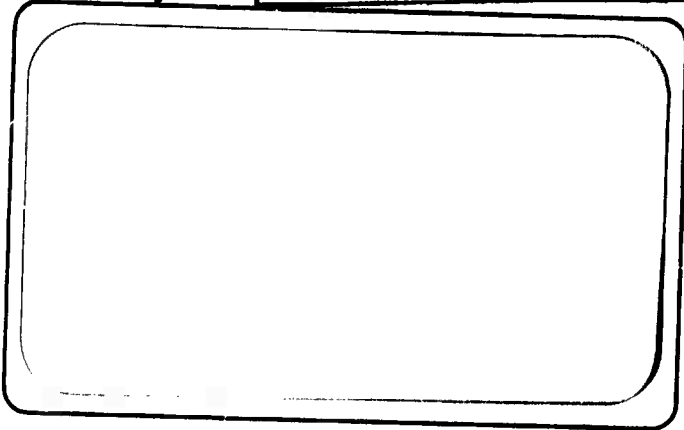


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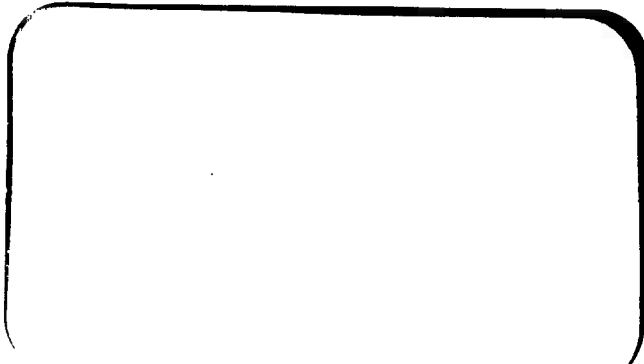
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Vibration Transmission of Flexible Pipe Couplings
Influenced by the Elements of
a Trim-Pump System

Assignments 67 102 & 72 105
MEL R&D Report 263/66
September 1966

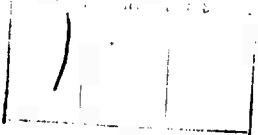
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ABSTRACT

An analytical-experimental study was performed to determine the sound transmission characteristics of a flexible coupling in a trim-pump system. Mechanical impedance methods were used to compare the structural path to the hull via resilient mounts and foundation to the structural path via flexible coupling and pipe hanger. Also investigated were the vibration characteristics of an Electric Boat and MEL flexible coupling under pressurized and unpressurized conditions.

Results showed that the sound transmission via flexible coupling and pipe hanger exceeded the sound transmission via the mount and foundation and would negate the isolation effectiveness of the primary mounting system. The results of comparing the two types of flexible coupling showed that the Electric Boat Coupling exhibited better total noise attenuation characteristics than did the MEL coupling. However, the MEL coupling showed a unique inherent characteristic of improved attenuation capabilities with increased pressure loading.

ADMINISTRATIVE INFORMATION

This investigation was authorized by reference (a) under NAVSHIPS (formerly BUSHIPS) Sub-project S-F013 11 08, Task 01352, and by reference (b) under NAVSHIPS Sub-project S-F113 11 09, Task 03955.

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Appendix A - Development of System Response Matrix (3 pages)

Appendix B - Development of Mount-Foundation-Hull Vibration Response Equation

Appendix C - Free Mobility and Impedance Measurements of System Elements (7 pages)

Appendix D - Technical References

VIBRATION TRANSMISSION OF FLEXIBLE PIPE COUPLINGS
INFLUENCED BY THE ELEMENTS OF
A TRIM-PUMP SYSTEM

1.0 INTRODUCTION

Designing submarine machinery components to withstand deep submergence, yet maintaining uncompromising efforts in effecting proper structureborne vibration isolation, is a problem of dominant importance to the Navy.

1.1 Problem: The Noise Transmission Path. Submarine radiated waterborne noise caused by vibration transmission from machinery to hull via foundations has been effectively controlled by use of resilient vibration isolation mounts. Due to the effectiveness of these isolators, alternate sound transmission paths, such as electrical power cable bundles, flexible pipe couplings, flexible exhaust stacks, and pipe hangers, were revealed. The relatively high rigidity of these elements "short circuit" and negate the effectiveness of the principal isolators to an extent sufficient to cause the transmitted noise to dominate a portion of the radiated noise spectrum.

1.2 Prior Work. Experimental evaluation of isolation devices is deemed possible^{1,2} if the measured transfer impedance across the isolation device is small compared to the driving point impedance at both interfaces where the isolation device terminates. Vibration transmission studies of two isolation devices, a flexible snorkel exhaust joint³ and an electrical power cable bundle⁴ have been made. The transfer impedance of the elements, when compared to the transfer impedance of the principal isolation mount used in the system, showed that both the exhaust joint and the power cable bundle possessed sufficient rigidity to become the controlling factor in the isolation effectiveness of the resiliently mounted machinery system.

1.3 Objectives. Deep submergence submarines impose greater reliability demands on sea-water piping systems because they must withstand the stresses induced by the depth pressures. These demands require structurally stronger flexible pipe couplings. Strength, in turn, tends to increase the vibration transmission properties of this type of isolation device. One objective of this investigation was to perform a system analysis to determine the overall vibration transmission properties of a flexible coupling inserted in a typical trim-pump system. Another objective was to compare the vibration transmission characteristics of two 4-inch flexible pipe coupling assemblies under pressurized and unpressurized conditions. A third objective was to verify, experimentally, the analytical-experimental approach used to fulfill the first two objectives.

2.0 INVESTIGATION

The experimental-analytical approach, based on the work of Smith,^{5,6} was used to investigate the vibration transmission properties of flexible pipe couplings when

¹Superscripts refer to similarly numbered entries in Appendix D.

mathematically inserted in a machinery system, the vibration properties of which are known or assumed, parameters.

2.1 Mathematical Mobility Model. Figure 1 is a schematic representation of the discharge side of a typical trim pump machinery system composed of a mounted pump, flexible pipe coupling, and a run of pipe transporting the pressurized discharge liquid. The numbered terminals located at the interfaces of each free-body represent orthogonal axes at which uncoupled forces and velocities are developed. The vibrational characteristics of each element in Figure 1 can be represented by free-body equations which involve the uncoupled velocities and forces at each numbered terminal and the elements' experimentally measured translational mobilities. The development of these equations is derived in Appendix A. In the derivation, the uncoupled elements were connected into a machinery system by application of mathematical boundary conditions to the response equations defining the free elements. Solutions of the resulting set of linear simultaneous equations yielded the coupled forces and velocities developed at each interface.

2.2 Utilization of Solutions. The coupled forces and velocities provide a means for defining the vibration transmission characteristics of the entire system. Transfer impedances from flexible coupling input, Terminals 4, 5, and 6, to pipe hanger-hull interface, Terminal 15, were calculated and power summed to define the system response for a combined velocity excitation applied to the flexible coupling input. Refer to Figure 1 for location of numbered terminals. The power summed system transfer impedance is

$$Z_{15}^* = \sqrt{\left| \frac{F_{15}}{V_4} \right|^2 + \left| \frac{F_{15}}{V_5} \right|^2 + \left| \frac{F_{15}}{V_6} \right|^2} \dots\dots(1)$$

The force ratios were obtained by means of the computed forces at the pipe hanger-hull interface divided by the power - summed forces developed at the flexible coupling input interface. The force ratios of the system are defined as

$$T_{15}^* = \frac{F_{15}}{F_{i^*}} \text{ where } F_{i^*} = \sqrt{\left| F_4 \right|^2 + \left| F_5 \right|^2 + \left| F_6 \right|^2} \dots\dots(2)$$

They give an indication of the system vibration amplification or attenuation and are expressed in decibels. Use of this artificial integration with respect to time and space of the input responses to the flexible coupling yielded simplified yet meaningful system vibration characteristics.

2.3 Resilient Mount-Foundation-Hull Transmission Path. Paragraph 2.2 summarized the method of characterizing the vibration response of the flexible coupling-pipe hanger transmission path, the derivation of which is developed fully in Appendix A. The other sound transmission path which exists in mounted machinery items is the path described by the resilient mount, foundation and hull. This path, until recently, had been considered the dominant sound transmission path. As an extension of the work of Wright, 1 Darby⁷ proposed a means of evaluating the acoustic radiation of mounted machinery items, taking into consideration the sound path from machinery foot to hull via isolation mount and foundation, and the paths to the hull by way of "short circuiting" isolation devices. A less rigorous attack is used in this report due to the limitation of accessible

data. However, a meaningful comparison of the structural noise transmission between the two paths can be made to show the drastic "short circuiting" effects of flexible pipe couplings and to provide possible criteria for future flexible coupling design. The development of the vibration response equations defining the mount-foundation-hull path is presented in Appendix B. Figure 2 is a sketch of a trim pump mounted at four points on a symmetrical foundation. The approach used the known transfer impedance characteristics of the resilient mount, Z_{ab} ; measured impedance characteristics of the foundation, Z_{bc} (between foundation input, Terminal b, and foundation - hull interface, Terminal c); and the driving point impedance of the hull, Z_{cc} . Solution of the equation resulted in a power-summed transfer impedance, defined by the forces developed at the foundation - hull interface by a unity velocity excitation applied to the four mount inputs,

$$Z_T^* = 2 \left[\frac{Z_{ab} Z_{cc}}{Z_{bc}} \right] \quad \dots(3)$$

Therefore, comparison of Equation (3) with the power-summed transfer impedances of the flexible piping system, Equation (1), indicates the controlling sound path and provides an estimate of the increase in waterborne noise as a result of the domination of one of the paths.²

2.4 Data Acquisition. Solutions of Equations (1) through (3) were obtained using experimentally measured mobility responses of the flexible pipe coupling, trim pump and pipe hanger, and assumed piping impedance parameters. Representative hull termination and foundation impedances were collected from published material. All experimental and assumed data appears in Appendix C.

2.4.1 Experimental Measurements. One of the two flexible pipe couplings, arranged in the three planar configuration, is shown in Figure 3 resiliently suspended and free from end constraints. Sinusoidal excitation was applied along three orthogonal axes designated by Terminals 4, 5, and 6. The three free driving-point mobilities, m_{44} , m_{55} , and m_{66} , were measured with a Wilcoxon Model 820 impedance head. The nine remaining transfer mobilities were obtained using the force-measuring element of the impedance head and Clevite 25D21 accelerometers oriented at the output end of the configuration at the orthogonal terminals, 7, 8, and 9. The acceleration signals were time-integrated, ratioed with the force signals, and processed by the electronic system, the schematic of which is shown in Figure 4. Digitized mobility magnitudes and phase angles were recorded by the analog to digital instrumentation. In a similar manner, the three driving point mobilities at the trim-pump output flange were measured and digitized. The magnitudes and phase angles were used in the digital programming technique⁶ to solve the coupled system response matrix, Equation (A8) and Equations (1) and (2).

2.4.2 Given Data. The hull and foundation data necessary for the solution of Equation (3), i.e., Z_{cc} and Z_{bc} , are representative of measurements on a Fleet-type submarine. However, the curves were smoothed and averaged and do not represent responses for any particular submarine trim-pump foundation - hull combination. The blocked transfer impedance of the 6E2000 resilient mount, loaded to 1150 pounds, was used to define Z_{ab} . The foundation, hull, and mount data are shown in Appendix C.

3.0 COMPARISON OF TWO TYPES OF FLEXIBLE COUPLINGS

3.1 Approach. A method similar to that summarized in paragraph 2.1 and detailed in Appendix A was used to compare, quantitatively, the vibration characteristics of two types of flexible couplings. Free mobility measurements were taken of each flexible coupling type, unfilled at atmospheric pressure and water filled at 600 psig.* To maintain consistency with the philosophy reported in other studies of isolation devices,^{2,3,4,8} the output end of each coupling type was blocked. In this report, the blocking was mathematically accomplished by setting the velocity response equations, defining the output interface, to zero. Solutions of the blocked forces, developed at the output interface, ratioed to velocity and force inputs, yielded blocked transfer impedances and blocked force ratios.

3.2 Flexible Pipe Coupling Description. Two types of 4-inch-diameter flexible pipe couplings were investigated. One type, developed by Electric Boat (EB) and shown in the three-planar configuration, Figure 3, offered limited flexibility by means of a "ball and socket" arrangement, the "ball" or inner pipe spool being isolated and water sealed from the "socket" or outer casing by rubber. The other type, developed by MEL, was similar in external appearance to the EB coupling and provided a pressure equalization feature to maintain coupling operation if seal rupture occurred. Flexibility of the coupling was provided by a thin film of sampled pressurized fluid and two rubber "O" ring seals located between "ball" and nylatron lined "socket." Three MEL flexible couplings were arranged in a three-planar configuration.

