DEVELOPMENT OF A WIDE BAND, TEN KILOWATT ACOUSTIC NOISE SOURCE

By: W.R. Miller, Ling Electronics

Mr. Miller received his Bachelor of Science Degree in Electrical Engineering from California State Polytechnic College, San Luis Obispo, California, in 1961. After two years in the magnet and power supply design field he joined Ling Electronics. His current duties include design and development responsibility for various shakers and acoustic transducers. He is a member of the Orange County Chapter of the IES.

INTRODUCTION

The technique of modulating an air stream to produce controlled sound is not new. The first practical device was a phonograph-stylus-operated device which appeared in France shortly before 1900. Development of a wide frequency range, high power device did not come about until 1955 when Stanford Research Institute developed a small generator, similar to present designs[1*]. Then in 1959 Altec Lansing developed the 2000 watt generator which is still in wide use today. A 4000 watt, improved version of this early generator is presently being manufactured by Ling Electronics.

In addition to the electropneumatic transducer, there are other modulated air stream devices being used for generation of high intensity acoustic power. The Northrop Noraircoustic Transducer[2], discrete frequency sirens[3] and "random" sirens[4] are examples.

State-of-the-Art

In mid-1964 the commercially available electropneumatic transducers were limited to two basic units. The small unit had a 2000 watt acoustic output and could be modulated fully from 0 to about 500 Hz. The large unit, an adaptation of an existing shaker design, had a 20 000 watt acoustic output and could be modulated fully from 0 to about 350 Hz. Both units required compressed air at approximately 40 psig and realized pneumoacoustic conversion efficiencies of 10%.

New Requirement

At that time a specific acoustic test requirement arose, calling for an electropneumatic transducer which would generate 10 kilowatts of acoustic power and have a flat or correctable output pressure response over the frequency range of 10 to 5000 Hz. Also desired was the ability to produce discrete frequency energy at the full 10 kilowatt level, over the frequency range of 25 to 1000 Hz.

Ling Electronics undertook the task of designing, building, and testing an electropneumatic transducer which fulfilled the requirements of the above-mentioned specification. The description of the design and development program of this wide band, 10 kilowatt electropneumatic transducer is presented below.

DESIGN GOALS

The object of this project was to design and build an electropneumatic transducer which would meet or exceed the tentative specification set down below:

Acoustic Output Power.	10 000 watts
Pneumatic Power. . .	27.6 N/cm^2 (40 psig)
	0.855 $\frac{Kg}{s}$ (1500 scfm)
Electrical Drive Power.	500 VA max.
Frequency Range. . .	Max. Power 25-1000 Hz
Weight	890N (200 lbs)
Pneumoacoustic Efficiency	10% minimum

To realize these goals it was necessary to advance the state-of-the-art in electropneumatic transducer design, especially in the area of wide frequency range operation. Existing designs were limited in frequency range by: 1) excessive mass of the moving element, 2) the velocity limit of the guidance system, or 3) inadequate drive coil cooling. The new design must allow circumvention of these limits.

*Superscript numbers refer to the references cited in Appendix I.

DESCRIPTION OF THE DESIGN

Principle of Operation

The electropneumatic transducer is basically an electrodynamically actuated pneumatic sleeve valve. The relatively massive magnetic structure serves as the main structural assembly and also as a reaction mass for the driver. Figure 1 illustrates the valve-coil-magnetic structure assembly. The compressed air is contained and guided by a part of the main structure. The unit may contain integral air filters to protect the valve members.

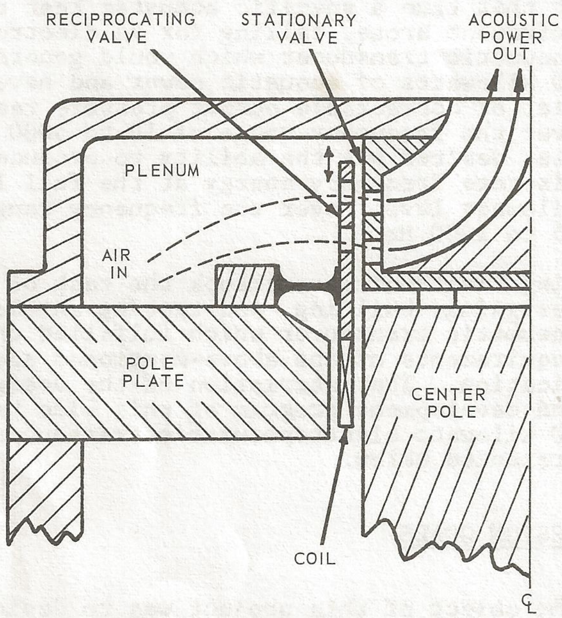

Fig. 1 Major Componets of Electropneumatic Transducer

As shown in Figure 1, the valve assembly is comprised of two members, a reciprocating valve and a stationary valve. In the quiescent position the slots of the stationary valve are overlapped by those of the reciprocating valve in such a manner as to present a half opened slot to the air stream.

"Full modulation" of the valve assembly is accomplished by allowing the reciprocating valve to fully open and then fully close the slot. This type of operation is analogous to the operation of a class A electronic amplifier, so it has been termed class A operation. Class B and C operations are possible, however, they produce an acoustic output which is similar to the full-wave rectification of the drive signal (double the input frequency).

The modulation of the air stream, produced by the valve action, produces pressure fluctuations upstream and downstream of the valve. These pressure fluctuations are propagated as acoustic pressure waves.

