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MERADCOM/OSU HYDRAULIC SYSTEMS RESEARCH PROGRAM

SECTION I

HYDRAULIC NOISE ATTENUATION

PREPARED BY THE CONTRACTOR

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

FEBRUARY 1977

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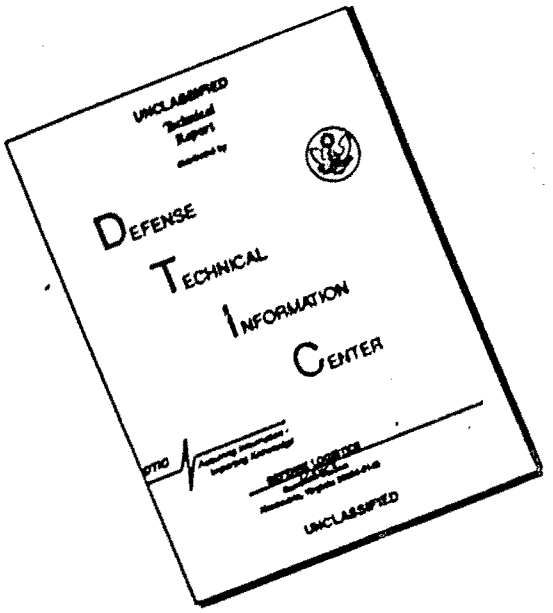
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SECTION I.

HYDRAULIC NOISE ATTENUATION.

PREPARED BY PERSONNEL OF

FLUID POWER RESEARCH CENTER
OKLAHOMA STATE UNIVERSITY
STILLWATER, OKLAHOMA

(10) G. E. Maroney

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The purpose of the Oklahoma State University/U.S. Army Mobility Equipment Research and Development Command Program is to provide the military with tools for the scientific appraisal of fluid power systems. The activities of this year's noise program are a continuation of the efforts of the previous years and make full use of the preceding program.

This section presents a detailed account of the project activities in the area of hydraulic noise. Specific test procedures are recommended for determining the fluidborne noise generation potential of hydraulic pumps and the acoustical performance characteristics of fluidborne noise attenuators. Experimental data are shown to indicate typical results that are obtained with the proposed test codes.

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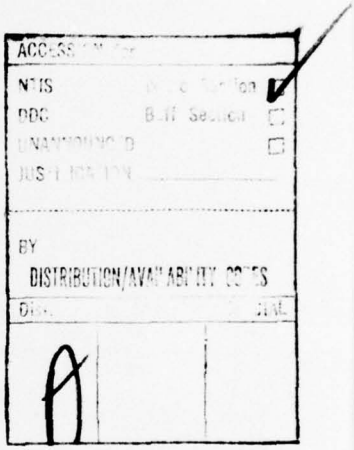
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SECTION I

HYDRAULIC NOISE ATTENUATION



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FOREWORD

This section presents a detailed account of the project activities in the area of hydraulic noise. Specific test procedures are recommended for determining the fluid-borne noise generation potential of hydraulic pumps and the acoustical performance characteristics of fluidborne noise attenuators. Experimental data are shown to indicate typical results that are obtained with the proposed test codes.

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- FOREWORD -

This report was prepared by the staff of the Fluid Power Research Center of the Division of Engineering, Technology & Architecture, Oklahoma State University of Agriculture and Applied Sciences. The study was initiated by the Mobility Equipment Research and Development Command, Fort Belvoir, Virginia. Authorization for the study reported herein was granted under Contract No. DAAK02-75-C-0137. The time period covered by this report is from 1 February 1976 to 31 January 1977.

The Contracting Officer's Representative was Mr. Hansel Y. Smith, and Mr. John M. Karhnak served as the Contracting Officer's Technical Representative. In addition, Mr. Paul Hopler has effectively represented the Contracting Officer both technically and administratively through various phases of this contract. The active participation of Messrs. Smith, Karhnak, and Hopler during critical phases of work contributed significantly to the overall success of the program.

The studies represented by this report were conducted under the general guidance of Dr. E. C. Fitch, Program Director. The details of each study are presented in a self-contained section of this report. The titles of the various sections together with their respective Project Managers are listed below:

SECTION I.	HYDRAULIC NOISE ATTENUATION — G. E. Maroney
SECTION II.	CYLINDER STRUCTURAL INTEGRITY ASSESSMENT — S. K. R. Iyengar
SECTION III.	HYDRAULIC SYSTEM OPERATIONAL SEVERITY ASSESSMENT — R. L. Decker/S. K. R. Iyengar
SECTION IV.	ON-BOARD MONITOR STUDY — R. L. Decker
SECTION V.	HYDRAULIC SYSTEM DIAGNOSTICS — R. K. Tessman
SECTION VI.	PUMP CONTAMINANT TOLERANCE VERIFICATION — L. E. Bensch

CHAPTER I

INTRODUCTION

The need to control the noise emitted from fluid power equipment has stimulated many manufacturers and users in the fluid power industry to take a "fresh look" at noise. There has been, in recent years, a more intensive look at the sources of noise in order to understand and model these sources. Instead of concentrating only on airborne noise, research has been extended into the areas of fluidborne and structure-borne noise since the control of these two forms of noise can have a significant impact on the overall airborne noise. This report is limited to the area of fluidborne noise (FBN).

Fluidborne noise can be controlled by either changing the source of the noise (usually a pump, motor, or valve), installing a FBN attenuator, or otherwise modifying the system. However, to change the source, it is necessary to understand the important parameters that constitute the fluidborne noise generation potential of the source. In this way, say, two pumps can be compared to see which one has the largest (or smallest) fluidborne noise potential. Also, if the important parameters of a noise source are determined (either analytically or experimentally), then it is possible to predict how the pump will affect a given system.

A similiar procedure can be used for FBN attenuators. The use of attenuators in fluid power systems has often been with a "fly by the seat-of-your-pants" philosophy. The person installing an attenuator in a system hopes that a given attenuator will be the "one" that will work for his system. With an adequate understanding of the performance of attenuators, it would be possible for manufacturers and users to compare attenuator A with attenuator B and predict what effect they would have on the system.

Chapter II of this report discusses a fluidborne noise attenuator evaluation procedure. The basic theory and important parameters are outlined, as well as an actual test procedure. This procedure is applied to an actual attenuator to evaluate its performance characteristics as compared to the theoretical model.

Chapter III covers the area of FBN sources, in particular pumps. A basic test procedure is outlined for the evaluation of the FBN potential of fluid power pumps to include source impedance and flow ripple. These are compared to the theoretical model.

Chapter IV is a brief discussion of the OSU Six Transducer Data Reduction Computer Program. This program has added tremendously to the OSU-FBN program by providing a more accurate estimation of standing wave parameters such as incident pressure, reflection factor, and the position of maximum pressure.

CHAPTER II

FLUIDBORNE NOISE ATTENUATOR EVALUATION

Accompanying the increased availability and use of fluidborne noise attenuators, there is an even greater need for an objective, reliable and reputable method of evaluating the effectiveness of these attenuators. Since attenuators are designed to reduce the fluidborne noise (FBN) or pressure ripple in a system, then, to critically evaluate an attenuator, a fundamental understanding of FBN is essential. This section of the report will outline some of the basic equations and assumptions needed for evaluating attenuators outline a test procedure for FBN attenuator evaluation and present test results from a sample attenuator.

THEORY

To model fluidborne noise or wave propagation in fluid power systems, plane wave theory can be utilized by making the following assumptions:

1. Wave propagation is one-dimensional. This condition is satisfied as long as the diameters of pipes, conduits, etc., are less than one-fourth the wave length of the highest frequency of interest. For instance, in a 2.54 cm (1 inch) i. d. pipe containing a fluid having a sonic velocity of 1220 m/s (4000 ft/sec), plane wave theory can be used up to frequencies of 12,000 Hz.

2. Viscous Dissipation is negligible. For very high flow rates in small diameter tubes, plane wave theory has to be modified to include a viscous dissipation term.

3. The amplitude of fluctuation in flow and pressure is small in comparison with mean flow and pressure. This assumption linearizes the differential equations used to derive plane wave theory. If this condition is not satisfied, as it may be in some fluid power systems, then a nonlinear model must be used.

Given these assumptions, the acoustic power flow per unit of area in an infinitely long pipe is the rms pressure, p, times the rms particle velocity. The rms particle velocity is p/Z, where Z is the characteristic impedance of the pipe which is defined as $\rho C/S$, with S being the cross-sectional area of the pipe. The power in the pipe is [7]:

$$W = S p^2 / \rho C \qquad (2\text{-}1)$$

Infinitely long lines seldom exist in the field hydraulic systems, so there is usually a right traveling wave, $(S p_i^2 / \rho C)$, and a left traveling wave, $(S p_r^2 / \rho C)$. The net power flow is:

$$W_{net} = (S/\rho C)\ (p_i^2 - p_r^2) \qquad (2\text{-}2)$$

where p_i and p_r, respectively, are associated with the incident and reflected pressure waves. Eq. (2-2) can be shown to be equivalent to:

$$W_{net} = (S/\rho C)\ (p_{max})\ (p_{min}) \qquad (2\text{-}3)$$

where p_{max} is the maximum pressure (rms) in the line and p_{min} is the minimum pressure in the line. The ratio p_{max}/p_{min} is defined as the standing wave ratio (SWR). The following equations show relationships between SWR, p_i, p_r, p_{max}, and p_{min}:

$$SWR = p_{max}/p_{min} = (p_i + p_r)/(p_i - p_r) \qquad (2\text{-}4)$$

or rearranging:

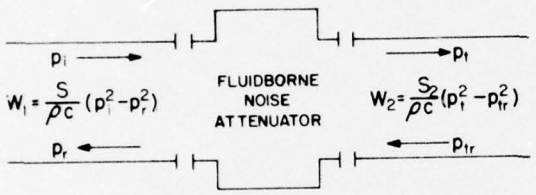
$$p_r/p_i = (SWR - 1)/(SWR + 1) = (p_{max} - p_{min})/(p_{max} + p_{min}) \qquad (2\text{-}5)$$

IMPORTANT TEST PARAMETERS

A test code for the evaluation of fluidborne noise attenuators should consider the following characteristics: transmission loss, input impedance, output impedance, and flow resistance. The transmission loss indicates how much of the incident pressure ripple is emitted at the outlet of the attenuator. The input impedance indicates how much of the incident pressure is reflected by the attenuator toward the source. The output impedance indicates how much of the incident pressure is reflected by the attenuator toward the source. The output impedance indicates what percentage of the pressure incident on the attenuator outlet is reflected back to the downstream portion of the system. The flow resistance of the attenuator indicates the power dissipated by a given value of mean flow through the unit. These four characteristics adequately define the performance characteristics needed to acoustically rate the attenuator and to compare

the power dissipated by the attenuator relative to the acoustical effectiveness of the unit.

