electropneumatic transducer and to its application in active noise control is due to Raida and Bschon [12], who show that the device can, under particular conditions, exhibit a highly directional characteristic and behave as a tripole. This directional characteristic is valuable for the active control of noise in ducts because it reduces the acoustic feedback between the secondary source and the primary sensor.

This brief review suggests that there is a need for further research on the theoretical and practical aspects of electropneumatic transducers. For example, hardly any work appears to have been done on the use of the subsonic source in active control. The efficiency of electropneumatic transducers also does not appear to have been considered in detail by many researchers. One of the purposes of this paper is to compare the performance and properties of sonic and subsonic sources. The integration of electropneumatic transducers into active noise control systems and its consequences on the system performance are also major issues. This forms the stimulus for the work presented here. The paper is arranged as follows.

Section 2 is devoted to the derivation of the fundamental equations of both sonic and subsonic sources, whereas section 3 deals with the problem of linearity of these sources. Section 4 considers the important issue of the efficiency of electropneumatic transducers. Finally, the merits of sonic and subsonic transducers are compared to that of common electrodynamic loudspeakers in section 5, in view of their application as sound generators or as secondary sources.

2. FUNDAMENTAL EQUATIONS OF ELECTROPNEUMATIC TRANSDUCERS

2.1. THE SONIC ELECTROPNEUMATIC TRANSDUCER

As stated in the introduction, the behaviour of the sonic electropneumatic transducer has already been analyzed theoretically by Meyer [9] and by Chapman and Glendinning [10]. This section presents a brief summary of the main results of this analysis, which is required for the calculation of the efficiency of this device in section 4. The electropneumatic transducer considered in the analysis is shown schematically in Figure 1. It consists of a plenum chamber that is supplied with compressed air which is separated from the source output by a valve. The analysis of the device is based on the following assumptions. (1) A one-dimensional flow is assumed; (2) compressed gas from the plenum chamber takes the route marked 1, 2 and then emerges into the atmosphere; (3) by the end of the throat (station 2), the flow is uniform; (4) the variation of the throat area is a prescribed function of the time $A_1(t)$: (5) the oscillation frequency is assumed low enough to allow a quasi-static approximation of the flow between the plenum chamber and station 2 (the validity of this assumption is discussed by Meyer [9]); (6) the fluid used in the device is air, which is assumed to be a perfect gas; (7) in the plenum chamber the air is in stagnation conditions; (8) the plenum pressure p_{pl} assumed to be large enough to guarantee a sonic flow at station 1: M_1, the Mach number at this station is equal to 1 (in other words the transformation between the plenum chamber and station 1 is isentropic [13]); (9) the transformation between station 1 and 2 is adiabatic: no external heat is added from outside the walls.

Under these assumptions, it can be demonstrated [10] that the equation linking the pressure p_2 and the particle velocity v_2 at station 2 of the source can be written as

$$p_2 u_2 = \beta A_1, \quad \text{with} \quad \beta = \left(\frac{5}{6}\right)^3 \frac{c_0 p_{pl}}{A_2}, \tag{1}$$

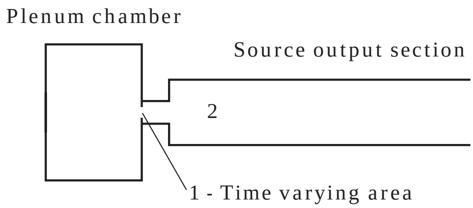

Figure 1. Simplified model of the electropneumatic transducer, as suggested by Meyer [9].

where A_2 is the cross-sectional area of the source output section. Because of assumption (5) above, this equation can be used not only to compute the steady state operation of the source (for which A_1 is fixed permanently to its mean value) but also to characterise the operations of the source as A_1 changes with time. One can, for example, write

$$p_2 = \bar{p}_2 + \delta p_2,\ u_2 = \bar{u}_2 + \delta u_2,\ A_1 = \overline{A}_1 + \delta A_1, \tag{2}$$

where $\bar{p}_2$ is the average value of p_2 and δp_2 is the time varying pressure at station 2. In practice, the factor $\delta u_2 \delta p_2$ is negligible compared to $\bar{p}_2 \bar{u}_2$. For example, considering the numerical example provided by Chapman and Glendinning [10], one computes:

$$\bar{p}_2 u_2 = 12 \times 10^4\ \text{N/ms} \quad \text{and} \quad \delta p_2 \delta u_2 = 12 \times 10^2\ \text{N/ms}.$$

Equation (1) can then be written

$$\bar{p}_2 \delta u_2 + \bar{u}_2 \delta p_2 \cong \beta \delta A_1. \tag{3}$$

Upon defining $\bar{p}_2/A_2\bar{u}_2$ and R_i, the internal resistance of the source and $\delta p_2/A_2\delta u_2$ as Z_l, the acoustic load impedance seen by the source, equation (3) becomes

$$\delta p_2 = A_2 \beta \delta A_1 / \bar{p}_2 \left(\frac{1}{Z_l} + \frac{1}{R_i} \right), \tag{4}$$

or, if one is interested in volume velocity variations, this is equivalent to

$$\delta u_2 = \beta \delta A_1 / A_2 \bar{u}_2 \, (R_i + Z_l). \tag{5}$$

Equations (4) and (5) are the basic operating equations of sonic electropneumatic transducers. The output of the device is a linear function of the alternating throat area, δA_1, and the device has a very high internal impedance, $R_i = 10^8 \times$ N s/m^5 for the device described by Chapman and Glendinning [10]. Note that when the source is acting as a secondary source in an active noise control system (in a duct for example), the pressure in front of the source will be cancelled so that Z_l goes to zero and equation (4) and (5) respectively become

$$\delta p_2 \cong 0 \quad \text{and} \quad \delta u_2 \cong \beta \delta A_1 / \bar{p}_2. \tag{6, 7}$$

2.2. THE SUBSONIC ELECTROPNEUMATIC TRANSDUCER

The compressed air source considered here is very similar to the one considered in the sonic case. The main difference is that smaller plenum pressures are used here, so that the Mach number at the throat is smaller than 1 and hence the flow through the throat is subsonic. More particularly the case will be considered for which $M_1 < 0{\cdot}6$, so that the fluid can be considered as incompressible, at least in a first approximation. The assumptions required for the analysis are therefore very close to those in the sonic case, except for assumption (8), which becomes: (8) the transformation between the plenum chamber and station 1 is adiabatic, and the fluid is assumed to be incompressible.