Due to the nonlinear nature of the propagation of high intensity sound waves (SPL's greater than 160 dB re. 2×10^{-5} N/m^2), the saw-tooth waveform is the stable waveform[5,6]. Figure 2 shows typical high intensity pressure waveforms as a function of propagation distance. This nonlinear effect aids in the generation of wide band random noise spectra. The high frequency components of the saw-tooth fill in the high frequency area where direct control of the electropneumatic transducer is impossible, due to the extremely high drive power requirements. Some requirements for high intensity sound energy require low distortion, discrete frequency operation at high sound pressure levels (SPL). This is just not possible in most cases. In fact, it just doesn't happen in an actual service environment, although there may still be an experimental need for low distortion sine wave energy.

Inductive Drive System

In order to produce high frequency acoustic power from an electropneumatic transducer it was necessary to develop a new valve design. The valve slot design incorporated 0.031 inch high slots (the 4 kilowatt unit has 0.060 inch high slots). Full modula-

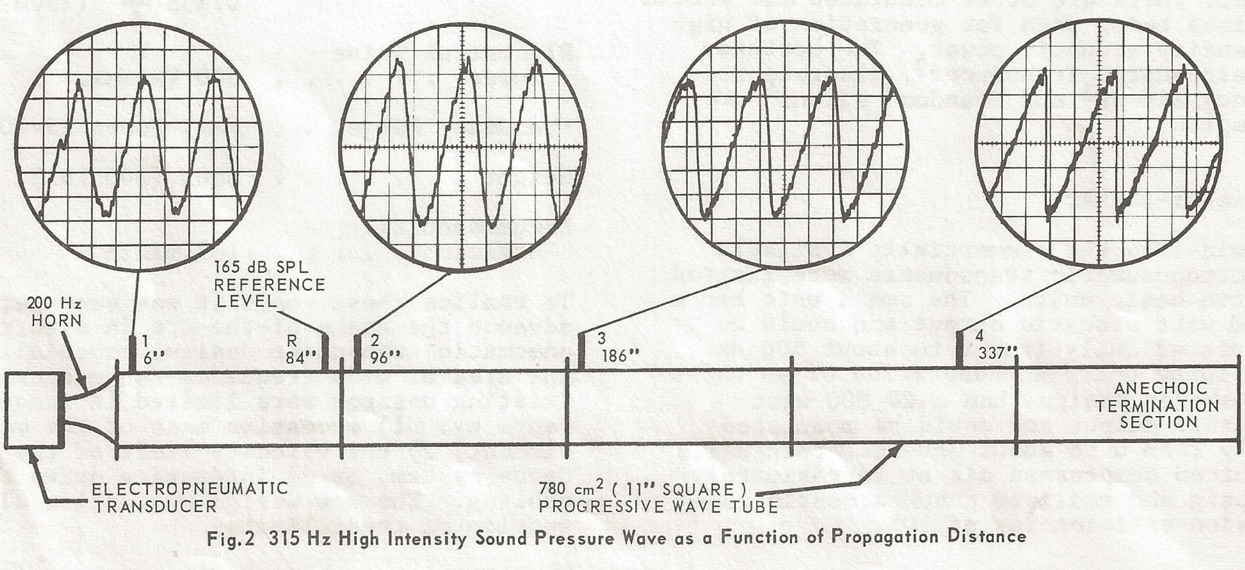

Fig. 2 315 Hz High Intensity Sound Pressure Wave as a Function of Propagation Distance tion at a frequency of 1000 Hz would require an acceleration level as follows:

$$g = 0.0511\ f^2 D,$$

where g is peak acceleration in gravity units (386 inches/second^2), f is frequency in hertz, and D is peak-to-peak displacement in inches.

Substituting 1000 Hz and 0.031 inch gives an acceleration of 1580 peak g's. This high acceleration level required that the reciprocating valve be very strong and rugged. Additionally, the electrical drive power for a given frequency range would depend directly on the square of the effective mass of the reciprocating valve. This is true because the force necessary for constant valve displacement is directly proportional to the effective mass ($F = M\omega^2 d$), and force is proportional to current. Power is proportional to current squared, therefore mass squared. At this point the decision was made to use the inductive drive system. In the inductive drive system (as opposed to the conventional conductive system used in loudspeakers, shakers, etc.) the drive current is induced into the moving member by transformer action. Figure 3 shows the mechanical details of an inductive drive system. Its major advantages are the lack of flexible current leads and the simple one-piece mechanical design which eliminates bonded joints. As a result, the inductive drive system can be operated at higher temperatures than the conductive system. The Ling design uses two primary windings (driver coils) and one single-turn secondary winding (shorted turn). Since the magnetic structure is operated at or near saturation the device is essentially an air core transformer.

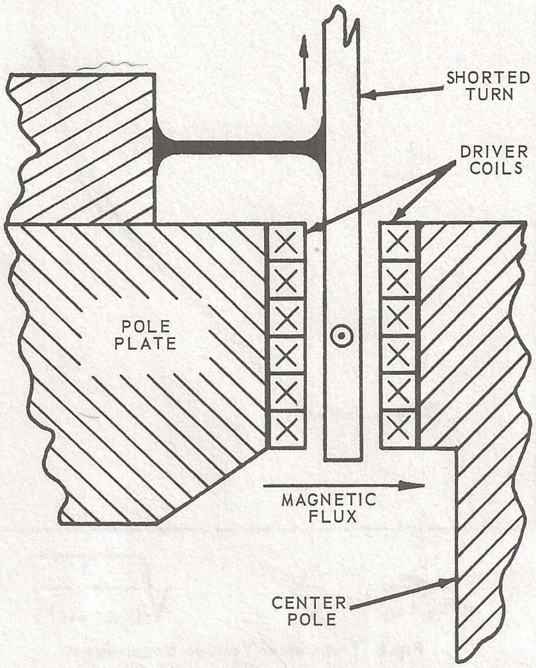

Fig.3 Mechanical Detail of Inductive Drive System

Figure 4 is the electrical analog circuit of the inductive drive system. It is convenient to combine the electrical and mechanical impedances of this device into a single electrical analog circuit for purposes of analysis. The mobility analogy is used as its force-current equivalence is fundamental in electrodynamic action. Appendix II presents a table of the interrelationships of the various parameters in the mobility analog system. In this analog system current is proportional to force, voltage to velocity, inductance to reciprocal spring stiffness, resistance to reciprocal damping coefficient, and capacitance to mass (weight). L_3 and C in Figure 4 represent the flexure and mass of the valve assembly, respectively.