4.0 VERIFICATION OF EXPERIMENTAL - ANALYTICAL TECHNIQUE

To verify the approach used in this investigation, it was decided to compare the measured transfer impedance of the EB flexible coupling when terminated into a finite impedance to the computed transfer impedance using the measured free mobility characteristics of the coupling and termination.

4.1 Direct Experimental Measurement. To measure the transfer impedance directly, a method previously presented^{4,8} was used. One end of the flexible coupling was bolted to a force measuring fixture in which four EB force gages were arranged in such a manner as to sense the forces normal to the plane passing through the flange interface defined by Terminal 7. The fixture, in turn, was attached to a high-impedance termination. A Goodmans vibration generator was attached to the free end of the coupling, Terminal 4, and sinusoidal velocity signals were measured by a Wilcoxon Model 820 impedance head. The force signals developed at each of the four force gages were individually ratioed to the velocity signals developed at the excited free end, Terminal 4. The resulting four transfer impedance curves were power summed to give one overall transfer impedance, characteristic of the system, and to maintain consistency with the technique employed in previous isolation device investigations.

*Abbreviations used in this text are from the GPO Style Manual, 1959, unless otherwise noted.

4.2 Experimental-Analytical Method. The experimental-analytical technique utilized the previously measured free driving point and transfer mobilities of the EB flexible couplings. In a manner similar to that described in paragraph 2.1, the velocity response equations involving the free mobility measurements of the flexible coupling and driving point impedance of the termination fixture were mathematically coupled at their interface, and a transfer impedance of the system was computed from the solutions of the coupled forces and velocities.

5.0 RESULTS

Figures 5 through 7 are force ratios (output force/power summed input force) of the pressurized EB flexible coupling. Figures 8 and 9 are the force ratios and transfer impedances of the trim pump system, influenced by the pipe hanger and piping parameters. In Figures 10 through 15, the blocked force ratios and transfer impedances of the EB and MEL flexible couplings, pressurized and unpressurized, are compared.

5.1 Flexible Coupling Force Ratios. The force ratios are observed to be influenced by the pipe hanger and piping constraints imposed upon the flexible coupling output terminals. The piping response characteristics were simulated by assigning constant impedance magnitudes to the piping input terminals to create an envelope representing piping antiresonant and resonant conditions. The impedance values are shown in Appendix C, Figure 7-C. Figures 5 and 7 show force ratios, T_7 and T_9 . Since the response of the flexible coupling in the directions indicated by Terminals 7 and 9 is independent of the pipe hanger constraints, Figure 16, the differences in force ratios are attributed solely to the piping parameters. Figure 6 depicts the force ratios, T_8 , defining Terminal 8. Little difference can be noticed between force ratios for resonant and antiresonant pipe responses throughout the frequency spectrum. This characteristic is explained when it is noted, Figure 7-C, Appendix C, that the pipe hanger driving point impedance, Z_{1313} , is consistently higher than the resonant piping impedance magnitude. The constraint offered the flexible coupling by the pipe hanger dominates that contributed by the piping. The dynamic forces developed at Terminal 8 are, therefore, relatively independent of the piping parameters, but are predominantly dependent upon the high impedance characteristics of the pipe hanger.

5.2 System Force Ratios. The force ratios, T_{15}^* , are shown in Figure 8. When comparing Figures 6 and 8, it is evident that the forces developed at Terminal 8, associated with the low (resonant) piping impedance parameter, undergo negligible attenuation between 20 and 200 Hertz (Hz). When the system force ratio associated with the high (antiresonant) piping response is considered, however, sufficient differences exist in impedances between pipe hanger and pipe, Figure 7-C, Appendix C, to provide considerable attenuation from 20 to 800 Hz. For frequencies greater than 800 Hz, the forces developed at Terminal 8, for both values of pipe impedances, are attenuated comparably. This indicates that the system force ratios, in this frequency range, are relatively independent of either resonant or antiresonant piping responses and are predominantly a function of the pipe hanger characteristics.

5.3 System Transfer Impedances. Figure 9 is a superposition of transfer impedances of the two sound transmission paths. Comparison of the impedance magnitudes yields a quantitative evaluation of the performance of the two isolation systems. The vibrational characteristics of the mount-foundation-hull sound path follow a dynamic stiffness line of approximately 3500 lb per in. from 20 to 170 Hz. At frequencies above

170 Hz, the response is predominantly influenced by the foundation-hull characteristics. Reasoning similar to that used in paragraph 5.2 explains the factors controlling the characteristics of the flexible coupling system transfer impedances. The importance of Figure 9 is to show the negating influence of the flexible coupling part of the system on the isolation effectiveness of the primary mounting system. It is seen that the two impedance curves of the flexible coupling-hanger system form an envelope describing the maximum and minimum impedance values possible with the assumed piping impedance parameters. At 100 Hz, the curve associated with the resonant piping condition indicates that maximum forces developed at the pipe hanger-hull interface exceed those developed at the hull-foundation interface by approximately 43 db. At frequencies above 1000 Hz, both curves coincide and extend below the impedance curve describing the primary mounting system.

5.4 Comparison of Flexible Coupling Types. Figures 10 through 15 are the blocked force ratios and blocked transfer impedances of the two types of flexible pipe coupling, pressurized and unpressurized. The curves were smoothed by computer to simplify comparison. It must be understood, however, that the results give only a comparative evaluation of the two types of flexible coupling and do not indicate performance when incorporated in a system, since rarely is the coupling terminated by blocked conditions.

5.4.1 The blocked force ratios, Figures 10 through 12, show that pressurizing the MEL flexible coupling decreases transmitted forces in the low- and mid-frequency range. This phenomenon can be explained by realizing that the MEL coupling's pressure equalizing feature provides a film of pressurized water between "ball and socket." When the film is considered as a resilient layer in series with the elastic nature of the nylatron "socket," a combined resilience of lower dynamic spring constant would result, thereby providing some vibration isolation. Above 1000 Hz, however, pressurization causes relatively high force transmission to extend to higher frequencies. The EB flexible coupling force ratio is relatively unaffected by pressure from 20 to 800 Hz. However, high force transmission extends into the higher frequency region, similar to that which occurred for the MEL coupling. Comparison of the flexible coupling blocked force ratios, pressurized and unpressurized, shows that the EB coupling yields lower force transmission characteristics across the frequency spectrum; the greatest difference between the two couplings, approximately 35 db, occurring between 400 and 5000 Hz for the unpressurized condition.

5.4.2 Both flexible couplings present similar transfer impedances, Figures 13 through 15, characterized by high antiresonant or blocked force magnitudes at approximately 1400 Hz and rapidly decreasing to minimum impedance magnitudes. Antiresonances or high impedances should be emphasized since, in these regions, maximum blocked forces are developed. If a comparison of Z_7^* , Z_8^* , and Z_9^* for each flexible pipe coupling configuration of like loading conditions is made, it is seen that the forces developed at Terminal 7 predominate over most of the frequency spectrum except at the high antiresonant region centered at 1000 Hz, where the forces developed at the three terminals are comparable in magnitude.

5.5 Comparison of Transfer Impedance Methods. Figure 16 is a comparison of transfer impedances of the EB flexible coupling configuration determined by experimental techniques and by the computational methods discussed in this report. Little corroboration of the two methods is apparent between 20 and 50 Hz. There is justification,

however, in stating that reasonable correlation exists between the two curves when the fact that the direct measurement involved translational and rotational responses, while the computed techniques consisted only of translational responses at Terminal 7. Lack of better corroboration can also be attributed to the generalized response characteristics which result from power summing the four individual transfer impedances in contrast to the point-by-point computed technique which results from the ratioing of discrete forces and velocities at Terminals 7 and 4. It is unfortunate that time did not allow for a more rigorous comparison of techniques. Assuming reciprocity and symmetry, to define the flexible coupling's mobility characteristics accurately, for 6 degrees of freedom, a minimum of 39 free mobility measurements would be required: the 12 translational mobilities, used in this report, plus 12 rotational and 15 translational-rotational mobilities. A similar series of measurements of the termination fixture would be necessary, defining the fixture by a minimum of 12 free mobilities. A system of equations of formidable size would result, but it is believed that solution of the matrix and computation of F_7/V_4 would better corroborate the experimentally determined transfer impedance, particularly between 20 and 50 Hz.

6.0 DISCUSSION

Wright's¹ philosophy regarding the generation of waterborne sound as a function of the dynamic forces transmitted to the hull is reflected in this investigation, where the characteristics of the system are defined in terms of its developed terminal forces.

6.1 Limitation. Darby² developed an approximation to predict the increase in radiated waterborne sound due to the controlling effect of one isolation device of high impedance when it was installed in a resiliently mounted machinery system. Applying the design curve of reference 2 at 100 Hz, a 36-db increase in radiated waterborne noise results when the flexible coupling transmission path is included compared to the primary mounting transmission path. Tempering this increase with the fact that hull radiation characteristics at the terminations of foundation - hull and pipe hanger - hull are apt to be dissimilar, it is still apparent that a serious limitation is imposed upon the effectiveness of the primary isolation system by the flexible coupling - pipe hanger transmission path.

6.2 System Corrective Measures. Assuming that, for strength, vibration, and shock purposes, the flexible coupling is of optimum design, certain corrective efforts can be applied to the system elements to reduce the forces transmitted to the hull, particularly for those frequencies at which the piping undergoes resonant conditions. Figure 7-C shows that the constant impedance magnitude describing the lower limit (resonance condition) of the piping response envelope lies entirely below the driving point impedance of the rubber pipe hanger, Z1313. The constraint offered by the hanger, therefore, results in the development of relatively high forces at the pipe hanger-pipe interface. Since, for all practical purposes, the force ratio of a hanger is unity, high forces are transmitted to the hull. To reduce the forces at the hull, one must use a pipe hanger of sufficiently low impedance to minimize the forces developed at the pipe hanger input terminal. As an example, a hanger exhibiting the impedance properties of a 6E100 resilient mount loaded to 100 pounds would exhibit a dynamic spring constant of approximately 310 lbs per in., Figure 7-C. Its impedance magnitudes lie below that describing the piping resonant condition, thereby offering an isolation potential to the forces developed at the flexible coupling output terminal.

6.3 Flexible Coupling Modifications. It is evident, from Figures 10 through 15, that neither type of flexible coupling is a satisfactory isolation device. The MEL coupling, however, exhibiting improved isolation characteristics when pressurized, shows potential if modifications are made. Improved isolation properties would be feasible if the nylatron lining forming the bearing surface in each casing or "socket" were replaced by a block rubber lining. The combined effect between pressurized water film and rubber would not only improve damping, but also result in a lower dynamic stiffness than presently exists between water film and nylatron liner.

7.0 CONCLUSIONS AND RECOMMENDATIONS

7.1 The conclusions derived from this study are:

- The block rubber pipe hanger and piping responses predominantly affect the vibration characteristics of the flexible coupling system. With the piping undergoing resonant conditions, the pipe hanger is ineffective in reducing the system force ratios between 20 and 200 Hz.
- Greater reduction of the system-transmitted forces would result if the block rubber pipe hanger were replaced by an isolation device of low dynamic stiffness terminated directly to the hull frame or a rigid structural member.
- The transfer impedance of the flexible coupling system exceeds that of the primary isolation mounting system, thereby negating the noise attenuation characteristics of the primary mounts.
- The EB flexible coupling exhibited better vibration characteristics than did the MEL flexible coupling.
- The MEL flexible coupling offers the unique feature of improved vibration attenuation with increased pressure loading. The combined effect of replacing the nylatron lining with rubber and the pressure sensitive attenuation feature, should improve the flexible coupling vibration isolation characteristics.

7.2 Recommendations are as follows:

- On the basis of the results of the system analysis, an isolation device of low impedance and properly terminated should be used in lieu of the existing block rubber pipe hanger, provided that measures are employed to constrain the system against excessive shock excursions.
- Since the MEL flexible coupling exhibited favorable pressure-dependent attenuation properties, efforts should be continued in its development for shipboard use, provided the coupling also presents favorable strength and shock-resistant characteristics.

8.0 FUTURE PLANS

It is planned to continue the investigation of flexible couplings as influenced by system elements. Since the trim pump system lends itself to realistic modeling, a study to optimize the system elements for minimum sound transmission would include the insertion of mobility measurements of various types of flexible couplings and hoses into the system matrix, along with improved pipe hanger characteristics.