TRANSMISSION LOSS



$$TL = 10 \log_{10} \frac{W_i}{W_t} \text{ (dB)}$$

$$W_i = \frac{S_1}{\rho c}(p_i^2)$$

$$W_t = \frac{S_2}{\rho c}(p_t^2)$$

Fig. 2-1. Definition of Transmission Loss and Relationship Between Energy Flows to and from a FBN Attenuator.

Figure 2.1 shows the relationship between p_i and p_r in a section of conduit connected to a fluidborne noise attenuator. Extending the same concept to the pipe downstream of the attenuator gives a relationship for the transmitted pressure, p_t, and the reflected pressure, p_{tr}, in the downstream pipe. One performance parameter for the attenuator is transmission loss:

$$TL = 10 \log_{10} (W_i/W_t) \qquad (2\text{-}6)$$

If the areas of the upstream and downstream conduits are equal and the system has equal densities and sonic velocities in both sections, then it follows that:

$$TL = 10 \log_{10} (S_1 p_i^{\,2}/\rho_1 C_1)(\rho_2 C_2/S_2 p_t^{\,2})$$

$$TL = 10 \log_{10} (p_i^{\,2}/p_t^{\,2}) \qquad (2\text{-}7)$$

$$TL = 20 \log_{10} (p_i/p_t)$$

Thus, the transmission loss can be determined by evaluation of experimental data using Eqs. (2-6) and (2-7).

INPUT IMPEDANCE

Ignoring phase relationships for this report, we can consider only the real part of the input impedance of the attenuator that is the absolute value reflection coefficient, the reflection factor, R. The reflection factor is p_r/p_i. R can be evaluated using Eq.(2-5). The data for evaluating the input reflection factor is the same data used to evaluate the transmission loss.

OUTPUT IMPEDANCE

If the attenuator will accept flow through the unit from either direction, it is possible to evaluate the output impedance using the output reflection factor, R_o. The data for calculating R_o is obtained by inverting the attenuator in the test circuit and obtaining the necessary measurements by repeating the test procedure used for obtaining the transmission loss and the imput impedance.

If the pressure drop through the attenuator is less than or equal to an equivalent length of tubing, then the attenuator is essentially non-dissipative. For non-dissipative

units, $p_t^2 = p_i^2 - p_r^2$, and Eq. (1-7) can be rewritten:

$$TL = 10 \log_{10} (p_i^2/(p_i^2 - p_r^2)$$

$$TL = 10 \log_{10} (1/(1 - R^2) \qquad (2\text{-}8)$$

Thus, the reflection factor for non-dissipative elements can be written in terms of the TL:

$$R^2 = 1 - 10^{-(TL/10)} \qquad (2\text{-}9)$$

PRESSURE DROP

The efficiency of the attenuator can be evaluated by ratioing the attenuation of the unit to the pressure drop of the unit at the desired flow. To obtain an estimate of the efficiency of the attenuator, data are needed which show the pressure drop versus flow of the unit. These data can be obtained using an acceptable test procedure.

FREQUENCY RANGE

One reason for using a fluidborne noise attenuator is to reduce the amplitude of system pressure ripple at the pumping frequency. Since most conventional pumps have

between 9 and 15 pumping elements (n) and are usually operated above 600 revolutions per minute (rpm) (N) but at or below 2500 rpm, a fundamental pumping frequency range is:

$$f_1 = Nn/60$$

$$f_{low} = (600)\ 9/60 = 90\ Hz \qquad (2\text{-}9)$$

$$f_{high} = (2500)\ 15/60 = 625\ Hz$$

If the proposed procedure requires that the first three pumping harmonics are to be measured, this will accommodate a frequency range of interest spanning 100 Hz and 2000 Hz. Constraining the frequency range to 100 Hz - 1000 Hz allows testing to be conducted in the average fluid power laboratory, using maximum pump speeds in the vicinity of 2300 rpm for a pump with nine pumping elements, thus providing information about the attenuator over a frequency range which is important to the largest number of users.

Because the pressure level at the fundamental pumping frequency is usually more repeatable than the levels at higher harmonies and because most attenuators will probably respond best to the highest pressure level in the system, it is recommended that data for attenuator evaluation be taken at the first three pumping frequencies. Data at individual frequencies are obtained by varying the pump speed. At a minimum, the procedure requires measuring attenuator performance at the eleven third-octave frequencies between 100 and 1000 Hz.

TEST PROCEDURE

Figure 2-2 outlines the test procedure. (Appendix A contains a more detailed and specific procedure.) Table 2-1 gives the test frequencies that should be used to evaluate the FBN attenuator if the first three harmonics of a pump are used. This yields data points at the eleven third-octave frequencies between 100 and 1000 Hz plus four overlapping data points.

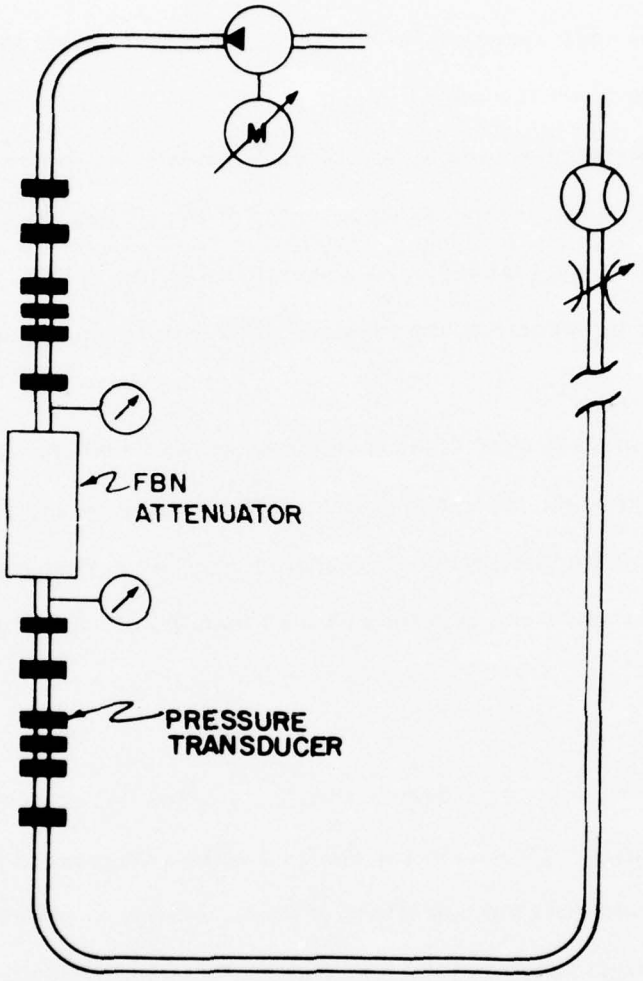
TABLE 2-1

Table of Test Frequencies Using Test Plan for Obtaining 15 Data Points for Evaluating Attenuator Performance

Desired 1/3 Octave Center Frequency (Hz)	TEST NUMBER				
	1	2	3	4	5
100	100*				
125		125*			
160			160*		
200	200				
250		250		(267)*	
315	(300)		(320)		315*
400		(375)			
500			(480)	(533)	
630					630
800				800	
1000					(945)

* Fundamental pumping frequencies

NOTE: Frequencies in parentheses deviate from desired frequencies.



Step 1: Verify anechoic termination and measure "tare" P.
Step 2: Measure pressure drop across attenuator.
Step 3: Determine energy incident to attenuator $(p_i)^2$.
Step 4: Determine energy transmitted through attenuator $(p_i)^2$.
Step 5: Measure sonic velocity.
Step 6: T. L. = $20 \log_{10} p_i/p_t$.
Step 7: Calculate pressure drop.

Fig. 2-2. Summary of Test Procedure Showing Schematic of Measurement and Loading Circuit

TEST SYSTEM

The critical components of the test circuit are illustrated in Fig. 2-2. The maximum flow of the pump at a speed to provide the 1000 Hz test pressure ripple should be less than the maximum allowable flow of the attenuator. The hydraulic lines from the pump to the load valve should be steel tubing of the source diameter, connected where necessary with unions that minimize discontinuities in the line. The transducer sensing surfaces should be tangent to the inside diameter of the steel conduit.

The range of attenuator sizes that can be accommodated by a given system can extend above and below the system conduit size if controls are placed on the transition lengths of porting adapters. Expansion chamber design curves [7] indicate that porting adaptor transitions with lengths less than 0.25λ should not affect the test results.

A test system, such as shown in Fig. 2-2, which has several meters of conduit between the measurement region and the load valve will provide a quasi-anechoic termination. This means that the variations of measurements as a function of x will be greatly reduced. If enough conduit is used so that $R \approx 0$, then, from Eq. (2-1), $p(x) = p_i$. If the measurement variations are less than 2 dB at any frequency, the reading can be averaged on a dB basis to obtain a value representative of p_i. Since large variations of the pressures will introduce the possibility of significant errors in obtaining p_i, it is reasonable to set some upper limit on the range of dB readings at a given operating condition. The test system should meet the qualification that, at any test condition, the range of dB readings shall not exceed 4 dB. This limit still allows an appreciable standing wave; the standing wave ratio is 1.6, or, p_{max} is equal to $1.6\, p_{min}$.

The transducer configuration should include at least three pressure transducers. It is recommended that six transducers be used with the spacing shown in Fig. 2-3. The configuration shown in Fig. 2-3 provides better accuracy than three transducers. The spacing is based on the Renard Series.

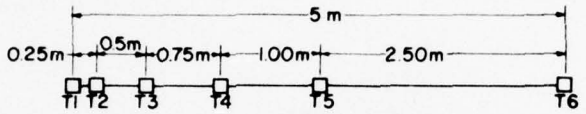


Fig. 2-3. Transducer Spacing for FBN Attenuator Performance Evaluation

TEST RESULTS

The attenuator selected for evaluation was an expansion chamber, which is a device with an increase in diameter at the inlet and a corresponding reduction in diameter at the outlet. The unit chosen for the test was symmetrical as described in Appendix E. The unit was tested for attenuation characteristics, input impedance and flow resistance. The results of the tests are described in the following paragraphs.

Transmission Loss

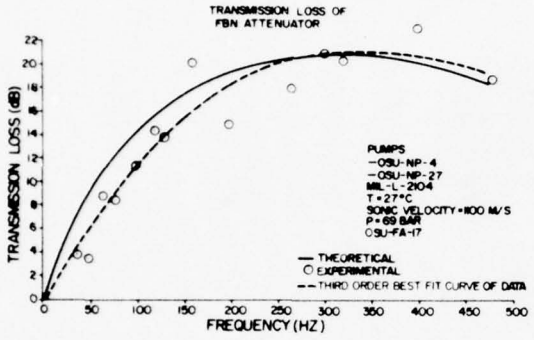


Fig. 2-4. Transmission Loss Curve for FBN Attenuator

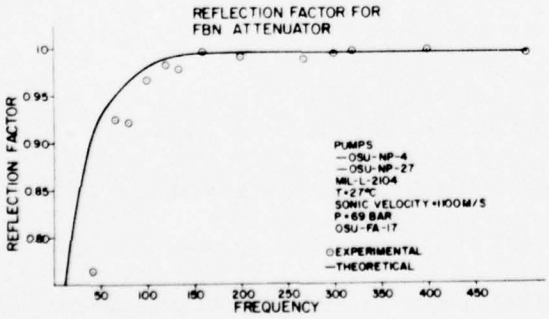


Fig. 2-5. Reflection Factor Curve for a FBN Attenuator

Fig. 2-4 compares the results of the transmission loss evaluation with the theoretical transmission loss for the expansion chamber. The experimental transmission loss was evaluated by using the OSU Six Transducer Data Reduction Program. The transducer configuration was the same as suggested in Fig. 2-3.

