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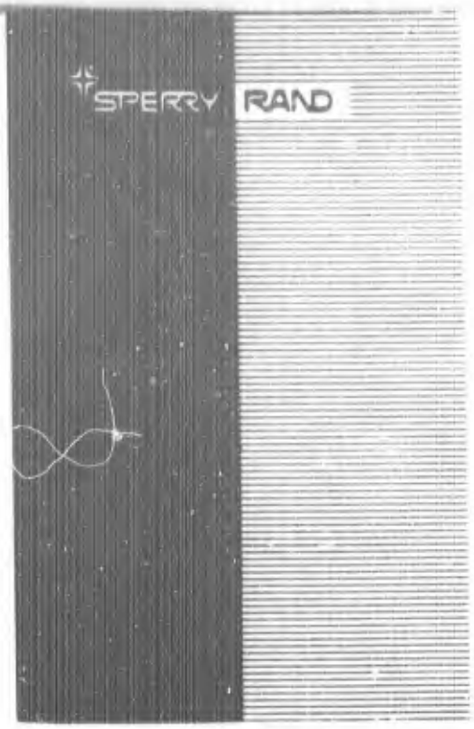
INTERIM REPORT

ANALYSIS OF ACOUSTIC BAFFLES FOR UNDERWATER NOISE REDUCTION

CONTRACT N00014-67-C-0303

Prepared for

OFFICE OF NAVAL RESEARCH
DEPARTMENT OF THE NAVY
WASHINGTON, D.C. 20360



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SGD-4230-0476
DECEMBER 1971

SPERRY GYROSCOPE DIVISION
GREAT NECK, NEW YORK 11020

FOREWORD

This is an interim report on the study of underwater noise reduction techniques being conducted by the Sperry Division for the Office of Naval Research, Acoustic Programs, under Contract No. N00014-67-C-0303. Included in this report is a description of the work being done on spring-mass type acoustic baffles. Specifically, the effects of rigid and compliant outer plates are analyzed, theoretical results are compared with experimental data, and good agreement is obtained. It is concluded that significant gains can be achieved at the lower frequencies through the use of compliant outer plates in the baffle design.

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SECTION 1

INTRODUCTION

Baffles are structures that shield acoustic sensors from unwanted noise sources and provide means for modifying the response characteristics of hydrophones. Thus, the acoustic baffle is a primary tool for underwater noise reduction and directivity pattern control in sonar arrays.

The transmission of acoustic energy in water can be impeded by a compliant baffle. To be effective, such a baffle must be sufficiently stiff to withstand hydrostatic pressures, yet it should be dynamically soft to perform as a highly reflective structure. A reflective characteristic is acceptable for the side of the baffle facing the interfering noise source, but the portion facing the hydrophone presents a major design challange. Unless this side is nonreflective, the interaction between the incident signal and the reflected signal from the baffle can alter the hydrophone array's response, often seriously degrading it.

In previous phases of this program, Sperry has developed a set of analytical models for predicting the reflectivity, transmittivity, and other acoustic parameters of planar multilayered baffle structures. Computer programs based on the analytical models have been utilized for determining baffled hydrophone response. The results of this program have shown high correlation with empirical data (References 1 and 2).

More recent work has been concerned with multilayer cylindrical baffles. Analytical expressions have been developed for the calculation of sound pressure level at any point exterior to the baffle, and a computer program to solve these equations was written. The calculated results show good agreement with experimental data (Reference 3).

Additional analysis has been done on baffle structures consisting of steel plates separated by pins. These structures are non-homogeneous and thus cannot be analyzed by the methods previously used. Lumped-parameter models, designed to simulate the behaviour of spring-mass baffles, have been developed. Performance calculations have been made which give good agreement with measured values (Reference 3).

Spring-mass baffles, by proper choice of spring and plate constants, offer the advantage of operating under higher hydrostatic pressures than other types of acoustic baffles.

SECTION 2

SPRING-MASS BAFFLES

A. RIGID OUTER PLATES

A typical spring-mass baffle configuration consisting of four plates is shown in figure 2-1. The outer plates are of moderate thickness and remain effectively rigid under pressure. These plates transmit the pressure, via pins, to the relatively thin inner plates that deform sufficiently to create the desired baffle compliance. The baffle structure is placed behind a heavy steel plate that supports the hydrophones and provides means for modifying their response characteristics.

The schematic representation shown in figure 2-2 illustrates how the analytical model of the baffle was developed.

The analysis was based on the premise that each of the non-deflecting outer plates is rigidly coupled to half of an inner plate which can deform. This mass is then linked by a spring to the other half of the inner plate; the two inner plate halves are rigidly linked together. Mass M_1 represents the mass of plate No. 1 plus half of plate No. 2. Mass m equals the sum of the halves of plates No. 2 and No. 3, while Mass M_2 represents the remaining half of plate No. 3 plus plate No. 4.

Based on the above model, an equivalent circuit of the spring-mass baffle has been developed and is shown in figure 2-3. L represents the associated viscous resistance of the springs, which is due mainly to the rubber pads associated with the separating pins. The characteristic inpedance of the water, Z, is numerically equal to 5.84 in an inch-pound-second system. Because of the symmetrical construction of the baffle, $M_1 = M_2 = M$ and $K_1 = K_2 = K$.

An examination of this circuit shows that at the lower frequencies, the effect of the outer plate mass is not significant because the term -jWM is much less than the value of Z. In this frequency region, the insertion loss is dominated by the spring term jK W and thus rises at 6 dB per octave. As the frequency increases, the dissipative term, L, dominates and the insertion loss flattens. At still higher frequencies, the mass terms become significant and the insertion loss again begins to rise.

B. COMPLIANT OUTER PLATES

The performance of the spring-mass baffle has been extended to lower frequencies. Conceptually, this calls for greater compliance in the interior plate-springs of the baffle, which might be obtained by adding further layers of these plate-springs. However, as was noted previously, additional compliance can be obtained, without increasing the baffle thickness, by making the outer plates thinner and thus less rigid.

An improved model for calculating baffle performance has been developed that realistically allows for the deformation of the outer plates. The model described in Section 2A assumes that the outer plates do not deform. The modified model approximates the deformation of the outer plates by dividing them into three regions and allowing each region to be displaced by a spring-like action from the adjacent region. (See figure 2-4.) One region encompasses the area of the outer plate which is linked by pins to the inner plate deformed in one direction. The second region, of equal area to the first, covers the area linked by pins to the inner plate deformed in the opposite direction. The various regions are linked by springs. Section 3 presents the detailed analysis of the compliant outer plate baffle model.