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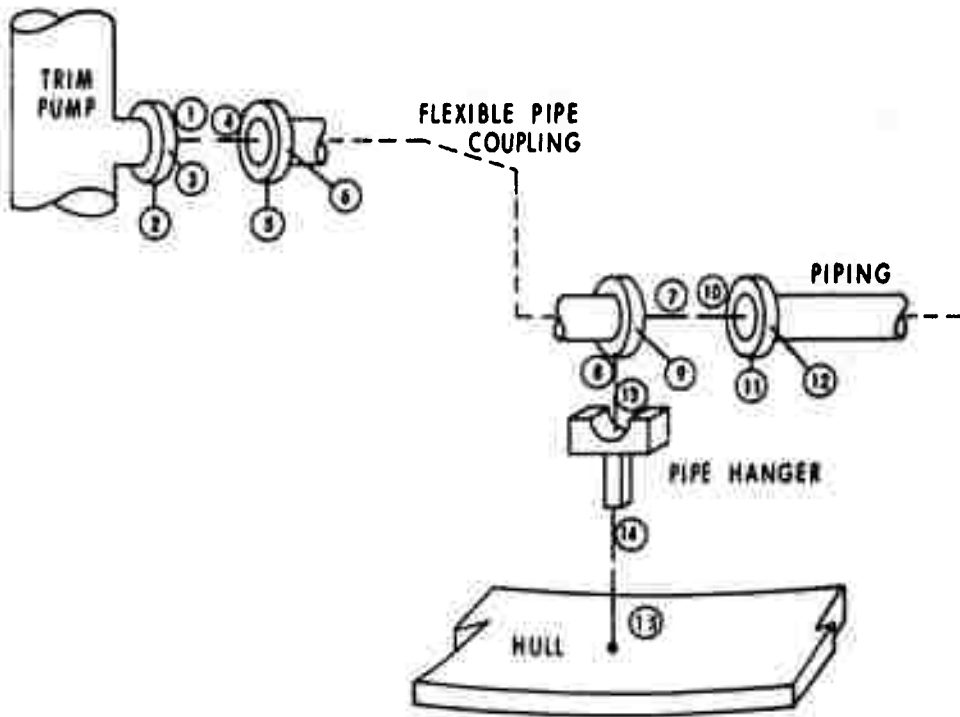
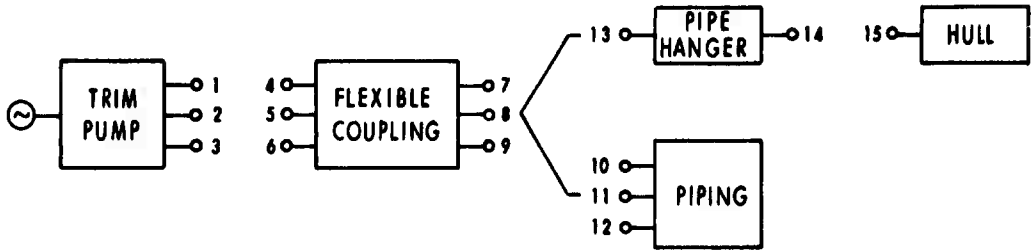


Figure 1

Schematic of Discharge Side of
Trim Pump System

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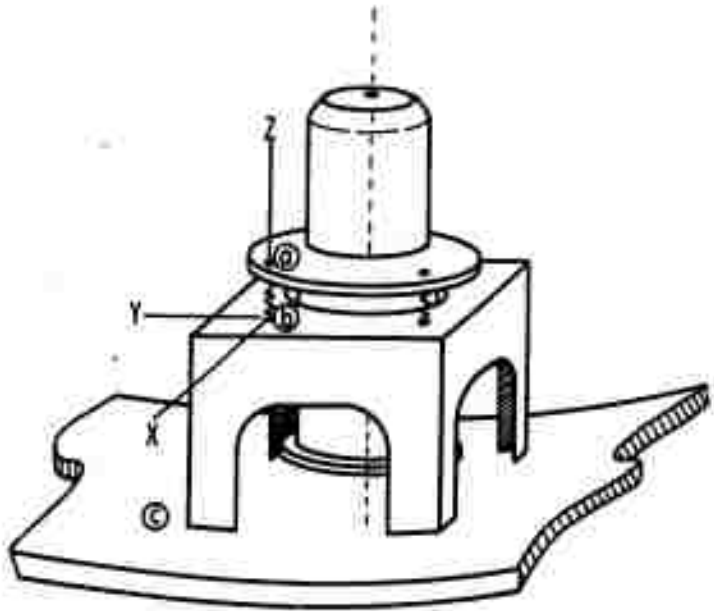


Figure 2

Mounted Trim Pump and Foundation

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NOTE:

- 1 - Goodmans Vibration Exciter
- 2 - Impedance Head
- 3 - Suspension Cord
- 4 - EB Flexible Coupling (3-planar Configuration)
- 5 - Accelerometers (Orthogonally Oriented)

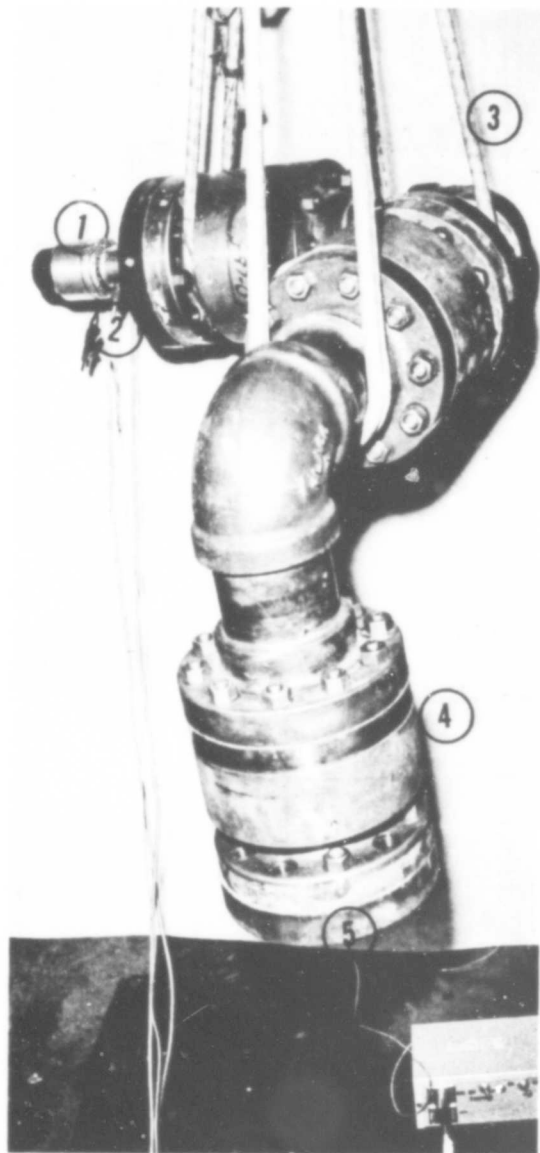


Figure 3

Flexible Pipe Coupling Configuration

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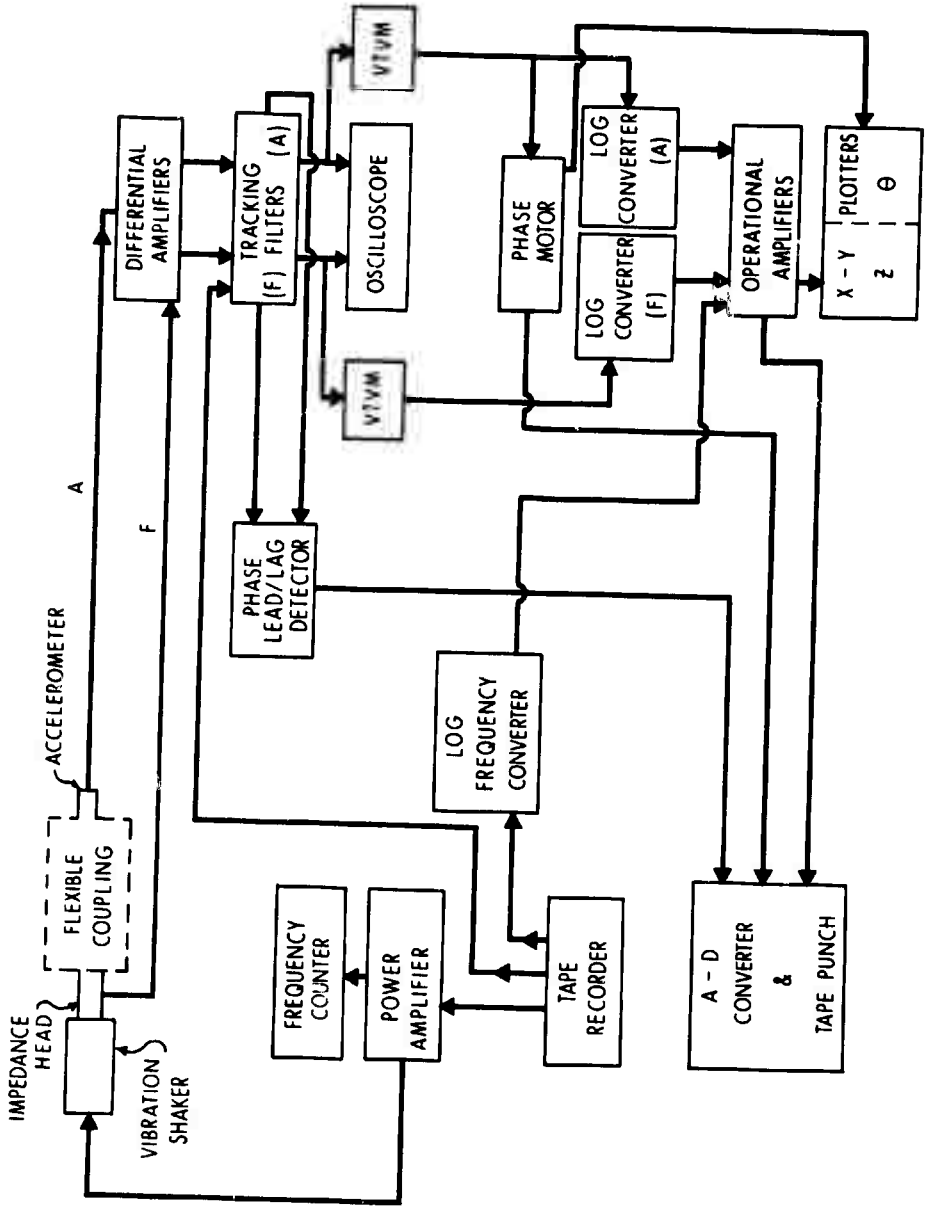


