A DOUBLE-ACTING MOVING ARMATURE TRANSDUCER

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1. INTRODUCTION

A dominant trend in active sonar systems is toward lower operating frequencies, greater output levels, and larger bandwidths. The challenge for the transducer designer is to satisfy these needs economically, recognizing that all three objectives imply increased size and expense for the projector and its drive electronics. The transducer project described in this paper was motivated by the observation that when one attempts to meet the need for very low frequency transducers by scaling up the dimensions of a moderately low frequency, high power, body force projector to cover a lower frequency band, the usual result is an enormous transducer with excess power capability.

This is illustrated in the first two columns of Table 1 (see Section 5). The first column lists characteristics of the Sanders Model 40, a proven high power flextensional unit. Scaling this design nearly two octaves, from 400 to 110 Hz, yields the truly impressive source described in the second column. Not only is this transducer too large and heavy for most uses, its source level rating (and associated power consumption) is too high for most applications. Of course the power rating can be lowered artificially by simply replacing some ceramic by inert material, but this is inelegant since it also reduces coupling and, concomitantly, bandwidth. Clearly another approach is needed.

In addition to meeting specific source level and bandwidth requirements, projectors designed for the very low frequency region (VLF, 20 to 200 Hz) should also strive for

- Economy (reflected in the initial transducer purchase expense; costs of auxiliary equipment such as amplifiers, tuning networks, and handling gear; and maintenance expenses for both transducers and amplifiers)
- Highest possible efficiency (which is related both to thermal design considerations and to usage expense since high-power systems consume large amounts of prime power, therefore higher efficiency yields considerable savings in energy costs)
- Reliability (for greater mission effectiveness as well as additional cost savings through less frequent repairs, less preventative maintenance or refurbishment, less need to send source maintenance technicians on sea trials, etc.)
- Low distortion (an increasingly important requirement in recent sonar system specifications).

Electroacoustic theory [1] indicates that miniaturized (very small compared to a wavelength) underwater sound projectors cannot achieve both large bandwidths and high efficiencies. Since the radiating area must be large to achieve respectable acoustic performance, why not let the size be what it has to be, then minimize cost through use of a more economical method of transduction? In this context it seems sensible to consider the moving armature (also known as "variable reluctance") transduction technique, since this is regarded as being underpowered relative to piezo- and magnetostriction, and would, thereby, tend to mitigate the excess power dilemma. Furthermore, it could be cheaper than alternate technologies for a given depth, frequency, and source level because the basic mechanism does not rely on expensive piezo- or magnetostrictive transduction materials; it can be made out of regular steel core laminations and insulated copper wire.

Additional advantages of the moving armature driver in this role are that it is suitable for low frequency work (eddy current losses are negligible in the VLF region), offers reasonably good efficiency and high coupling, and retains the traditional advantages of the moving armature driver, simplicity and reliability. Standard objections to this transduction mechanism, high distortion and performance changes with depth, can be overcome by innovative design.

This paper will describe a moderately high-powered 110-Hz sonar projector which has been designed on these principles. Dimensions, weights, and performance predictions for the prototype and for similar designs for other frequencies and power levels will be presented.

2. TECHNICAL OVERVIEW

Perhaps the simplest example of a moving armature driver is an electric buzzer or bell ringer: an electromagnet attracts a hinged armature which, upon moving from its rest position, interrupts the electric circuit and stops the magnetic attraction. A spring then returns the armature to its rest position. When this humble mechanism is refined for use in an electroacoustic transducer, the position of the armature does not influence the electric signal; and, in the double acting case we shall be examining, opposing electromagnets pull the armature first one way and then the other. A centering spring provides dynamic stability and helps return the armature to its equilibrium position midway between the two magnet faces.

This section presents salient features of the moving armature transducer, as related to its use in sonar projectors. Most of the technical antecedents cited here were taken from the 1988 review report by McLaughlin and Moffett [2].