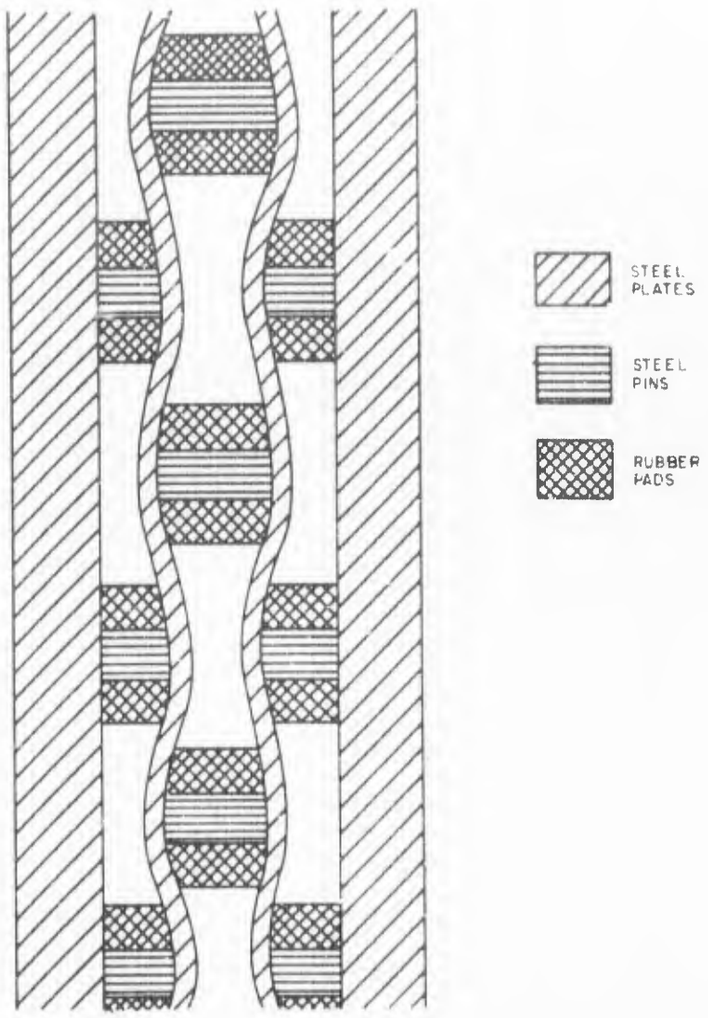

Figure 2-1. Spring-Mass Baffle Configuration Under Hydrostatic Pressure

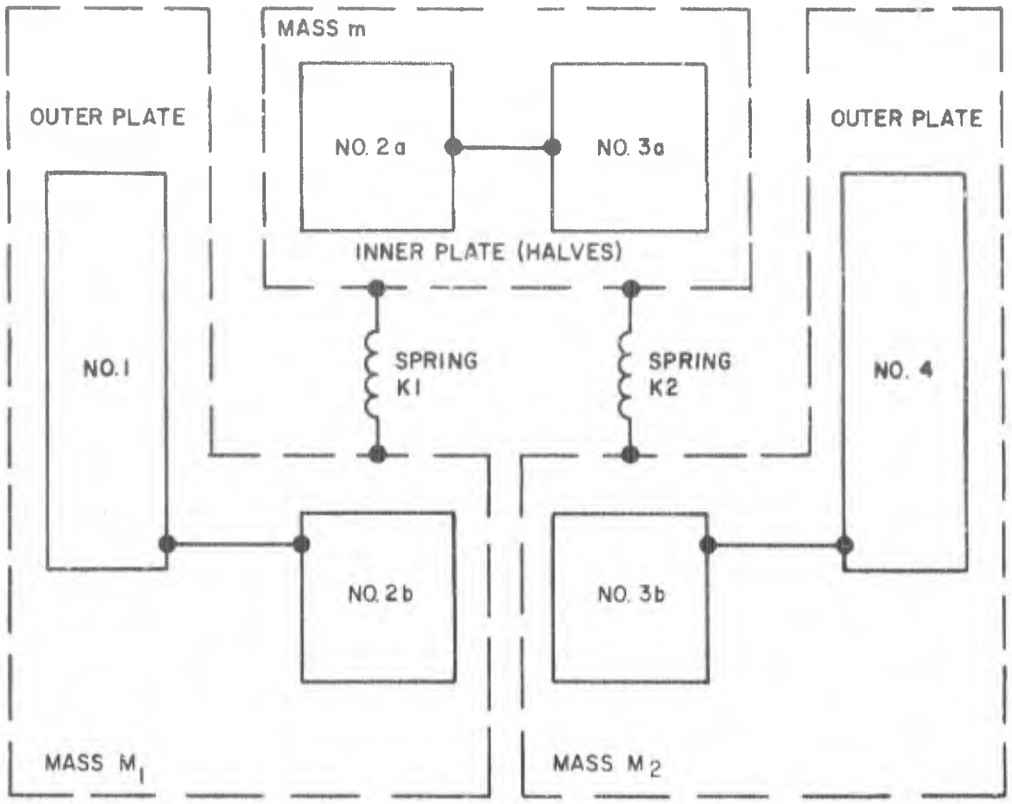

Figure 2-2. Analytical Model of
Spring-Mass Baffle with Rigid Outer Plates

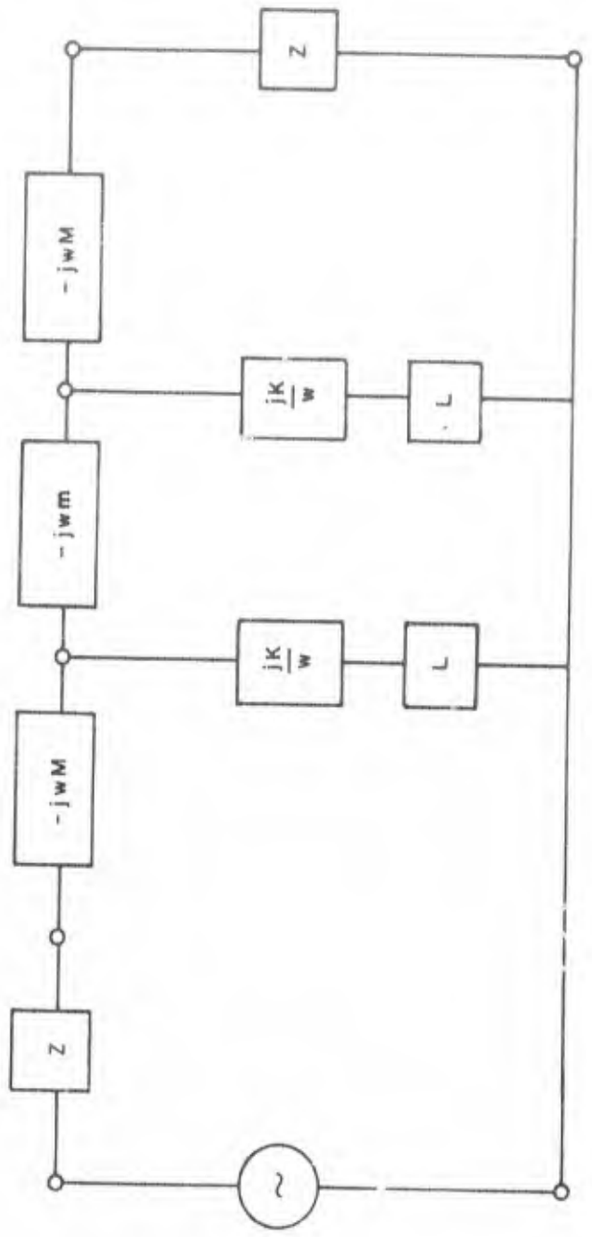

Figure 2-3. Equivalent Circuit of Spring-Mass Baffle with Rigid Outer Plates

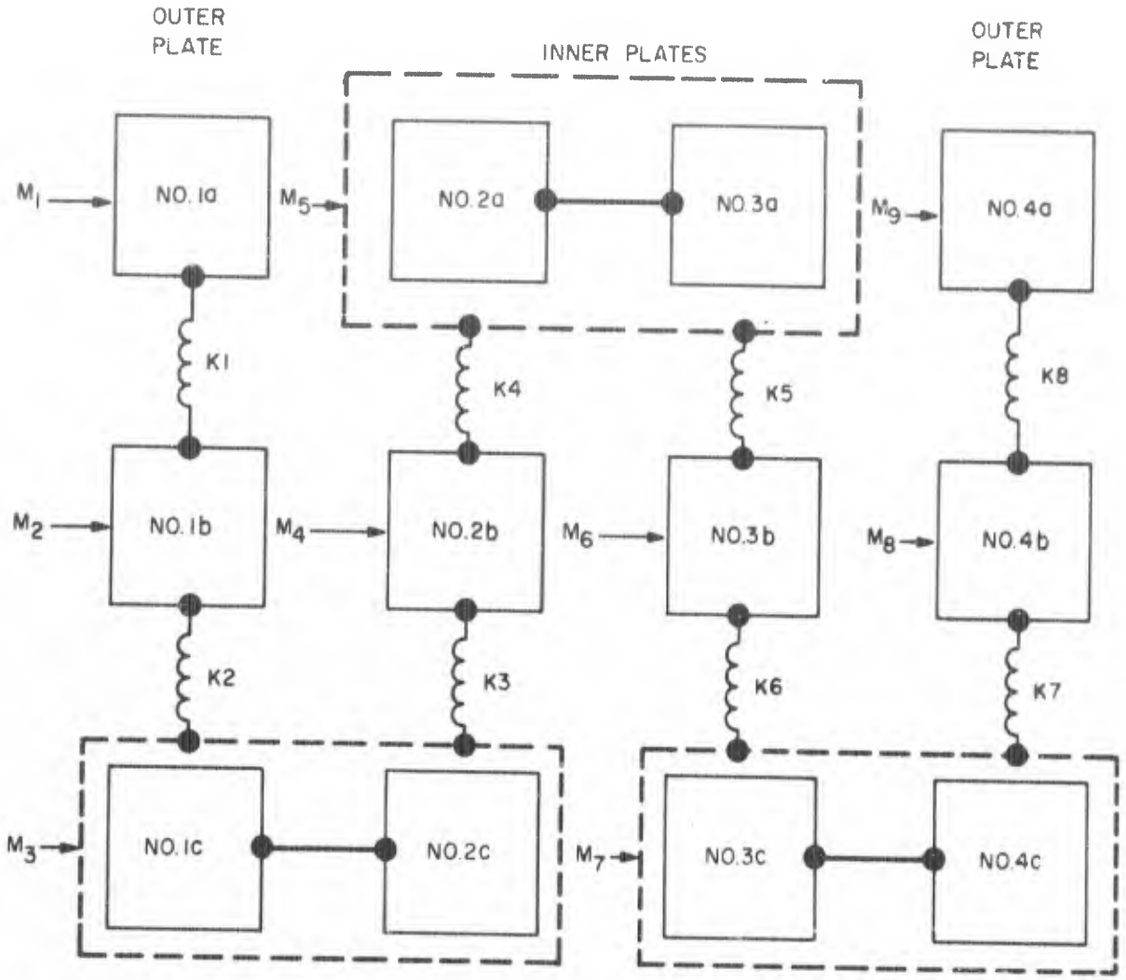

Figure 2-4. Analytical Model of Spring-Mass Baffle with Compliant Outer Plates

SECTION 3

ANALYSIS OF COMPLIANT SPRING-MASS BAFFLE

The area of each baffle plate is divided into three regions as shown in figure 3-1. The interiors of the squares marked A constitute region A, the interiors of the squares marked C constitute region C, and the area exterior to the squares constitutes region B. The centers of the A squares lie over the pins supporting the plate from below; if the plate supports another set of pins, these stand on the C squares. The squares are of such a size that region A and region C each constitute 25 percent of the plate area. Each region has its own amplitude and phase of vibration normal to the plane of the plate. The complex velocities of these vibrations will be denoted by VA, VB, and VC, respectively.

To study the interaction between the vibrating plate and sound waves in the adjacent water, the velocity at the plate surface will be expanded in the form

$$\sum\sum V_{mn}\, e^{iq\,(mx + ny)}$$

where $q = \pi/(2^{1/2}d)$ and d is the side of the squares as shown in figure 3-1. The first step is to expand the characteristic functions of regions A and C. (The characteristic function of a region equals 1 inside the region and 0 outside.)

Let the origin be at the center of an A square. A single period of the pattern in figure 3-1, showing only the A squares, has the appearance of figure 3-2. It can be shown that the characteristic function of region A is

$$CFA = (1/4) \qquad \sum\sum S_m\, S_n\, e^{iq\,(mx + ny)}$$

where terms having m + n odd are omitted and

$$S_a = \frac{\sin aqd/2}{aqd/2} .$$

When only the C squares are shown, a single period of the pattern in figure 3-1 has the appearance of that shown in figure 3-3. The characteristic function CFC of region C has the same expansion as CFA, except that odd subscript terms are changed in sign.

The velocity at the baffle face may be expressed as

$$VB + (VA - VB)\,CFA + (VC - VB)\,CFC$$

so that

$$Voo = .25\,(VA + 2VB + VC),$$

while for even m and n not both zero

$$Vmn = .25\,(VA - 2VB + VC)\,Sm\,Sn$$

and for odd m and n

$$Vmn = .25\,(VA - VC)\,Sm\,Sn$$

When m + n is odd, Vmn vanishes.

In the water next to the baffle face, the velocity may be expressed as a series

$$V = \sum\sum Zmn\, e^{iq\,(mx + ny)}$$

where the coefficients Zmn are functions of z and must satisfy the wave equation

$$Z''/Z = q^2\,(m^2 + n^2) - k^2$$

where k is the wave number.

Two kinds of boundary condition will be considered: in the first, the baffle faces a heavy steel plate across a narrow water gap; in the second, the extent of the water is unlimited.

Let the width of the water gap be H and take z = o at the face of the heavy steel plate, with the positive z-axis pointing toward the baffle. Let Zmn have the form

$$Amn \sin(Wmnz) + Bmn \cos(Wmnz)$$

where $Wmn^2 = k^2 - q^2\,(m^2 + n^2)$. Both the real and imaginary parts of Wmn are assumed non-negative.

Assume that the velocity VP at the face of the heavy steel plate is independent of x and y. Except for m = n = o, Bmn must vanish. Matching velocities at the baffle surface give

$$Amn = Vmn \csc(WmnH).$$

The remaining coefficients are given by

$$Boo = VP$$
$$Aoo = Voo \csc(kH) - VP \operatorname{ctn}(kH).$$

For the second kind of boundary condition, let z = o at the baffle face with the positive z-axis pointing away from the baffle. Express Zmn in the form

$$C_{mn}\, e^{iW_{mn} z} + D_{mn}\, e^{-iW_{mn} z}.$$

For positive imaginary Wmn, the requirement of boundedness forces Dmn to vanish. For positive real Wmn, the requirement that any incident wave be planar again forces Dmn to vanish unless m = n = o; even then, Doo vanishes unless there is an incident wave. Thus Dmn vanishes and

$$C_{mn} = V_{mn}$$

except that, if there is an incident wave of (particle) velicity VI,

$$D_{oo} = VI$$

$$C_{oo} = V_{oo} - VI$$

Table 3-1 summarizes the forms taken by Zmn for various boundary conditions. Also shown are the coefficients in the expansion

$$P = \sum\sum P_{mn}\, e^{iq\,(mx + ny)}$$

where P is the pressure in the water. The pressure can be obtained from the velocity using the basic relation

$$\partial P/\partial z = i\omega\rho V$$

The average pressure on an A square can be calculated by integrating P in the x, y, plane. The result is

$$P_A = \sum\sum P_{mn}\, S_m\, S_n,$$

with Pmn evaluated at the baffle face. The expression for PC, the average pressure on a C square, is the same except that odd subscript terms are changed in sign. The average pressure PB on region B is given by the relation

$$P_{oo} = .25\,(P_A + 2\,P_B + P_C).$$

The pressures PA, PB, PC may be expressed in terms of the velocities VA, VB, VC by replacing the coefficients Pmn in the pressure expansions by their values taken from table 3-1 and then replacing the Vmn by their expressions in terms of the velocities.