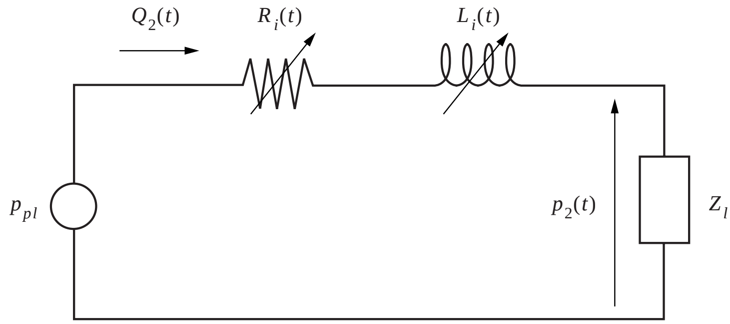

Figure 2. Equivalent acoustic diagram of the subsonic compressed air source. In this diagram, p_{pl} is the steady plenum pressure, $R_i(t)$ and $L_i(t)$ respectively represent the time varying resistance and inductance of the opening formed by the throat, Z_l is the acoustic impedance of the acoustic system which loads the source, $Q_2(t)$ is the volume flow into the duct and $p_2(t)$ is the time varying pressure at the duct input.

The equivalent acoustic diagram of the source is given in Figure 2. In this model the instantaneous flow through the throat is related to the instantaneous pressue across it by a time-varying impedance $R_i(t) = P(t)/Q_2(t)$. This diagram is equivalent to the diagram given by Clark Jones [14], who was interested in the study of acoustic sirens, a problem similar to that of electropneumatic transducers. According to several authors [15, 16], the reactance $L_i(t)$ can be considered as negligible compared to the resistance $R_i(t)$, at least for small throat openings. According to Sivian [17] the measured resistance R_i of a small rectangular orifice under the conditions above is closely approximated by the following equation, assuming that the flow across the orifice is fully turbulent,

$$R_i = (\rho/C_d)(Q_2/A_1^2), \tag{8}$$

where Q_2 is the volume velocity through the orifice, A_1 is the area of the orifice, C_d is the discharge coefficient of the orifice and ρ is the density of the fluid involved.

One can also write

$$R_i = p/Q_2, \tag{9}$$

where p is the pressure difference across the orifice. The combination of these two equations leads to

$$Q_2 = A_1\sqrt{C_d(p_{pl} - p_2)/\rho}. \tag{10}$$

This last equation can also be derived directly from the simple application of the Bernouilli equation to an incompressible fluid. Upon using the quasi-static approximation, equation (10) continues to hold when the throat area varies in time:

$$Q_2(t) = A_1(t)\sqrt{C_d(p_{pl} - p_2(t))/\rho}. \tag{11}$$

This last equation is the fundamental equation of subsonic electropneumatic transducers. The theoretical value of the discharge coefficient is 2, and this was found to be a good approximation to that observed in practice, as discussed in the companion paper (Part II). For alternating quantities, one can write $p_2 = \bar{p}_2 + \delta p_2$, $u_2 = \bar{u}_2 + \delta u_2$, $A_1 = \overline{A}_1 + \delta A_1$, and equation (11) becomes

$$\delta p_2 = A_2 Z_l \left[\frac{\overline{A}_1 + \delta A_1}{A_2} \sqrt{\frac{C_d(p_{pl} - \bar{p}_2 - \delta p_2)}{\rho}} - \bar{u}_2 \right], \tag{12}$$

or alternatively if one is interested in volume velocity variations, then

$$\delta u_2 = \frac{\overline{A}_1 + \delta A_1}{A_2} \sqrt{\frac{C_d(p_{pl} - \bar{p}_2 - \delta p_2)}{\rho}} - \bar{u}_2, \tag{13}$$

The output of the subsonic source is thus not linearly dependant on the alternating throat area, and it cannot be guaranteed that the internal impedance is large. These equations are analyzed and discussed in more detail in the following sections.

3. LINEARITY OF ELECTROPNEUMATIC TRANSDUCERS

One of the major requirements of secondary sources used in active noise control is linearity. Weak non-linearities in the secondary transducer are likely to strongly reduce the efficiency of a conventional active noise control system by limiting the noise reduction, as discussed by Beltran [18]. Broadly speaking, electropneumatic transducers are likely to exhibit two source of non-linearities: the physical phenomenon of sound emission is likely to be non-linear and the link between the electrical current at the input of the electrodynamic shaker and the modulation of the airflow can also be non-linear. These phenomena are discussed below for both the sonic and the subsonic sources.

3.1. LINEARITY OF THE SONIC ELECTROPNEUMATIC TRANSDUCER

The fundamental equation for sonic electropneumatic transducers (equation (1)) is non-linear. However, as already stated, it is usually valid to neglect the non-linear term in this equation when perturbed quantities are concerned, which leads to a linearized approximation: the mechanism of sound generation in sonic sources is almost perfectly linear. The degree of linearity of the source therefore directly depends on the accuracy of its mechanical design: distortion in the movement of the valve producing the modulation of the airflow must be as weak as possible. In practice this distortion is likely to be due to aerodynamic effects or to mechanical friction, as illustrated in Figure 3. The reduction of friction between the sliding plate and the stator is a common problem, that can be solved by coating the valve faces with glass-fitted PTFE, by using a hydrocarbon lubricant or by using an aerostatic thrust bearing, as suggested by Glendinning *et al*. [11]. Another possibility is to increase the clearance between the sliding plate and the stator, but this solution leads to an increase in the leakage flow in the system, hence, to a decrease in the efficiency of the source. The problem of reduction of friction is discussed in more detail in Part II.

The influence of aerodynamic forces on the linearity of the sliding plate movement has been discussed by Rapier and Parkin [19] and Glendinning *et al*.

[11]. When the slots are aligned, the flow through the aperture results in a pressure difference between either side of the sliding plate, and the resulting net force acts to close the valve. The magnitude of this force will vary with valve opening position, and hence the non-linearity. This non-linear behaviour could, in principle, be compensated for by using electrical and/or mechanical means.

3.2. LINEARITY OF THE SUBSONIC ELECTROPNEUMATIC TRANSDUCER

Three potential causes of non-linearities can be immediately identified for the subsonic electropneumatic transducer.

(i) The movement of the valve that modulates the air flow across the source is likely to be non-linear. Upon assuming that the electrodynamic shaker is linear, the degree of linearity of the slider movement directly depends, as in the sonic case, on friction in the system and on aerodynamic forces. Due to the smaller value of plenum pressure, these problems are less important than in the sonic case: the pressure loading on the sliding plate is reduced, thus reducing friction. Moreover, the flow across the aperture is reduced, thus reducing the pressure difference between either side of the sliding plate and the net force on this plate. One can conclude that if the possibility exists of reducing mechanical non-linearities in the sonic case, it also exists in the subsonic case.

(ii) The discharge coefficient C_d is likely to vary with the throat area. The relationship between the discharge coefficient C_d and the throat area $A_1(t)$ is, theoretically, difficult to derive. In a seminal paper on acoustic sirens, Allen and Watters [20] experimentally demonstrated that this discharge coefficient can be assumed to be constant throughout the cycle, demonstrating the validity of the quasi-static approximation. This problem is reconsidered in the experimental analysis of the subsonic compressed-air source in Part II.