The first important distinction is that the electromechanical coupling coefficient is not governed by material properties but rather by the degree of magnetic bias and the size of the air gap in the magnetic circuit. Consequently one can achieve high coupling if the quiescent magnetic gap is made small. Achieving a reliable small gap requires clever mechanical design, accurate machining, and careful assembly and alignment. The magnetic gap must not only be small; it must be independent of changes in hydrostatic pressure. A solution for this problem is to use an internal reaction mass to provide a depth-independent reaction to the radiation force. This technique is well established in principle [3,4], but seldom seen in everyday devices.

The next distinguishing feature is that the basic transducing mechanism is non-linear, however there are quite effective means around this obstacle. These distortion reducing methods rely on using magnetic bias and on what has been termed a "push-pull" magnetic design. (Actually, "pull-pull" or "double-acting" seem preferable descriptions, since electromagnets can never push anything.) This feature has appeared in designs proposed by several inventors [5-8]. Many of these designs use permanent magnet bias since this eliminates the need for a DC power supply, minimizes coil space requirements, and halves the number of power cables needed. However, magnets do not permit bias field adjustment (to tweak

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coupling and electrical impedance) and they introduce another source of nonlinearity. In fact, Abbott [9] has stated that permanent magnets should be avoided to achieve maximum efficiency.

A colleague at BBN, Bruce Murray, has achieved excellent results using double-acting moving armature drive in high output *shakers*. These devices exhibit very low distortion (typically THD levels are 45 dB below the fundamental at high but non-saturating drive levels), and similar results are expected from acoustic projectors.

3. PROTOTYPE DESIGN

Main features of the transducer under development are shown in Figure 1, a mechanical schematic of one half of the projector. To balance dynamic forces, a mirror image of these parts is placed below it, or you can imagine the single-sided projector mounted on a large stiff baffle and radiating into an infinite half space. The external housing of the twin unit is a circular cylinder, a form that fits nicely in a torpedo-shaped tow body.