The result in the case of the narrow water gap is

$$PA = PH + H2\,(VA - 2VB + VC) + H1\,(VA - VC)$$

$$PB = PH - H2\,(VA - 2VB + VC)$$

$$PC = PH + H2\,(VA - 2VB + VC) - H1\,(VA - VC)$$

where

$$PH = ic\rho VP \csc(kH) - ic\rho \frac{VA + 2VB + VC}{4} \operatorname{ctn}(kH)$$

and H1, H2 are defined as subsums of the series

$$-i\omega\rho\,.25 \sum\sum (Sm^2\,Sn^2/Wmn) \operatorname{ctn}(WmnH).$$

H2 is the sum of all terms having m and n even but not both zero, and H1 is the sum of all terms having m and n odd.

When the water is of unlimited extent, the result is

$$PA = PF + F2\,(VA - 2VB + VC) + F1\,(VA - VC)\,.$$

$$PB = PF - F2\,(VA - 2VB + VC)$$

$$PC = PF + P2\,(VA - 2VB + VC) - F1\,(VA - VC)$$

where

$$PF = c\rho \frac{VA + 2VB + VC}{4} - 2c\rho\ VI$$

and F1, F2 are subsums of the series

$$\omega\rho\,.25 \sum\sum Sm^2\,Sn^2/Wmn.$$

F2 is the sum over terms having m and n even but not both zero, and F1 is the sum over terms having m and n odd.

Equations of Motion

The baffle model to be analyzed is shown in figure 3-4. The mounting plate is optional. Odd-numbered masses represent either single A and C squares of plate or two such squares on adjacent plates, pinned together. Even-numbered masses, except M10, represent plate regions B.

For use in the equations of motion, quantities Mn will represent mass area ratios, the masses all being divided by the area d^2 of a single A or C square.

The springs shown in figure 3-4 will be characterized by force per volume ratios Kmn equal to the force per unit distance, required to displace mass Mn relative to mass Mn, divided by the area d^2.

The equations of motion for the 10 masses of figure 3-4 are given in table 3-2. If the mounting plate is not used, the last equation should be omitted. The pressure terms may be eliminated using the pressure/velocity relationships developed above.

At the right end of the assembly, the boundary condition of unlimited water applies so that

$$P1 = PF + F2\,(V1 - 2V2 + V3) + F1\,(V1 - V3)$$

$$P2 = PF - P2\,(V1 - 2V2 + V3)$$

$$P3 = PF + F2\,(V1 - 2V2 + V3) - F1\,(V1 - V3)$$

where

$$PF = c\rho\,\frac{V1 + 2V2 + V3}{4} + 2PI$$

and $PI = -c\rho\ VI$ is the pressure of the incident wave from the right.

When the mounting plate is not used, the pressures P7, P8, and P9 are given by the equations for P1, P2, and P3, respectively, with V1, V2, and V3 replaced by -V7, -V8, and -V9, respectively, and PI replaced by the pressure PJ of the incident wave from the left.

With the mounting plate in place, the boundary condition of the narrow water gap applies at the left end of the assembly, and

$$P7 = PH + H2\,(V7 - 2V8 + V9) + H1\,(V7 - V9)$$

$$P8 = PH - H2\,(V7 - 2V8 + V9)$$

$$P9 = PH + H2\,(V7 - 2V8 + V9) - H1\,(V7 - V9)$$

where

$$PH = ic\rho\ V10\ \csc(kH) - ic\ \frac{V7 + 2V8 + V9}{4}\ \mathrm{ctn}\,(kH).$$

In the last equation of table 3-2, P10 is the value of Poo at the face of the mounting plate:

$$P10 = ic\rho\ V10\ \mathrm{ctn}\,(kH) - ic\rho\ \frac{V7 + 2V8 + V9}{4}\ \csc(kH).$$

The pressure PO is related to the incident-wave pressure PJ and the reflected-wave pressure PR by the equations

$$PO = PJ + PR$$

$$c\rho\ V10 = PJ - PR$$

which show PO = 2PJ - $c\rho$ V10. This is the quantity to be measured when the mounting plate is in place. With the mounting plate removed, the desired quantity is the value of Poo at the baffle face. To summarize, the quantities to be measured are:

$$2PJ - c\rho\ V10 \qquad \text{(mounted)}$$

$$2PJ - c\rho\ \frac{V7 + 2V8 + V9}{4} \qquad \text{(isolated).}$$

TABLE 3-1

FORMS OF Zmn AND Pmn

$Wmn^2 = k^2 - q^2 (m^2 + n^2)$

NARROW WATER GAP

Zmn = Vmn sin (Wmnz)/sin (WmnH)

Pmn = $-i\omega\rho$ (Vmn/Wmn) cos (Wmnz)/sin (WmnH)

Zoo = (Voo csc (kH) - VP ctn (kH)) sin (kz) + VP cos (kz)

Poo = $ic\rho$VP sin (kz) - $ic\rho$ (Voo csc (kH) - VP ctn (kH)) cos (kz)

UNLIMITED WATER

Zmn = Vmn exp (iWmnz)

Pmn = $\omega\rho$ (Vmn/Wmn) exp (iWmnz)

Zoo = (Voo - VI) exp (ikz) + VI exp (-ikz)

Poo = $c\rho$(Voo - VI) exp (ikz) - $c\rho$VI exp (-ikz)

(if no incident wave, set VI = 0)