Figure 4 - Schematic of Instrumentation System

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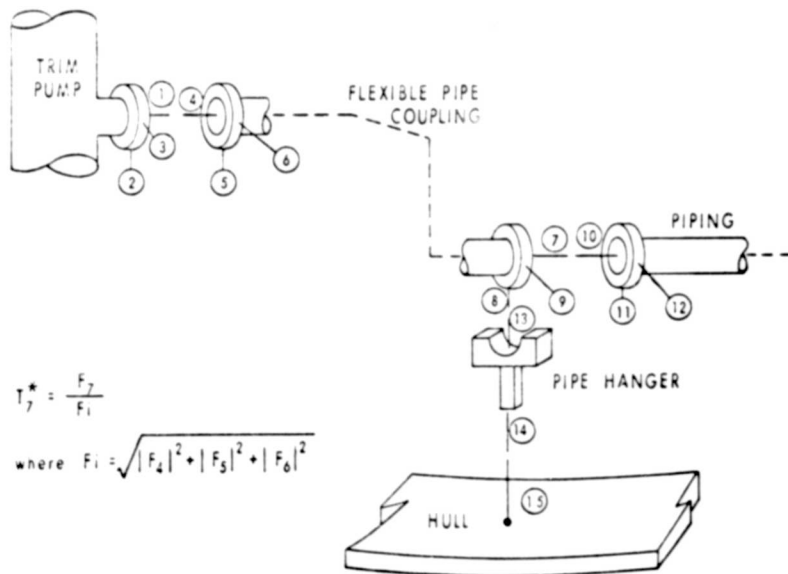
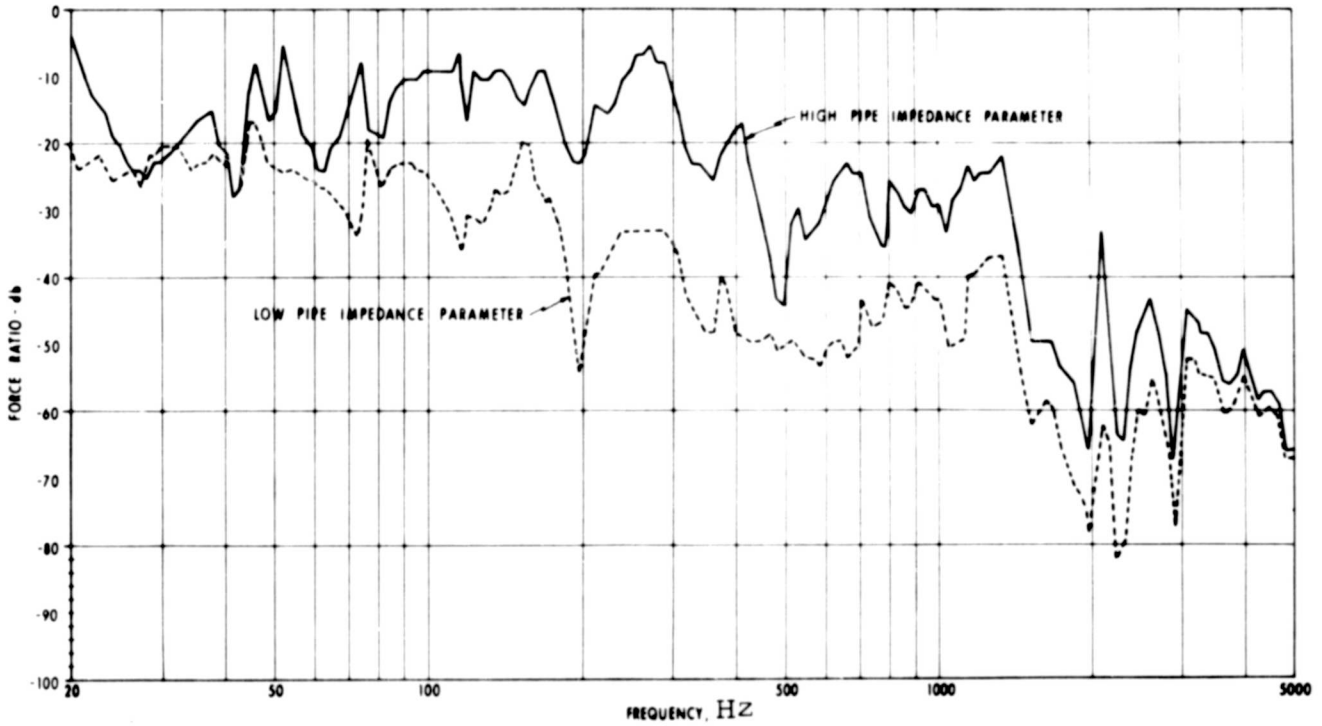
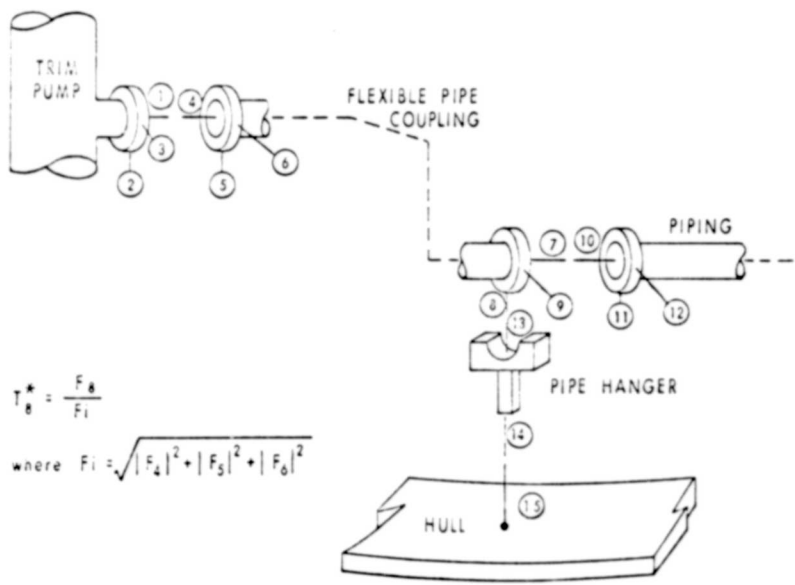
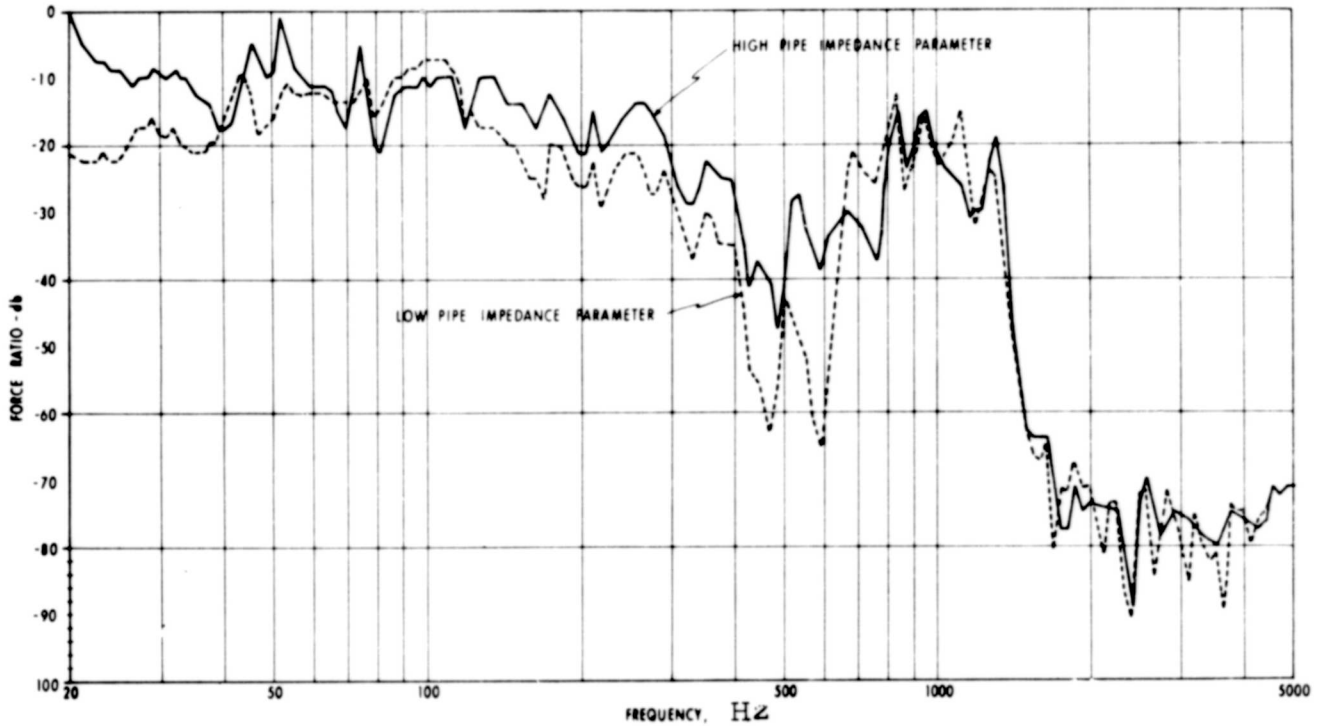


Figure 5

Flexible Coupling Force Ratios T_7^*

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$$T_8^* = \frac{F_8}{F_i}$$

where $F_i = \sqrt{|F_4|^2 + |F_5|^2 + |F_6|^2}$

Figure 6

Flexible Coupling Force Ratios T_8^*

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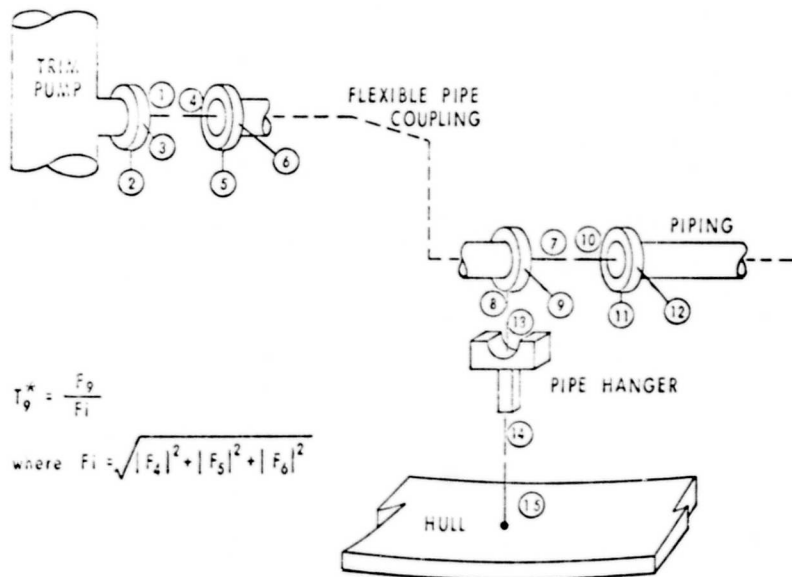
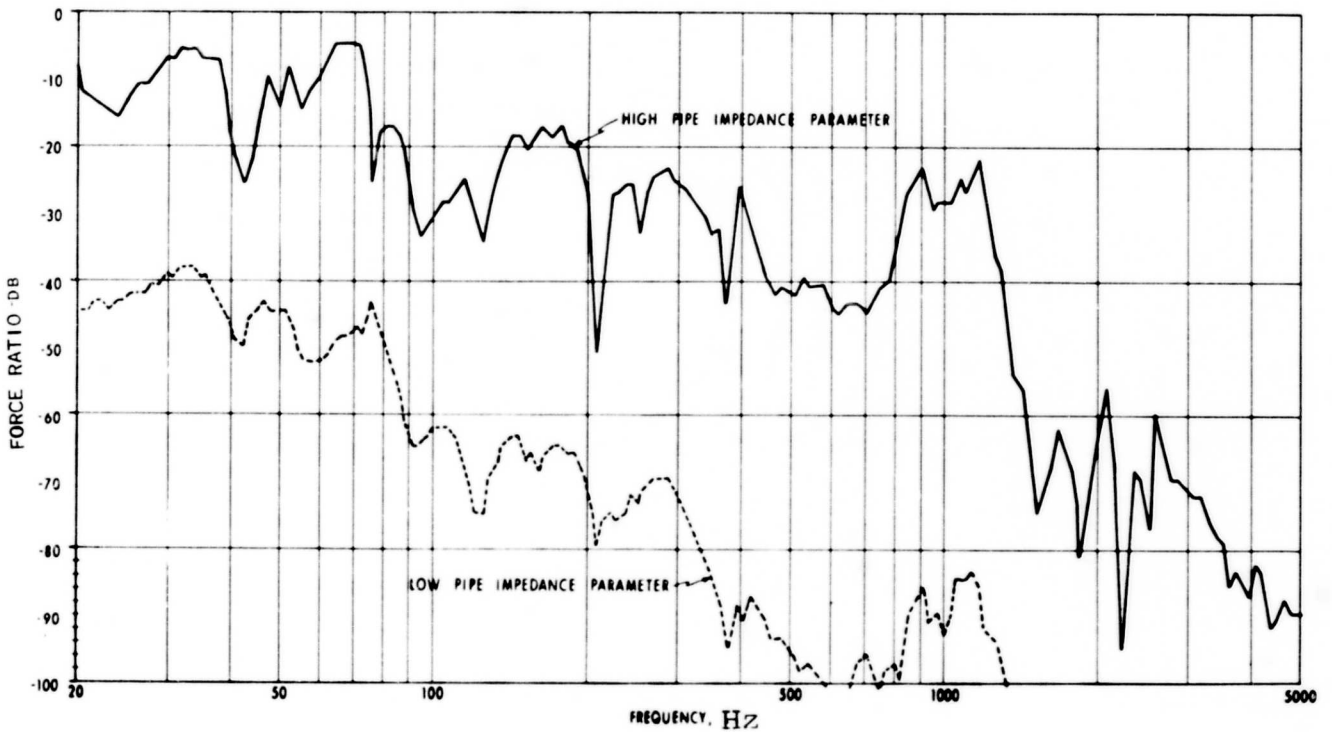