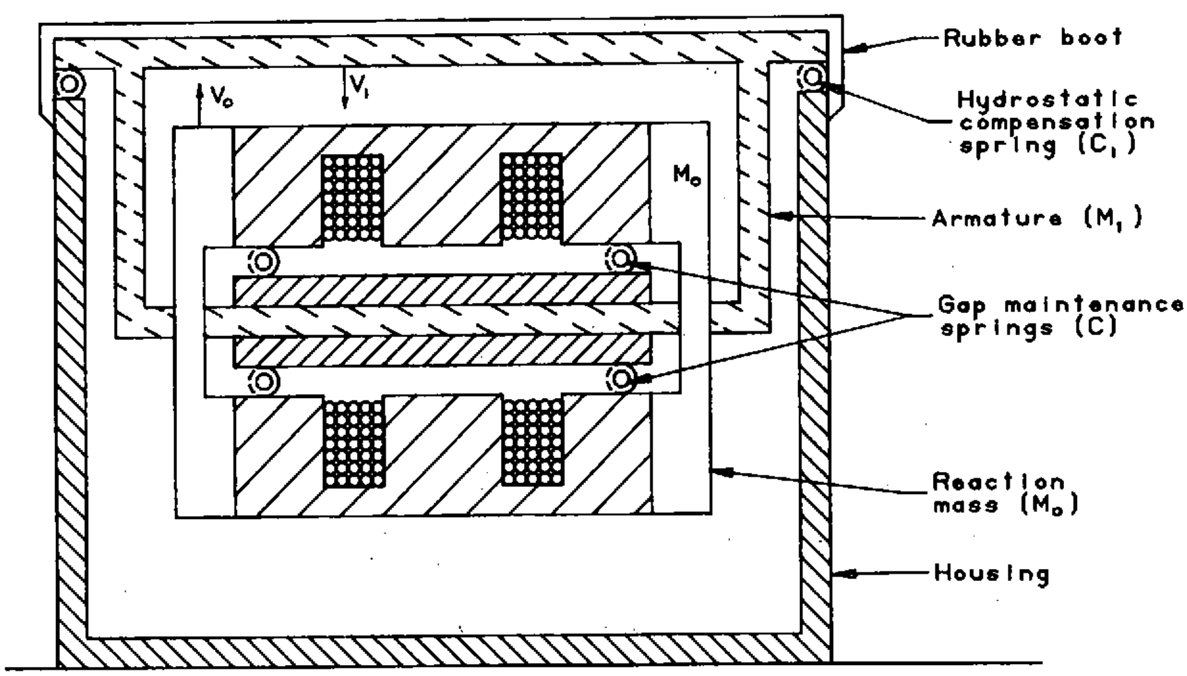

Figure 1: Mechanical Concept, One Side of Source

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The magnetic circuit is formed by two sets of E-shaped laminations straddling back-to-back I-shaped laminations and held in position by gap maintenance springs. The E's are each fitted with two sets of windings wound on the same core; one for the DC bias current, the other for the AC signal current. The DC windings in the two cores are connected in series and the air gaps are adjusted to be the same thickness so that when the DC bias current is applied the static magnetic attraction is equal on the two sides and the armature assembly remains in place. Unequal initial gaps or a too-high bias current can cause one gap to snap shut when DC is applied.

The AC windings from each electromagnet are also connected in series, but with opposite polarity so that the magnetic attraction between E's and I's alternates between acting to close first one gap then the other. The magnetic circuit is completed through the rigid keepers which maintain a fixed distance between the two E's. The gap springs, shown schematically as small rings in the gaps themselves, are actually located in the spaces behind the E cores in our design. The mass of the two E laminations, their windings, and the keepers is much greater than that of the central I-laminations, its support bracket, the radiator plate and the radiation mass of the water load, therefore oscillations of the relative positions of the E's and I's produce useful deflections of the radiator plate. The radiator plate is isolated from the housing by a second set of springs, the hydrostatic compensation springs, which serve to maintain a fixed magnetic gap independent of depth. The housing is motionless, due either to the balancing vibrations of the lower half of the projector or by being attached to an infinite mass. To summarize, at the operating frequency the stator (the reaction mass, the parts connected to the E's) stays relatively stationary due to its higher inertia, while the lower-mass armature assembly (the parts connected to the I's) vibrates with larger amplitude and radiates sound into the acoustic medium.

Two features of this design are essential to its proper operation: one is the double-acting back-to-back arrangement of the magnetic forces which improves the linearity of the driver; the other is the use of an internal reaction mass and an independent suspension system for the vibrator, an arrangement which prevents hydrostatic pressure changes from affecting the magnetic gap. Reference [2] indicates that US patents have been issued for transducers which include both of these features: one to Frank Massa, Jr. in 1967 and another to George Pida in 1972. Both transducers used permanent magnet bias and were dipole radiators ("shaker boxes") rather than having one vibrating face isolated from the acoustic medium. Massa's device has the E-laminations attached to the armature. The Pida transducer has the E's attached to the stator, as does our design, but has a reaction mass/armature mass ratio of 1:1 in contrast to our 7:1 ratio.

An equivalent circuit for one half of the device is shown in Figure 2. Most circuit elements can be easily identified with one or more mechanical components in Figure 1. The others are: the radiation resistance and mass, R_r and M_r; the mechanical losses, R_m and R_1, associated with motion of the gap and hydrostatic compensation springs, respectively; the blocked resistance and inductance, R_b and L_b, representing purely electrical properties of the windings when all motion is blocked; and C_m the compliance of the gap spring C after magnetic softening. $V_R = V_1 - V_o$ is the relative velocity between masses M_o and M_1, and α is the electromechanical gyrator ratio.

Our arrangement has $M_o >> M_1$, so little velocity flows in the M_o shunt (the reaction mass is nearly stationary). Mechanical losses are partitioned between R_1, associated with motion V_1, and R_m, associated with motion V_R. Performance predictions are based on the equivalent circuit, with stiffnesses and masses calculated for a given parts layout, α and L_b calculated from theory (and verified by measurements on an instrumented E/I pair in a test stand), and losses simply guessed at.