TABLE 3-2

BASIC EQUATIONS OF MOTION

$-i\omega M1V1 = iK12/\omega (V2 - V1) - P1$

$-i\omega M2V2 = iK23/\omega (V3 - V2) + iK12\ \omega (V1 - V2) - 2P2$

$-i\omega M3V3 = iK34/\omega (V4 - V3) + iK23\ \omega (V2 - V3) - P3$

$-i\omega M4V4 = iK45/\omega (V5 - V4) + iK34\ \omega (V3 - V4)$

$-i\omega M5V5 = iK56/\omega (V6 - V5) + iK45\ \omega (V4 - V5)$

$-i\omega M6V6 = iK67/\omega (V7 - V6) + iK56\ \omega (V5 - V6)$

$-i\omega M7V7 = iK78/\omega (V8 - V7) + iK67/\omega (V6 - V7) + P7$

$-i\omega M8V8 = iK89/\omega (V9 - V8) + iK78\ \omega (V7 - V8) + 2P8$

$-i\omega M9V9 = iK89/\omega (V8 - V9) + P9$

$-i\omega M10V10 = 4 (PO-P10)$

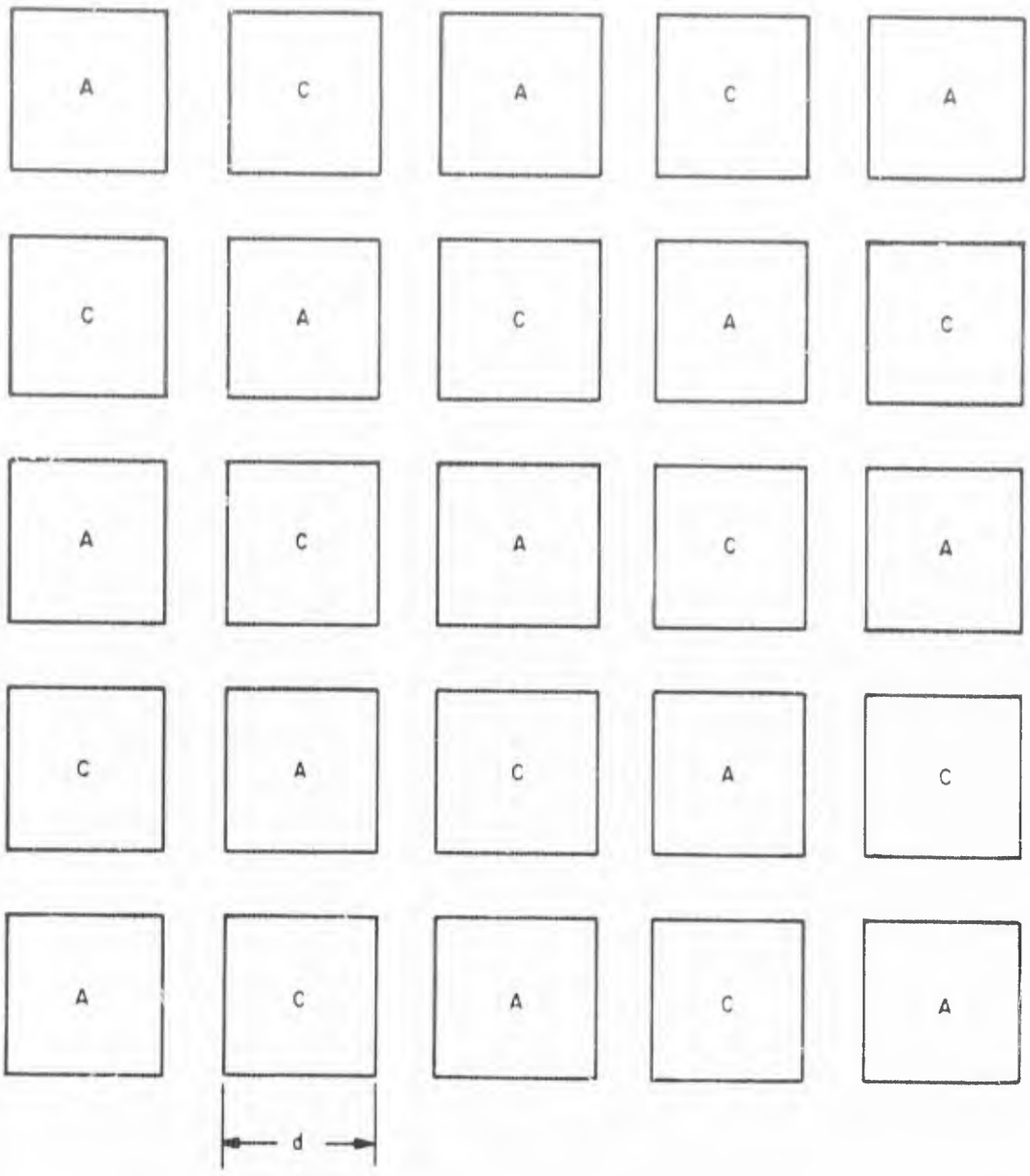

Figure 3-1. Regions on Surface of Baffle Plate

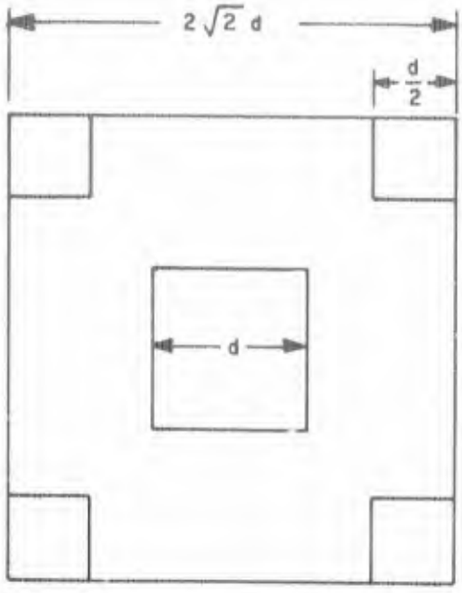

Figure 3-2. Pattern of A Squares

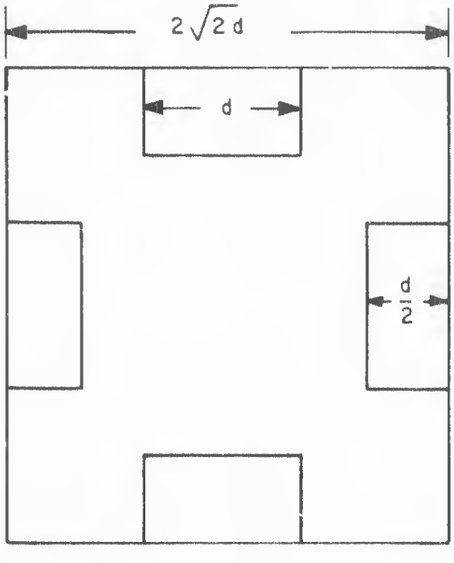

Figure 3-3. Pattern of C Squares

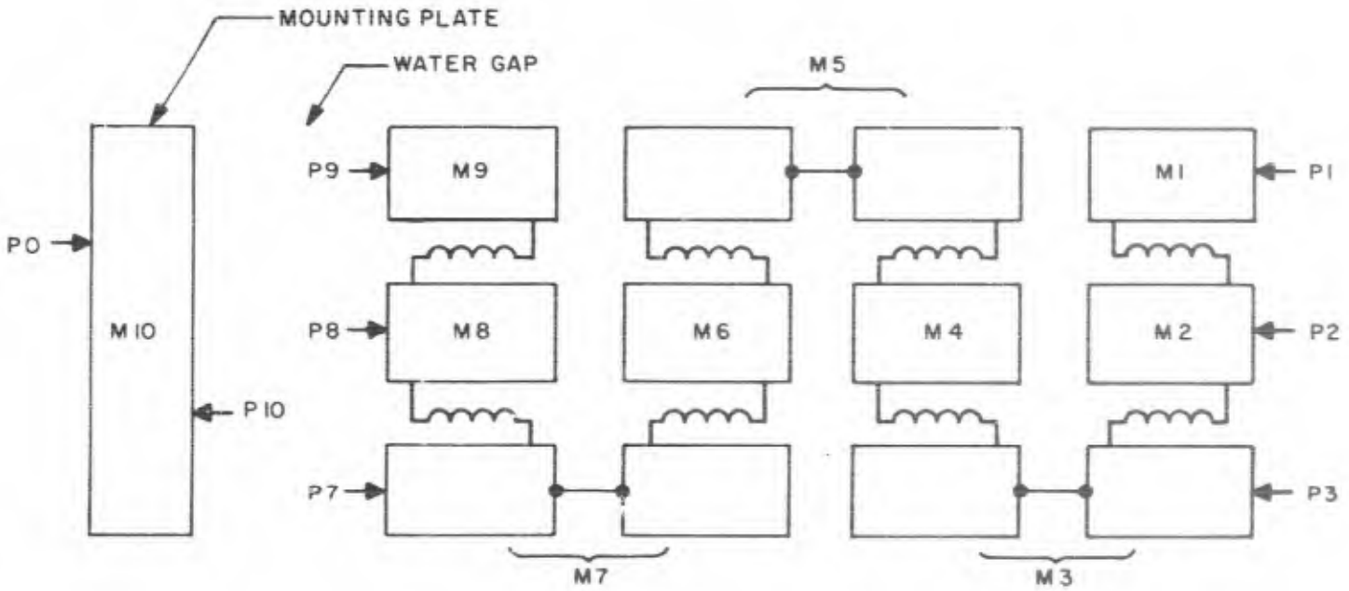

Figure 3-4. Analytical Model of Baffle Assembly Including Hydrophone Mounting Plate

SECTION 4

RESULTS

The equations derived in this report were developed into a computer program and a number of cases were examined. A complete listing of the program is given in table 4-1. The results of the computer runs are illustrated in figures 4-1 and 4-2.

An explanation of the symbols used in the figures is as follows: in all cases the figures pertain to a configuration consisting of a spring-mass baffle mounted a short distance behind a 1-1/2-inch thick steel plate. The plotted points show the pressure at the front of this steel plate as a function of frequency for the two cases of a sound wave incident at the front and the back of the plate-baffle combination. With sound incident from the front, the resulting curve shows the receiving response of a hydrophone mounted on the steel plate. With the sound incident from the back, the resulting curve shows the effectiveness of the baffle in shielding the hydrophone from noise originating behind the baffle. The latter curve is related to insertion loss but is a more practical evaluation of baffle performance since it includes the mounting arrangement for the hydrophone.

In figure 4-1 GAP refers to the distance (in inches) between the steel plate and the baffle. The designations AM, AK and AL pertain to the mass, spring constant and spring damping, respectively, of the masses and springs which comprise the baffle model as shown in figure 2-3. The relatively large mass value(.001104) is that of the 1-1/2-inch steel hydrophone mounting plate. The third spring, whose value is given as 0 in the figures, does not appear in figure 2-3. This spring was placed in the model to account for any compliance between the steel plate and the baffle contributed by the mounting arrangement. It was found in practice, however, that mounting arrangements produce little such compliance. The frequency scale is logarithmic to allow a wide frequency range to be covered on one plot. The reference for 0 dB is the incident sound wave.

In figure 4-2 (4 sheets), the label PIN SPACE refers to the spacing (in inches) between the pins which separate the various plates in the baffle. WATER GAP (in inches) is the distance between the steel plate and the baffle. The MASSES, SPRINGS, and DAMPING are the mechanical parameters of the baffle. The 10 mass values are those for the 10 masses that comprise the baffle model as shown in figure 3-4. Similarly, the eight values for spring constants and damping are for the eight springs in the model. The legend for the plotted points is given at the bottom of the page. Since there are two frequency scales, four curves are needed to display the data. As in the other figures 0 dB is referenced to the level of the incident wave.

Figure 4-1 shows the performance of a baffle in which the outer plates are assumed noncompliant. The mass values correspond to 1/8-inch plastic outer plates and 1/16-inch plastic inner plates. The spring values correspond to operation at low hydrostatic pressures. Note that the crossover point, at which the pressures corresponding to front and back incidence become equal, lies at 45 Hz.

Figure 4-2 (sheet 1 of 4) shows the same configuration, this time analyzed, using the new program written for compliant outer plates, but constrained so as to simulate the original program. Only two of the eight springs are assumed appreciably compliant. As might be expected, the results are effectively the same as in figure 4-1.

Figure 4-2 (sheet 2 of 4) shows the results when two additional springs are allowed to be appreciably compliant, and agrees with figure 4-1. Because of differences in model structure and normalization, the spring and damping parameters are generally eight times as large in the figures generated by the compliant outer plate program than in those generated by the rigid plate program, but the identical physical situation of 1/8-inch plastic outer plates and 1/16-inch inner plates is represented.

Figure 4-2 (sheet 3 of 4) shows the situation when all springs are assumed appreciably compliant. This is the case of compliant outer plates. Since the stiffness varies as the cube of the thickness, the outer plates are eight times as stiff as the inner plates. The insertion loss is improved perhaps by 1 dB at lower frequencies and 4 dB at higher frequencies. Note the dip near 2000 Hz caused by the compliance of the outer plate.

Figure 4-2 (sheet 4 of 4) shows the results for the case where the compliance of the outer plates is increased to match that of the inner plates. Further improvements are evident in the insertion loss, and the dip due to outer-plate compliance has moved down to 700 Hz. From this data it can be concluded that small but definite gains in performance can be made using compliant outer plates in the baffle.

Figures 4-3 through 4-5 compare theoretical data calculated by the compliant outer plate computer program with experimental data obtained from tests of a spring-mass baffle and associated hydrophone array. The array tested consisted of a 6 by 6 foot, 1-1/2-inch thick steel plate in which the hydrophones were mounted; a 1-1 2-inch thick rubber sheet was bonded to the front of the plate, and the spring-mass baffles were placed behind the hydrophone mounting plate.

The figures show the receiving response of the hydrophone mounted in the center of the steel plate at water depths of 100, 400 and 700 feet. The parameters that describe the baffle in the computer program were varied to take into account the effect of depth on the baffle. Agreement at the lower frequencies is very good. At the higher frequencies, the experimental data exhibit ripples which are probably due to the finite size of the array. Especially noteworthy is the correct prediction by the theory of a dip in the response in the vicinity of 4.5 kHz.

TABLE 4-1
PROGRAM LISTING

```
0
 01000      PRØGRAM BAFIII(INPUT,ØUTPUT,TAPE1,TAPE6)
 01010      DIMENSIØN DB(64,2,2),DBO(2),FDATA(3,10),FHEAD(7,10)
 01020+     ,GLAB(2),IFMT(2),ISCALE(3),ISYM(10),LINE(72),LINREF(72)
 01030+     ,LEG(2,2,2),MLAB(2),NF(2),MARK(2,2,2),VAL(10,3),XA(3,3,3)
 01040      CØMPLEX A(10,12),HF(3,2),Z
 01050      CØMMØN /PARAM/WN,GEØM(2) /SS/SS(50)
 01060      DATA PI,CC,RC/3.14159265358979,60630.,5.84/
 01070      DATA FDATA/1.84510,.0166667,2.84510, 2.84510,.0166667,3.84510,
 01080+     24*0./
 01090      DATA ISCALE/3,0,0/
 01100      DATA ISYM/8H11335577,9*0/
 01110      DATA XA/1.,0.,-1., 0.,0.,0., -1.,0.,1., 1.,-2.,1., -2.,4.,-2.,
 01120+     1.,-2.,1., 1.,2.,1., 2.,4.,2., 1.,2.,1./
 01130      T1= .5*PI*SQRT(.5)  $  SS(1)= 1.
 01140      DØ 90 N=1,49
 01150   90SS(N+1)= (SIN(N*T1)/(N*T1))**2
 01160      REWIND 1  $  REWIND 6  $  NPLØT= 0
 01170  100READ(1,500) (LINE(L),L=1,8)  $  IF(EØF,1) 260,110
 01180      500FØRMAT(8A10)
 01190  110DECØDE(10,510,LINE) I,J  $  IF(I.EQ.1H*) GØTØ 100
 01200      510FØRMAT(3XA1,I1)
 01210      GØTØ(120,120,120,140,150,260) J
 01220*     READ MASSES, SPRINGS, DAMPING
 01230  120ENCØDE(20,520,IFMT) ISCALE(J)
 01240      520FØRMAT(1H(,I2,17HP,5X3I1,8(I2,F6)))
 01250      DECØDE(80,IFMT,LINE) K,L,M,(N,VAL(N,J),NN=1,K)
 01260      IF(L.EQ.0) GØTØ 160
 01270*     SYMMETRY FEATURE IS ØPTIØNAL
 01280      ENCØDE(10,530,IFMT) ISYM(L)
 01290      530FØRMAT(A10)
 01300      DECØDE(10,540,IFMT) (LINE(N),N=1,10)
 01310      540FØRMAT(10I1)
 01320      KK= 9-J/2
 01330      DØ 130 N=1,KK  $  NN= LINE(N)
 01340  130VAL(N,J)= VAL(NN,J)  $  GØTØ 160
 01350*     READ GEØMETRIC DATA
 01360  140DECØDE(80,550,LINE) K,L,M,(N,GEØM(N),NN=1,K)  $  GØTØ 160
 01370      550FØRMAT(5X3I1,2(I2,F6))
 01380*     READ MØDE/PLØT DATA
 01390  150DECØDE(80,560,LINE) NUMF,MØDE,LØGLIN,(NF(N),DBO(N),N=1,NUMF)
 01400      560FØRMAT(5X3I1,2(I2,F6))
 01410  160IF(I.NE.1HE) GØTØ 100
 01420      NPLØT= NPLØT+1
 01430*     FREQUENCY LØØP
 01440      DØ 200 NN=1,NUMF  $  K= NF(NN)
 01450      F1= FDATA(1,K)  $  F2= FDATA(2,K)
 01460      NUMI= 1.5+(FDATA(3,K)-F1)/F2
 01470      DØ 200 I=1,NUMI
 01480      T1= F1+(I-1)*F2
 01490      W= 2.*PI*10.**T1*(2-LØGLIN)+2000.*PI*T1*(LØGLIN-1)
 01500      WN= W/CC  $  WR= WN*RC
 01510      CALL KWIKØ(HF)
 01520      HF(1,1)= HF(1,1)*CMPLX(0.,WR)
 01530      HF(2,1)= HF(2,1)*CMPLX(0.,WR)
 01540      HF(3,1)= CMPLX(0.,.25*RC/TAN(WN*GEØM(2)))
```