Figure 7

Flexible Coupling Force Ratios T_9^*

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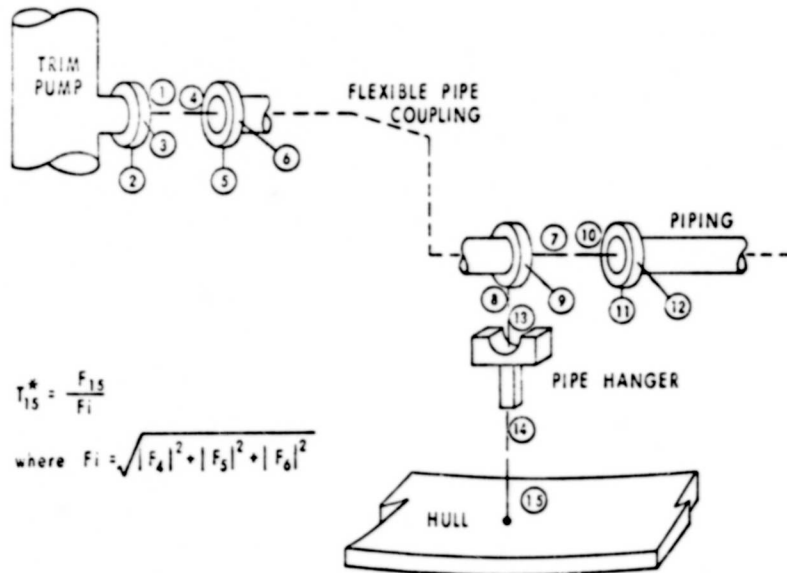
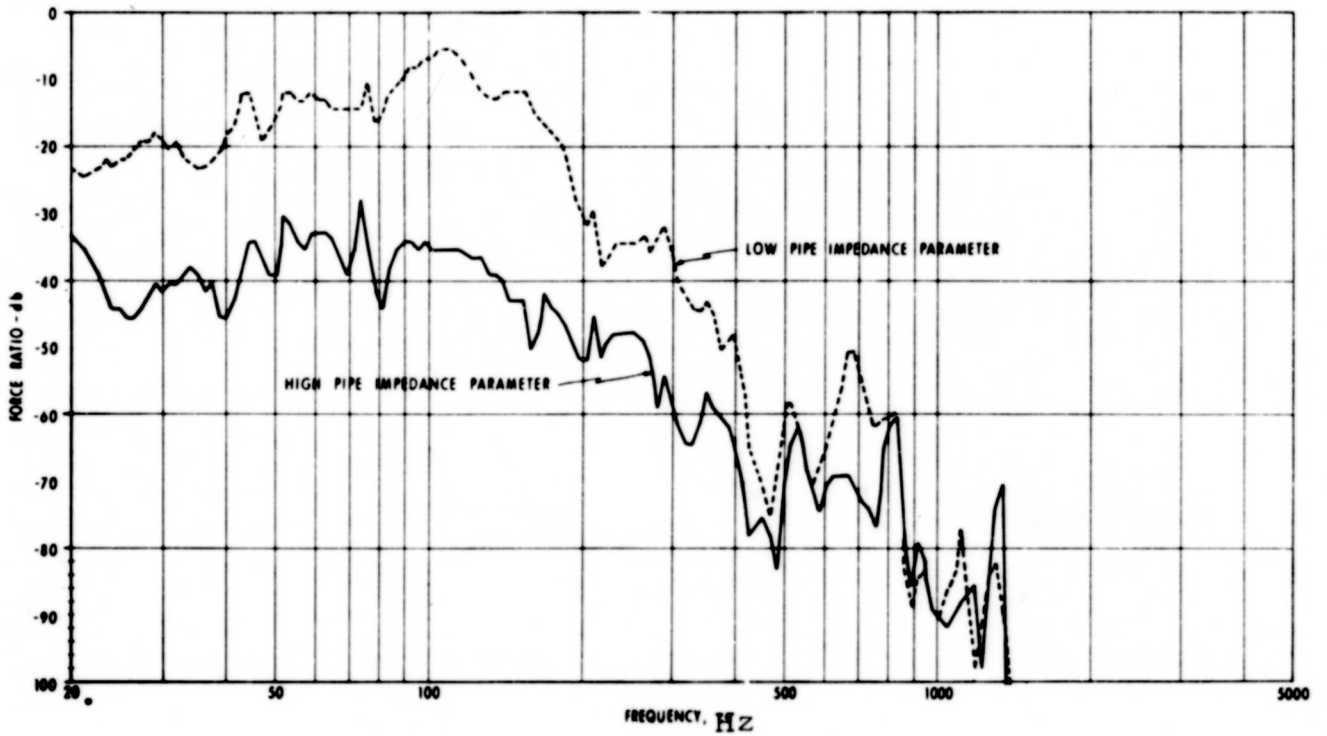


Figure 8

System Force Ratios T_{15}^*

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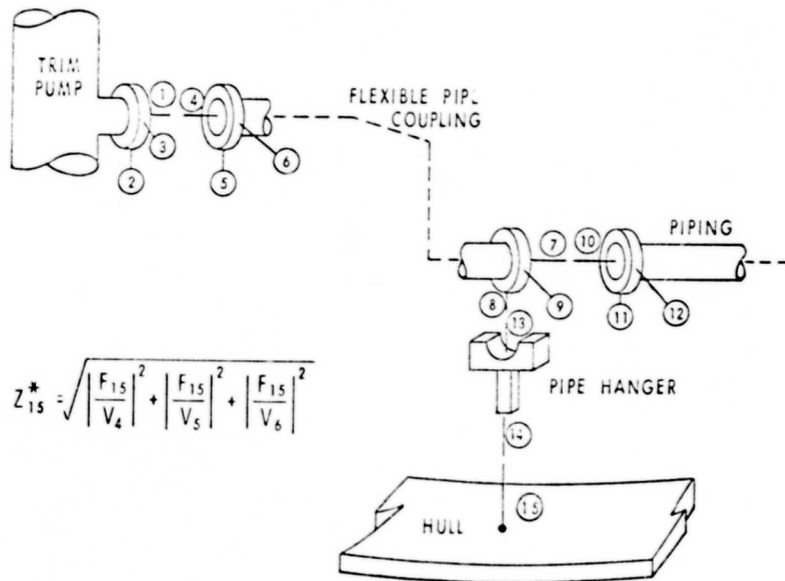
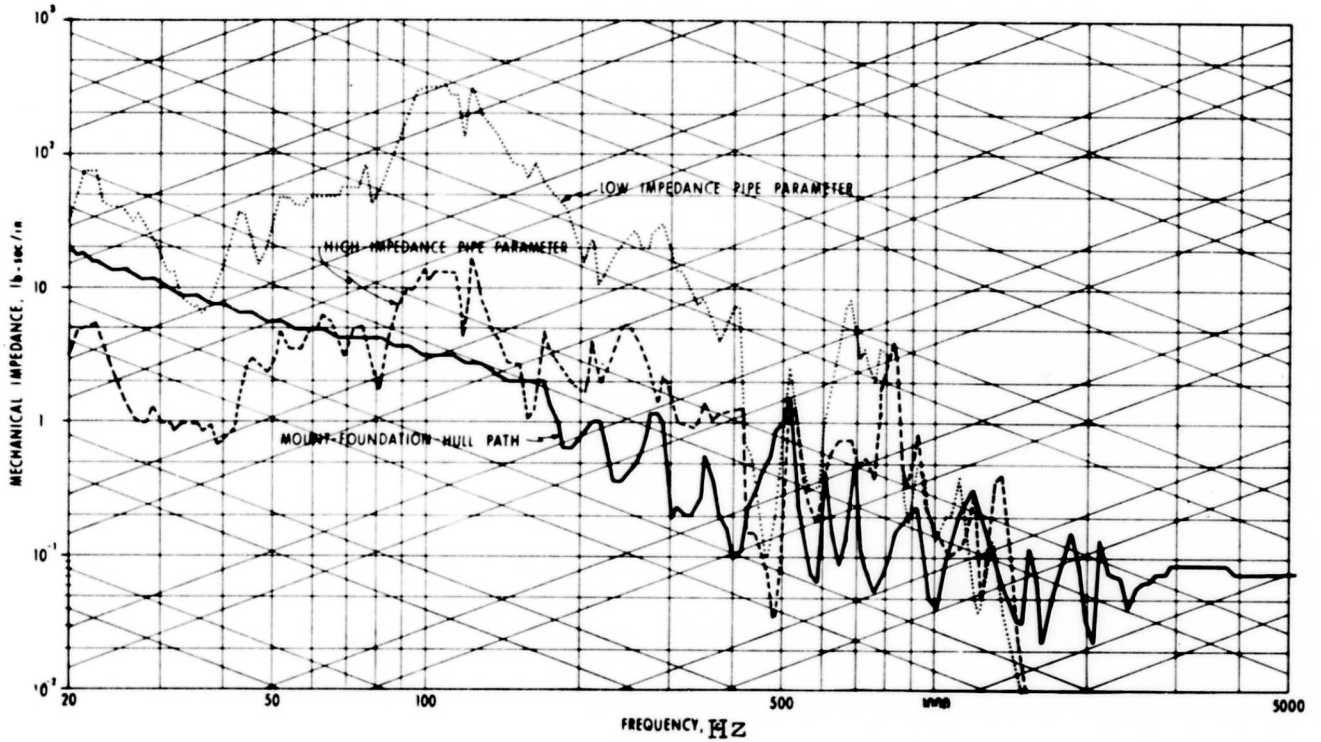
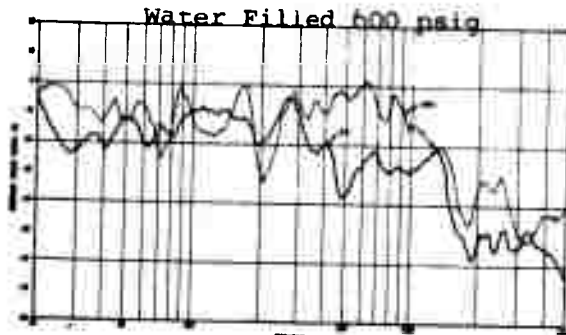


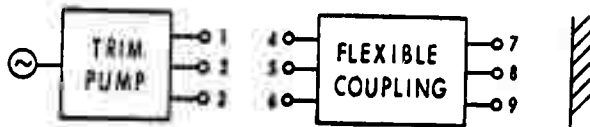
Figure 9

System Transfer Impedances

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(Applies to the two curves)



$$T^*7 = \frac{F7 \text{ (Blocked)}}{F_i}$$

$$\text{where } F_i = \sqrt{|F_4|^2 + |F_5|^2 + |F_6|^2}$$

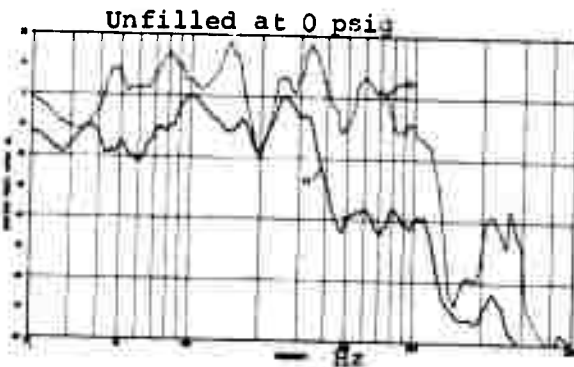
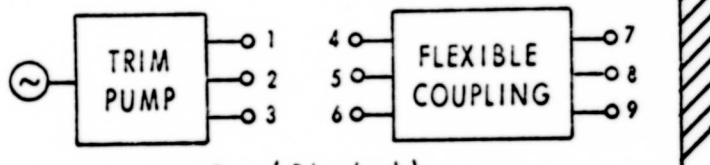
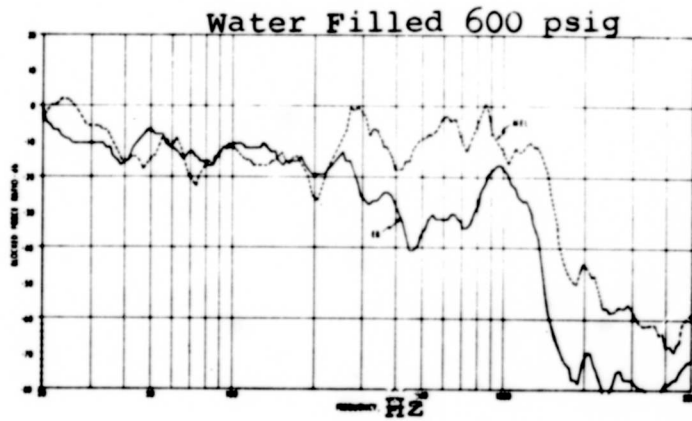


Figure 10

Flexible Coupling Blocked Force
Ratio T^*7

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$$T_8^* = \frac{F_8 \text{ (Blocked)}}{F_i}$$

(Applies to the two curves)

$$\text{where } F_i = \sqrt{|F_4|^2 + |F_5|^2 + |F_6|^2}$$

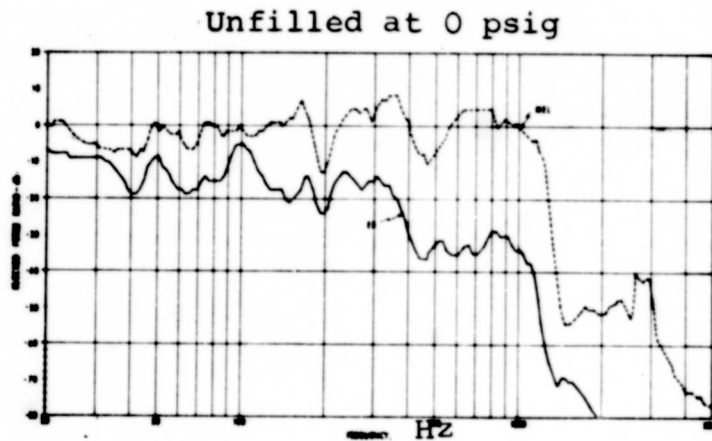
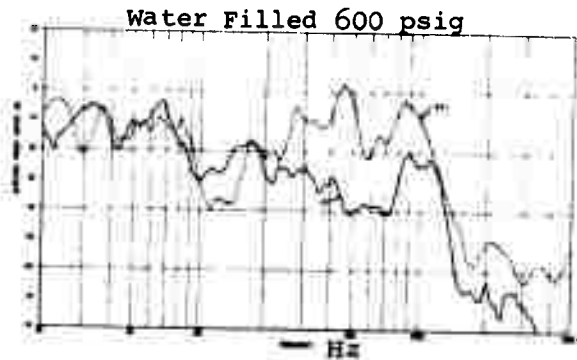