(iii) The physical mechanism of sound production is non-linear. The fundamental equation of subsonic sources (11) shows that the volume velocity produced by the source and hence the acoustic pressure at its output depend non-linearly on the throat area $A_1(t)$. The typical waveform at the output of a pneumatic source is plotted in Figure 4, where the are of the orifice is varied sinusoidally.

The sharp dips in the waveform can be explained by a phenomenon of equalization. During a large proportion of the movement of the valve controlling

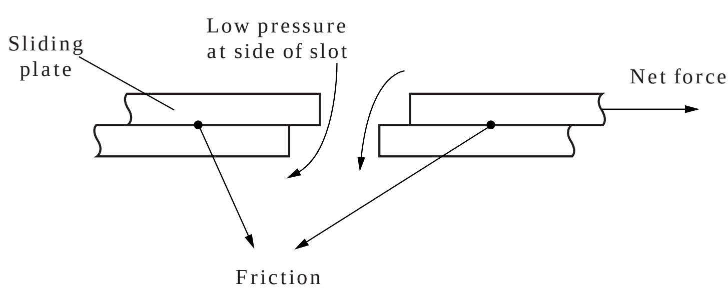

Figure 3. Illustration of the potential causes of a non-linear movement of the sliding plate in an electropneumatic transducer.

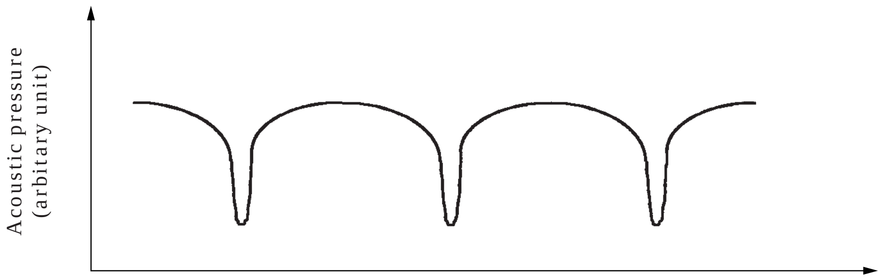

Figure 4. Typical waveshape of acoustic pressure at the output of a subsonic source for a sinusoidal modulation of the throat area.

the airflow, the valve opening is large enough so that the pressure at the source output is of the same order as the pressure in the plenum chamber since $R_i < R_l$. Hence, there is little pressure drop across the orifice. When the valve is closing, the flow across it is restricted ($R_i > R_l$), producing the dip of pressure seen in Figure 4, whose width is small compared to the period of the signal.

One way of reducing this non-linear behaviour is to connect the source to an output section of very low acoustic impedance. Considering the equivalent acoustics of the subsonic source (Figure 2), and neglecting the time varying inductance at the throat, one can write the relationship between $p_2(t)$ and p_{pl} as

$$p_2(t) = p_{pl}\left(\frac{R_l}{R_l + R(t)}\right), \tag{14}$$

where the load impedance has been assumed purely real and frequency independent ($Z_l = R_l$). The smaller the acoustic resistance of the duct, the smaller the acoustic pressure variations at the source output. In the case in which R_l is always small compared with $R(t)$, $p_2(t)$ will thus be small compared with P_{pl} and the equation for the alternating volume velocity of the source (equation (11)) can be rewritten, at least in a first approximation, as

$$Q_2(t) \cong A_1(t)\sqrt{C_d p_{pl}/\rho}, \tag{15}$$

so that the output is again proportional to alternating throat area. In respect to the computation of the pneumatic efficiency of the source, it is clear that this solution suffers from a major drawback in that it reduces the efficiency: if the system is to have a large efficiency, the load resistance R_l must be large compared to the resistance of the throat opening $R(t)$, because the acoustic power derived by the source is the power dissipated in R_l.

Another way of reducing the non-linear behaviour of the source is to predistort the movement of the valve in such a way that the desired output waveform is achieved. This method was used to great effect by Allen and Watters [20] in the design of an acoustic siren. Linearization of a physical system is generally a complex task that requires the identification of both the linear and non-linear parts of the transfer function of the system as well as their inversion. In the case of the subsonic compressed air source, however, we have the significant advantage that the analytic form of the non-linearity is known. The whole phase of identification can therefore be by-passed and the inversion of the system characteristic can be performed analytically. If $p_{2\,required}(t)$ is the required acoustic pressure at the source output, then upon using equation (11) the throat area is given by

$$A_1(t) = Q_2(t)\sqrt{\frac{\rho}{C_d(p_{pl} - p_{2\,requried}(t))}}\,. \tag{16}$$

Equation (16) does not allow a direct computation of $A_1(t)$ because volume velocity waveform $Q_2(t)$ is unknown. This waveform is however linked to that of the acoustic pressure $p_{2\,required}(t)$ by

$$Q_2(t) = \int_0^\infty Y_a(\tau)p_{2\,required}(t-\tau)\,\mathrm{d}\tau, \tag{17}$$

where $Y_a(t) = F^{-1}[Y_a(\mathrm{j}\omega)]$ is the inverse Fourier transform of the acoustic input admittance of the acoustic load to which the source is connected. An expression of $Y_a(t)$ is however difficult to derive theoretically and equations (16) and (17) cannot in general be solved analytically.

If the acoustic pressure at the source output is requested to be purely sinusoidal at frequency f, the situation is, however, somewhat different. In this case one can write

$$p_{2\,required}(t) = P\sin(2\pi ft) \quad \text{and} \quad Q_2(t) = \alpha p_{2\,required}(t-\theta), \tag{18, 19}$$

where α and θ are the amplitude and phase of the acoustic load admittance at the frequency f. The combination of equations (16), (18) and (19) leads to

$$A_1(t) = K\sin\{2\pi f(t-\theta\}\sqrt{\frac{1}{P_{pl} - \{p\sin(2\pi ft)\}}}, \tag{20}$$

where $K = \alpha P\sqrt{\rho/C_d}$ is constant throughout the cycle. Equation (20) gives the throat area versus time for producing a sinusoidal acoustic pressure at frequency f. Note that when solving this equation one must account for two constraints,

$$A_1(t) \geqslant 0 \quad \text{and} \quad A_1(t) \leqslant A_{1\,\max}, \tag{21}$$

where $A_{1\,\max}$ is the area of the valve when it is fully open. The computation of $A_1(t)$ requires only the knowledge of the delay θ. If the acoustic impedance of the duct is purely resistive, then $\theta = 0$. For high efficiency, the amplitude of $p_{2\,required}(t)$ must be close to the plenum pressure. Equation (20) however shows that if $P = p_{pl}$, then $A_1(t)$ goes to infinity: it is therefore necessary to make a compromise between efficiency and valve size. A simulation was carried out to investigate the influence of the value of P on the shape of $A_1(t)$. The result is illustrated in Figure 5: to avoid equalization, the value must remain almost closed during a large part of the cycle. The throat area versus time becomes sharper as the amplitude of the acoustic pressure tends to the plenum pressure.