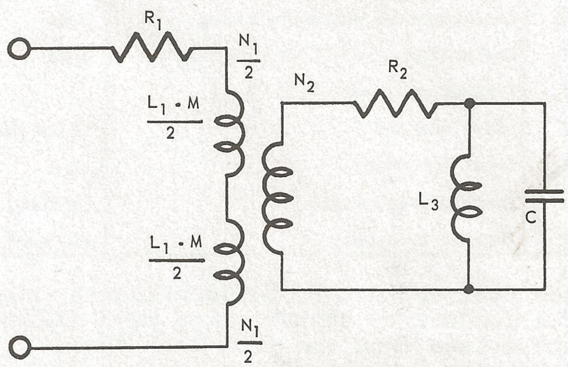

Fig.4 Electrical Analog Circuit of Transducer

Figure 5 is an equivalent analog circuit of the system shown in Figure 4. In this case all of the electrical elements in the secondary were referenced to the primary side of the transformer by multiplying by the square of the turns ratio.

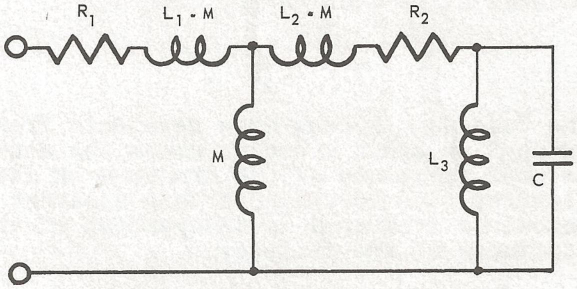

Fig.5 Equivalent Analog Circuit of Transducer

By setting up the appropriate loop equations the equivalent circuit can be solved for the various impedance and drive requirement functions. In order to solve the equations, certain simplifying assumptions are made. As a result, the expressions are not exact. However, for purposes of analysis they are sufficient. The symbols presented in Table I are used in the analysis.

Figure 6 shows the Bode plot of the input impedance. In all cases discussed in this paper, the axial structural resonance frequency of the moving element has been neglected, as it is above the operating frequency range. Note that the input imped-

TABLE I

SYMBOL	DEFINITION	UNITS
C	Analog Capacitance (dynamic mass)	farads
d	Peak displacement	centimeters
I	Drive Current	amperes rms
k	Coupling constant	dimensionless
L_1	Self inductance of primary winding	henrys
L_2	Self inductance of secondary	henrys
L_3	Analog inductance (spring compliance)	henrys
m	Mass	kilograms
M	Mutual inductance (primary-secondary)	henrys
N_1	Number of turns of primary winding	dimensionless
N_2	Number of turns of secondary winding	dimensionless
R_1	Resistance of primary winding	ohms
R_2	Resistance of secondary winding	ohms
s	Laplace operator ($\alpha + j\omega$)	
V	Drive voltage	volts rms
Z_{11}	Input impedance	ohms
μ	Force/current ratio	Newtons/ampere
ω	Angular frequency	radians/second

ance plot appears to be identical to that of a conductive shaker. The main difference arises from the fact that the expression for the fundamental armature resonance frequency (normally spring-mass resonance),

$$\omega = \frac{L_2 + L_3}{L_2 L_3 C},$$

simplifies to the following expression because L_3 is much larger than L_2:

$$\omega = \frac{1}{L_2 C}.$$

The "apparent spring-mass resonance frequency" is about a decade above the actual natural frequency of the armature on its flexures, and furthermore this apparent resonance frequency is independent of the stiffness of the flexures.

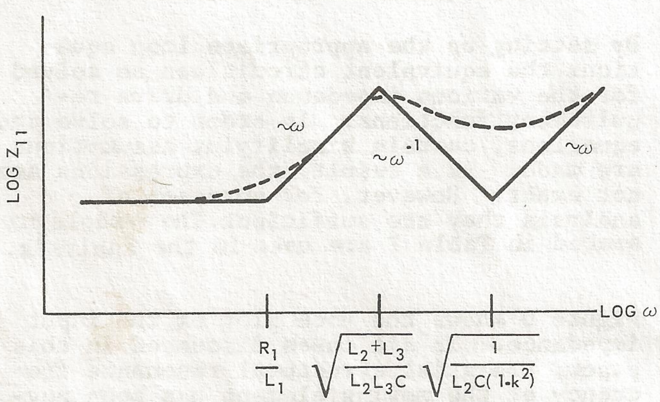

Fig.6 Transducer Input Impedance

Figure 7 shows the drive current requirement for constant valve displacement, while Figure 8 shows the drive voltage requirement. Note that the current and voltage requirements both rise with decreasing frequency at very low frequencies. At frequencies below the apparent spring-mass resonance frequency the primary-to-secondary current ratio is decreasing with increasing frequency. Above this resonance it is constant. Figure 9 is a Bode plot of the volt-ampere requirement of the transducer for constant valve displacement. Also shown in dashed lines is the real power requirement. Note that above the apparent spring-mass resonance frequency the real power requirements increase as the fourth power of frequency. It is apparent that any increase in full level frequency range is hard to obtain because of the power dissipation limitation of the cooling system.