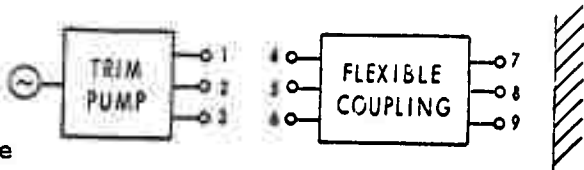
Figure 11

Flexible Coupling Blocked Force
Ratio T_8^*

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(Applies to the two curves)



$$T_9^* = \frac{F_9 \text{ (Blocked)}}{F_i}$$

$$\text{where } F_i = \sqrt{|F_4|^2 + |F_5|^2 + |F_6|^2}$$

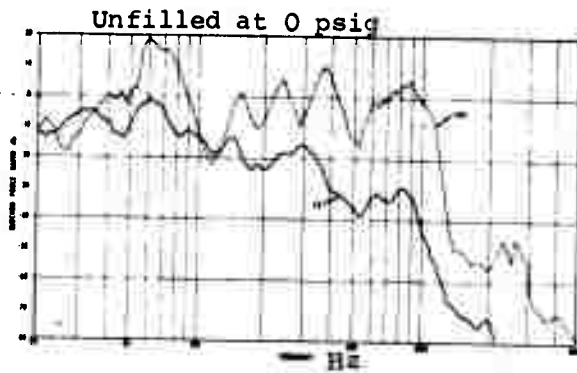
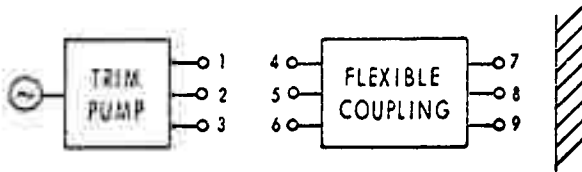
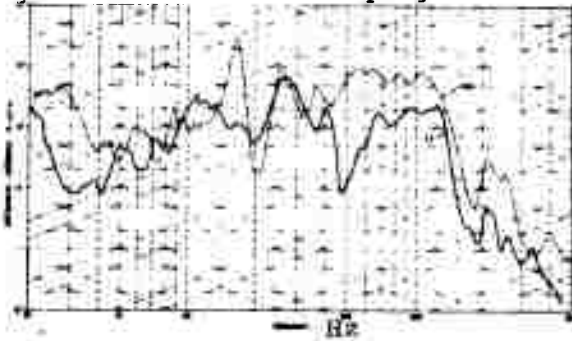


Figure 12

Flexible Coupling Blocked Force
Ratio T_9^*

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water Filled 600 psig



(Applies to the two curves)

$$Z_7^* = \sqrt{\left| \frac{F_7}{V_4} \right|^2 + \left| \frac{F_7}{V_5} \right|^2 + \left| \frac{F_7}{V_6} \right|^2}$$

where F_7 is blocked force

Unfilled at 0 psig

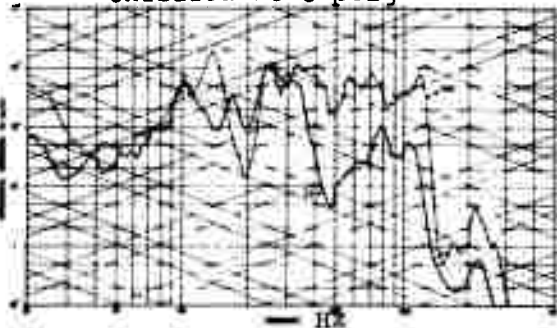
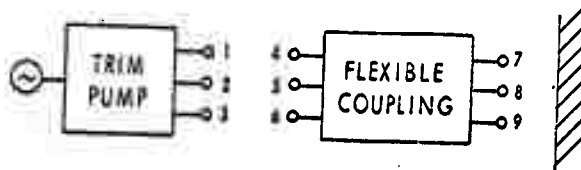
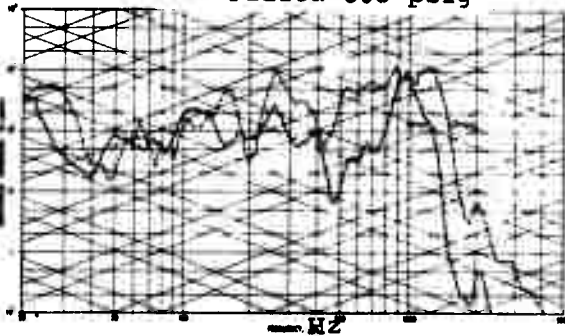


Figure 13

Flexible Coupling Blocked Transfer Impedances, Z_7^*

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Water Filled 600 psig



(Applies to the two curves)

$$Z_0^* = \sqrt{\left| \frac{F_0}{V_4} \right|^2 + \left| \frac{F_0}{V_5} \right|^2 + \left| \frac{F_0}{V_6} \right|^2}$$

where F_0 is blocked force

Unfilled at 0 psig

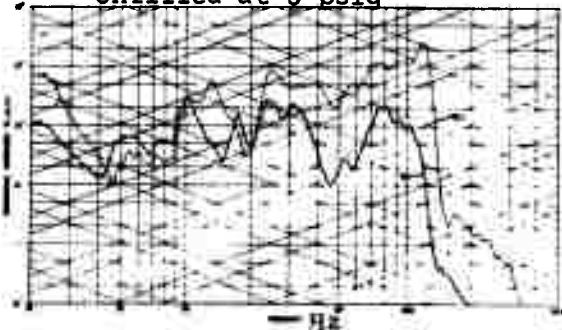
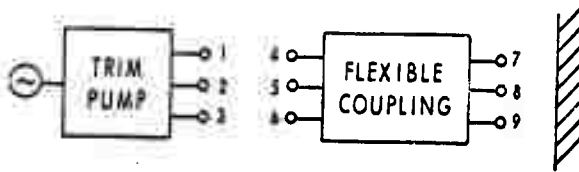
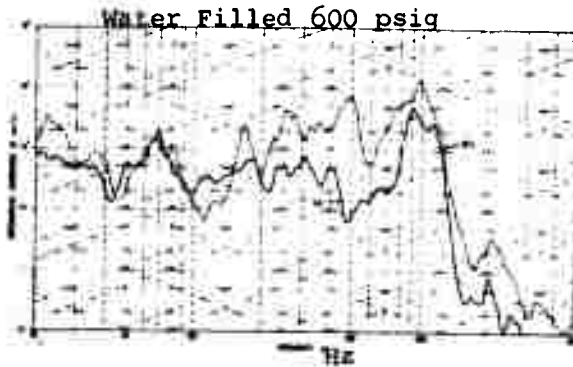


Figure 14

Flexible Coupling Blocked Transfer Impedances, Z_0^*

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$$Z_g^* \sqrt{\left| \frac{F_g}{V_4} \right|^2 + \left| \frac{F_g}{V_5} \right|^2 + \left| \frac{F_g}{V_6} \right|^2}$$

(Applies to the two curves)

where F_g is blocked force

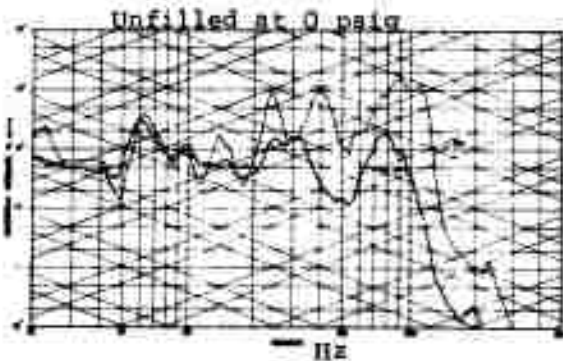


Figure 15

Flexible Coupling Blocked Transfer
Impedances, Z_g^*

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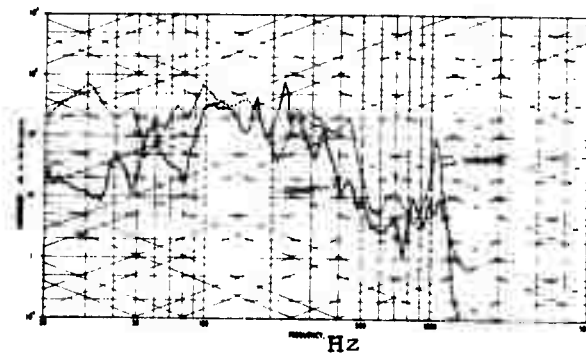


Figure 16

Experimental Verification
of Computed Results Z $7/4$

APPENDIX A

Development of System Response Matrix

REFERENCE

- (a) Smith, J. E., "Vibration Transmission through Rubber Block Pipe Hangers,"
MEL rept (in preparation)

The uncoupled velocities and forces at each numbered terminal, and the measured free driving point and transfer mobilities (translational only) of the elements of the system are defined according to the following sets of equations:

Trim Pump at Outlet Flange

$$v_1 = f_1 m_{11} + v_{01}$$

$$v_1 = f_2 m_{22} + v_{02} \quad \dots(A1)$$

$$v_2 = f_3 m_{33} + v_{03}$$

The v_0 's are the free velocities that exist at the pump output flange when it is free of connections. They will represent the velocity excitations by the pump on the system when the system elements are united.

Flexible Coupling at Input Interface

$$v_4 = f_4 m_{44} + f_5 m_{45} + f_6 m_{46} + f_7 m_{47} + f_8 m_{48} + f_9 m_{49}$$

$$v_5 = f_4 m_{54} + f_5 m_{55} + f_6 m_{56} + f_7 m_{57} + f_8 m_{58} + f_9 m_{59} \quad \dots(A2)$$

$$v_6 = f_4 m_{64} + f_5 m_{65} + f_6 m_{66} + f_7 m_{67} + f_8 m_{68} + f_9 m_{69}$$

Flexible Coupling at Output Interface

$$v_7 = f_4 m_{74} + f_5 m_{75} + f_6 m_{76} + f_7 m_{77} + f_8 m_{78} + f_9 m_{79}$$

$$v_8 = f_4 m_{84} + f_5 m_{85} + f_6 m_{86} + f_7 m_{87} + f_8 m_{88} + f_9 m_{89} \quad \dots(A3)$$

$$v_9 = f_4 m_{94} + f_5 m_{95} + f_6 m_{96} + f_7 m_{97} + f_8 m_{98} + f_9 m_{99}$$

Piping Input Interface

$$v_{10} = f_{10} m_{1010}$$

$$v_{11} = f_{11} m_{1111} \quad \dots(A4)$$

$$v_{12} = f_{12} m_{1212}$$

Pipe Hanger Input Interface

$$v_{13} = f_{13} m_{1313} + f_{14} m_{1314} \quad \dots(A5)$$

Pipe Hanger Output Interface

$$v_{14} = f_{14}m_{1414} + f_{13}m_{1413} \quad \dots(A6)$$

Hull Input Interface

$$v_{15} = f_{15}m_{1515} \quad \dots(A7)$$

The measured transfer mobilities between terminals located at the pump outlet interface are low compared to the driving point mobilities; therefore, it is assumed no coupling exists between terminals. A similar assumption was applied to the hull mobility (inverse impedance), therefore, one measurement, m_{1515} , was used in a direction normal to the hull. The free mobility measurements of the typical block rubber pipe hanger, the vibration properties of which were reported by Smith, reference (a), were assumed to be unidirectional.

The vibration transmission properties of the flexible coupling in shipboard systems are largely a function of the termination offered by the piping system, the physical arrangement of which is often dependent on available compartment space. No typical impedance measurements are available, therefore, to define this boundary condition. To compensate for the lack of detail, response values, defining the envelope between antiresonant and resonant conditions, were assigned to the fictional piping system by assuming constant impedance magnitudes as parameters. Terminals 11 and 12 were designated as radial orthogonal axes having equal impedance response magnitudes, and Terminal 10 was assigned as the longitudinal axis of the piping defined by a different and higher constant impedance magnitude. Two sets of constant values were assigned to the terminals, thereby covering a dynamic range of 5000 to 1.0 lb-sec per in., a reasonable response envelope for an arbitrary piping system.