The influence of the time delay θ on the shape of the throat area was also investigated. For convenience, equation (20) can be rewritten as

$$A_1(t) = K \sin\{\omega t - \varphi\} \sqrt{\frac{1}{P_{pl} - \{p \sin(\omega t)\}}}, \qquad (22)$$

with $\varphi = 2\pi f\theta$. The throat area versus time was computed for various values of φ. The result is illustrated in Figure 6. In each of the Figures 6(a–d), the three representations of the throat area versus time are almost merged, since a small variation in the value of φ has a negligible influence on the throat area versus time. This result shows that the efficiency of the predistortion should not be affected by small errors in the determination of φ and that this parameter thus need not be measured very accurately in an experimental system. Experiments have been carried out to linearize a subsonic compressed-air source and the results are presented in Part II.

The efficient production of controlled waveforms by using subsonic electropneumatic transducers into an arbitrary load impedance thus requires a suitable predistortion. When the subsonic source is used as a secondary actuator in an active noise control system, however, this conclusion must be reconsidered. Broadly speaking, the aim of any active noise control system is to minimize the acoustic pressure at a location close to the secondary source output. The acoustic pressure experienced by the source, p_2 in equation (10), is thus significantly reduced when active control is being applied, in which case Q_2 becomes an almost linear function of A_1 in equation (10). This conclusion is important because it suggests

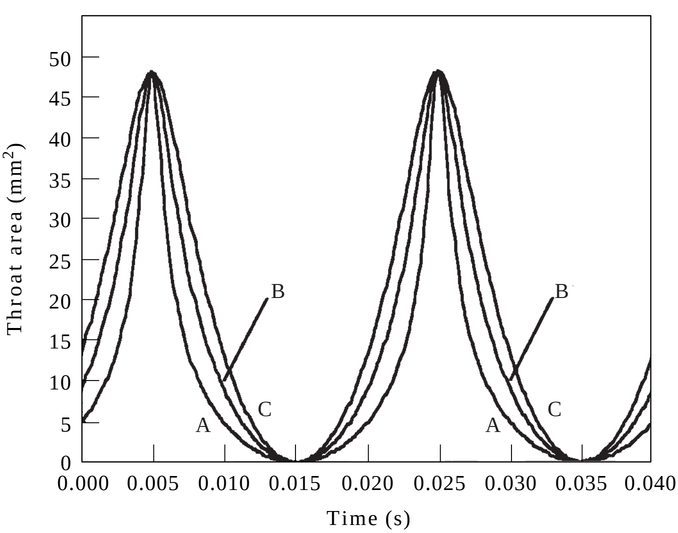

Figure 5. Throat area versus time required to produce a sinusoidal output acoustic pressure for a subsonic source, in the following conditions: $f = 50$ Hz, $p_{pl} = p_{atm} + 200$ Pa, $\theta = 0$ s, amplitude P of the output acoustic pressure: curve A, 195 Pa; curve B, 180 Pa; curve C, 150 Pa.

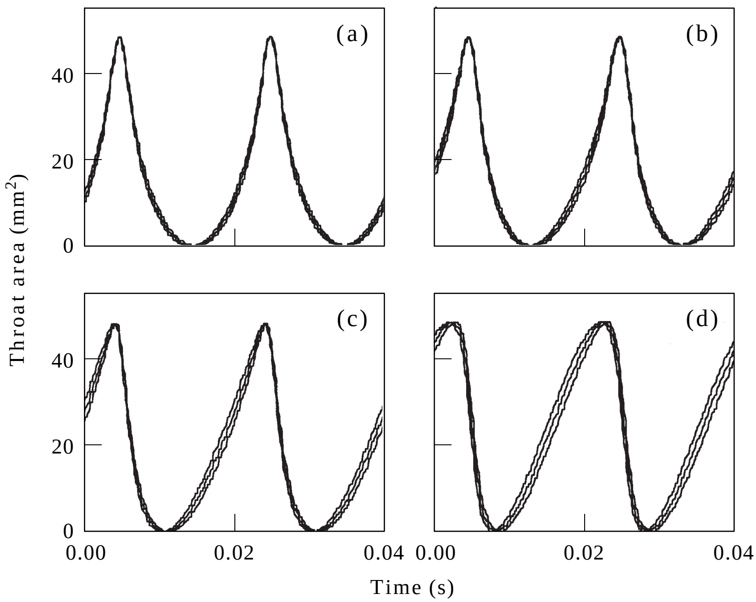

Figure 6. Throat area versus time for producing a sine output acoustic pressure. Assumed conditions: $f = 50$ Hz, $p_{pl} = p_{atm} + 200$ Pa, amplitude A of the output acoustic pressure = 180 Pa. In each graph, the throat area is plotted for three different values of angle φ. (a) $\varphi = 0°$, 5° and 10°; (b) $\varphi = 25°$, 30° and 35°; (c) $\varphi = 50°$, 55° and 60°; (d) $\varphi = 80°$, 85° and 90°.

that provided a robust controller is used during the transient phase, i.e., a controller whose convergence is unaffected by the initial non-linear behaviour of the secondary source, the mechanism of non-linear sound emission is subsonic electropneumatic transducers disappears in the steady state. In this case, the degree of linearity of the source only depends, as in the sonic case, on the degree of linearity of the valve movement.

4. EFFICIENCY OF ELECTROPNEUMATIC TRANSDUCERS

4.1. DEFINITION

After linearity, efficiency is probably the second major parameter to quantify the performance of sources used as secondary actuators in active noise control systems. Compressed air transducers are characterized by the pneumatic efficiency η_p (sometimes referred to as the efficiency of modulation) that quantifies the extent to which the steady plenum pressure is transformed in an acoustic flow, and this is defined as

$$\eta_p = W_a / W_p \text{ (} \times \text{ 100 if this value is to be given in \%)} \tag{23}$$

where W_a is the acoustic power at the source output and W_p is the pneumatic power required to feed the plenum chamber with compressed air. Upon assuming for convenience that the source is connected to a purely resistive acoustic load R_l, the acoustic power generated is given by

$$W_a = Q_{2\,rms}^2 R_l, \tag{24}$$

where $Q_{2\,rms}$ is the root mean square value of the acoustic volume velocity at the source output. The pneumatic power W_p is assumed to be equal to the power required to isentropically compress the air which flows through the transducer from atmospheric pressure to plenum pressure. It is given by [9]

$$W_p = \bar{\rho}_1 \bar{u}_1 \overline{A}_1 w_{comp}, \tag{25}$$

where w_{comp} is the work done per unit mass to compress air from the atmospheric pressure p_{atm} to the plenum pressure, given by [21]

$$w_{comp} = \frac{\gamma}{\gamma - 1} \frac{p_{atm}}{\rho_{atm}} \left[\left(\frac{p_{pl}}{p_{atm}} \right)^{\gamma - 1/\gamma} - 1 \right], \tag{26}$$

where γ is the ratio of specific heats (1·4 for air). Using equations (23)–(26) yields the pneumatic efficiency of electropneumatic transducers as (if the source is connected to a purely resistive acoustic load)

$$\eta_p = \frac{Q^2_{2\,rms} R_l}{\bar{\rho}_1 \bar{u}_1 \overline{A}_1 \dfrac{\gamma}{\gamma - 1} \dfrac{p_{atm}}{\rho_{atm}} \left[\left(\dfrac{p_{pl}}{p_{atm}} \right)^{\gamma - 1/\gamma} - 1 \right]}. \tag{27}$$