TABLE 4-1 (Cont'd)
PROGRAM LISTING

```
0
 01550      HF(1,2)= HF(1,2)*CMPLX(WR,0.)
 01560      HF(2,2)= HF(2,2)*CMPLX(WR,0.)
 01570      HF(3,2)= CMPLX(.25*RC,0.)
 01580*     ENTER MATRIX VALUES
 01590      DØ 170 J=1,10   $  DØ 170 K=1,12
 01600  170A(J,K)= (0.,0.)
 01610      DØ 180 J=1,10   $  IF(J.GT.8) GØTØ 180
 01620      Z= CMPLX(VAL(J,3),VAL(J,2)/W)
 01630      A(J,J)= A(J,J)+Z   $  A(J,J+1)= -Z
 01640      A(J+1,J+1)= A(J+1,J+1)+Z   $  A(J+1,J)= -Z
 01650  180A(J,J)= A(J,J)+CMPLX(0.,-W*VAL(J,1))
 01660      Z= CMPLX(0.,-RC/SIN(WN*GEØM(2)))
 01670      DØ 190 J=1,3
 01680      A(J,12)= -XA(J,2,3)   $  A(J+6,11)= (MØDE-1)*XA(J,2,3)
 01690      A(J+6,10)= Z*XA(J,1,3)   $  A(10,J+6)= Z*XA(J,1,3)
 01700      DØ 190 K=1,3   $  DØ 190 L=1,3
 01710      A(J,K)= A(J,K)+XA(J,K,L)*HF(L,2)
 01720  190A(J+6,K+6)= A(J+6,K+6)+XA(J,K,L)*HF(L,MØDE)
 01730      A(10,11)= 8.   $  A(10,10)= A(10,10)+16.*HF(3,1)+16.*HF(3,2)
 01740*     FIND PRESSURES
 01750      CALL SØLV(A,11-MØDE)
 01760      DØ 200 J=1,2   $  T1= 4-2*J
 01770      Z= T1-RC*((2-MØDE)*A(10,J+10)+(MØDE-1)*.25*(A(7,J+10)
 01780+     +2.*A(8,J+10)+A(9,J+10)))
 01790  200DB(I,J,NN)= -2000.+2.*AINT(1000.5+10.*ALØG10(CABS(Z)))
 01800*     PLØT
 01810      DATA FHEAD/40H .07       .10       .15     .20        .,
 01820+     24H30          .50       .70,
 01830+                 40H .70      1.00      1.50    2.00       3.,
 01840+     24H00          5.00      7.00,56*0./
 01850      DATA GLAB/10H PIN SPACE,10H WATER GAP/
 01860      DATA (LEG= 10HFRØM FRØNT,10HFRØM  BACK,0,0,
 01870+     2(10HFRØM FRØNT,10HFRØM  BACK))
 01880      DATA MLAB/7HMØUNTED,8HISØLATED/
 01890      DATA MARK/1H+,1H*,0.,0.,1HX,1H#,1H+,1H*/
 01900*     WRITE HEADING
 01910      II= 4+NUMF-MØDE   $  JJ= 6+MØDE
 01920      ENCØDE(110,600,LINE) II,JJ
 01930      600FØRMAT(10H(4H----I2,I1,29H(/)25X20HPRESSURE AT FRØNT ØF
 01940+     5H/21XA11,25H,21H BAFFLE CØNFIGURATIØN
 01950+     27H/25X19H(DB//INCIDENT WAVE)))
 01960      WRITE(6,LINE) NPLØT,MLAB(MØDE)
 01970      II= 3-MØDE   $  JJ= 11-MØDE
 01980      ENCØDE(100,610,LINE) II,JJ
 01990      610FØRMAT(1H(I1,24H(/26XA10,F6.2)/6HMASSES/I2,12H(3PF7.4)/7HS
 02000+60HPRINGS/8(0PF7.0)/7HDAMPING/8(0PF7.2)//28X15HFREQUENCY (KHZ)))
 02010      WRITE(6,LINE) (GLAB(I),GEØM(I),I=1,II),(VAL(J,1),J=1,JJ)
 02020+     ,((VAL(J,L),J=1,8),L=2,3)
 02030      DØ 210 I=1,NUMF   $  J=NF(I)
 02040      ENCØDE(70,620,LINE) (FHEAD(K,J),K=1,7)
 02050      620FØRMAT(7A10)
 02060      WRITE(6,630) (LINE(K),K=1,7)
 02070      630FØRMAT(4X7A10)
 02080      DECØDE(70,640,LINE) LINE
 02090      640FØRMAT(70A1)
```

TABLE 4-1 (Cont'd)
PROGRAM LISTING

0

```
      DØ 210 L=2,70
  210LINREF(L)= LINE(L).AND.(LINE(L).EQ.1H.)
      NN= 0   $  KK= NUMF+NUMI   $  LL= 2+NUMI
      LINE(1)= 4H(DB)   $  LINE(LL)= 5H (DB)
      T1= DBO(1)   $   T2= DBO(2)
      ENCØDE(20,650,IFMT) NUMI   $  GØTØ 230
      650FØRMAT(4H(A5,I2,6HA1,A5))
*     WRITE ØNE LINE (LØØP)
  220IF(AMØD(DBO(1),10.).NE.0.) GØTØ 230
      ENCØDE(4,660,LINE) DBO(2)   $  LINE(LL)= LINE(1)
      660FØRMAT(F4.0)
      ENCØDE(4,660,LINE) DBO(1)
  230DØ 240 I=1,NUMI
      LINE(I+1)= LINREF(I+1)
      IF(LINE(1).NE.0) LINE(I+1)= 1H.
      DØ 240 J=1,2   $  DØ 240 N=1,NUMF
      IF(DB(I,J,N).EQ.DBO(N)) LINE(I+1)= MARK(J,N,NUMF)
  240CONTINUE
      WRITE(6,IFMT) (LINE(K),K=1,KK)
      LINE(1)= 0   $  LINE(LL)= 0
      DBO(1)= DBO(1)-2.   $  DBO(2)= DBO(2)-2.
      NN= NN+1   $  IF(NN.LT.36) GØTØ 220
*     WRITE TAIL
      WRITE(6,670)
      670FØRMAT()
      IF(NUMF.EQ.2) WRITE(6,680)
      680FØRMAT(5X17HUPPER/LEFT SCALES,[illegible]/RIGHT SCALE [illegible]
      JJ= 5-NUMF   $  ENCØDE(40,690,LINE) JJ
      690FØRMAT(28H2(8XA10,1XA1,31XA10,1XA1,/),I1,4H(/)))
+     I1,31H(19H INBØARD INCIDENCE A1,16X),I1,4H(/)))
      WRITE(6,LINE)  ((LEG(I,J,NUMF),MARK(I,J,NUMF),J=1,2),I=1,2)
      DBO(1)= T1   $  DBO(2)= T2   $  GØTØ 100
*     EXIT
  260WRITE(6,700)
      700FØRMAT(4H----)
      ENDFILE 6   $  REWIND 6
      END
      SUBRØUTINE KWIKØ(HF)
      CØMPLEX HF(3,2)
      DIMENSIØN R(2)
      CØMMØN /PARAM/WN,S,H /SS/SS(50)
      DATA ACC,PI/.0001,3.14159265358979/
      DATA R/.309325,.088187/
      Q= (PI/S)*SQRT(2.)   $  QQ= 2.*Q**2
      HF(1,1)= CMPLX(-R(1)/Q,0.)   $  HF(2,1)= CMPLX(-R(2)/Q,0.)
      HF(1,2)= CMPLX(0.,-R(1)/Q)   $  HF(2,2)= CMPLX(0.,-R(2)/Q)
      K= 0   $  IZ= 0   $  E= .75   $  WW= WN*1.E-6
  100K= K+1   $  N= 2
      DØ 130 II=IZ,K
      I= K-II
      T1= (2-I/K-(K-I)/K)*SS(K-I+1)*SS(K+I+1)
      T3= [illegible](K**2+I**2)
      T2= WN**2-T3   $  T3= SQRT(T3)
      IF(T2.LT.0.) GØTØ 110
      T2= SQRT(T2)   $  IF(T2.LT.WW) T2= WW
```

TABLE 4-1 (Cont'd)
PROGRAM LISTING

```
0
 08180     HF(N,2)= CMPLX(T1/T2,T1/T3)+HF(N,2)
 08190     HF(N,1)= CMPLX(T1/T3+T1/T2/TANH(H*T2),0.)+HF(N,1)
 08200     GOTO 120
 08210 110T2= SQRT(-T2)    IF(T2.LT.WW) T2= WW
 08220     HF(N,2)= CMPLX(0.,T1/T3-T1/T2)+HF(N,2)
 08230     HF(N,1)= CMPLX(T1/T3-T1/T2/TANH(H*T2),0.)+HF(N,1)
 08240 120E= E-T1
 08250 130N= 3-N
 08260     IF(T2.LE.WW) GOTO 100
 08270     IF(ABS(E/T3-E/T2/TANH(H*T2))
 08280+    .GT.ACC*CABS(HF(1,1)+HF(2,1))) GOTO 100
 08290     IF(ABS(E/T3-E/T2).GT.ACC*CABS(HF(1,2)+HF(2,2))) GOTO 100
 08300     RETURN
 08310     END
 09000     SUBROUTINE SOLV(A,M)
 09010     COMPLEX A(10,12),SUM,TOT
 09020     DIMENSION KR(10,2)
 09030     DATA KR/1,1,1,3,4,5,6,7,7,7, 3,3,4,5,6,7,10,10,10,10/
 09040     DO 150 N=1,M
 09050     SUM= (0.,0.)
 09060     J1= KR(N,1)   $  J2= N-1
 09070     IF(J1.GT.J2) GOTO 110
 09080     DO 100 J=J1,J2
 09090 100SUM= SUM+A(N,J)*A(J,N)
 09100 110A(N,N)= 1./CSQRT(A(N,N)-SUM)
 09110     K1= N+1   $  K2= MINO(M,KR(N,2))
 09120     IF(K1.GT.K2) GOTO 150
 09130     DO 140 K=K1,K2
 09140     SUM= (0.,0.)   $  TOT= (0.,0.)
 09150     J1= KR(K,1)   $  J2= N-1
 09160     IF(J1.GT.J2) GOTO 130
 09170     DO 120 J=J1,J2
 09180     SUM= SUM+A(K,J)*A(J,N)
 09190 120TOT= TOT+A(N,J)*A(J,K)
 09200 130A(K,N)= (A(K,N)-SUM)*A(N,N)
 09210 140A(N,K)= (A(N,K)-TOT)*A(N,N)
 09220 150CONTINUE
 09230     DO 170 I=11,12
 09240     DO 170 L=1,2
 09250     DO 170 K=1,M
 09260     SUM= (0.,0.)
 09270     N=  (2-L)*K       +(L-1)*(M+1-K)
 09280     J1= (2-L)*KR(N,1)+(L-1)*(N+1)
 09290     J2= (2-L)*(N-1)   +(L-1)*MINO(M,KR(N,2))
 09300     IF(J1.GT.J2) GOTO 170
 09310     DO 160 J=J1,J2
 09320 160SUM= SUM+A(N,J)*A(J,I)
 09330 170A(N,I)= (A(N,I)-SUM)*A(N,N)
 09340     RETURN
 09350     END

 LENGTH =  215 LINES
```