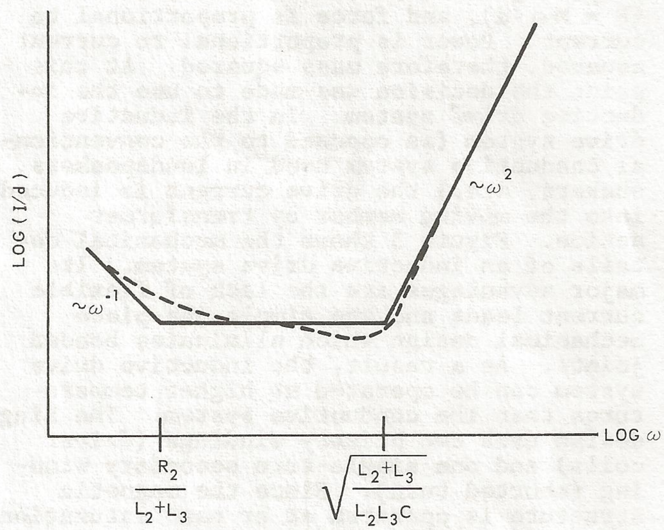

Fig.7 Transducer Current Requirement

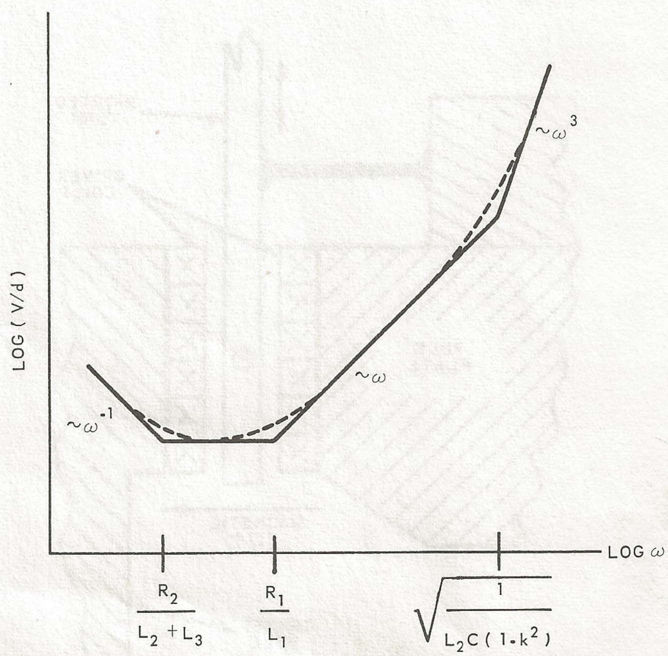

Fig.8 Transducer Voltage Requirement

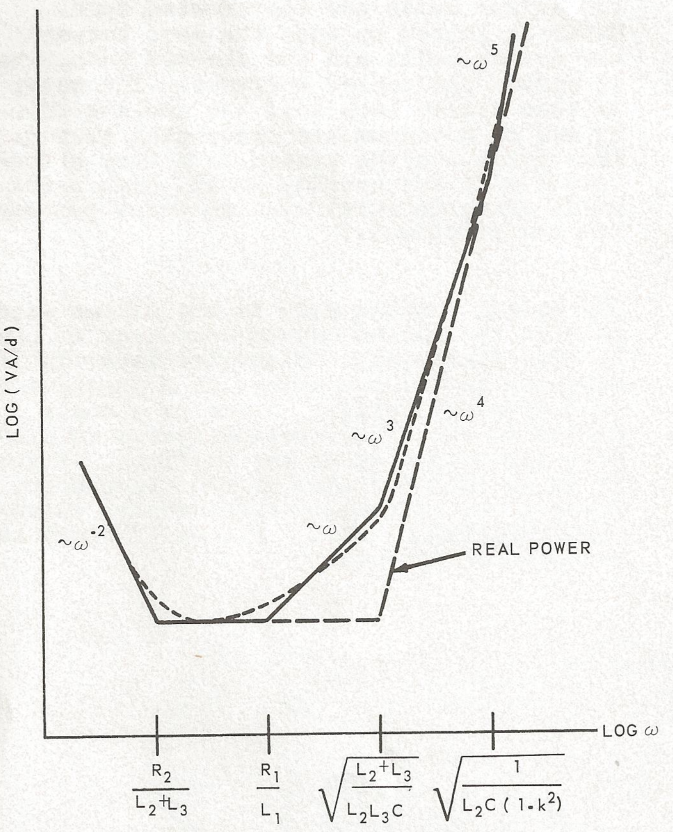

Fig.9 Transducer Volt-Ampere Requirement

Modulating System

The reciprocating, slotted sleeve valve type modulating system (refer to Figure 1) was designed to have a total modulated area of 23.8 cm^2 (3.7 in^2). This value was obtained by directly scaling existing designs. The final design incorporated 24 columns of ten slots each. Each slot is 0.89 mm (.035 in) high.

A method of controlling the neutral point of the reciprocating valve slots relative to the stationary valve slots (bias point) was necessary due to the net force caused by the "Bernoulli forces" acting on the slots. Simply, this force is due to the difference in velocity of the air passing over the top and bottom of each slot. The force is always in such a direction as to close the valve. The "biasing system" is shown in Figure 10. The control valve (CV) meters the air flow which has leaked up between the reciprocating and stationary modulating valves. This causes the pressure, p_b, above the top flexure to be higher than atmospheric pressure, p_o, but lower than the plenum supply pressure, p_x. The pressure p_b exerts a net force on the reciprocating valve and upper flexure, opposite in direction to the Bernoulli force. The relative position of the modulating valve slots can then be varied by changing the setting of the control valve, CV. There are actually four of these control valves on the transducer.