The scatter using the six transducers is significantly less when compared to the data scatter obtained from three transducers. The six transducer technique appears to give more accurate values of the transmission loss for the frequencies measured. The procedure for determining the transmission loss of a FBN attenuator is given in Appendix A.

Fig. 2-5 shows the reflection factor for the input to the expansion chamber. These values were determined using the OSU Six Transducer Data Reduction

Program. For a non-dissipative FBN attenuator, the reflection factor of the attenuator can be deduced from the transmission loss with the use of Eq. 2-9. These values are also shown in Fig. 2-5.

As is shown in Fig. 2-5, it appears that the OSU Six Transducer Data Reduction Program can directly provide accurate information about the reflection factor up to approximately 325 Hz. Above 325 Hz the program (and data) seem to give values of R that are smaller than expected. This might be more of a limitation on the measurement technique itself than the computer program. The data was not taken in "real time" so that any sonic velocity changes during the tests would have affected the measured pressure values. This would affect the fit that the computer program would have for the data.

Another possible source of error is the decrease in wavelength with a corresponding increase in frequency. For instance, at 100 Hz there are six transducers covering slightly less than half a wavelength. At 1000 Hz there are five wavelengths between the first and last transducer. With 5 "peaks" and "valleys" it is harder to locate the exact location of maximas and minimas since they are so much closer together. The ideal, of course, would be to have infinite number of transducers in order to directly measure the maximas and minimas. However, this is not realistic. More study should be done to determine the maximum and average error an investigator might expect from using the six stationary transducer technique.

CHAPTER III

PUMP FLUIDBORNE NOISE GENERATION POTENTIAL

With the growing concern for the reduction of the overall airborne noise level in hydraulic systems, attention has begun to be focussed on understanding the interactions that contribute to the overall airborne noise level. In particular, this has meant a more "in-depth" look at fluidborne noise and structureborne noise [1, 3, 4, 5, 7, 9, 10].

A major source of fluidborne noise in hydraulic systems is the positive displacement pump. Since the fluidborne noise level can contribute significantly to the overall airborne noise level in a system, the need has developed to adequately model and rate components that generate fluidborne noise; namely, the positive displacement pump.

The fluidborne noise (FBN) generated by fluid power pumps has been the subject of several papers and reports in recent years [2, 3, 4, 5, 9, 10]. Ichikawa and Yamaguchi [2] developed a model for the variation in flow and pressure (FBN) generated by a gear pump. From the data given, the model proved satisfactory for predicting the flow ripple of the gear pump. They consider the leakage (or source) impedance in calculations to determine the pressure ripple delivered to the line.

Willikens [3] presents a model for a gear pump and presents data to support his model. He, also, shows that the pressure ripple measured is dependent on both the flow ripple generated by the pump and the leakage impedance of the pump.

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Unruh [5, p.2] recognizes the need to ultimately relate the measured pressure ripple back to the flow ripple of the pump. He states: ". . . the basic phenomenon which actually needs to be established for determining the fluidborne noise characteristics of a pump is the amount of flow ripple that a pump generates due to its inherent construction characteristics . . . ."

If a pump can be considered as being analogous to a constant current source in an electrical circuit, then the pump will deliver a constant flow ripple to the system at a given load condition. Just as the load on a constant current source determines the voltage in the circuit, so the load conditions on a fluid power pump will determine the magnitude of the pressure ripple in the system. The pump also has a source impedance analogous to that of a constant current source. For a pump, the source impedance will depend on the geometry (or volume) of the pump casing and the amount of leakage flow [2,3].

The two important parameters in defining an electrical current source are the strength of current and the source internal impedance. Similarly, for a hydraulic pump, the inherent flow ripple of the pump and the source impedance are needed to adequately describe the pump in order to model and predict the fluidborne noise level in a hydraulic system. A general approach for evaluating these parameters of fluid power pumps is discussed in the next section.

TEST PROCEDURE CONCEPT

To determine the flow ripple and impedance of a fluid power pump, several different measurements must be taken. The flow ripple can be determined by measuring the pressure ripple.

The basic concept behind the proposed test procedure for determining the fluidborne noise potential of fluid power pumps involves two phases:

1. Determining the Flow Ripple - With today's technology, it is not industrially practical to directly measure the flow ripple of fluid power pumps. Pressure ripple measurements must be taken near the pump with an anechoic termination, and then converted to flow. This is discussed in the section, Flow Ripple Determination.
2. Determining the Pump Impedance - This phase involves coupling a known load impedance within 1/20 of a wavelength of the pump and measuring the pressure ripple between the pump and known load impedance. From this information, the source impedance can be calculated as is discussed in the section. Pump Impedance Determination. There are two distinct steps in the phase. The first is the evaluation of the load impedance and second, the evaluation of the source impedance.

Flow Ripple Determination

As stated in the previous section, determination of the flow ripple of the pump involves taking pressure ripple measurements near the pump, which is connected to anechoic termination. An anechoic termination would be a termination which, ideally, would produce no reflections. The noise in the fluid would propagate down the tube and not be reflected back toward the source. Since fluid power pumps have a wide range of pumping frequencies, an anechoic termination needs to be "non-reflective" over a wide range of frequencies.

The difficulty in obtaining a true, non-reflective termination has been discussed by Heymann [6]. He states: "*It is . . . difficult to obtain a practical reflection-free* termination in liquid systems, most approaches realize at best a termination in which reflections . . . are weak . . . ."

One approach to constructing an anechoic termination is to use very long lengths of line so that the pressure ripple is eventually dissipated as it travels down the line. For pipe diameters of one inch and flows above 3.5 gpm, a hydraulic circuit line length of between 35 - 70 meters is required. For larger diameters, the length required will be longer, since the frictional dissipation will decrease. Also, for larger flows, the length of line required will decrease as the flow increases. Generally, as the frequency increases, the length required should decrease. The dissipation term is thus:

$$D = F(d, Q, f) \qquad (3\text{-}1)$$

where: D = dissipation term

d = pipe diameter

Q = flow

f = frequency

The basic concept of measuring pump pressure ripple in a system with an anechoic termination is shown in Fig. 3-1. In Fig. 3-1, a pump is connected to an anechoic termination ahead of the system load valve and filter. If there are no reflections at the termination, then the pressure at any point in the hydraulic line will be the same, since there will be no standing waves. In such a case, the impedance at any point in the line

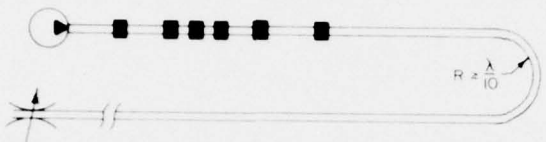


Fig. 3-1. Basic Anechoic Test System & Equation for Evaluating the Flow Ripple of a Fluid Power Pump

is just the characteristic impedance, Z_o. Thus, the flow ripple in an anechoic termination is related to the pressure as:

$$\tilde{Q}_o = \tilde{P}_o / Z_o \qquad (3\text{-}2)$$

where:

$\tilde{P}_o$ = measured pressure ripple

Z_o = characteristic impedance of the line = $\rho C/S$

ρ = density of the fluid

C = sonic velocity

S = cross-sectional flow area

The general relationship between the free volume flow ripple generated by the pump, Q_f, and flow ripple measured in the line, Q_L, is given by [18]:

$$Q_L = Q_f \frac{Z_S}{Z_S + Z_L} \qquad (3\text{-}3)$$

where Z_s and Z_L are the impedances of the pump and load, respectively. If an anechoic termination is used as the load, then $Z_L = Z_o$ and $Q_L = Q_o$, in which case:

$$Q_o = Q_f \frac{Z_S}{Z_S + Z_o} \tag{3-4}$$

There is a similar relationship between the pressure ripple measured in a line, P_L, and the pressure ripple generated, P_B, by the pump. This is [18]:

$$P_L = P_B \frac{Z_L}{Z_S + Z_L} \tag{3-5}$$

For an anechoic termination, this becomes:

$$P_O = P_B \frac{Z_o}{Z_S + Z_o} \tag{3-6}$$

A look at Eq. (3-4) shows, that if $Z_S >> Z_o$, then the flow ripple measured in an anechoic termination will be approximately the same generated by the pump--i.e.,

$$Q_o \approx Q_o \tag{3-7}$$

Also from Eq. (3-6), if the load impedance is much greater than the source impedance, $Z_o >> Z_{S1}$, then:

$$P_o \approx P_6 \tag{3-8}$$

As mentioned previously, a truly anechoic termination can only be approached in liquid systems. Since there will be small reflections, the easiest way to measure the incident pressure is by the use of multiple transducers. The recommended stationary transducer configuration is shown in Fig. 2-2. With the use of the OSU Six Transducer

Data Reduction Program in Appendix E, the reflection factor of the termination and the incident pressure from the pump can be determined. The maximum allowable reflection from the termination is a reflection factor of 0.23 which corresponds to a stadning wave ratio of 4 dB.

With the incident pressure from the pump known and the sonic velocity either measured or estimated by the computer program, the pump flow ripple can be calculated using Eq. (3-2).

To compare the magnitude of flow ripple for one pump with that of another, the flow ripple of the first and second harmonics should be added together for a particular speed. If this is done for several speeds, and a linear regression performed that yields a curve passing through the origin, then a single number rating can be used. This number would be:

$$\text{Rating} = q_{ToT}/N \tag{3-9}$$

where:

$q_{ToT} = q_1 + q_2$

N = speed

So, in order to calculate the approximate flow ripple for a certain pump, this number would only have to be multiplied by the speed. Obviously, this number does not give the purchaser of the pump enough information about the pump to adequately predict what it will do in a system; however, it does give the purchaser a number to compare with other pumps. A pump with a lower number would be one that would generate less flow

ripple and vice-versa.

The test procedure for determining the flow ripple of a pump is given in Appendix C.

Pump Impedance Determination

To evaluate the impedance of a pump, another series of tests must be performed in which a known load impedance is inserted in the system with the pump. A simple device that can be used for a load impedance is the orifice.