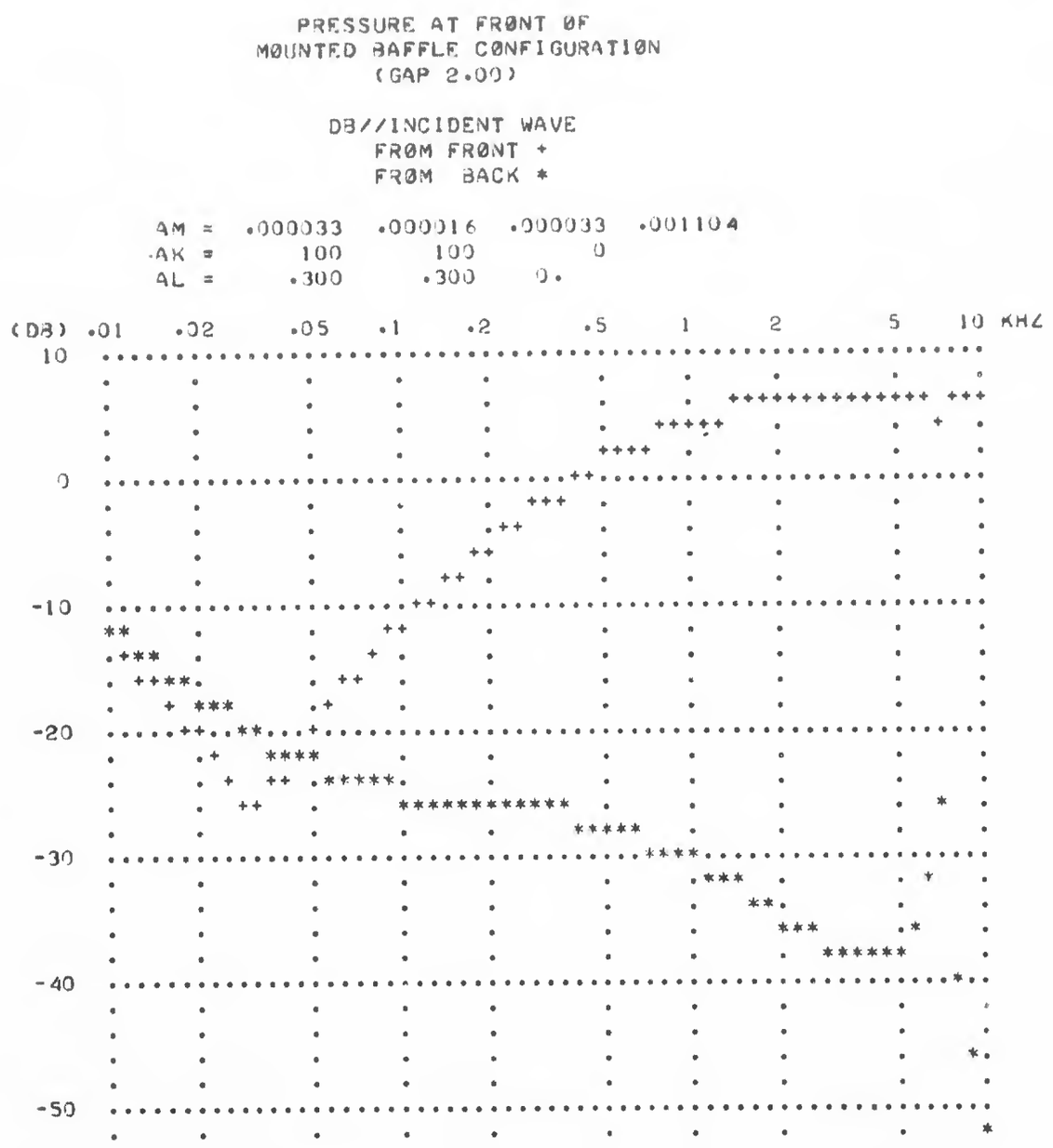

Figure 4-1. Computed Performance of Spring-Mass Baffle (Rigid Outer Plates)

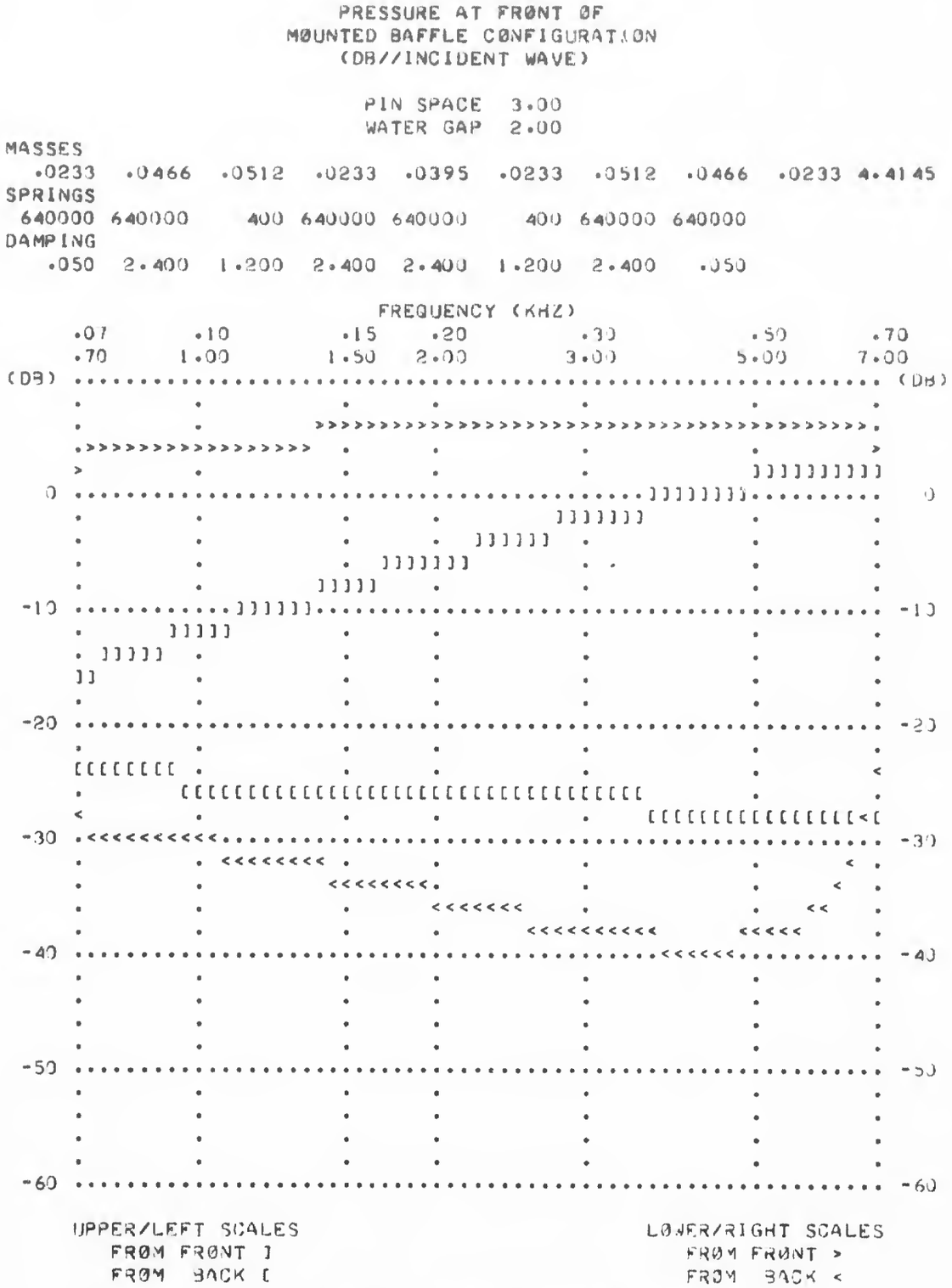

Figure 4-2. Computer Performance of Spring-Mass Baffle (Compliant Outer Plates) (Sheet 1 of 4)