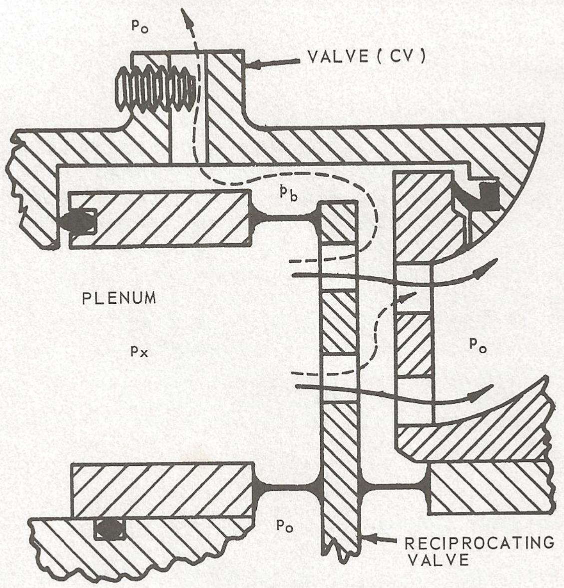

Fig.10 Modulating Valves and Bias System

The clearance between the valve assemblies was made nominally 25 µm (0.001 inch) to reduce the leakage from slot to slot between the valves. This leakage can significantly reduce the efficiency of the unit. The outside diameter of the stationary valve (mating surface) is coated with teflon to reduce friction and eliminate the possibility of galling. The stationary valve is made of 300 series stainless steel. Figure 11 shows a complete modulating valve assembly.

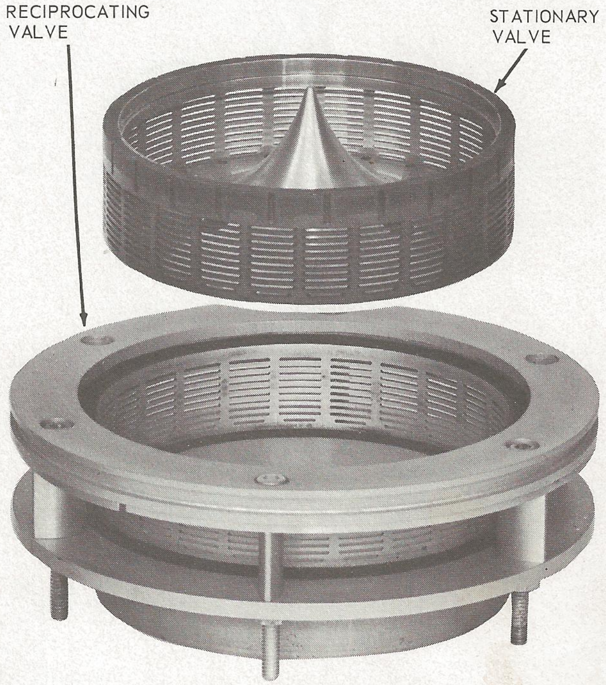

Fig.11 Modulating Valve Assembly

The modulating valves are fed with compressed air from a cast aluminum plenum housing surrounding them. The air is filtered to 141 μm adjacent to the plenum housing, by four screen filters.

Magnetic Structure

The magnetic structure (body) of the transducer consists of a gap-end pot similar to that of an electrodynamic speaker. Its component parts are a body ring - bottom plate - centerpole casting, a machined top plate, an outer driver coil assembly, and an inner driver coil assembly. The radial length of the magnetic air gap is 0.59 cm (0.23 inch). The gap flux density is 1.14 T (11400 gauss) averaged over the shorted turn of the reciprocating valve assembly. The field is excited by a 15 000 ampere-turn coil which dissipates about 1500 watts.

Cooling System

There are two basic cooling paths in the transducer. Both use pressurized distilled water as the coolant. The first cools the two driver coils and the shorted turn. Water is jetted up into the gaps between the driver coils and the shorted turn, where it washes the cooled surfaces. The water is then pulled back into the cooling unit by use of a vacuum air scavenging system. This spray cooling system is a very effective device for cooling small, high power units where integral cooling water passages are not practical.

The second cooling path is the direct water path in the hollow copper conductor of the 52-turn field coil. Distilled water is supplied to the transducer at a pressure of 13.8 N/cm^2 (20 psig). There is a small filter in the spray cooling system to prevent contamination of the orifices. Figure 12 is an exploded view of the transducer, showing all of the important design inovations described above. Figure 13 shows the assembled transducer.