As previous tests indicate [9], it is desirable to experimentally determine the characteristics of the load impedance instead of just assuming it a purely resistive component. To evaluate the load impedance, the test procedure outlined in Appendix A should be followed. In this case, the load impedance is treated as a fluidborne noise attenuator with the transmission loss and reflection factor being determined from the data taken from the test.

For the general case of a load impedance, Z_T, downstream a distance, L, from the pump (See Fig. 3-2.), the pressure at any point, d (from the load), in the circuit is given by [7]:

$$P(d) = \frac{P_o e^{-\gamma L}}{(1 - \rho_s \rho_t e^{-2\gamma L})} (e^{\gamma d} + \rho_t e^{-\gamma d}) \qquad (3\text{-}10)$$

where:

P_o = pressure that would occur in an anechoicly terminated system

ρ_t = termination reflection factor

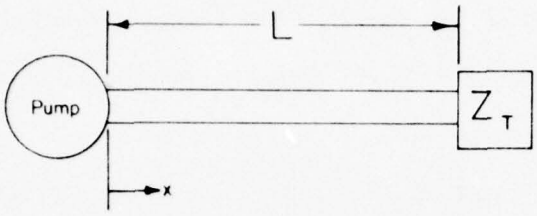


Fig. 3-2. Basic Test System Used to Evaluate the Source Impedance of the Pump

L = distance from pump to the load

ρ_s = source reflection factor

γ = propagation coefficient = $\frac{\omega}{c}$

ω = frequency (radians/sec)

Equation (3-10) can be rearranged to solve for ρ_s, the source reflection factor. Thus:

$$\rho_s = \left(1 - \frac{P_o e^{-\gamma L}}{P(d)} \left(e^{\gamma d} + \rho_t e^{-\gamma d} \right)\right) \frac{e^{2\gamma L}}{\rho_t} \qquad (3\text{-}11)$$

If it is assumed that the length of the line, L, is less than one-twentieth of the shortest wavelength of interest, then all the exponential terms in Eq. (3-11) drop out. In such a case, Eq. (3-11) becomes:

$$\rho_s = \left(1 - \frac{P_o}{P(d)} (1 + \rho_t)\right) / \rho_t \qquad (3\text{-}12)$$

Since P_o, P(d) and ρ_t are either measured or experimentally determinable quantities, ρ_s is the only unknown in Eq. (3-11) or Eq. (3-12). The source reflection coefficient, ρ_s, is related to the source impedance by the following equations:

$$Z_s = Z_o (1 + \rho_s) / (1 - \rho_s) \qquad (3\text{-}13)$$

where: Z_s = source impedance

Z_o = characteristic line impedance

The source impedance has both a magnitude and phase angle. The test procedure for determining the pump impedance is given in Appendix D. With both the source impedance and flow ripple determined, the fluidborne noise characteristics of a fluid power pump are sufficiently defined to predict the resultant pressure ripple in a system.

TEST SYSTEMS

A schematic of the anechoic termination test system was shown in Fig. 3-1. The anechoic termination was constructed out of 41 meters of 2.54 cm O. D. steel tubing and 39 meters of 2.54 cm O. D. steel hose. With long lengths of tubing and hose, an anechoic termination is accomplished by dissipating the pressure ripple. A valve, to allow pressurization of the system, and a filter are located downstream of the anechoic termination. Since the pressure ripple is significantly dissipated by the time it reaches the valve and filter any reflections from these components are minimized. Though there is dissipation in the line, it is assumed, for the flow rates used, that the effect of dissapation between transducer positions is negligible.

The transducers are piezoelectric pressure transducers. Each transducer is calibrated to within ± 1% accuracy. The transducers are mounted in individual mounts so that the tip of the transducer lies flush with the interior wall of the tubing. The transducer spacing is the same as shown in Fig. 2-3. Because of the generally long wavelengths of pressure waves in hydraulic oil, the transducers must be spaced far enough apart to adequately measure the standing waves in the system. A more thorough discussion of the use of stationary pressure transducers for fluidborne noise analysis can be found in Ref. [9 - 11].

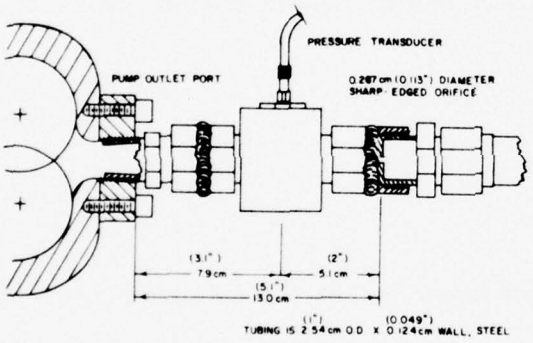


Fig. 3-3. "Close-Coupling" of Pump, Pressure Transducer & Orifice

The system used to measure the transmission loss and reflection characteristics of the orifice is the same as that used in the previous chapter (Fig. 2-1). Measurements are taken both upstream and downstream of the orifice. With the aid of the OSU-Six Transducer Data Reduction Program (See Appendix E.) the incident and transmitted pressure can be determined, as well as the reflection factor and the location of the standing wave maxima and minima (which yields the reflection factor phase angle) upstream of the orifice.

Shown in Fig. 3-3 is the orifice, pump, and pressure transducer arrangement used to evaluate the pump impedance. The orifice is "close-coupled" to the pump, being less than 1/20 of a wavelength away from the pump.

Test Results

The procedures outlined in this report for measuring the fluidborne noise characteristics of a hydraulic pump require the use of an anechoic termination. This section presents the results of evaluations of the anechoic termination, the flow ripple of several pumps, the reflection characteristics of an orifice and the internal impedance of a pump for several frequencies.

Anechoic Termination

Plotted in Fig. 3-4 is the reflection factor for the anechoic termination as a function of frequency. Pressure measurements were taken with six stationary pressure transducers. The reflection factor was determined from the pressure measurements using the OSU Six Transducer Data Reduction Program.

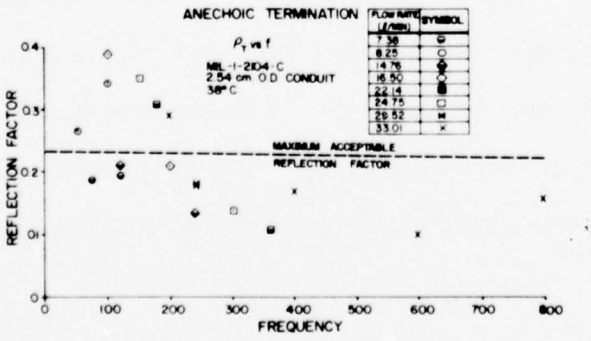


Fig. 3-4. Reflection Factor for Anechoic Termination Configuration Showing the Recommended Maximum Acceptable Reflection Based on 4 dB Standing Wave

This program reconstructs the standing wave in the system given the six pressure measurements and frequency. The incident pressure, reflection factor, sonic velocity, and position of standing wave maxima are varied until a "best fit" combination is found for the data input.

The largest experimental value of ρ_T was 0.3878 at 100 Hz. This, of course, is over the maximum acceptable value for the reflection factor. This was true of several other frequencies as shown in Fig. 3-4.

As the flow rate and frequency increases, the general trend is for the reflection factor to decrease. The frictional dissipation is both a function of the flow rate and frequency. As either the flow rate or frequency increase so does the frictional dissipation. Since the performance of this type of anechoic termination depends significantly upon frictional dissipation, an increase in this factor should decrease the reflection factor as shown in Fig. 3-4.

Flow Ripple in the Line

Three identical gear pumps were used to obtain an average value of experimental flow ripple in the line, Q_o, from a particular model pump and to determine a variation between units that should be "identical." The flow ripple is zero at zero RPM, and theoretically should increase linearly as the speed of a gear pump is increased [2].

Based upon what the theoretical values should be, straight lines were fitted through the origin for the data. Fig. 3-5 and Fig. 3-6 show the data and the fitted lines for the first two harmonics for the three pumps used. Because of the high reflection

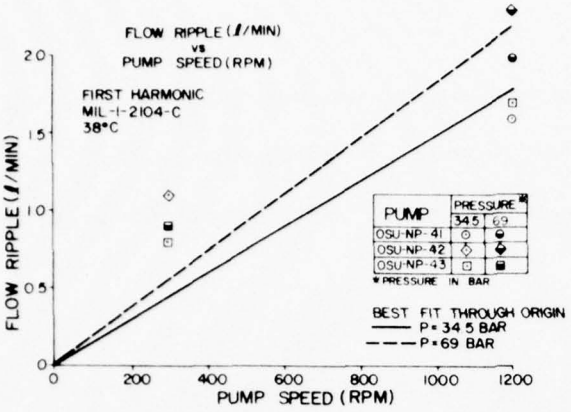


Fig. 3-5. Flow Ripple Versus Pump Speed Showing Experimental Values and Best Fit Line Through Origin for the First Harmonic

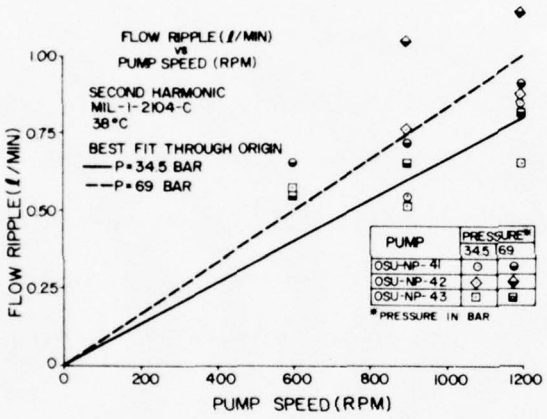


Fig. 3-6. Flow Ripple Versus Pump Speed Showing Experimental Values and Best Fit Through Origin for the Second Harmonic

encountered at the low flow rates generated by the pumps, several data points for both the first and second harmonics were not used. Since the termination would not be considered "anechoic" at those particular combinations of flow rate and frequencies, the experimental values of flow ripple would normally be erroneous.

The calculated values for the slopes of the best fit lines through the origin are listed in Table 3-1. A "t" test was performed on the data to test the hypothesis: "the fitted line goes through the origin." The results of the test are also shown in Tables 3-1 and 3-2. In all cases, the hypothesis could not be rejected which indicates that it is acceptable to fit this set of data with a line that goes through the origin.

The fitted line for a system pressure of 69 BAR in both cases had a slope larger than the ones for system pressures of 34.5 BAR. However, because of the small number of data points in each case and the large variation between units, it was not possible to distinguish any significant difference between the two lines. For instance, the limit on the slope of the second harmonic, 34.5 BAR line was 2.65×10^{-5} (ℓ /min)/RPM for the lower limit and 1.308×10^{-3} (ℓ/min)/RPM for the upper limit. The slope of the second harmonic 69 BAR line was, from Table 3-1, 8.52×10^{-4} (ℓ /min)/RPM which is well within the limits of the accepted slope.