Equations (A1) through (A7) describe the uncoupled responses of each element unconstrained at its interfaces. To connect the elements mathematically into a machinery system, boundary conditions were imposed at the interfaces, the assumption being made that all interface connections were rigid. Therefore, referring to Item (b), Figure 1 of the text, it can be stated that $v_1 = v_4$, $f_1 + f_4 = 0$; $v_2 = v_5$, $f_2 + f_5 = 0$; $v_3 = v_6$, $f_3 + f_6 = 0$. Since the terminals of the flexible pipe coupling, piping, and pipe hanger have common axes when connected, $v_8 = v_{11} = v_{13}$ and $f_8 + f_{11} + f_{13} = 0$. Also $v_7 = v_{10}$, $f_7 + f_{10} = 0$; $v_9 = v_{12}$ and $f_9 + f_{12} = 0$. When the connection between pipe hanger and hull are formed, it can be stated that $v_{14} = v_{15}$ and $f_{14} + f_{15} = 0$. When these boundary conditions are applied to Equations (A1) through (A7) and when unity free velocity is assigned to v_{01} , v_{02} and v_{03} , (since the investigation was made with the pump not operating) a system of simultaneous linear equations evolve from which solutions of the coupled forces and velocities at each terminal can be obtained. The matrix form of this system of linear equations can be written as $[v] = [m] \times [f]$ or in expanded form as:

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APPENDIX B

Development of Mount- Foundation -
Hull Vibration Response Equation

REFERENCE

- (a) Wright, D. V., and A. C. Hagg, "Practical Calculations and Control of Vibration Transmission through Resilient Mounts and Basic Foundation Structures," Westinghouse Research Lab, BUSHIPS Contr NObs-72326, Dec 1959

Figure 2 of the text is a sketch of a trim pump mounted at four points on a symmetrical foundation. Lack of data prevents inclusion of three degrees of freedom at the mount-foundation interfaces, therefore, it is assumed that translational forces, developed along the x, y axes, generate negligible forces at the hull, Terminal c. Referring to the dynamics along the z axis only, the forces generated at the foundation-mount interface, Terminal b, can be defined as a product of the velocity of the machine foot and the blocked transfer impedance of the isolation mount, reference (a),

$$f_b = v_a Z_{ab} \quad \dots(B1)$$

The insertion factor or ratio of dynamic forces at the hull, Terminal c, to the forces at the foundation input, Terminal b, can be derived from the ratio of the measured hull driving point impedance, Z_{cc} , without the foundation attached, to the transfer impedance of the foundation, Z_{bc} , when attached to the hull. In equation form

$$\frac{Z_{cc}}{Z_{bc}} = \frac{f_c}{v_c} \times \frac{v_c}{f_b} = \frac{f_c}{f_b} \quad \dots(B2)$$

Combining Equations (B2) and (B1) results in a transfer impedance which defines the path from mount input to hull-foundation interface,

$$Z_{ca} = \frac{f_c}{v_a} = \frac{Z_{ab} Z_{cc}}{Z_{bc}} \quad \dots(B3)$$

Equation (B3) is a valid estimate of the vibration transmission through each foundation leg if the following assumptions are made:

- The hull driving point impedance at each hull-foundation interface is low compared to the transfer impedances between the locations.
- Attachment of the foundation to the hull, at Terminal c, applies negligible constraint to the hull response.
- Dynamic coupling between mounting locations is negligible.
- The four isolation mounts have identical transfer impedance characteristics.

With the above assumptions, a description of the total sound transmission path can be estimated by a power summation of the four identical transfer impedances, resulting in

$$Z_T^* = 2 \left[\frac{Z_{ab} Z_{cc}}{Z_{bc}} \right] \quad \dots(B4)$$

Appendix C

Free Mobility and Impedance
Measurements of System Elements

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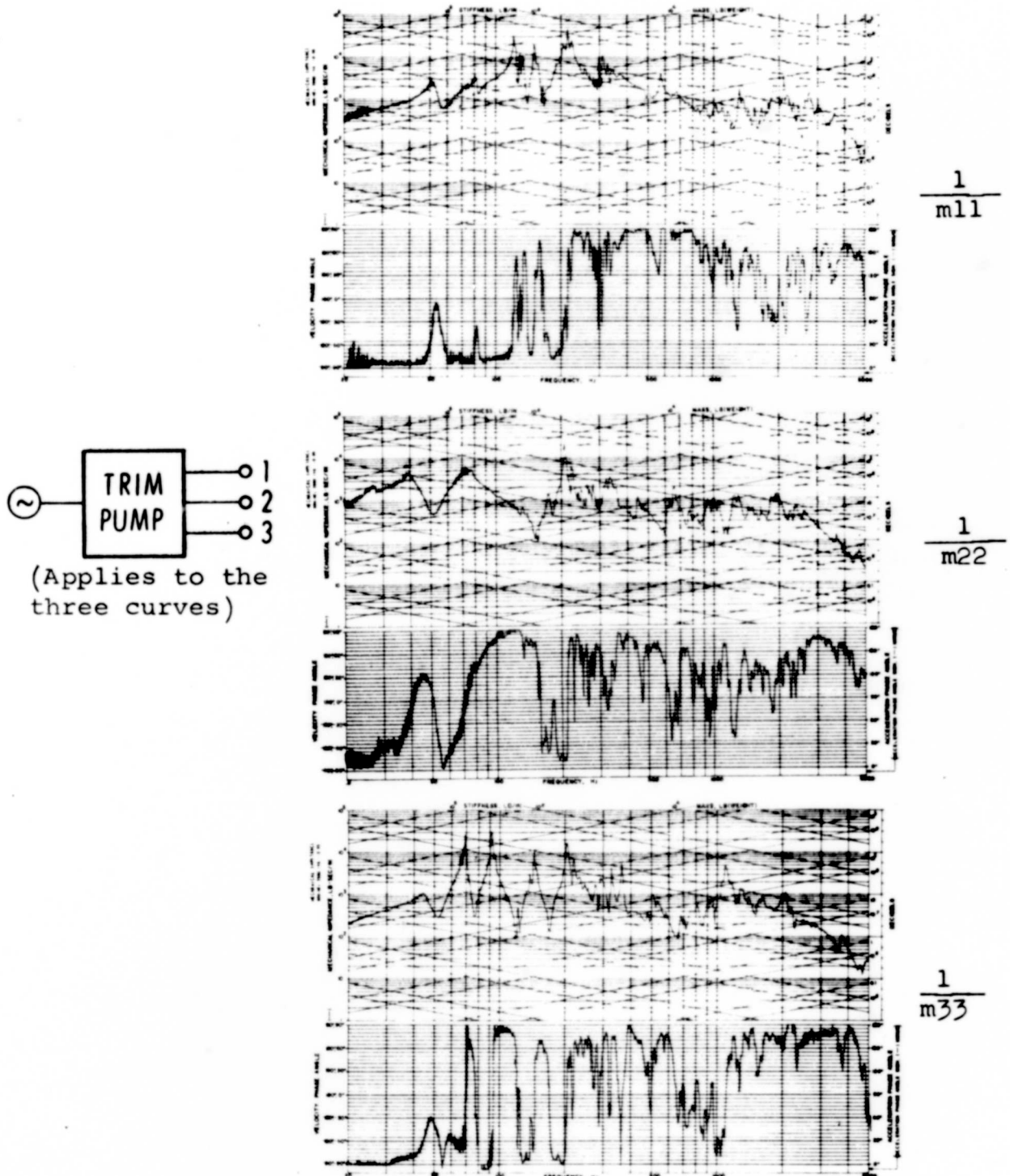
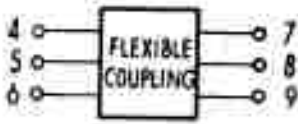
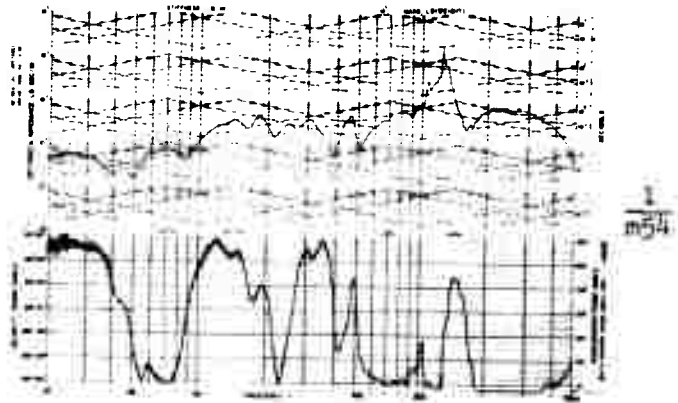


Figure 1-C
Free Mobility Measurements of Trim Pump
Pump at Outlet Flange

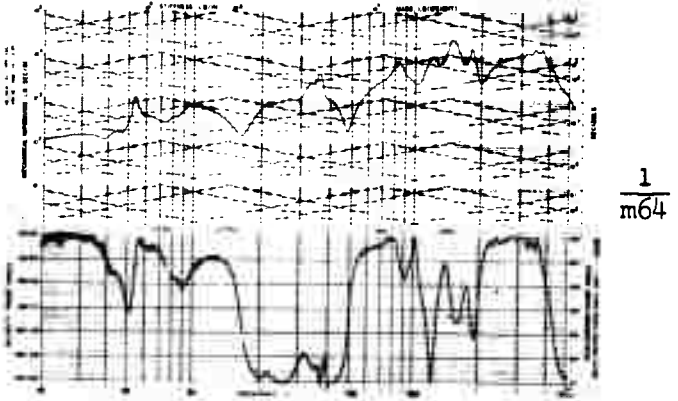
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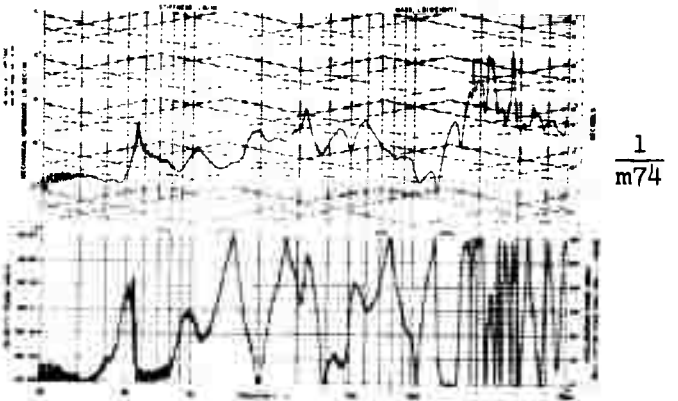
(Applies to the three curves)



$\frac{1}{m54}$



$\frac{1}{m64}$



$\frac{1}{m74}$

Figure 3-C, Free Mobility Measurements of EB Flexible Coupling Configuration, Water Filled at 600 psig

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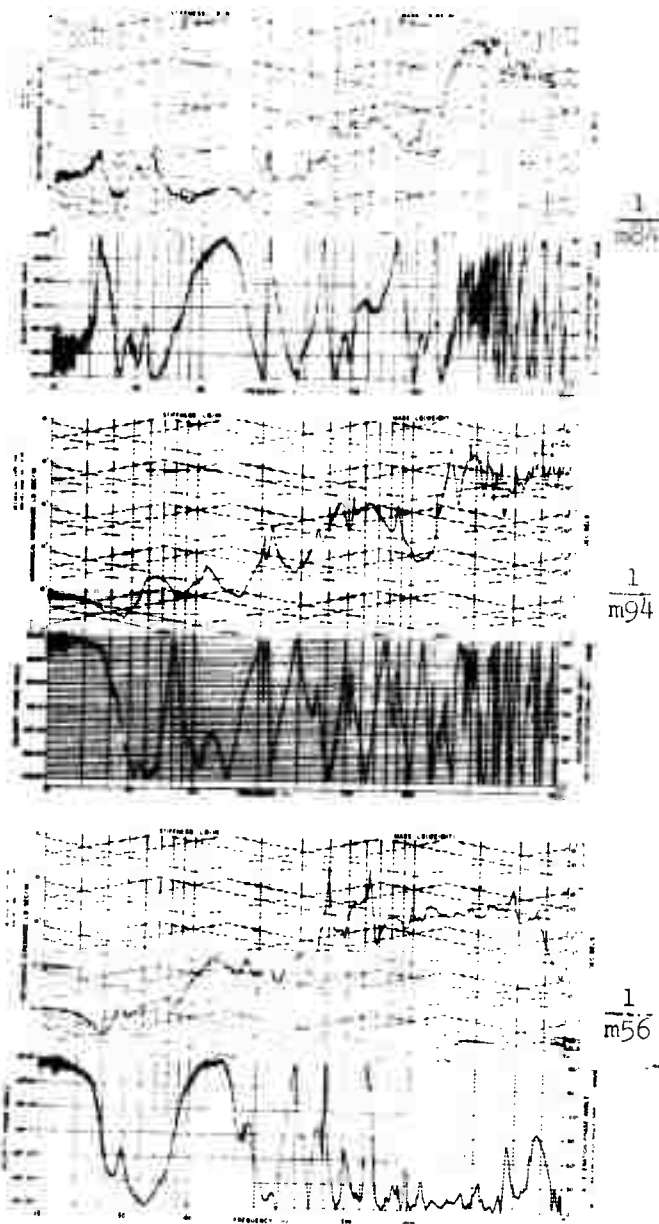
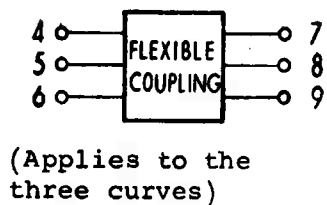