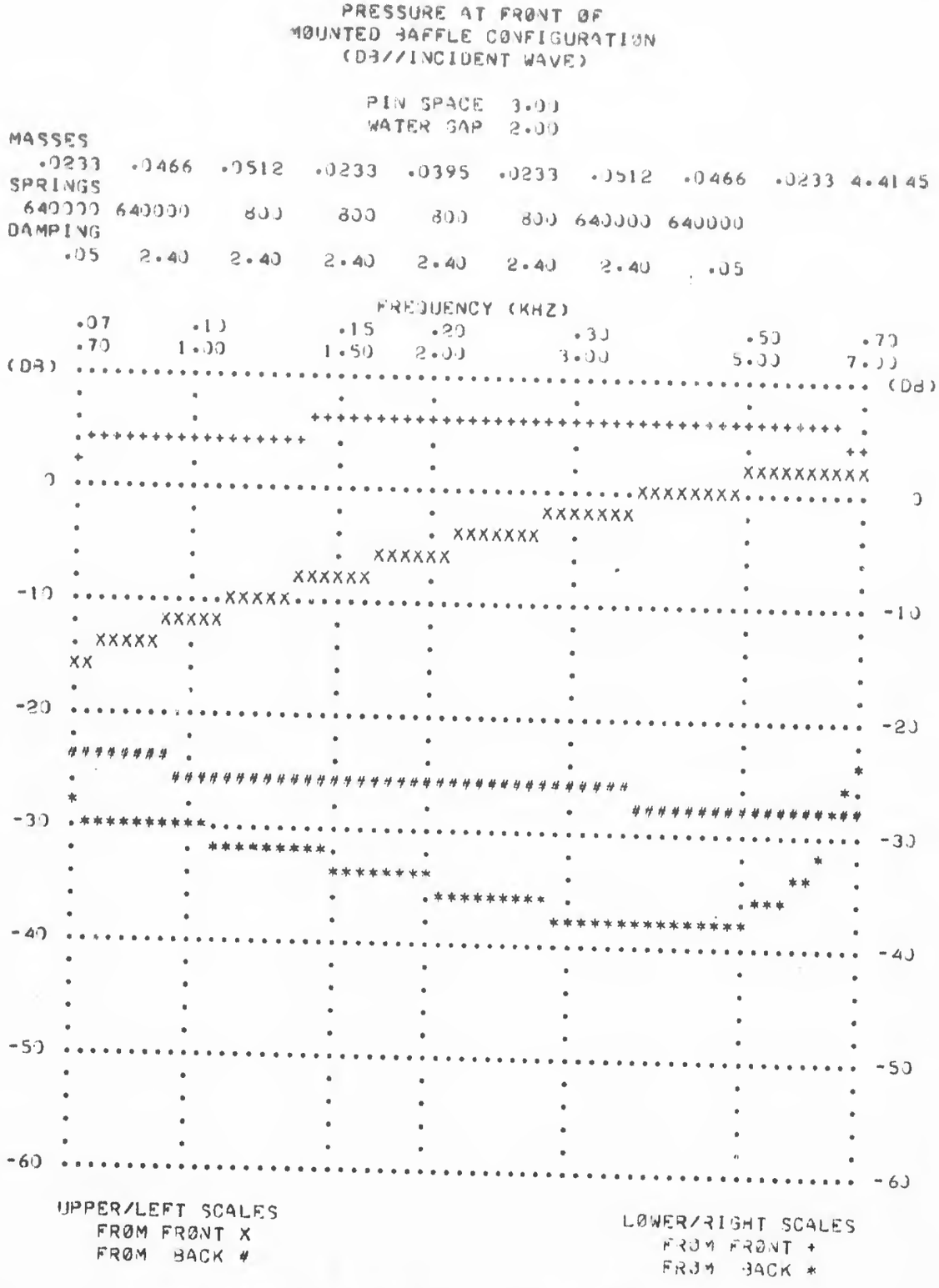

Figure 4-2. Computer Performance of Spring-Mass Baffle (Compliant Outer Plates) (Sheet 2 of 4)

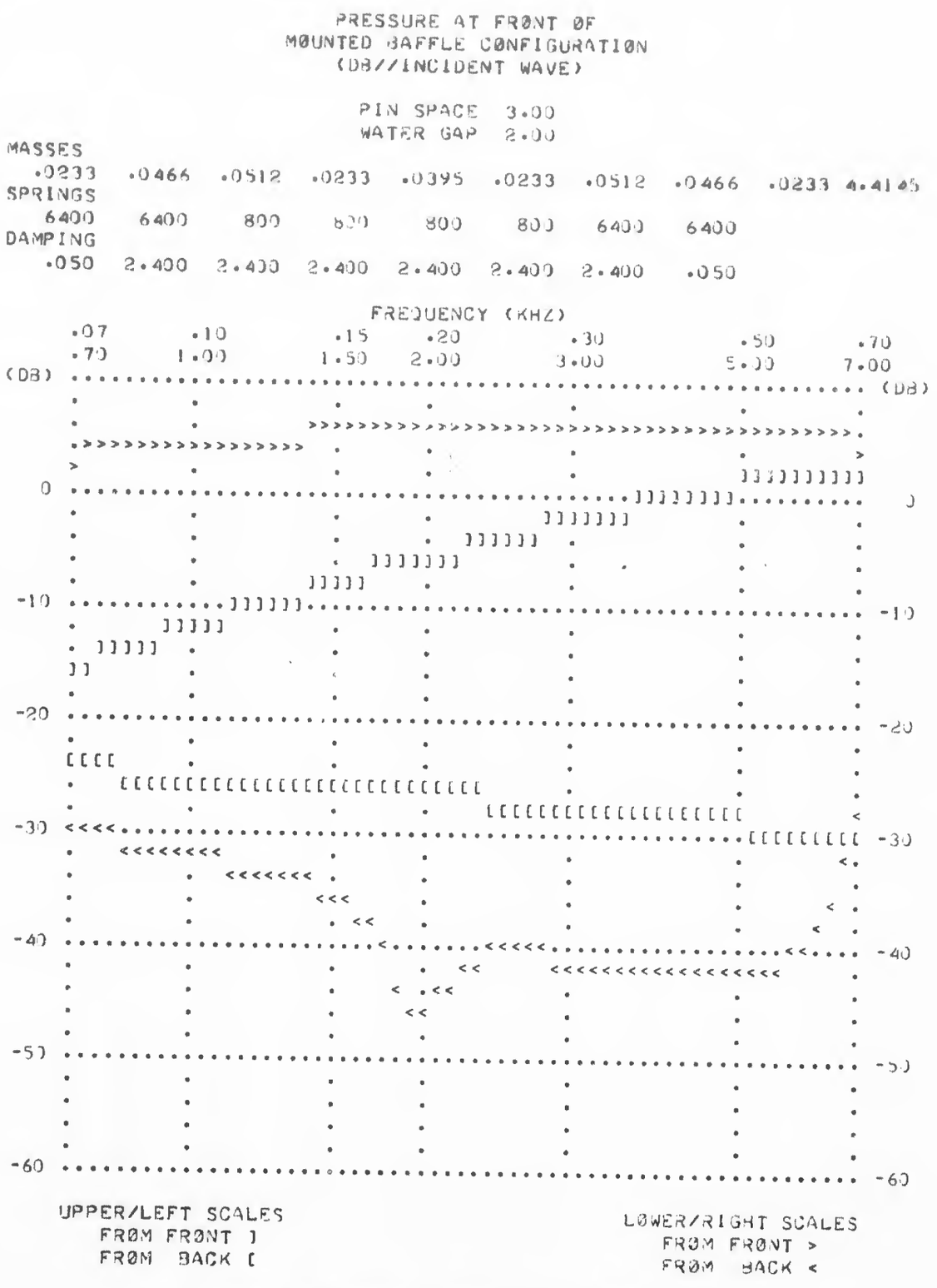

Figure 4-2. Computed Performance of Spring-Mass Baffle (Compliant Outer Plates) (Sheet 3 of 4)

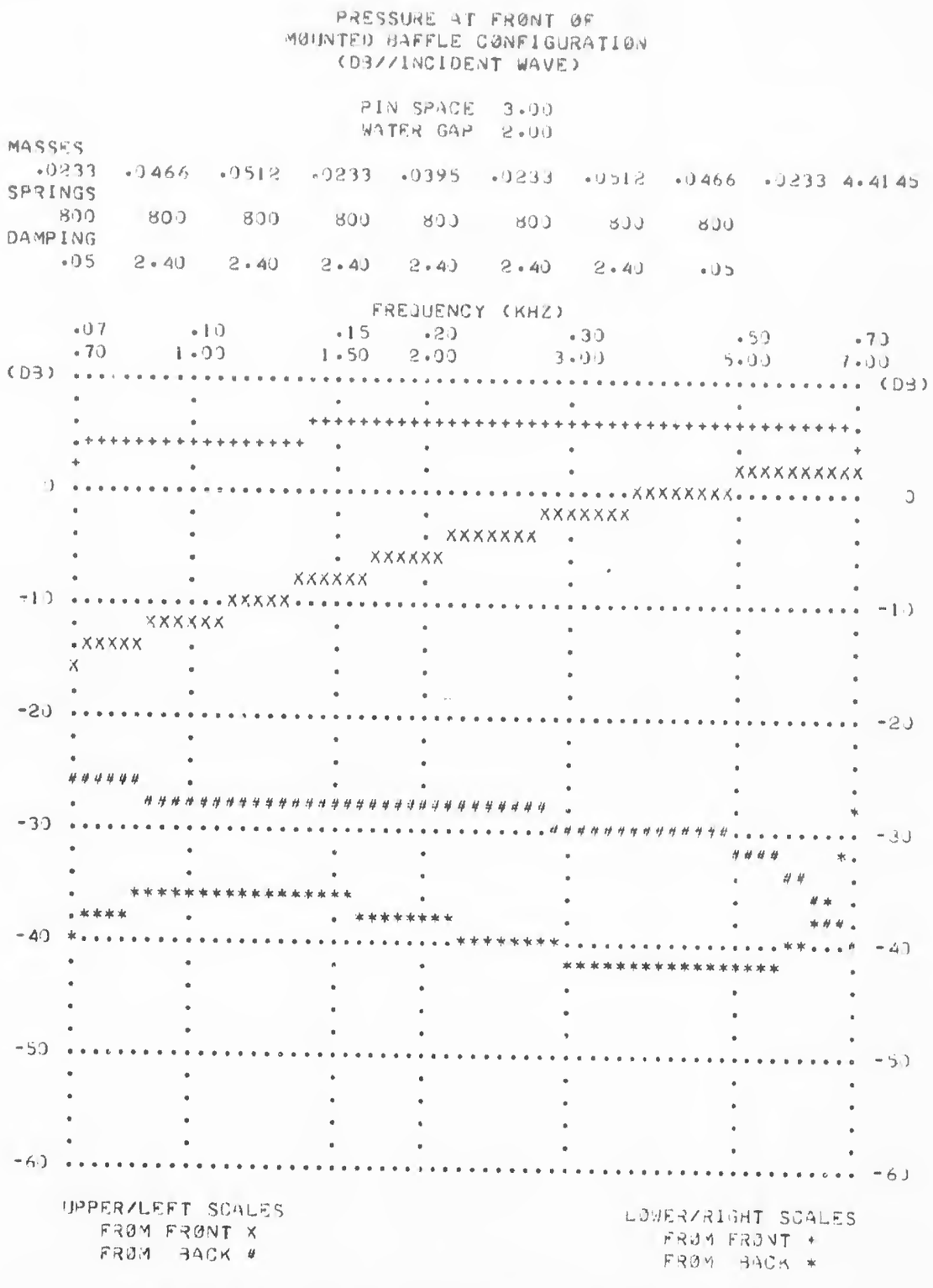

Figure 4-2. Computed Performance of Spring-Mass Baffle (Compliant Outer Plates) (Sheet 4 of 4)