HARMONIC	SYSTEM PRESSURE (BAR)	SLOPE OF LINE (ℓ/MIN)/RPM	CALCULATED 't'
1	34.5	1.48×10^{-3}	0.436
	69.0	1.82×10^{-3}	0.602
2	34.5	6.67×10^{-4}	0.707
	69.0	8.52×10^{-4}	0.662

TABLE 3-1. Slope of Flow Ripple Line and Statistical Information for for Two Harmonics and Two Pressures

FREQUENCY (Hz)	REFLECTION FACTOR	PHASE ANGLE (DEGREES)
200	0.833	1.49
400	0.9477	5.78
600	0.9914	17.59
800	0.9450	6.68

TABLE 3-2. Reflection Factor and Phase Angle for the Orifice

These preliminary flow ripple tests seem to indicate, that, for a given pump model, there can be a large amount in variation of flow ripple between units. As wide a variation as 4.3 dB (64.8%) was observed in the 900 RPM second harmonic, 69 BAR, flow ripples between units OSU-NP-42 and OSU-NP-43. The largest observed standard deviation for any one data point was less than 0.6 dB which seems to indicate that the differences between the various pumping units was due to something inherent in the pumping units and not measurement error.

Pump Impedance

The experimental reflection factor and phase angle of the orifice are listed in Table 3-2. The phase angle of the orifice was calculated by determining how far the standing wave maxima was shifted away from the orifice toward the pump. The position of the standing wave maxima was calculated in the OSU Six Transducer Data Reduction Program (See Appendix E.).

The experimental reflection factor of the orifice is significantly larger than that calculated by only considering the resistive component of the impedance. For instance, at a flow rate of 33.01 ℓ/min. and pressure drop of 56.4 BAR, the expected reflection factor would be 0.7421 if only resistance were considered.

With the reflection factor and phase angle of the orifice known, the orifice was "close-coupled" to the pump as was shown in Fig. 3-4. The pressure was measured using the transducer, and its value was used in Eqs. (3-5) and (3-6) to determine the source reflection factor and impedance. These values are listed in Table 3-3. The first four harmonics were used at the fastest test speed (1200 RPM) of the pump.

FREQUENCY (Hz)	P_s	Φ^*	Z_s^{**}	θ^*
200	0.976	14.110	2.498	84.4
400	0.601	31.367	0.829	44.34
600	0.522	37.177	0.678	40.90
800	0.702	64.483	0.447	68.17

TABLE 3-3. Reflection Factor and Impedance of Pump OSU-NP-43 For Various Frequencies

The impedance of the pump appears to decrease with increasing frequency as shown in Fig. 3-7 and as expected with an analytical model of a positive displacement pump as proposed by Davidson [18]. In the analytical model the pump impedance is modeled proportional to the inverse of the frequency squared and the internal volume of the pump as given by the expression:

$$Z_s = \frac{\rho c^2}{j \omega V} \tag{3-14}$$

where:

ρ = density of fluid medium

c = sonic velocity

ω = frequency in rad/sec

V = internal volume of pump

An internal volume of the pump OSU-NP-43 of 34.4 cm^3 (2.1 $in.^3$) and the average sonic velocity of 1207.5 m/sec was used to calculate the theoretical impedance of the pump shown in Fig. 3-7. The largest observed deviation was at 400Hz where the

experimental and theoretical values of impedance were off by less than 3.5. (This is a dimensionless number because the logarithim of the pump impedance divided by the characteristic impedance of the line is a dimensionless quantity.)

Pump Flow Ripple

With the impedance of the pump and the flow ripple in the anechoic termination known, it was then possible to determine the pump flow ripple using Eq. (3-4). This was done for pump CSU-NP-43 at a system pressure of 69 BAR for two different anechoic terminations. The results are shown in Figs. 3-8 and 3-9 for the first and second harmonics, respectively.

Again, due to excessive reflections at some frequencies, it was not possible to use all the data. That is why, for instance, at the first harmonic, data is not plotted for the 2.16 cm I. D. conduit anechoic termination at pump speeds of 600, 900, and 1200 RPM. One anechoic termination was constructed of 2.16 cm I. D. steel hose and tubing, while the other was constructed of 1.08 cm I. D. steel hose and tubing.

A linear regression was performed on the data for both harmonics. For the first harmonic, the correlation coefficient, r, was 0.9789. The values of the coefficients, A and B, for the regression line, $A + B_x$, were A = .410 ℓ/min and B = .0015 ℓ/min/RPM. For the second harmonic, r = 0.9485, A = .400 ℓ/min, and B = .00031 ℓ/min/RPM. Because of the large correlation coefficient, it would appear that a linear relationship exists in the range studied between the free volume flow ripple (what the pump actually generates) and the speed of the pump. For the two different anechoic terminations, that data at any one point for the free volume flow ripple was, in all cases, within 5% of each other. This would indicate the validity of Eq. (3-4) which states that the free

volume flow ripple of the pump is independent of the load impedance. Since, in the data taken, two different load impedances were used with an equivalent free volume flow ripple determined.

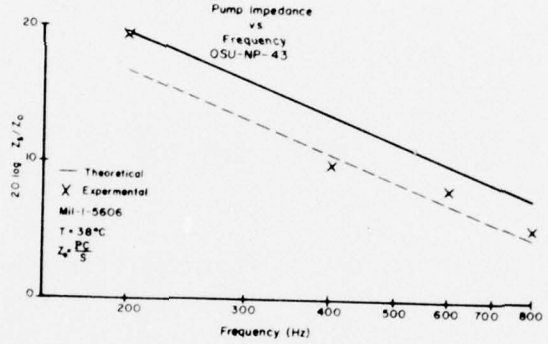


Fig. 3-7. Pump Impedance as a Function of Frequency (OSU-NP-43) Nondimensional

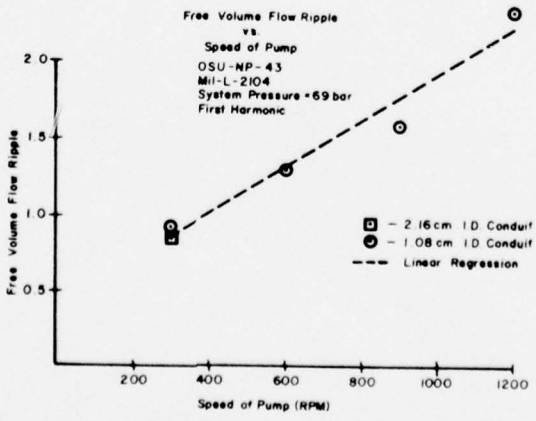


Fig. 3-8. Free Volume Flow Ripple as a Function of Pump Speed--First Harmonic

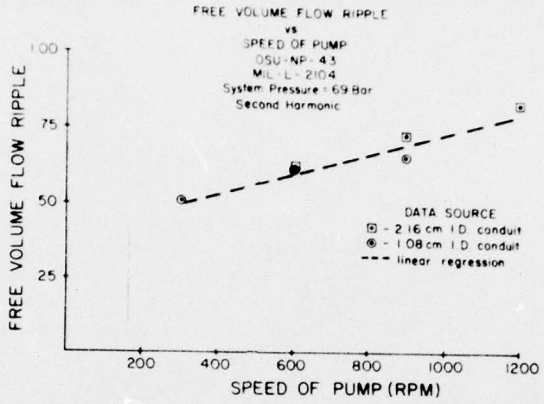


Fig. 3-9. Free Volume Flow Ripple as a Function of Pump Speed--First Harmonic

CHAPTER IV

COMPUTER PROGRAM

The purpose of this chapter is to discuss the theory behind the OSU Six Transducer Data Reduction Program and outline how this program can be used to assist in analysis of the fluidborne noise (FBN).

Theory

In many hydraulic systems, wave propagation in lines often can be considered one dimensional and frictionless. For a simple system such as that shown in Fig. 4-1, there is a FBN source, such as a pump, a transmission line, and some downstream impedance, Z_T, that has a reflection factor, ρ_t.

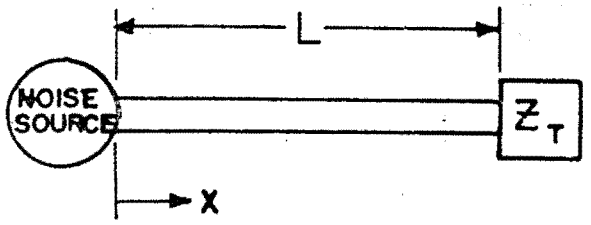


Fig. 4-1. Schematic of Simple Acoustic System Consisting of Source, Transmission Line, and a Lead Impedance, Z_T

The pressure ripple that the FBN source generates at a given frequency can be written in the trigonometric form:

$$P = P_o \text{Cos}(\omega t - kx) \tag{4-1}$$

where:

P_o = amplitude of pressure ripple

ω = angular frequency in radians/sec

t = time

$k = \frac{\omega}{c}$

c = sonic velocity

x = distance

Equation (4-1) describes a pressure wave of amplitude, P_o, traveling in the positive x-direction.

When the pressure wave reaches the termination impedance, part of it is reflected back toward the source, and part of it will continue traveling in the "positive x-direction." The portion reflected back toward the source can be described as:

$$p_r = \rho_t P_o \text{Cos}(\omega t + Kx) \tag{4-2}$$

Now there is a "left" and "right" traveling wave in the system shown in Fig. 4-1. The wave described by Eq. (4-2) will travel to the left until it reaches the source, at which time a portion of it will be reflected back toward the right. It will have a magnitude of $\rho_T \rho_s P_o$. This wave will then travel down the pipe until it is reflected, etc.