4.2. EFFICIENCY AND ACTIVE NOISE CONTROL—THE EFFICIENCY PARAMETER

The pneumatic efficiency of electropneumatic transducers depends on the acoustical load to which the source is connected. When the transducer is used as a secondary source in active noise control, the acoustic load impedance and hence, the acoustic power derived by the source, is close to zero so that the pneumatice efficiency, as defined in equation (23), is meaningless. In this case, the source efficiency can be characterized by using the volume velocity efficiency parameter ε, defined as

$$\varepsilon = |Q_{rms}|^2 / W_p \;\; ((\mathrm{m}^3\,\mathrm{s}^{-1})^2/\mathrm{W}) \tag{28}$$

By using equations (23), (24) and (28) the link between ε and η_p can be expressed as

$$\varepsilon = \eta_p / R_l. \tag{29}$$

In active control, R_l goes to zero and equation (29) is replaced by

$$\varepsilon = \lim_{Z_l \to 0} (\eta_p / \mathrm{Re}\,[Z_l]). \tag{30}$$

The efficiency parameter is independent of the acoustic load impedance.

The efficiency parameter suffers from a major drawback: unlike the pneumatic efficiency it is a dimensional parameter. The efficiency parameter is hence more difficult to interpret than the pneumatic efficiency. Equation (30) shows however that the efficiency parameter is proportional to the pneumatic efficiency and so the more efficient the source when connected to a given acoustical load, the more efficient this source will be when used as a secondary source in an active noise control system, if the load impedance is small enough that ε has reached a limiting value in equation (30). A comparison of the efficiencies of various transducers used as secondary sources in active noise control systems is therefore possible on the basis of the pneumatic efficiency and we will therefore continue to characterize pneumatic sources used in active noise control systems by using the pneumatic efficiency.

4.3. THEORETICAL MAXIMUM EFFICIENCY

The maximum theoretical pneumatic efficiency of subsonic electropneumatic transducers can be estimated by using the simplified equivalent circuit of the device given in Figure 2. Assuming that the output of the source is sinusoidal and that the throat area is modulated from zero to a maximum value, one can split the volume flow across the source $Q(t)$ into its alternating and steady components:

$$Q(t) = Q_{d.c.} + Q_{a.c.} \sin(\omega t), \quad \text{with} \quad Q_{d.c.} = Q_{a.c.}. \tag{31}$$

If the acoustic load is purely resistive ($Z_l = R_l$), the acoustic output power W_a is equal to

$$W_a = R_l (Q_{a.c.}/\sqrt{2})^2. \tag{32}$$

The pneumatic power injected into the system W_p is equal to the power dissipated when the valve in its position, without modulation. For maximum acoustic output the flow resistance $R_i(t)$ must be small compared to the load resistance R_l. If the source is optimized for maximum acoustic output, the power dissipated when the valve is placed in its mean position is hence mainly dissipated in the load. On can therefore write

$$W_p \cong R_l Q_{d.c.}^2, \tag{33}$$

where it has been assumed that the acoustic load resistance at dc is the same as that at ω. Combining equations (23), (32) and (33) yields the maximum theoretical pneumatic efficiency $\eta_{p,\,max}$ as

$$\eta_{p,\,max} = 100\,\frac{R_l (Q_{a.c.}/\sqrt{2})^2}{R_l Q_{a.c.}^2} = 50\%. \tag{34}$$

This last result shows that the electropneumatic transducer has a behaviour similar to that of class A amplifiers: the operating point and the input signal are such that the fluid in the output circuit flows at all times. It is important to recall that the above result assumes that the output of the source is sinusoidal. For example, if the output of the source is a square wave, equation (32) becomes

$$W_a = R_l Q_{a.c.}^2 \tag{35}$$

and the computation leads to a maximum theoretical pneumatic efficiency equal to 100%.

4.4. PNEUMATIC EFFICIENCY OF THE SONIC ELECTROPNEUMATIC TRANSDUCER

Upon assuming that the source is connected to a purely resistive acoustic load R_l and a sinusoidal modulation of the valve, the acoustic power at the source output becomes:

$$W_a = (\delta p_2)^2 / 2R_l. \tag{36}$$

According to equation (4),

$$\delta p_2 = \frac{A_2 \beta \delta A_1}{\bar{p}_2 \left(\dfrac{1}{R_l} + \dfrac{1}{R_i} \right)} = \frac{A_2 \beta \delta A_1 R_l R_i}{\bar{p}_2 (R_l + R_i)}, \tag{37}$$

where $R_i = \bar{p}_2 / (A_2 \times \bar{u}_2)$ is the internal resistance of the source. The pneumatic power W_p is computed by using equations (25) and (26). The computation of the mean values $\bar{u}_1$ and $\bar{\rho}_1$ can be directly performed from the results of Chapman and Glendinning [10]:

$$\bar{\rho}_1 = \frac{\gamma p_{pl}}{c_0^2} \left(\frac{5}{6} \right)^{5/2}, \qquad \bar{u}_1 = c_0 \left(\frac{5}{6} \right)^{1/2} \tag{38, 39}$$

Using equations (27), (38) and (39) hence yields the pneumatic efficiency as

$$\eta_{p,\,sonic} = \left(\frac{5}{6} \right)^3 \frac{p_{pl} c_0^3 (\delta A_1)^2 R_l R_i^2}{2 \overline{A}_1 \bar{p}_2^2 (R_l + R_i)^2 \dfrac{\gamma^2}{\gamma - 1} \times \dfrac{p_{atm}}{\rho_{atm}} \left[\left(\dfrac{p_{pl}}{p_{atm}} \right)^{\gamma - 1/\gamma} - 1 \right]}. \tag{40}$$

The pneumatic efficiency is proportional to the square of the swept area. The pneumatic efficiency of a sonic source was estimated by using the following data, based on Chapman and Glendinning [10]: $\overline{A}_1 = 2{\cdot}4 \times 10^{-5}$ m^2, $\delta A_1 = 2{\cdot}4 \times 10^{-5}$ m^2 (full modulation of the valve), $c_0 = 343$ m/s, $\rho_{atm} = 1{\cdot}2$ kg/m^3, $p_{atm} = 1{\cdot}02 \times 10^{-5}$ Pa, $p_2 = 1{\cdot}02 \times 10$ Pa, $\gamma = 1{\cdot}4$, p_{pl} ranging from $p_{atm} = 5 \times 10^4$ Pa to $p_{atm} = 3 \times 10^5$ Pa. Three different values where chosen for the acoustic load resistance: $R_l = 4{\cdot}88 \times 10^5$ N S/m^5, $R_l = 2{\cdot}44 \times 10^6$ N S/m^5 and $R_l = 4{\cdot}63 \times 10^6$ N S/m^5 (the reason why these three values are chosen relates to Part II). The source pneumatic efficiency for the conditions described above is plotted in Figure 7.