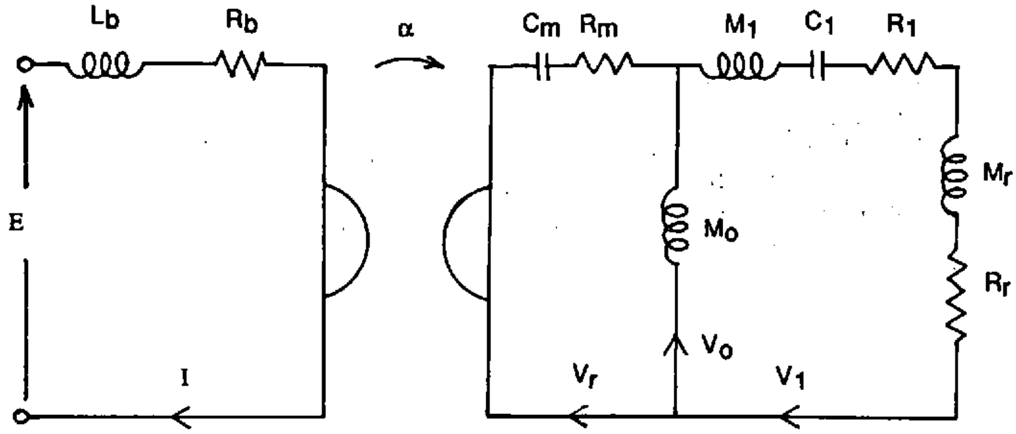

Figure 2: Equivalent Circuit for One Side of Source
(Force/Voltage Analogy)

Figure 3, a wide band plot of TCR and TVR, shows two distinct resonances since this is a two degree-of-freedom mechanical system. The sharp low frequency one represents large amplitude lightly damped motion of the stator (reaction mass) with the armature relatively still. The broader high frequency one is the desired mode in which the roles of the two masses are reversed. Useful output at the low frequency resonance is not practical because the amplitude is limited by gap size, an effect not accounted for when calculating the TCR. The maximum source level curve in Figure 4 focuses on the upper resonant region and shows that we expect an output level of 194 dB re 1 μPa at 1 m at the constant current resonance of 110 Hz. Mechanical Q and efficiency forecasts are less certain since these depend on mechanical losses which are difficult to predict in a new design. These curves are based on a single-element loaded Q_m of 10.

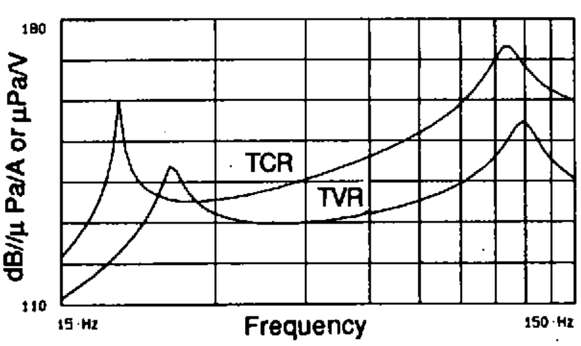

Figure 3: TCR & TVR Over a One
Decade Frequency Range

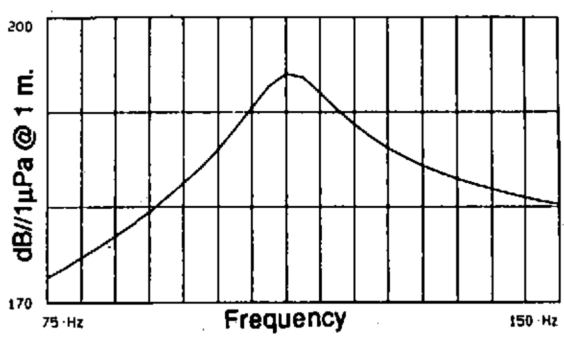

Figure 4: Maximum SPL For Constant
Current Drive

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Figure 5 shows that the portion of static gap taken up by armature motion at full power peaks at 60%. Included on the same graph and to the same scale is the deflection of the reaction mass, demonstrating that it remains essentially motionless. The size of the prototype is 18 inches in diameter by 45 inches long, and the ka of the radiator piston is .068 at resonance. This is smaller than desired (remember: the first design principal was to make it BIG for efficient broadband operation), but we were constrained to using existing components and to keeping the cost of the first article low. The weight of the device is 1000 lbs in air and 600 lbs in water. Operating depth range at full output is expected to be 70 to 840 ft, with survival depth 1200 ft.