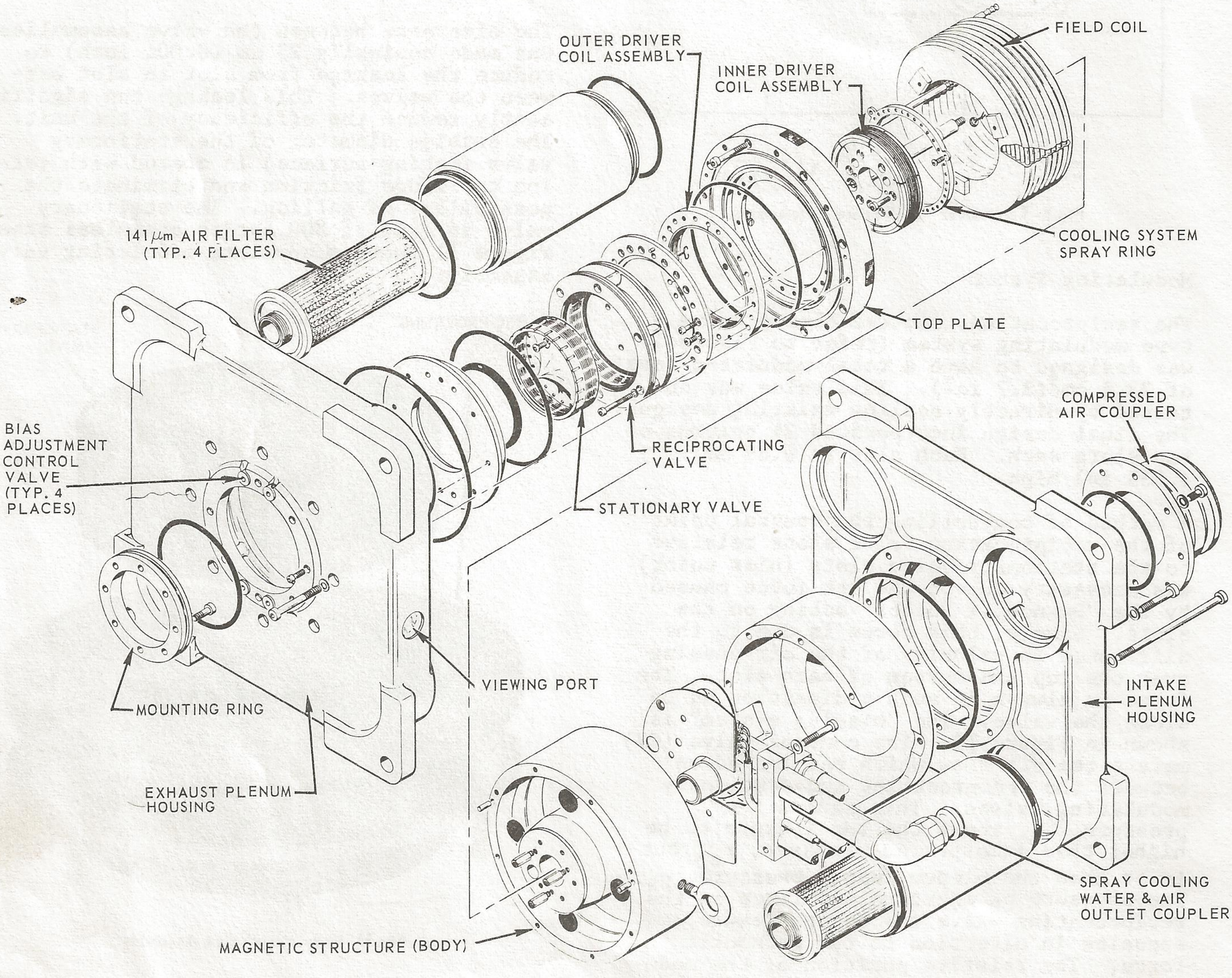

Fig.12 Electropneumatic Transducer Exploded View

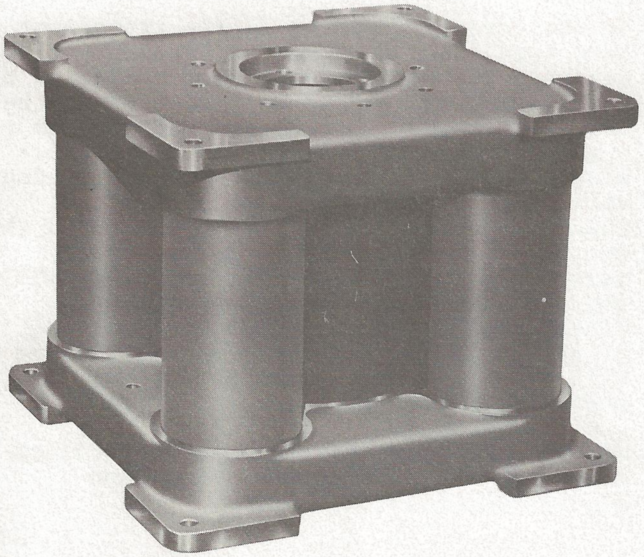

Fig. 13 Complete Electropneumatic Transducer

PROTOTYPE TEST PROGRAM

During the winter and early spring of 1965, the prototype transducers were evaluated. The tests included pneumatic, hydraulic, electrical, mechanical, and optical measurements. The tests evaluated the following parameters:

1. Air gap flux density.
2. Spray cooling system dissipation capability.
3. Field power requirements.
4. Electrodynamic drive power requirements as a function of frequency.
5. Cooling system water flow rate and pressure requirement.
6. Valve flexure stiffness.
7. Compressed air flow rate and pressure requirement.
8. Reciprocating valve bias system capability.
9. Acoustic power output.
10. Full power frequency range.
11. Random noise spectrum shaping capability.
12. Reliability of moving parts.

It was determined that the spray cooling system would allow drive coil currents of 60 amperes rms with about a 100 F° temperature rise. This 60 ampere limit allowed a maximum full modulation frequency of 1250 Hz, and a minimum of about 20 Hz. At a field excitation of 15 000 ampere-turns the gap flux density was above the knee of the saturation curve, where greater excitation would not significantly increase the flux in the gap.

The effective (dynamic) weight of the reciprocating valve was found to be 0.53N (0.12 pound). This meant that the force produced by the transducer at 1250 Hz and 0.45 mm vector displacement (.0175 inch vector) would be 1490 N vector (335 pounds vector), or better, 1053 N rms (237 pounds rms), which will apply for all waveforms.

The electrical drive power requirements conformed to the predicted shapes, as shown in Figures 6 through 9. The low frequency corner $(\frac{R_2}{L_2+L_3})$ occurred at about 40 Hz, the $(\frac{R_1}{L_1})$ corner at about 120 Hz, the apparent spring-mass resonance frequency at about 700 Hz and the high frequency corner ("antiresonance") at about 2000 Hz. The valve displacement was monitored with a calibrated microscope during the tests.