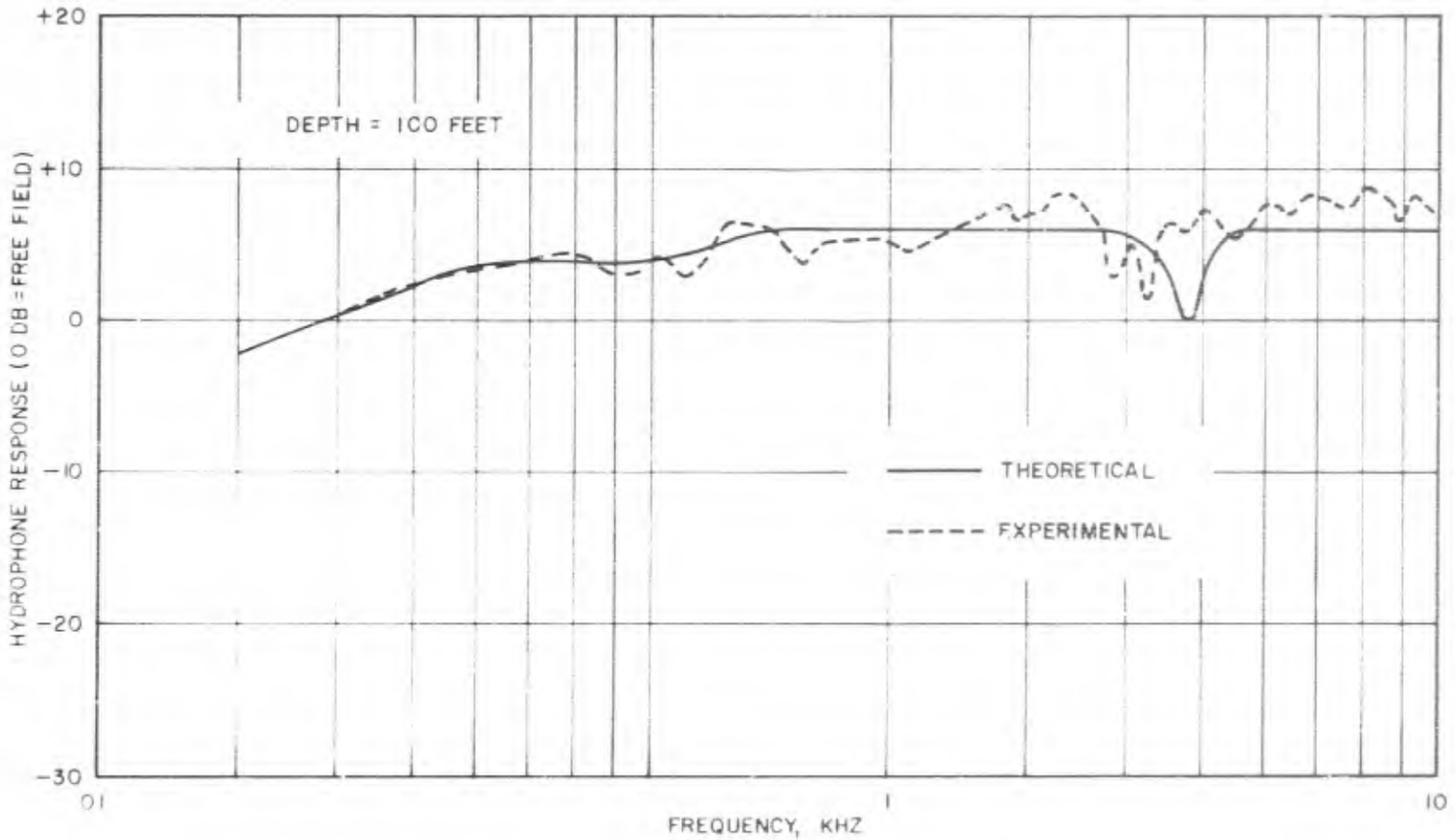

Figure 4-3. Comparison of Computed and Experimental Baffled Hydrophone Response (Depth = 100 Feet)

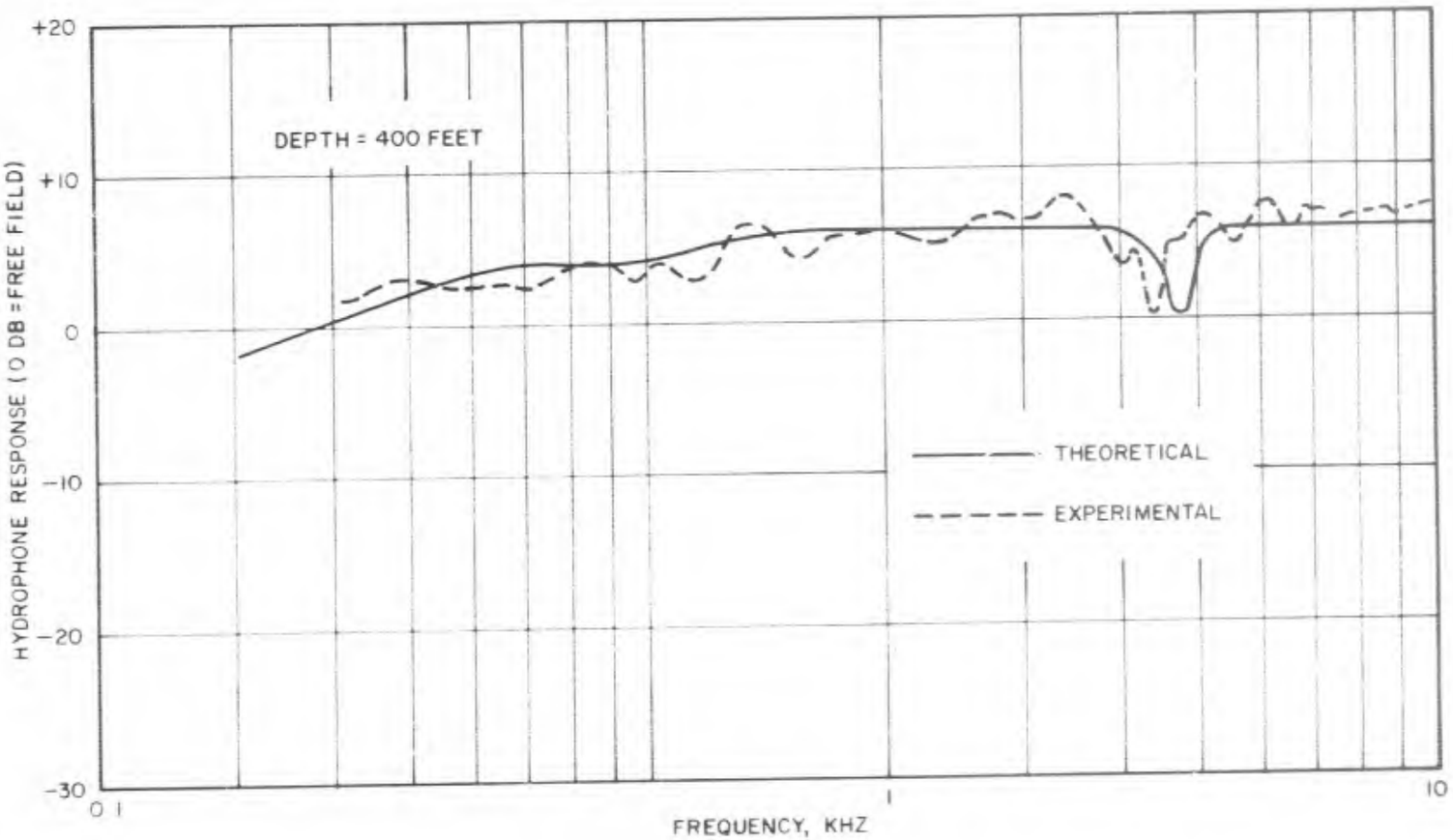

Figure 4-4. Comparison of Computed and Experimental Baffled Hydrophone Response (Depth = 400 Feet)

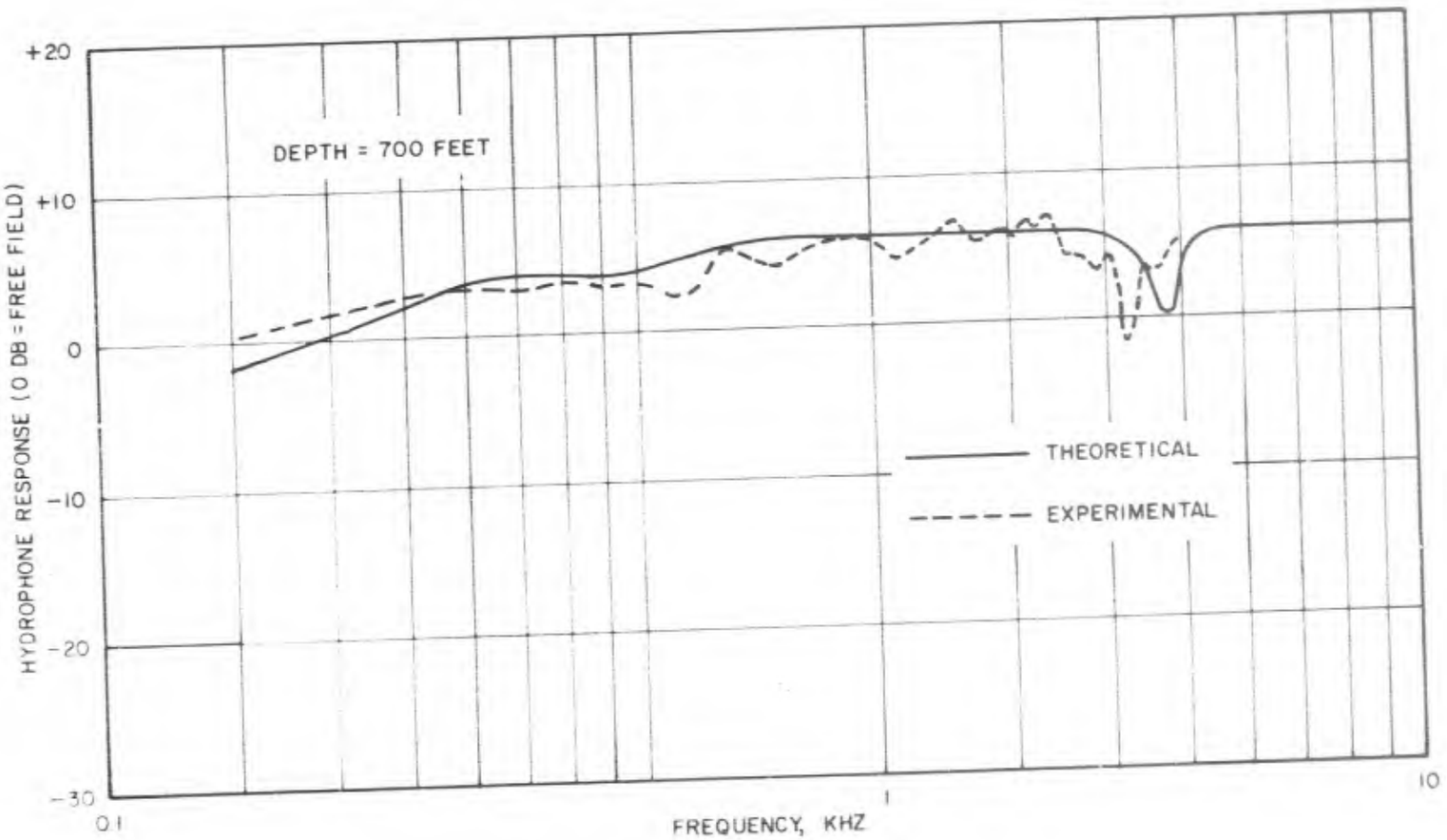

Figure 4-5. Comparison of Computed and Experimental Baffled Hydrophone Response (Depth = 700 Feet)

SECTION 5

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3. Analysis of Acoustic Baffles for Underwater Noise Reduction, Sperry Gyroscope Division, SGD-4230-0430, June 1971.

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13. ABSTRACT

This is an interim report on the study of underwater noise reduction techniques being conducted by the Sperry Division for the Office of Naval Research, Acoustic Programs, under Contract No. N00014-67-C-0303. Included in this report is a description of the work being done on spring-mass type acoustic baffles. Specifically, the effects of rigid and compliant outer plates are analyzed, theoretical results are compared with experimental data, and good agreement is obtained. It is concluded that significant gains can be achieved at the lower frequencies through the use of compliant outer plates in the baffle design.

14 KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
Acoustic baffles Waterborne noise reduction						

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