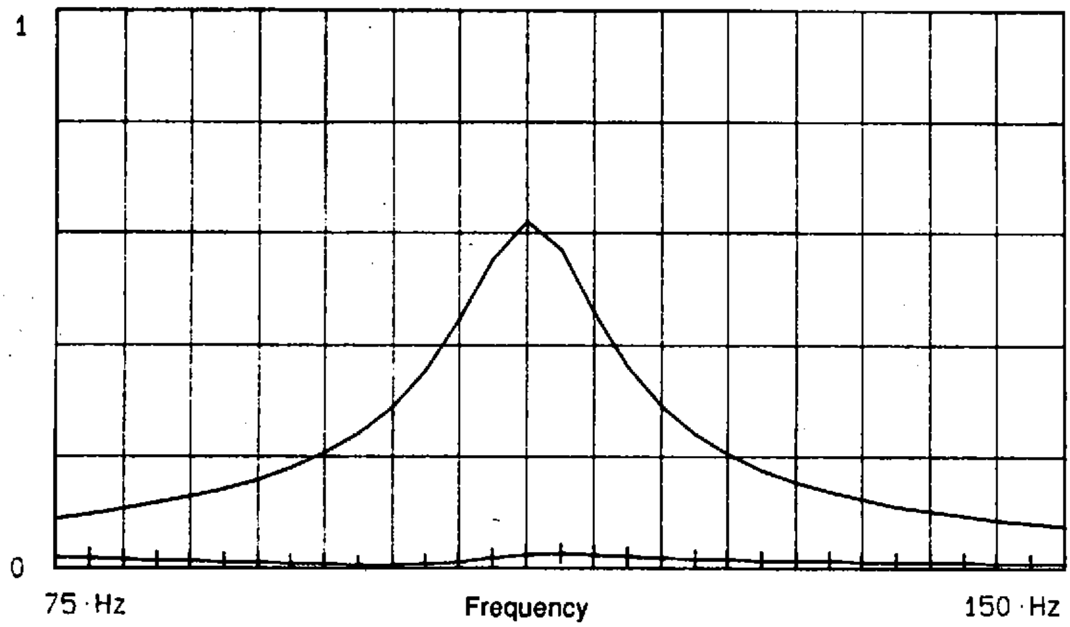

Figure 5: Peak Gap and Reaction Mass Displacements as a Fraction of Static Gap for Constant Current Drive at Maximum Level

Figure 6 presents the predicted electrical impedance of the unit. The moderate phase angle variation around resonance suggests that electrical tuning may not be necessary. Figure 7 shows the variation of input power and input volt-amperes (untuned) at maximum drive level, while Figure 8 presents the electroacoustical efficiency, first considering the motional circuit only (acoustical power out divided by AC power in), then including DC bias field losses as well.

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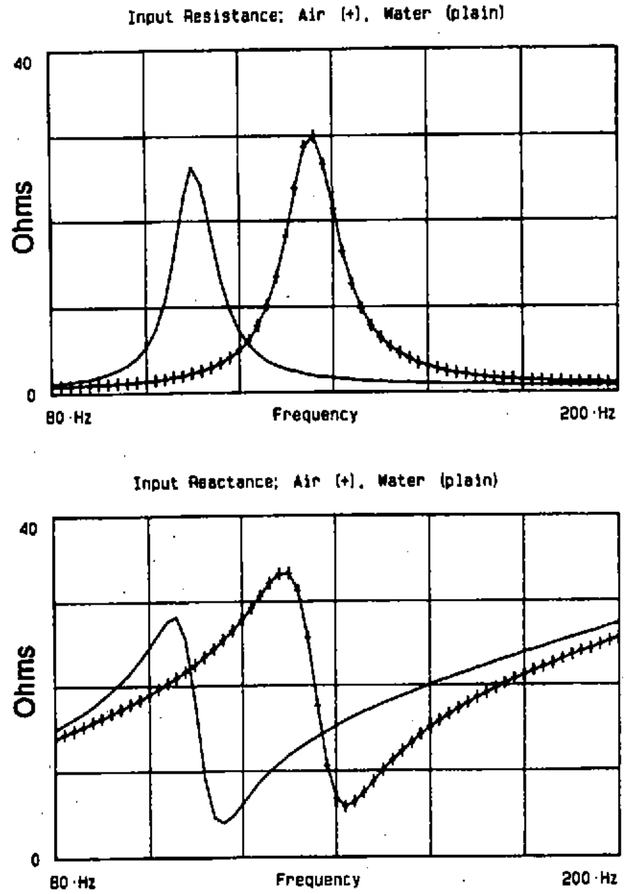