The sum of the "right" traveling waves incident on the termination is given by:

$$P_w = P_I \cos(\omega t - kx) \tag{4-3}$$

where:

$$P_I = P_o(1 + \rho_T \rho_s + (\rho_T \rho_s)^2 + (\rho_T \rho_s)^3 + \ldots) \tag{4-4}$$

Similarly the sum of the "left" traveling wave is:

$$P_L = P_R \cos(\omega t - kx) \tag{4-5}$$

where:

$$P_R = \rho_T P_I = \rho_T P_o(1 + \rho_T \rho_s + (\rho_T \rho_s)^2 + (\rho_T \rho_s)^3 + \ldots) \tag{4-6}$$

The pressure ripple at any one point in the system is just the combination of the "left" and "right" traveling waves:

$$P(x,t) = P_I \cos(\omega t - kx) + P_R \cos(\omega t + kx) \tag{4-7}$$

It would be desirable to be able to describe the magnitude of the standing wave in the system as just a function of x. In other words, to separate Eq. (4-7) into a magnitude and phase component of the form [12]:

$$P(x,t) = P(x) \cos(\omega t + \psi) \tag{4-8}$$

If the trigonometric identities:

$$\text{Cos}\,(\omega t - kx) = \cos\omega t\ \cos kx + \sin\omega t\ \sin kx$$

and

$$\text{Cos}\,(\omega t + kx) = \cos\omega t\ \cos kx - \sin\omega t\ \sin kx$$

are substituted into Eq. (4-7), and both sides squared, the resultant reduces to:

$$P(x,t) = [P_I^{\,2} + P_R^{\,2} + 2P_I P_R \cos 2kx]^{1/2} \cos(\omega t + \psi) \qquad (4\text{-}9)$$

where:

$$\psi = \tan^{-1}\frac{P_I - P_R}{P_I + P_R} \tan kx \qquad (4\text{-}10)$$

Equation (4-9) is in the form required by Eq. (4-8). The magnitude P(x) of the standing wave is:

$$P(x) = [P_I^{\,2} + P_R^{\,2} + 2P_I P_R \cos 2kx]^{1/2} \qquad (4\text{-}11)$$

Since $\rho_T = \frac{P_R}{P_I}$ this substitution can be made into Eq. (4-11) to obtain:

$$P(x) = P_I^{\,2}[1 + \rho_T^{\,2} + 2\rho_T \cos 2kx]^{1/2} \qquad (4\text{-}12)$$

If stationary pressure transducers are used to measure the FBN in a system, then, at least three measurement locations are needed to solve the unknowns (P_I, ρ_T and x) in Eq. (4-12). If six transducers are used in the configuration shown in Fig. 4-2,

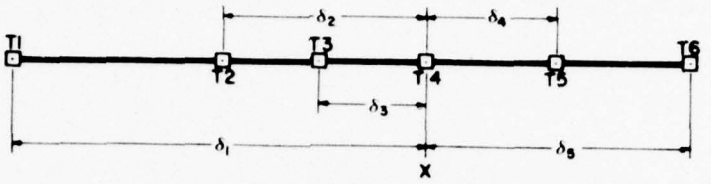


Fig. 4-2. Six Pressure Transducers Arbitrarily Spaced in a FBN Measurement System

then, the pressure ripple measured at each transducer can be related to x, and R by the following set of equations:

$$P_1(x) = P_I^2 [1 + \rho_T^2 + 2 \rho_T \cos 2k (x - \delta_1)]^{1/2} \quad (4\text{-}13)$$

$$P_2(x) = P_I^2 [1 + \rho_T^2 + 2 \rho_T \cos 2k (x - \delta_2)]^{1/2} \quad (4\text{-}14)$$

$$P_3(x) = P_I^2 [1 + \rho_T^2 + 2 \rho_T \cos 2k (x - \delta_3)]^{1/2} \quad (4\text{-}15)$$

$$P_4(x) = P_I^2 [1 + \rho_T^2 + 2 \rho_T \cos 2k (x - \delta_4)]^{1/2} \quad (4\text{-}16)$$

$$P_5(x) = P_I^2 [1 + \rho_T^2 + 2 \rho_T \cos 2k (x - \delta_5)]^{1/2} \quad (4\text{-}17)$$

$$P_6(x) = P_I^2 [1 + \rho_T^2 + 2 \rho_T \cos 2k (x - \delta_6)]^{1/2} \quad (4\text{-}18)$$

The choice of lettering transducer 4(b) at x was completely arbitrary. The value of x gives the distance that transducer 4 is relative to the maxima of the standing wave.

Generally speaking, the computer program finds the best fit of Eq. (4-13) through Eq. (4-18) to the data by varying x, ρ_T, P_I, and sonic velocity.

The Program

The following are needed as input to the computer program:

(1) the measured values of the FBN at the six transducer positions

(2) the frequency in cycles/sec

(3) the distances δ_1, δ_2, δ_3, δ_4, and δ_5 as shown in Fig. 4-2.

The first computational step in the program is converting the pressure in dB to PSI. This is accomplished in the subroutine PSI. This particular subroutine is designed for use with the calibration of instrumentation used here at the Fluid Power Research Center. A certain reading is taken straight from the charts and this number fed as input into the program. This number needs 138.3 added to it to get the actual pressure in dB. This is built into the PSI subroutine. If this program is to be used in other facilities, then, this subroutine can be altered to suit the particular needs of that facility. The pressure in PSI is, then, calculated by the subroutine.

The program, then, locates the measured maximum and minimum pressure readings from the data given. These are calculated for two reasons:

(1) It gives an initial estimated value of the reflection factor, ρ_T. The reflection factor is related to the actual maximum and minimum pressures in the pipe at a certain frequency by equation:

$$\rho_T = \left(\frac{p_{max}}{p_{min}} - 1\right) / \left(\frac{p_{max}}{p_{min}} + 1\right) \qquad (4\text{-}19)$$

If a certain p_{max} and p_{min} are measured, then, the actual value of the reflection factor must, at least, be as large as that corresponding to the measured p_{max} and p_{min}.

(2) The second reason p_{max} and p_{min} are located is to allow an initial "guessing" point for the value of x, which is the distance that transducer 4 is from the actual maximum of the standing wave.

As has been stated previously, the four variables, x, ρ_T, P_I, and the sonic velocity are varied in order to best fit the expected curve to the data. The best fit is found by "nesting" four do-loops within each other similar to that shown in Fig. 4-3.

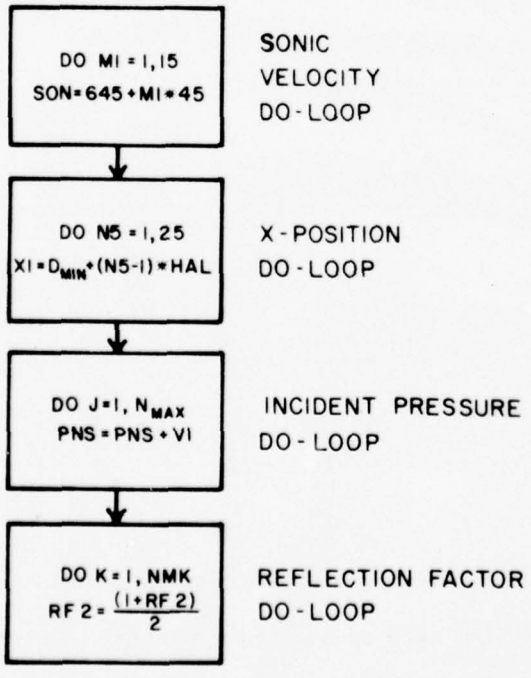


Fig. 4-3. Simple Flow Diagram of Important Computational "Nested" Do-Loops in Program

A four-dimensional matrix, in essence, is searched in fixed increments until a best fit is found. The incremental size step on the sonic velocity is 45 m/sec starting at 645 m/sec and moving up to 1320 m/sec. The position, x, is varied 25 equal increments over a wavelength.

The reflection factor can be varied with equal increments or unequal increments dependent on the initial value of the estimated reflection factor. For a reflection factor $\rho_T \leq 0.75$ the program will break the value $1 - \rho_T$ into twenty equal segments. If $\rho_T > 0.75$, the program will use a value:

$$\rho_{T_{n+1}} = 1 - \left[\frac{1 - \rho_{T_n}}{2}\right]$$

as incremental values of the reflection factor.

The incident pressure begins with an estimated value:

$$P_I = \frac{(P_{max} + P_{min})}{4}$$

It is, then, varied as:

$$P_{I_{r+1}} = \left[\frac{(P_{max} + P_{min})}{2} - P_{I_r}\right] * 0.4$$

The use of the computer program provides an approximation of the parameters needed to define the standing wave in a system. It is only as good as the theory behind it and the accuracy of the pressure transducers. If there is a significant amount of dissipation in a hydraulic line, then, the basic equations would have to be modified to include

dissipation. In that case, a new unknown would be entered into the equations.

However, for the case where dissipation is not a problem, the program can be used with only minor mocifications to suit a particular facility to estimate the standing wave parameter.

CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

Both of the proposed test procedures have been tested in the laboratory and modified to incorporate the results of the experimental validation as well as the inputs from industry. Both procedures form valid nuclei for industrially acceptable test codes.

The test code for evaluating the fluidborne noise generation potential of hydraulic pumps is based on the use of an anechoic termination in the hydraulic load system. This recommendation does not preclude the use of a procedure which requires the use of a non-anechoic termination in the test system.

A test code under consideration by the NFPA Project Group T3.9.24 incorporates two non-anechoic terminations to obtain the flow ripple and impedance of a pump. The code was developed by Davidson [18]. Tests are currently being conducted by several individuals on the NFPA committee to test the repeatability of the test code. The pumps used for the tests were provided by the FPRC.

It is recommended that the test code developed by Davidson for the NFPA committee be compared to the test code developed by FPRC personnel. The basis of the comparison should be tests performed on the same pumping units. Any differences in either the flow ripple or impedance for the two codes should, then, be resolved.

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Both the fluidborne noise attenuator and pump fluidborne noise test procedure use the digital computer in data reduction. It is recommended that the computer program continue to be developed to provide direct computation of the transmission loss of FBN attenuators and FBN characteristics of pumps.

For all fluidborne noise measurements and predictions in hydraulic systems, a note of caution should be observed regarding the magnitude of the oscillatory pressures, which are usually significant relative to the mean pressure levels. Nonlinearities resulting because of the high noise/mean pressure ratios will become more evident as we gain more knowledge in this area. They may not be a major concern.

The MERADCOM-OSU Fluidborne Noise Attenuator Test Procedure outlined in this report should be submitted to industry for review.

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APPENDIX A

METHOD FOR EVALUATING THE PERFORMANCE OF A FLUIDBORNE NOISE ATTENUATOR

INTRODUCTION

In fluid power systems, power is transmitted and controlled through a fluid under pressure within an enclosed circuit. Pumps convert mechanical power into fluid power. Sound is created in the fluid during the power conversion process. In fluid power systems, fluidborne noise can be controlled with the use of fluidborne noise attenuators. The attenuator can reflect sound energy back toward the source, and absorb sound energy within it.

The need for a technique to evaluate the performance of fluidborne noise attenuators has long been recognized [12]. One of the primary considerations in the selection of a fluidborne noise attenuator is its effectiveness in reducing the fluidborne noise in a system.

1. Scope
   To include rating basis, test methods and method of reporting transmission loss, pressure drop, and impedance of fluidborne noise attenuators.

2. Purpose
   To establish a uniform standard for measuring, reporting and accurately comparing the transmission loss, pressure drop, and impedance of fluidborne noise attenuators.

3. Terms and Definitions
   3.1 For definition of Fluid Power terms used in this document, see Ref. [13].

   3.2 For definition of Acoustical Terms used in this document, see Ref. [14].

   3.3 Anechoic Termination. A termination in a hydraulic system which essentially reflects none of the incident sound energy over the frequency range of interest.

4. Unit of Measurement
   The International System of Units (SI) is used in accordance with Ref. [15] dB relative to 20 μPa.

5. Graphic Symbols
   Graphic symbols used are in accordance with Refs. [16] and [17]. Where Refs. [16] and [17] are not in agreement, Ref. [17] governs.