The minimum value of plenum pressure for sonic flow at the throat can be computed by considering the condition for shocked flow at station 2, which can be written as [13]

$$\bar{p}_2 / p_{pl} \cong p_{atm} / p_{pl} < 0{\cdot}528. \tag{41}$$

For shocked conditions at station 1, p_{pl} must thus be larger than $1{\cdot}90\ p_{atm}$ (if viscous effects are neglected). Values of efficiencies for plenum pressures below this level are not shown in Figure 7. This figure shows that for acoustic loads considered, the efficiency ranges from about 3 to 15%. The pneumatic efficiency increases for decreasing plenum pressures down to limit set by equation (41), but decrease as the load impedance is reduced.

4.5. PNEUMATIC EFFICIENCY OF THE SUBSONIC ELECTROPNEUMATIC TRANSDUCER

The various parameters in equation (27) are computed as follows. First

$$\bar{\rho}_1 \cong \rho_{atm} \quad \text{(fluid is assumed to be incompressible)} \tag{42}$$

The mean velocity $\bar{u}_1$ is computed by using the flow equation (11) and the mass conservation equation:

$$\bar{u}_1 \overline{A}_1 = \bar{u}_2 A_2. \tag{43}$$

Combining equation (27), (42) and (43), and assuming the source is connected to a purely resistive acoustic load, one finds that the pneumatic efficiency $\eta_{p,\,subsonic}$ is given by

$$\eta_{p,\,subsonic} = \frac{(\delta p_{2\,rms})^2/R_l}{\sqrt{\dfrac{C_d(p_{pl} - \bar{p}_2)}{\rho}}\,\overline{A}_1 \times \dfrac{\gamma}{\gamma - 1} \times p_{atm}\left[\left(\dfrac{p_{pl}}{p_{atm}}\right)^{\gamma - 1/\gamma} - 1\right]}. \tag{44}$$

The pneumatic efficiency obviously depends on the root mean square value of the acoustic pressure at the source output and hence on the waveshape of the acoustic signal derived. To allow a fair comparison between the pneumatic efficiency in sonic and subsonic electropneumatic transducers we computed the pneumatic

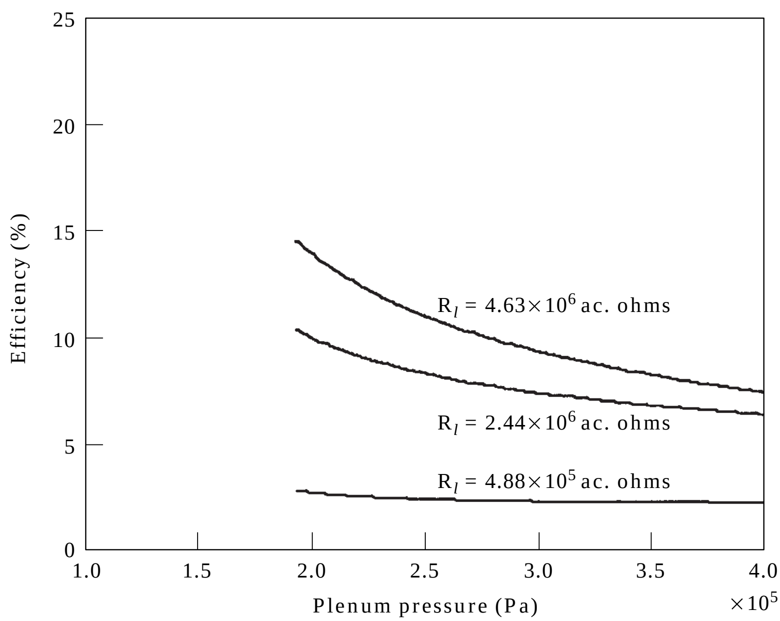

Figure 7. Pneumatic efficiency of the sonic source versus the plenum pressure for three acoustic load resistance. For each curve the solid line corresponds to sonic conditions, whereas the dashed line corresponds to a Mach number at the throat smaller than 1 (1 ac. ohm = 1 N s/m^5).

efficiency in the subsonic case assuming a sinusoidal output acoustic pressure. In other words the movement of the valve that controls the flow is assumed to be distorted in such a way that a sinsusoidal output is produced, as explained in section 3. The various parameters in equation (44) are then computed as follows, for fixed values of R_l and p_{pl}. First

$$\delta p_{2\,rms} = \delta p_{2\,peak}/\sqrt{2} = (\bar{p}_2 - p_{atm})/\sqrt{2}. \tag{45}$$

The mean value of the throat area, $\overline{A}_1$, is computed with reference to the waveform shown in Figure 5. Finally, the mean pressure $\bar{p}_2$ is computed by using the quasi-static assumption:

$$\text{if} \quad A_1(t) = \overline{A}_1 + \delta A_1(t) = 0, \quad \text{then} \quad p_2(t) = p_{atm}. \tag{46}$$

The combination of condition (46) with equations (12) and (43) leads, after a little algebra, to

$$\frac{\overline{A}_1 \sqrt{\dfrac{C_d}{\rho}(p_{pl} - \bar{p}_2)} + \dfrac{p_{atm} - \bar{p}_2}{R_l}}{\sqrt{\dfrac{C_d}{\rho}(p_{pl} - p_{atm})}} = 0. \tag{47}$$

In normal conditions the denominator of equation (47) is not equal to zero, and so equation (47) reduces to

$$\overline{A}_1 \sqrt{\frac{C_d}{\rho}(p_{pl} - \bar{p}_2)} + \frac{p_{atm} - \bar{p}_2}{R_l} = 0, \tag{48}$$

or

$$\frac{\bar{p}_2^2}{R_1^2} + \bar{p}_2\left(\frac{C_d}{\rho}\overline{A}_1^2 - \frac{C_d p_{atm}}{R_1^2}\right) + \frac{p_{atm}^2}{R_1^2} - \overline{A}_1^2 \frac{C_d p_{pl}}{\rho} = 0. \tag{49}$$