Figure 6: Electrical Impedance ,
Air (+'s) and Water

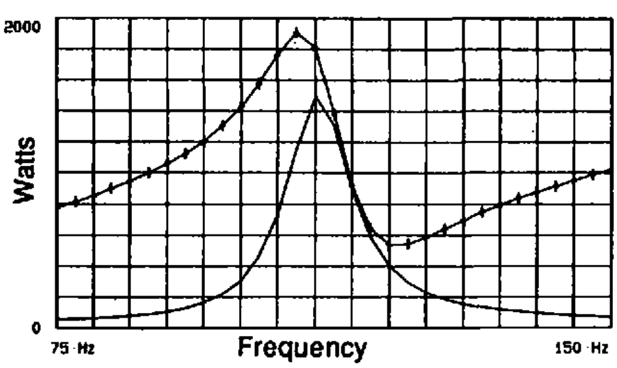

Figure 7: Electrical Input Power
and Volt-Amperes

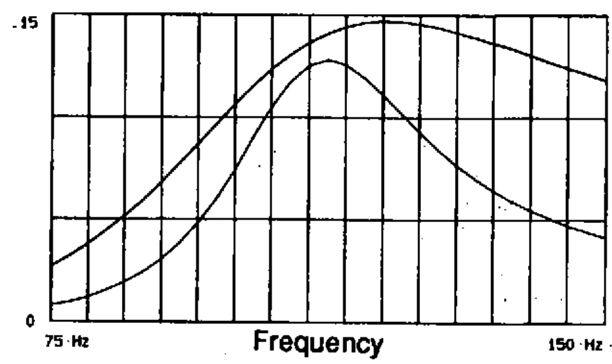

Figure 8: Electroacoustic Efficiency,
With and Without DC Losses

4. RELATED DESIGNS

Since the prototype is somewhat smaller than we would have liked, model studies were done to investigate performance changes with size. There are many ways to vary size. In the standard method all dimensions are scaled by the same ratio, and resonance frequency shifts by the reciprocal of that ratio. In the present study a different approach was taken. Using the same "motor" (moving armature vibrating mechanism) in all cases, the diameter of the radiator plate was varied. To maintain depth capability, the stiffness of the depth compensation spring was varied in proportion to the exposed area . To approximate the balance between that portion of the armature mass that serves as the radiator plate (mass varies as the cube of diameter to preserve static bending stresses in the plate) and that portion connected to the I-cores (mass unchanged by diameter changes), M_1 was varied proportional to area. The mechanical loss R_1 (essentially the boot loss) was varied with area, while the reaction mass, gap springs, and other losses were left unchanged.

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The results are given in Figures 9 and 10. The smallest case (11.5 in diameter) is the size of the current prototype, the others are larger variants.