6. Test Equipment
   6.1 Use standard measurement devices to measure oil flow, oil pressure, pump speed and oil temperature.
   6.2 Use one-third octave band or narrow band analyzer to measure fluid-borne noise levels.
   6.3 Use oscilloscope to measure sonic velocity.
   6.4 Use oil as the test fluid.

7. Test Conditions Accuracy
   7.1 Maintain these test conditions during the test within the limits specified in Table A-1.

TABLE A-1

Test Conditions Accuracy

TEST CONDITION	CUSTOMARY U. S. UNIT	S. I. UNIT	MAINTAIN WITHIN (±) OF ACTUAL
FLOW	USGPM	ℓ/MIN	± 2%
PRESSURE	PSI	BAR	± 2%
SPEED	RPM	REV MIN	± 2%
TEMPERATURE	° F	° C	38 ± 2°C
SONIC VELOCITY	FT/SEC	M/SEC	± 5%

8. Test Procedure
   8.1 Install Anechoic Termination
   8.2 Construct Hydraulic Circuit
      8.2.1 Install in the circuit oil filters, oil coolers, reservoirs, and control valve as needed downstream of the anechoic termination.
      8.2.2 Use test fluid and filtration in accordance with the manufacturers' recommendation.
      8.2.3 Install inlet and discharge line diameters in accordance with the manufacturers' recommended practice. Exercise extra care when assembling inlet lines to prevent air leaking into circuit.

8.3 Install Pressure Transducers
- 8.3.1 Install two sets of pressure transducers as shown in Fig. 2-2.
- 8.3.2 Insure that transducer spacing is the same as in Fig. 2-3 with spacing tolerance within ± 2%.
- 8.3.3 Place a section of test system, equivalent in length to the attenuator, between the two sets of pressure transducers.

8.4 Verify Anechoic Termination
- 8.4.1 Set system pressure to maximum test pressure for FBN attenuator.
- 8.4.2 Set speed of pump so that the fundamental pumping frequency is at 100 Hz.
- 8.4.3 Record pressure transducer measurements as a function of frequency in dB, for first three harmonics of fundamental pumping frequency.
- 8.4.4 Measure a "tare" pressure drop vs. flow rate.
- 8.4.5 Measure sonic velocity per OSU-FPRC procedure [12].
- 8.4.6 Repeat 8.4.2, 8.4.3, 8.4.4, and 8.4.5 for fundamental pumping frequencies of 125, 160, 267, and 315 Hz.
- 8.4.7 Insure that the range of dB readings $\leq$4 dB for all test frequencies.
- 8.4.8 Record all data in data table similar to Table A-2.

8.5 Measure Attenuator Performance
- 8.5.1 Remove the section of test system equivalent in length to the attenuator between the sets of pressure transducers.
- 8.5.2 Install FBN attenuator.
- 8.5.3 Repeat step 8.4.1.
- 8.5.4 Repeat step 8.4.2.
- 8.5.5 Repeat measurements in 8.4.3.
- 8.5.6 Measure flow and pressure drop across attenuator.
- 8.5.7 Relieve system pressure to a minimum pressure.
- 8.5.8 Increase system pressure to that in 8.5.3.
- 8.5.9 Repeat measurements in 8.5.5 and 8.5.6.
- 8.5.10 Repeat steps 8.5.7 and 8.5.8.
- 8.5.11 Repeat measurements in 8.5.9.
- 8.5.12 Repeat steps 8.5.4 up to and including 8.5.11 for fundamental pumping frequencies of 125, 160, 267, and 315 Hz as indicated in Table A-2.

8.6 Invert attenuator and repeat steps 8.5.3 through 8.5.12.

9. Calculations
- 9.1 Obtain pressure drop versus flow due to the attenuator by correcting 8.5.6 with 8.4.4.
- 9.2 Calculate the transmission loss and impedance of the attenuator from data taken in 8.5.

10. Data Presentation
- 10.1 Report pressure drop versus flow rate in a form similar to that in Fig. A-1.
- 10.2 Report attenuator transmission loss as a function of frequency as shown in Fig. A-2.
- 10.3 Report attenuator input reflection factor as a function of frequency as shown in Fig. A-3.
- 10.4 Report attenuator output reflection factor as a function of frequency as shown in Fig. A-3.

TABLE A-2. Data Table for FBN Attenuator Test.

Date: _____ Attenuator I.D. _____ Pump I.D. _____

Pump Speed: _____ System Pressure: _____ Flow Rate: _____

Fluid: _____ Fluid Viscosity: _____ Fundamental Pumping Frequency: _____

Tare Pressure Drop: _____ Pressure Diff. _____

Harmonic	Transducer Location	Measurement	Pressure Level at Transducer (dB)						Sonic Velocity
			P_1	P_2	P_3	P_4	P_5	P_6	
f_1	Up	(1) (2) (3)							
		μ							
	Down	(1) (2) (3)							
		μ							
f_2	Up	(1) (2) (3)							
		μ							
	Down	(1) (2) (3)							
		μ							
f_3	Up	(1) (2) (3)							
		μ							
	Down	(1) (2) (3)							
		μ							

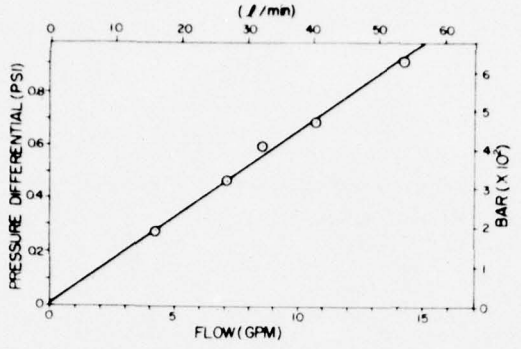


Fig. A-1. Pressure Drop Versus Flow Rate

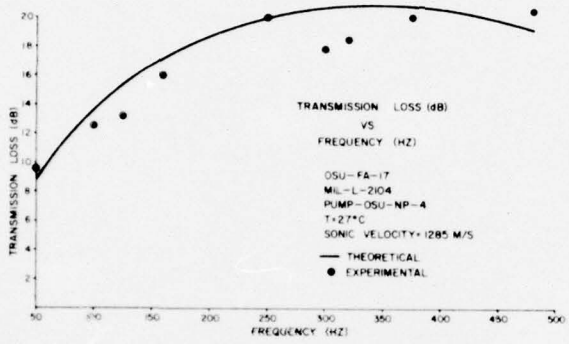


Fig. A-2. Transmission Loss Versus Frequency

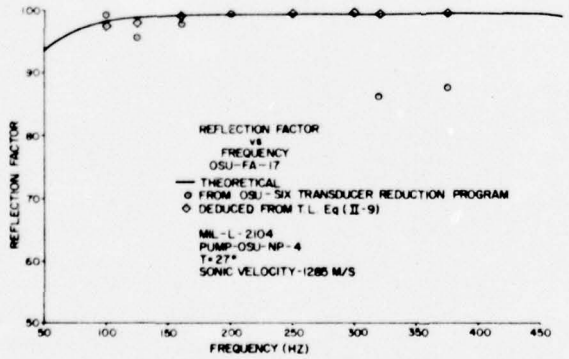


Fig. A-3. Reflection Factor as a Function of Frequency

11. Summary of Designated Information

The following designated information is needed when applying this recommended standard to a particular application or use.

11.1 Transmission loss spectrum

11.2 Pressure drop versus flow

11.3 Input reflection factor spectrum

11.4 Output reflection factor spectrum

APPENDIX B

PUMP FLUIDBORNE NOISE CHARACTERISTICS TEST PROCEDURE

As discussed in Chapter III, knowledge of two pump characteristics is necessary in order to adequately define the pump's acoustical performance in any given system. They are:

1. The flow ripple output of the pump (analogous to the current output of an electrical current source).
2. The pump's internal impedance (analogous to the internal impedance of an electrical current source).

The test procedure for evaluating the pump flow ripple output is given in Appendix C. Appendix D gives the test procedure for evaluating the pump's internal impedance.

APPENDIX C

PUMP FLOW RIPPLE EVALUATION TEST PROCEDURE

1. Mount pump and pressure transducers with anechoic termination.
2. Test anechoic termination:
   a. Set mean system pressure to maximum rating.
   b. Insure that the range of dB readings ≤ 4 dB at each frequency over the desired frequency range.
3. Measurement of pump flow ripple:
   a. After establishing condition 2(b), test the pump at the following speeds and maximum pressure: one-fourth, one-half, three-fourths, and maximum speed. Record the first three harmonics for each setting.
   b. Repeat 3(a) for one-half maximum pressure.
4. Reducing and reporting the data:
   a. From the tests in Part 2, use the OSU Six Transducer Data Reduction Program to determine the reflection factor for the anechoic termination. The data should be plotted in the same format as Fig. A-3.
   b. From the tests in Part 3, use the OSU Six Transducer Data Reduction Program to determine the pressure ripple in the pipe and the sonic velocity in the fluid.
   c. Determine the value of the pump flow ripple, Q_f, using Eq. (3-2) and Eq. (3-4).* The flow ripple should be plotted as a function of the speed of the pump with two different harmonics. The overall value of the flow ripple can be obtained for each speed by adding q_1 and q_2. A value of

* See Appendix D.

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q/N should be obtained by performing a linear regression on the plot of q_{TOT} vs. speed.

APPENDIX D

PUMP IMPEDANCE EVALUATION TEST PROCEDURE

1. Mount pump and pressure transducers with anechoic termination.
2. Test anechoic termination:
   a. Set mean system pressure to maximum rating.
   b. Insure that the range of dB readings ≤ 4 dB at each frequency over the desired frequency range.
3. Measurement of incident pressure, P_o:
   a. After establishing condition 2(b), test the pump with the anechoic termination at maximum pressure and one of the speeds used in the Flow Ripple Evaluation per Appendix C. Record the first three harmonics from each transducer.
4. Obtain an orifice which will produce a pressure drop equal to the maximum pressure rating at the pump flow corresponding to the speed chosen in 3(a). Install orifice in test system per Fig. 2-3.
5. Test orifice as an attenuator per Appendix A. Use pump speed per 3(a). Reduce data per Appendix A and obtain ρ_t for the first three harmonics.
6. "Close-couple" pump, transducer, and orifice per Fig. 3-4.
7. Test system at speed per 3(a). Record first three harmonics from transducer (P (d)).
8. Reducing and reporting the data:
   a. From the results of Part 2, use the OSU Six Transducer Data Reduction Program to determine the reflection factor (ρ_{tt}) for the anechoic termination at the first three harmonics. Record on Table D-1.

b. From the results of Part 3, use the OSU Six Transducer Data Reduction Program to determine the incident sound pressure from the pump, P_o, for the first three harmonics. Record on Table D-1.

c. Reduce data from Part 5 per Appendix A to determine ρ_{to} for the orifice for the first three harmonics. Record on Table D-1.

d. Record P(d) data from Part 7 on Table D-1.

e. Using Eq. (3-6) determine the pump reflection factor, ρ_s. Record on Table D-1.

f. Using Eq. (3-7) determine the pump impedance, Z_s. Record on Table D-1.