This equation in $\bar{p}_2$ has two solutions. The one we are interested in is such that: $\bar{p}_2 > p_{atm}$. The pneumatic efficiency (equation 44) can be calculated by using

$$\eta_{p,\,subsonic} = \frac{(\bar{p}_2 - p_{atm})^2}{2R_l\overline{A}_1 \times \dfrac{\gamma}{\gamma - 1} \times p_{atm}\sqrt{\dfrac{C_d(p_{pl} - \bar{p}_2)}{\rho}}\left[\left(\dfrac{p_{pl}}{p_{atm}}\right)^{\gamma - 1/\gamma} - 1\right]}. \tag{50}$$

with $\bar{p}_2$ calculated from equation (49). This expression was evaluated numerically, for the same data as used in the sonic case. This time, however, the source is required to work in the subsonic region. Since the fluid is assumed to be incompressible, the Mach number at station 1, M_1, must be smaller than say, 0·6. The plenum pressure leading to $M_1 = 0{\cdot}6$ is computed by using equations (11) and

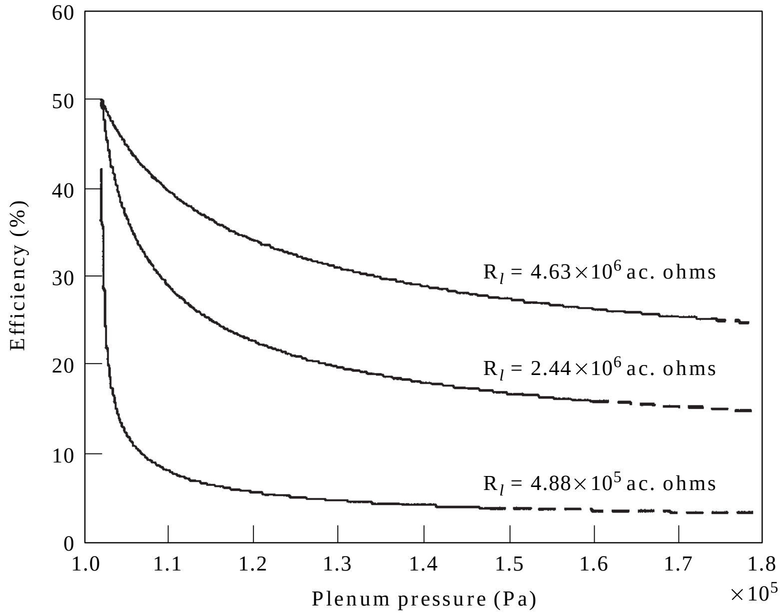

Figure 8. Pneumatic efficiency of the subsonic source versus the plenum pressure for three acoustic load resistance. For each curve the solid line corresponds to subsonic conditions, whereas the dashed line corresponds to a Mach number at the throat larger than 0·6.

(48), as well as the definition of the Mach number. After a little algebra the plenum pressure for $M_1 = 0{\cdot}6$ can be written as

$$p_{pl} = p_{atm} + 0{\cdot}6c_0\left(R_l\overline{A}_1 + \frac{0{\cdot}6}{C_d}\rho c_0\right). \tag{51}$$

The calculated pneumatic efficiency is plotted in Figure 8, for plenum pressures ranging from $p_{atm} + 300$ Pa to $1{\cdot}6 \times 10^5$ Pa and assuming $C_d = 2$ and $\overline{A}_1 = 2{\cdot}4 \times 10^{-5}$ m^2. This time, the pneumatic efficiency ranges from, say 8 to 50%, the theoretical maximum value.

5. ELECTROPNEUMATIC TRANSDUCERS VERSUS ELECTRODYNAMIC LOUDSPEAKERS: A COMPARISON

To assess the advantage of using a sonic or a subsonic electropneumatic transducer as a sound generator or as a secondary source in active noise control, we compared the performance of these sources so that of a typical electrodynamic loudspeaker, whose characteristics are assumed to be as follows: diameter d of the diaphragm 20·0 cm; maximum diaphragm linear excursion 10 mm p-p, which requires an electrical input power of 200 W. The maximum volume velocity Q_{max} that can be derived by this loudspeaker is [22]:

$$Q_{max} = \omega\delta\xi\, \pi d^2/4 = 0{\cdot}05 \text{ m}^3/\text{s}, \tag{52}$$

where $\delta\xi$ is the maximum displacement of the diaphragm.

The comparison is carried out in two cases. In the first one, the sources are assumed to work as sound generators and to be connected to a duct with a perfect anechoic termination. The diameter D of the duct is chosen to be equal to the diameter of the diaphragm of the loudspeaker, i.e., $D = d = 20$ cm. The input acoustic impedance of this duct is purely resistive and given by:

$$R_l = \rho c/S = 13000 \text{ N s/m}^5, \tag{53}$$

To guarantee a fair comparison, it is assumed that: (1) the sources are working at the same frequency ($f = 50$ Hz) and all generate a sinusoidal acoustic pressure (in other words the subsonic source is connected to a predistortion processor), and (2) the sources produce the same volume velocity, which is equal to the maximum volume velocity that can be emitted by the electrodynamic loudspeaker (0·05 m^3/s). The characteristics of interest in this first comparison are the efficiency and the linearity of the sources. The "area ratio" defined as the ratio of the area of the valve of the electropneumatic transducer when fully open to that of the diaphragm of the loudspeaker, was also estimated. In the second comparison, the sources are assumed to act as secondary sources in an active noise control system. It is assumed that the sources experience a purely resistive acoustic load resistance $R_{l\text{-}ANC}$ equal to 1 000 N s/m^5 (about a tenth of the previous case) and we are required to generate the same volume velocity as in the first case. The characteristics of interest for this second comparison are the efficiency parameter, the linearity and the capacity of the source to operate in extreme environments.

The electroacoustic efficiency $\eta_{e,a}$ of the electrodynamic loudspeaker is computed by using

$$\eta_{e,a} = 100\ W_a/W_e = 100\ (\delta p_{2,rms})^2/R_l W_e, \tag{54}$$

where W_e is the electrical power feeding the loudspeaker and $\delta p_{2,rms}$ is the rms value of sinusoidal acoustic pressure fluctuations at the source output:

$$\delta p_{2,rms} = \delta p_{2,\,max}/\sqrt{2} = R_l Q_{max}/\sqrt{2}. \tag{55}$$

The combination of equations (54) and (55) leads to

$$\eta_{e,a} = 100\ R_l Q_{max}^2/2W_e = 8\%. \tag{56}$$

Using equations (30) and (56) yields the efficiency parameter as $\varepsilon = 6{\cdot}25 \times 10^{-6}$ $m^6/s^2/W$.