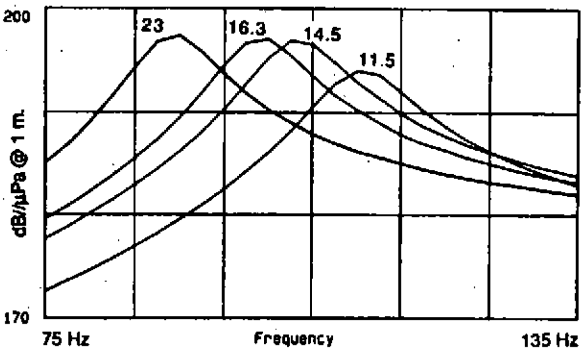

Figure 9: Maximum SPL for Four Radiating Plate Diameters. Parameter is Diameter (inches)

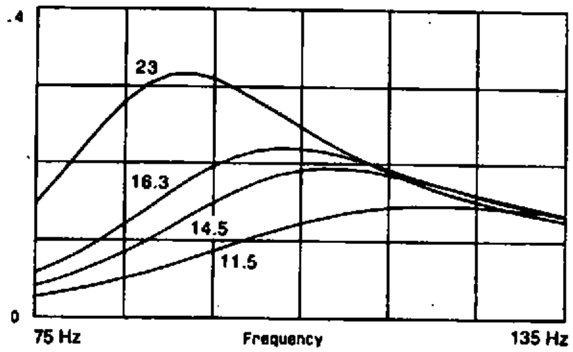

Figure 10: Electroacoustic Efficiency for Four Radiating Plate Diameters. Parameter is Diameter (inches)

Note first that increasing radiating area causes a uniform drop in main resonance; the decrease is about 22% (0.3 octave) for a quadrupling of radiating area. This frequency shift shows that the increase in M_r with size has more influence on resonance than the increase in depth compensation spring stiffness.

Next observe that the larger three examples appear to belong to a group having the same peak output and bandwidth at each of the three resonances, while the smallest one has lower output but comparable bandwidth. This phenomenon is due to the limiting constraint being different in the two regions: the smallest one is current limited, the larger three magnetic field limited. The efficiency curves peak at progressively larger values as radiating area increases, with a particularly large jump between the largest and next-to-largest cases.

These experiments can be summarized as follows: varying radiation area but not driver size (except to optimize the gap and bias at each stage) results in a moderate frequency shift, little change in bandwidth or peak source level, and substantial change in efficiency at resonance. In fact, the variation in efficiency with ka is essentially linear over the 64% change in ka represented by these four samples.

5. SUMMARY

Table 1 compares pertinent parameters for three low frequency projectors: our new moving armature design (designated USS190), a glass-shell flextensional scaled up to match the resonance of the moving armature device, and a large moving coil source (J15-3). We find, as expected, that the power level, efficiency, and power density of the moving armature design are fairly evenly bracketed by those of the ceramic-based design on the high side and the non-resonant electromagnetic design on the low side.

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Table 1: Comparison of Low-Frequency Projectors

	Sanders Model 40		BBN	NRL
	Regular	Scaled	USS 190	J15-3
Length (in)	20.9	75	45	18
Diameter (in)	19.1	69	18	29
Volume (cubic ft)	5.7	266	6.6	6.9
Air Weight (lbs)	180	8300	1000	375
Max Depth (ft)	500	500	800	590 *
Resonance (Hz)	400	110	110	(140)
Mechanical Q	5	5	10	(.4)
Bandwidth (Hz)	355-435	100-120	105-115	50-400
Max. Power Out (kW)	6.5	84	0.2	0.008
Max. SPL (dB//µPa)	209	220	194	180
Watts/lb	36	10	0.2	0.02
Watts/cu ft	1100	310	30	1.2
Max efficiency	0.9	0.9	0.15	0.01

* Performance degrades with depth; values in table are for shallow operation.

The technical objectives of this project were to develop a small, low frequency, moderately high power source having low distortion, high reliability, and low cost. Careful analysis of the design combined with experience with this technology in shakers leads us to believe we will meet all acoustic goals. As for reliability and cost, it's too early to tell, but we are hopeful.

The prototype unit is intended to serve as a test article for measuring losses and distortion, for providing data for refining the design equations and design methodology, and as a towed source in ongoing projects at BBN. The development schedule for this unit calls for deep water testing by December, and it is expected that test data will be available to show at the conference. We hope that disclosure and distribution of this data may bring us in contact with other users needing projectors with these qualities.

6. REFERENCES

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