Figure 4-C, Free Mobility Measurements of EB Flexible Coupling Configuration, Water Filled at 600 psig

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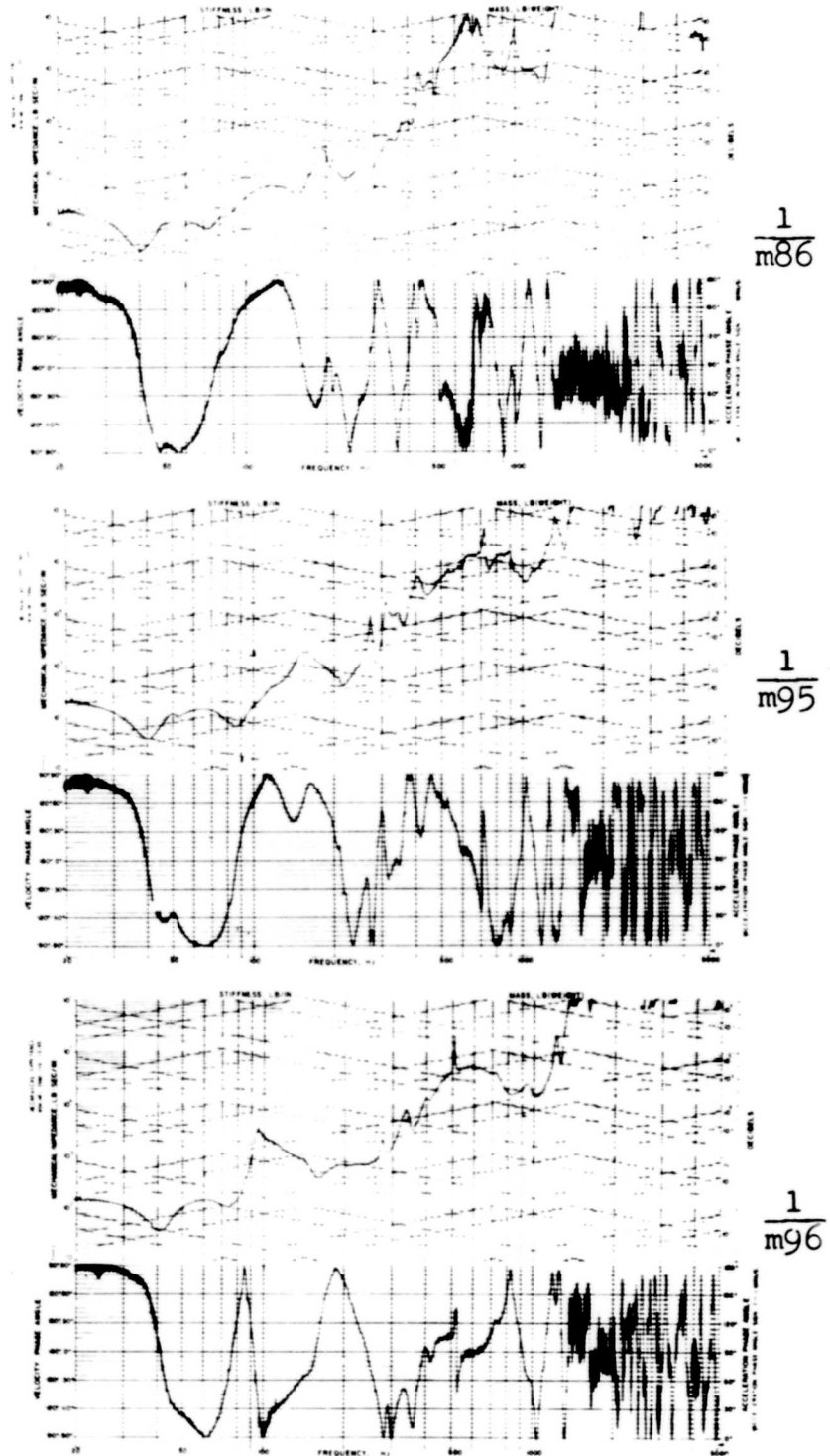
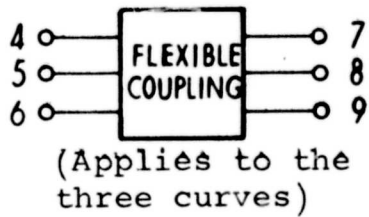


Figure 5-C, Free Mobility Measurements of EB Flexible Coupling Configuration, Water Filled at 600 psig

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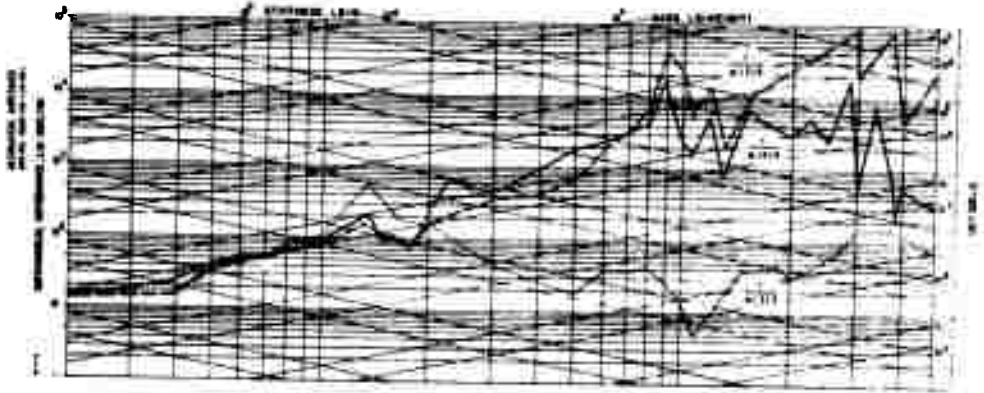


Figure 6-C

Free Driving Point and Transfer Mobility Measurements of Block Rubber Pipe Hanger

Legend applies to the two curves

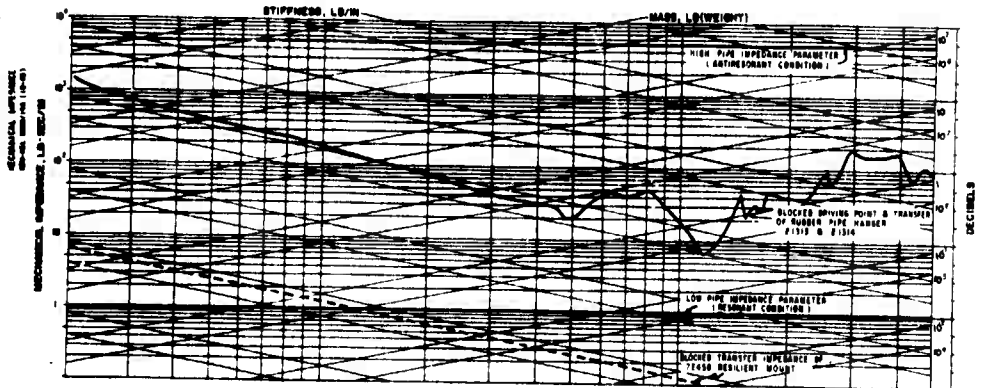


Figure 7-C

Comparison of Piping Parameters and Isolator Characteristics

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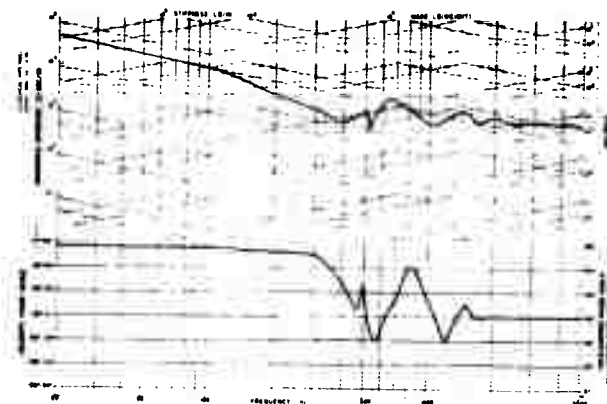
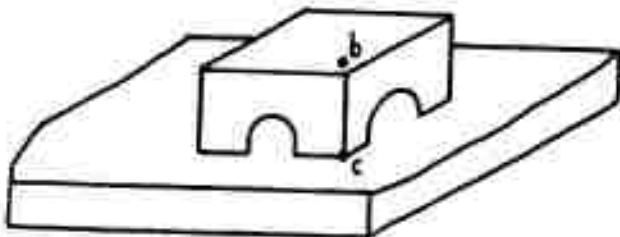


Figure 8-C

Hull Driving Point Impedance $\frac{1}{m1515}$ and Z_{cc}



(Drawing applies to Figures 8-C and 9-C)

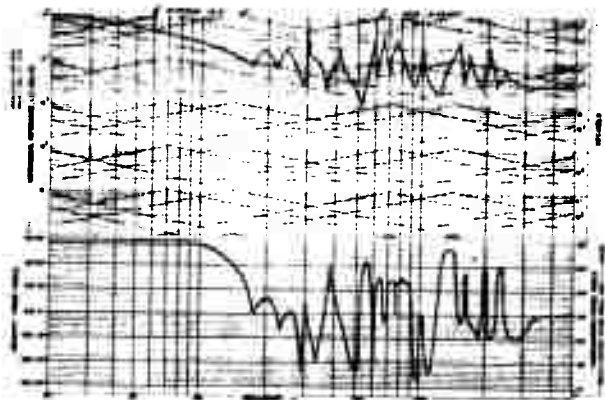


Figure 9-C, Power Summed Foundation Transfer Impedance, Z_{bc}

Appendix D

Technical References

- 1 - Wright, D. V., and A. C. Hagg, "Practical Calculations and Control of Vibration Transmission through Resilient Mounts and Basic Foundation Structures," Westinghouse Research Lab, BUSHIPS Contr NObs-72326, Dec 1959
- 2 - Darby, R. A., "The Quantitative Evaluation of Vibration Isolation Devices for Mounted Naval Machinery Installations," MEL Rept 82 287E, 29 Aug 1963
- 3 - Schwartz, J. I., and H. L. ReKate, "Behavior Study of a Curtiss-Wright Snorkel Flexible Exhaust Connection for Submarine Application," NAVENGRXSTA Rept 72 108, 13 Nov 1963
- 4 - Hirsch, R. C., "The Experimental Evaluation of an Electrical Power Cable Bundle as a Vibration Transmission Path," MEL Rept 107/65, May 1965
- 5 - Smith, J. E., "Final Report on the Development of Analysis Techniques for Submarine Machinery Installations by Mechanical Impedance Methods," PNS Test T 819-030, Sep 1963
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- 7 - Darby, R. A., "A Practical Method for Predicting Acoustic Radiation or Shock Excursions of Navy Machinery," 34th Shock and Vibration Bulletin, NRL, Dec 1964
- 8 - Hirsch, R. C., "Vibration Characteristics of Two Flexible Hose Types Arranged in a Right Angle Configuration," MEL Rept 288/65, Oct 1965
- 9 - Smith, J. E., "Vibration Transmission through Rubber Block Pipe Hangers," MEL Rept (in preparation)

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13. ABSTRACT <p>An analytical-experimental study was performed to determine the sound transmission characteristics of a flexible coupling in a trim-pump system. Mechanical impedance methods were used to compare the structural path to the hull via resilient mounts and foundation to the structural path via flexible coupling and pipe hanger. Also investigated were the vibration characteristics of an Electric Boat and MEL flexible coupling under pressurized and unpressurized conditions. Results showed that the sound transmission via flexible coupling and pipe hanger exceeded the sound transmission via the mount and foundation and would negate the isolation effectiveness of the primary mounting system. The results of comparing the two types of flexible coupling showed that the Electric Boat Coupling exhibited better total noise attenuation characteristics than did the MEL coupling. However, the MEL coupling showed a unique inherent characteristic of improved attenuation capabilities with increased pressure loading.</p> <p>(Author)</p>			

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KEY WORDS

LINK A

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LINK C

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Submarines
 Piping systems
 Structureborne vibration
 Flexible couplings
 Radiated waterborne noise
 Foundations, machinery
 Hull resonance
 Isolation mounts
 Trim pump systems