The sonic electropneumatic transducer considered for the comparison is assumed to work with a plenum pressure about three times larger than the atmospheric pressure: $p_{pl,\,sonic} = 3 \times 10^5$ Pa, which is a common value for conventional sonic sources [11]. The first step is to compute the fluctuation in the valve area δA_1 required to produce the specified volume velocity. Equation (5) is used to compute this value. If one notes that the source internal impedance R_i is in general very large compared to the duct input resistance [11], one can write

$$\delta A_1 = \frac{\delta Q_2 p_{atm}}{(5/6)^3 c_0 p_{pl}} = 0{\cdot}85 \times 10^{-4} \text{ m}^2. \tag{57}$$

For a full amplitude, sinusoidal, modulation of the valve the mean value of the throat area is $\overline{A}_1 = \delta A_1 = 0{\cdot}85 \times 10^{-4}\,\text{m}^2$. The efficiency of the source is computed by using equation (40), in which R_l is neglected compared to R_i. The computation leads to $\eta_{sonic} = 0{\cdot}35\%$. Finally, the efficiency parameter is computed by combining equations (30) and (40): $\varepsilon_{sonic} = 2{\cdot}7 \times 10^{-7}\,\text{m}^6/\text{s}^2/\text{W}$.

The characteristics of the subsonic electropneumatic transducer are computed by assumng that when fully open (area $A_{1\,max}$), the resistance of the throat opening is given by equation (8), where $A_1 = A_{1\,max}$ and $Q_2 = Q_{max}$. Upon assuming this resistance is equal to a half of R_1, $A_{1\,max}$ can be computed, upon assuming also that the discharge coefficient is equal to 2: $A_{1\,max} = 2{\cdot}15 \times 10^{-3}\,\text{m}^2$. The plenum pressure of the source when used as a sound generator is computed by equation (11):

$$P_{pl,\,sub} = p_{atm} + \delta p_{2,\,max} + \frac{\rho}{C_d}\frac{Q_{max}^2}{A_{1,\,max}^2}, \quad \text{where} \quad \delta p_{2,\,max} = R_l Q_{max}. \tag{58}$$

The pneumatic efficiency of the source is computed by using equation (50), where $\bar{p}_2 = p_{atm} + (\delta p_{2,\,\max}/2)$, $\overline{A}_1 \cong 0{\cdot}3\,A_{1\,\max}$ (the coefficient 0·3 is motivated by Figure 5). The pneumatic efficiency under these conditions is equal to 36%.

When the source is used in an active noise control system, the acoustic load resistance and hence $\delta p_{2\,max,\,ANC}$ decrease. The required plenum pressure also decreases: according to equation (58), it is in this case equal $p_{atm} + 325\,\text{Pa}$. The efficiency parameter is computed by combining equation (30) and (50):

$$\varepsilon_{subsonic} = \frac{(\bar{p}_2 - p_{atm})^2}{2R_{l,\,ANC}^2\overline{A}_1 \times \dfrac{\gamma}{\gamma - 1} \times p_{atm}\sqrt{\dfrac{C_d(p_{pl} - \bar{p}_2)}{\rho}}\left[\left(\dfrac{p_{pl}}{p_{atm}}\right)^{\gamma - 1/\gamma} - 1\right]}. \tag{59}$$

where $\bar{p}_2 = p_{atm} + (\delta p_{2\,max,\,ANC}/2)$, $\overline{A}_1 \cong 0{\cdot}5 A_{1\,max}$: the source is now reasonably linear, and a sinusoidal modulation of the valve produces a sinusoidal acoustic pressure, and hence the coefficient of 0·5 used above. One can thus compute in this case $\varepsilon_{subsonic} = 38 \times 10^{-6}\,\text{m}^6/\text{s}^2/\text{W}$. The results of the comparison are summarized in Table 1. Note that the frequency response of the sources is not considered in this comparison. This is because the frequency responses mainly depend on issues such as the volume of the loudspeaker enclosure, the frequency response of the electrodynamic loudspeaker and shaker, or the volume of the plenum chamber. If the sources are all properly designed they should have a similar frequency response.

6. CONCLUDING REMARKS

The properties of sonic and subsonic electropneumatic sources have been considered. It has been shown that the sonic source has a fundamentally linear operation and a high internal acoustic impedance, but can be very inefficient if the acoustic load impedance is not also high. In active sound control applications the effective load impedance becomes very small when the system is operating and

TABLE 1

Comparison of the characteristics of electropneumatic transducers with those of an electrodynamic loudspeaker; (1) *efficiency corresponds to the pneumatic efficiency for electropneumatic transducers and to the electro-acoustic efficiency for the loudspeaker*; (2) *non-linearities due to mechanical or electrical sources are assumed to be negligible*

Characteristic	Sonic source	Subsonic source	Electrodynamic loudspeaker
Volume velocity		$0{\cdot}05\ m^3/s$	
Sources are assumed to act as sound generators			
Plenum pressure	$p_{atm} + 2{\cdot}0 \times 10^5$ Pa	$p_{atm} + 975$ Pa	–
Ratio of area	0·005	0·07	1
Efficiency (1)	0·35%	36%	8%
Linearity of sound generation process (2)	Good	Poor, unless predistorted	Excellent
Sources are assumed to act as secondary sources in active control			
Plenum pressure	$p_{atm} + 2{\cdot}0 \times 10^5$ Pa	$p_{atm} + 325$ Pa	–
Efficiency parameter	$2{\cdot}7 \times 10^{-7}\ m^6/s^2/W$	$3{\cdot}8 \times 10^{-5}\ m^6/s^2/W$	$6{\cdot}25 \times 10^{-6}\ m^6/s^2/W$
Linearity of sound emission process (2)	Good	Good, after the transient phase	Excellent
Resistance to extreme environments	Good	Good	Poor
characteristic of controller	No particular requirement	Must be robust	No particular requirement

sinusoidal output waveform when used as a normal acoustic source, and the efficiency is then much larger than the sonic source. The pneumatic efficiency of subsonic electropneumatic sources driving sinusoidal signals can reach 50%. The linearity of the subsonic source improves as the acoustic load impedance becomes smaller, and hence is particularly well suited to active control applications. As in the sonic case, the pneumatic efficiency increases as the plenum pressure decreases but for the subsonic source a minimum plenum pressure is not required to maintain sonic conditions at the throat. Broadly speaking, for maximum efficiency the plenum pressure must be close to the maximum pressure required at the source output if much of the energy is not to be dissipated by turbulence in the throat. Hence, the advantage of the subsonic source over the sonic one. In an example calculation the efficiencies of the sonic and subsonic compressed air sources we calculated to be 0·35 and 36% respectively, compared with a typical efficiency for an electrodynamic loudspeaker of about 8%. The use of electropneumatic transducers can also be attractive when the source is required to be compact: Table 1 shows that for production of a given volume velocity, the area of modulation for an electropneumatic source is almost negligible compared to the area of the diaphragm of a loudspeaker. Both subsonic and sonic sources are candidates for acting as secondary sources in active noise control systems. These sources offer a useful alternative to common electrodynamic loudspeakers, especially when the secondary source has to act in an extreme environment, which may be hot, humid and/or corrosive. Because the subsonic electropneumatic source is non-linear during the period where the active control system is adapting, and the pressure in front of the device has not been completely cancelled, the use of the subsonic source in active noise control requires a robust controller. The design of such controllers is the subject of current work.

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