The remainder of the tests were performed with application of full pneumatic power. It was noted that the air flow added some loading to the transducer as seen in the drive power requirements. The upper and lower full power frequency limits were not changed, however. The reciprocating valve bias system worked very well. It allowed the relative opening of the slots to be varied from about 0.127 mm (.005 inch) up to about 0.508 mm (0.020 inch).

The evaluation of the acoustic power generation capabilities of the transducer was performed by coupling the transducer to a properly terminated 780 cm^2 cross section (11 inches square) progressive wave tube. Two different T = 0.6 hypex connectors were used during the tests: 1) a 13 foot long, 25 Hz cutoff unit, and 2) a 6.5 foot long, 50 Hz cutoff unit. Early tests showed that even though full valve modulation was attained up to 1250 Hz, the output acoustic power rolled down from the 10 kilowatt level at frequencies below 1250 Hz. It was further noted that the full acoustic power frequency range, as measured in the progressive wave tube (PWT) extended to a higher frequency with the 50 Hz horn than it did with the 25 Hz unit. This high intensity sound propagation attenuation phenomenon was verified by calculations[7] using existing theory[5,6]. A later test, using a 200 Hz, free field horn, showed that 10 kilowatts of acoustic output power could be produced to above 1000 Hz. This whole exercise dramatically pointed out the fact that the physical design of the acoustic system is the important parameter in the realization of high intensity sound energy at high frequencies. High level, wide band acoustic test system designs should give consideration to multiple narrow band sources (e.g. low frequency and high frequency transducers with appropriate low and high frequency cutoff horns).

Figure 14 shows several sine acoustic output power curves showing full capability with different cutoff frequency horns. Figure 15 shows several representative random noise spectra which were attained at the full level of 10 kilowatts overall. The random noise shaping capability is, of course, dependent on the characteristics of the acoustic system being driven by the transducer. Figure 2 graphically illustrates why this is true.

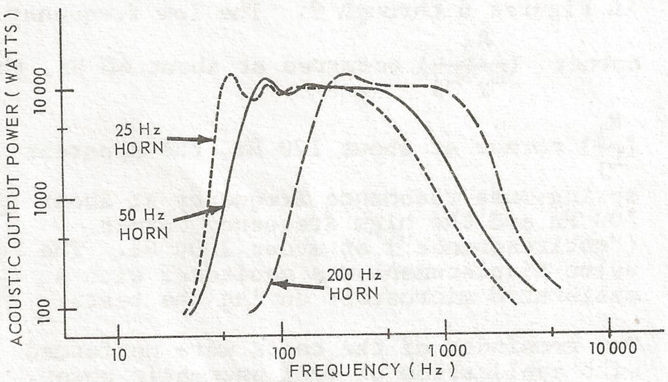

Fig.14 Sine Acoustic Output Power

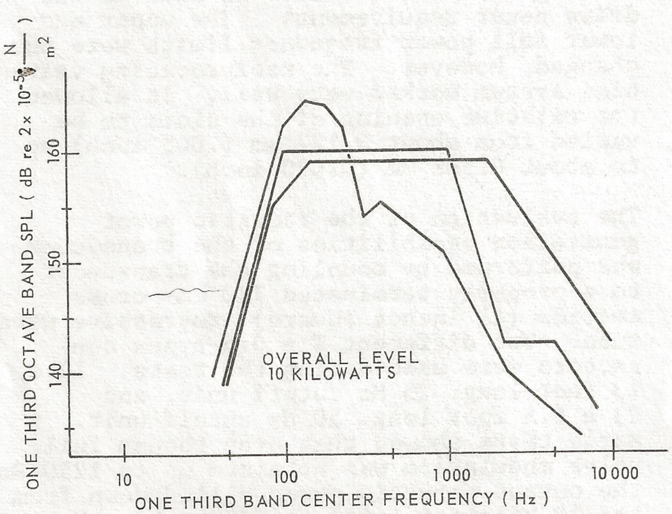

Fig.15 Typical Random Noise SPL Spectra

The final phases of the prototype test program involved running a 48 hour full power endurance test of the transducer. Some problems were encountered at first with the flexure rubber-to-aluminum bond. This problem was ultimately solved by going to a fortified natural rubber compound and taking special precautions in the prebonding cleaning processes. Since that time, valve assemblies in service have performed reliably for more than 100 hours.

LATEST CONFIGURATION

Several design features of the transducer have been changed since the final prototype design. One was the increased stiffness of the flexure system, to help counteract the Bernoulli force. The second was the use of a glass-filled teflon coating on the outside diameter of the stationary modulating valve to increase its life. The third change, presently being implemented, is the use of a high strength room temperature vulcanizing silicone rubber compound (RTV) for the flexures. This change will eliminate the high pressure, hot molding process which is presently being used and will also give far better resistance to hydrocarbon contamination.

Appendix III presents the latest specifications of this 10 kilowatt, wide band electropneumatic transducer.

APPLICATIONS

The most widespread use of electropneumatic transducers is in sonic fatigue, structure dynamics diagnosis, and simulated environment acceptance testing. These tests are usually performed in reverberation chambers of various sizes or in some form of progressive wave tube (i.e. conventional or shroud type) with a free field termination. Another type of acoustic test is the semi-reverberant, direct radiation approach, where the acoustic energy is radiated directly from a horn onto the test specimen. Some absorption material is placed in the test chamber to absorb part of the reflected energy. This method allows a great degree of spatial control of level and spectrum shape.

Speech communication for warning and propaganda distribution systems, using an electropneumatic transducer as the prime acoustic power source, are being investigated. There is even one application where a transducer is being used as a sonic cleaner for large missile parts. There is no reason why gases, or fluids, other than air can't be used with the electropneumatic transducer, as long as the proper safety precautions are adhered to.

CONCLUSIONS

As a result of this development program an electropneumatic transducer with a 10 kilowatt acoustic power output and wide frequency range capability was evolved. The frequency range of the new transducer extends an octave above that of other existing units.

As discussed in the text, the task of efficiently propagating the high frequency, high intensity acoustic energy from the transducer to the test section is one which requires a basic understanding of the attenuation mechanisms involved. The acoustic system designer must take this into account to fully realize the capabilities of this new transducer.

APPENDIX I

REFERENCES

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APPENDIX II MECHANICAL - ELECTRICAL RELATIONSHIPS MOBILITY ANALOG

	Spring-Inductance		Dashpot-Resistance		Mass-Capacitance	
	I	V	I	V	I	V
$\hat{d}(\hat{in})$	$\frac{8.85L\hat{I}}{\mu}$	$\frac{1.41\hat{V}}{\mu f}$	$\frac{1.41R\hat{I}}{\mu f}$	$\frac{1.41\hat{V}}{\mu f}$	$\frac{.224\hat{I}}{\mu C f^2}$	$\frac{1.41\hat{V}}{\mu f}$
$\hat{v}(\hat{in}/sec)$	$\frac{55.55L\hat{I}f}{\mu}$	$\frac{8.85\hat{V}}{\mu}$	$\frac{8.85R\hat{I}}{\mu}$	$\frac{8.85\hat{V}}{\mu}$	$\frac{1.41\hat{I}}{\mu C f}$	$\frac{8.85\hat{V}}{\mu}$
$\hat{a}(\hat{in}/sec^2)$	$\frac{349L\hat{I}f^2}{\mu}$	$\frac{55.55\hat{V}f}{\mu}$	$\frac{55.55R\hat{I}f}{\mu}$	$\frac{55.55\hat{V}f}{\mu}$	$\frac{8.85\hat{I}}{\mu C}$	$\frac{55.55\hat{V}f}{\mu}$
$\hat{g} = \frac{\hat{a}}{386}$	$\frac{.904L\hat{I}f^2}{\mu}$	$\frac{.144\hat{V}f}{\mu}$	$\frac{.144R\hat{I}f}{\mu}$	$\frac{.144\hat{V}f}{\mu}$	$\frac{2.29\times10^{-2}\hat{I}}{\mu C}$	$\frac{.144\hat{V}f}{\mu}$

$\frac{\hat{F}}{\hat{I}} = \mu \left(\frac{\hat{lbs}\ force}{\hat{amp}}\right)$

$\hat{d}$ ($\hat{inches}$)

$\hat{v}$ ($\hat{inches}$/sec)

$\hat{a}$ ($\hat{inches}/sec^2$)

f (cycles/sec)

$\hat{V}$ ($\hat{volts}$)

$\hat{I}$ ($\hat{amps}$)

L (Henries) $= \frac{.113\mu^2}{K}$, K (lbs/inch)

R (Ω) $= \frac{.113\mu^2}{D}$, D ($\hat{lbs}//\hat{inch}/sec$)

C (Farads) $= \frac{2.29\times10^{-2}W}{\mu^2}$, W (lbs)

L: $\hat{d} \sim \hat{I}$, $\hat{v} \sim \hat{V}$

R: $\hat{v} \sim \hat{I} \sim \hat{V}$

C: $\hat{v} \sim \hat{V}$, $\hat{a} \sim \hat{g} \sim \hat{I}$

APPENDIX III SPECIFICATIONS

Acoustic/Pneumatic

Acoustic Power Output:	10 kilowatts
Air Pressure Required at Exhaust Plenum Housing:	20.7 N/cm^2 (30 psig)
Air Flow Rate Required:.	0.855 kg/s (113 lbs/min)
Output Port Diameter:	9.09 cm (3.57 in)
Air Input Port:.	4" OPW 633 LAT Male Fitting
Frequency Range:	20 Hz to 5000 Hz, max power up to 1250 Hz
Maximum Allowable Plenum Pressure:	41.4 N/cm^2 gauge (60 psig).
Pressure Gauging Port:	¼" Female NPT on exhaust plenum housing

Mechanical

Valve Mean Diameter:	11.3cm (4.45 in)
Flexure Stiffness:	1137 N/cm (650 lbs/in)
Apparent Spring-Mass Resonant Frequency:	700 Hz
Maximum Armature Acceleration: . .	1.96 x 10^4 m/s^2 rms (2000g rms)
Slot Height:	0.89 mm (0.035 in)
Number of Slots:	240
Total Slot Area:	23.8 cm^2 (3.7 in^2)
Cooling System:	External water/air system independent of air supply
Rated Water Pressure:.	13.8 N/cm^2 (20 psig)

Drive Requirements

Maximum Drive Current:	60 amperes rms
Maximum Drive Voltage:	50 volts rms
Maximum Drive Power:	3000 volt-amperes

D.C. Armature Resistance (without cable):	0.24 ohm
Driver Coil-Armature Turns Ratio:	28:1
Effective Reciprocating Valve Weight:.	0.53 N (0.12 lb)

Field Requirements

Rated Field Current:	300 amperes d.c.
Rated Field Voltage (without cable):	4.6 volts d.c.
Field Resistance (without cable):	0.013 ohm
Field Protection:.	Discharge Rectifier

General

Overtravel:.	Excessive Valve Motion is limited by built-in snubbers
Position:.	May be operated in any position
Air Filtration:.	Internal filters provide 140 µm filtration
Interlocks:.	Normally closed thermal interlocks provided on driver coil and field coil
Mounting Ring:	Provided with transducer.

Dimensions

Length:.	0.457m (18.0 in)
Width:	0.457m (18.0 in)
Height:.	0.438m (17.25 in)
Weight (less cables and hoses):. .	754 N (170 lbs)