TABLE D-1. Data Recording Table for Pump Impedance Test

Harmonic	Frequency	ρ_{tt} Termination	P_o	ρ_{to} (Orifice)	P(d)	ρ_s (Pump)	Z_s (Pump)
1							
2							
3							

Date: _____ Fluid Temp: _____ Sonic Velocity: _____

Pump I.D.: _____ Fluid: _____ Max. Outlet Press.: _____

Max. Pump Speed: _____ Fluid Viscosity: _____

APPENDIX E

THE OSU SIX TRANSDUCER DATA REDUCTION PROGRAM

The OSU Six Transducer Data Reduction Program is a computer program which uses the sound pressure data supplied by the stationary pressure transducers to provide a best estimate of the sound pressure distribution within the system and the system's accoustical characteristics.

Inputs to the program are the sound pressure level readings (in dB) directly from the analyzer charts and the pumping frequency (fundamental or harmonic) under investigation. From this information the program deduces the incident sound pressure (P_i), the reflection factor (ρ_s), the estimated sonic velocity (c), the position of the standing wave maximum (X), the error squared between measured pressures and "best fit" pressures, and the "best fit" pressures at the transducer locations.

The program is listed on pp. 66 - 76. Page 77 is an example print-out of the program's output for two frequencies.

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```
1  RELEASE 2.0              MINP              DATE = 76335          12/55/53

      SUBROUTINE MINP(P,PMIN,DMIN,D)
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +     THIS SUBROUTINE LOCATES THE MEASURED MINIMUM OF THE          +
C     +     STANDING WAVE IN THE CONDUIT                                 +
C     +     PMIN=MINIMUM MEASURED PRESSURE                               +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
      DIMENSION P(10),D(10)
      IF(P(1).LE.P(2)) GO TO 1
      PMIN=P(2)
      DMIN=D(2)
      GO TO 2
    1 PMIN=P(1)
      DMIN=D(1)
    2 IF(PMIN.LE.P(3)) GO TO 3
      PMIN=P(3)
      DMIN=D(3)
    3 IF(PMIN.LE.P(4)) GO TO 4
      PMIN=P(4)
      DMIN=0.0
    4 IF(PMIN.LE.P(5)) GO TO 5
      PMIN=P(5)
      DMIN=-D(4)
    5 IF(PMIN.LE.P(6)) GO TO 6
      PMIN=P(6)
      DMIN=-D(5)
    6 RETURN
      END



      SUBROUTINE XMAX(C,S,YM,X1,FR,SON)
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +     THIS SUBROUTINE CALCULATES THE VALUE OF X, THE POSITION      +
C     +     OF THE MAXIMUM PRESSURE                                      +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      DIMENSION C(10),S(10)
      REAL K1
      W11=YM*(C(5)+C(4)-2)
      W12= 2*C(3) -C(2)- C(1)
      W21= YM*(S(5)+S(4))
      W22= S(2)+S(1)-2*S(3)
      W=(W11+W12)/(W21+W22)
      K1=(2*3.14159*FR/SON)
      X1=(ATAN(W))/(2*K1)
      RETURN
      END
```

```
G1  RELEASE 2.0              PRED              DATE = 76335          12/55/53

      SUBROUTINE PRED(RE,PR,SN,XN,PNR,FR,D)
      DIMENSION XN(10),RE(10),PR(10),SN(10),PNR(10),D(10),X(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                      +
C     +     THIS SUBROUTINE CALCULATES THE BEST FIT PRESSURES AT THE        +
C     +     SIX TRANSDUCER LOCATIONS TO BE PRINTED ON THE OUTPUT OF         +
C     +     THE PROGRAM                                                     +
C     +                                                                      +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      Z3=0.0
      IF(SN(1).EQ.Z3) GO TO 4
      R=RE(1)*RE(1)+1
      D1=4*3.14159 *FR/SN(1)
      X(1)=COS(D1*(XN(1)-D(1)))
      X(2)=COS(D1*(XN(1)-D(2)))
      X(3)=COS(D1*(XN(1)-D(3)))
      X(5)=COS(D1*(XN(1)+D(4)))
      X(4)=COS(D1*(XN(1)))
      X(6)=COS(D1*(XN(1)+D(5)))
      DO 1 I=1,6
      PN=2*X(I)*RE(1)+R
      IF(PN.LT.Z3) GO TO 2
      GO TO 3
    2 PN=-PN
    3 CONTINUE
      PN1=SQRT(PN)
      PNR(I)=PR(1)*PN1
    1 CONTINUE
      GO TO 21
    4 CONTINUE
      DO 20 J=1,6
      PNR(J)=0.0
   20 CONTINUE
   21 CONTINUE
      RETURN
      END
```

```
G1  RELEASE 2.0              PCAL              DATE = 76335          12/55/53

      SUBROUTINE PCAL(REF,X,PNS,PC)
      DIMENSION X(10),PC(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +    THIS SUBROUTINE CALCULATES THE ESTIMATED VALUE OF THE           +
C     +    PRESSURE AT THE SIX TRANSDUCER LOCATIONS GIVEN THE ESTI-        +
C     +    MATED VALUES OF INCIDENT PRESSURE, POSITION OF THE              +
C     +    STANDING WAVE MAXIMA, SONIC VELOCITY, AND THE REFLECTION        +
C     +    FACTOR.                                                         +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
C
      Y1=REF*REF+1
      DO 10 I=1,6
      Y2=2*REF*X(I)
      Y3=Y1+Y2
      IF(Y3.LE.0) GO TO 2
      Y4=SQRT(Y3)
      GO TO 3
    2 Y5=-Y3
      Y4=SQRT(Y5)
    3 CONTINUE
      PC(I)=PNS*Y4
   10 CONTINUE
      RETURN
      END



      SUBROUTINE CMPRE(PC,Z2,VR1)
      DIMENSION PC(10),Z2(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +        THIS SUBROUTINE COMPARES THE VALUES OF THE PRESSURE         +
C     +        FROM THE BEST FIT OPTIMIZATION WITH THE MEASURED            +
C     +        VALUES AND DETERMINES THE SUM OF THE VARIANCES FOR THAT     +
C     +        PARTICULAR FIT.  THIS WILL THEN BE COMPARED TO THE 25       +
C     +        PER CENT ERROR VARIANCE IN SUBROUTIN REFP OR IN THE         +
C     +        MAIN PROGRAM, DEPENDING UPON WHERE IT WAS CALLED FROM       +
C     +                                                                    +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      Y2=0.0
      DO 20 K=1,6
      Y=Z2(K)-PC(K)
      Y1=Y**2
      Y2=Y1+Y2
   20 CONTINUE
      VR1=Y2
      RETURN
      END
```

```
      XN(4)=XN(3)
      XN(3)=XN(2)
      XN(2)=X1
      GO TO 6
    2 IF(VR1.GE.U(3)) GO TO 3
      U(5)=U(4)
      U(4)=U(3)
      U(3)=VR1
      PR(5)=PR(4)
      PR(4)=PR(3)
      PR(3)=PNS
      RE(5)=RE(4)
      RE(4)=RE(3)
      RE(3)=RF2
      SN(5)=SN(4)
      SN(4)=SN(3)
      SN(3)=SON
      XN(5)=XN(4)
      XN(4)=XN(3)
      XN(3)=X1
      GO TO 6
    3 IF(VR1.GE.U(4)) GO TO 4
      U(5)=U(4)
      U(4)=VR1
      PR(5)=PR(4)
      PR(4)=PNS
      RE(5)=RE(4)
      RE(4)=RF2
      SN(5)=SN(4)
      SN(4)=SON
      XN(5)=XN(4)
      XN(4)=X1
      GO TO 6
    4 IF(VR1.GE.U(5)) GO TO 5
      U(5)=VR1
      PR(5)=PNS
      RE(5)=RF2
      SN(5)=SON
      XN(5)=X1
    5 CONTINUE
    6 CONTINUE
      RETURN
      END
```

```
G1  RELEASE 2.0              REGV              DATE = 76335          12/55/53

      SUBROUTINE REGV(VR1,U,PNS,PR,RF2,RE,SN,SON,X1,XN)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                      +
C     +    THIS IS THE SUBROUTINE WHERE THE BEST FIT VALUES OF INCI-         +
C     +    DENT PRESSURE(PIN), REFLECTION FACTOR(SON), ERROR SQUARED         +
C     +    (VR1), AND POSITION(X1) ARE STORED. ONLY THE BEST FIVE            +
C     +    VALUES ARE STORED.                                                +
C     +                                                                      +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      DIMENSION U(10),PR(10),RE(10),SN(10)
      DIMENSION XN(10)
      IF(VR1.GE.U(1)) GO TO 1
      U(5)=U(4)
      U(4)=U(3)
      U(3)=U(2)
      U(2)=U(1)
      U(1)=VR1
      PR(5)=PR(4)
      PR(4)=PR(3)
      PR(3)=PR(2)
      PR(2)=PR(1)
      PR(1)=PNS
      RE(5)=RE(4)
      RE(4)=RE(3)
      RE(3)=RE(2)
      RE(2)=RE(1)
      RE(1)=RF2
      SN(5)=SN(4)
      SN(4)=SN(3)
      SN(3)=SN(2)
      SN(2)=SN(1)
      SN(1)=SON
      XN(5)=XN(4)
      XN(4)=XN(3)
      XN(3)=XN(2)
      XN(2)=XN(1)
      XN(1)=X1
      GO TO 6
    1 IF(VR1.GE.U(2)) GO TO 2
      U(5)=U(4)
      U(4)=U(3)
      U(3)=U(2)
      U(2)=VR1
      PR(5)=PR(4)
      PR(4)=PR(3)
      PR(3)=PR(2)
      PR(2)=PNS
      RE(5)=RE(4)
      RE(4)=RE(3)
      RE(3)=RE(2)
      RE(2)=RF2
      SN(5)=SN(4)
      SN(4)=SN(3)
      SN(3)=SN(2)
      SN(2)=SON
      XN(5)=XN(4)
```

```
G1  RELEASE 2.0                COR                  DATE = 76335          12/55/53

      SUBROUTINE COR(X,FR,SCN,D,X1)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                      +
C     +     THIS SUBROUTINE CALCULATES THE VALUES OF THE COSINES OF          +
C     +     TRANSDUCER LOCATIONS RELATIVE TO THE POSITION OF THE             +
C     +     STANDING WAVE MAXIMA X1,ONCE X1 HAS BEEN ESTIMATED.              +
C     +                                                                      +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      DIMENSION X(10),D(10)
      D1=4*3.14159*FR/SCN
      X(1)= COS(D1*(X1-D(1)))
      X(2)= COS(D1*(X1-D(2)))
      X(3)= COS(D1*(X1-D(3)))
      X(4)= COS(D1*X1)
      X(5)= COS(D1*(X1+D(4)))
      X(6)= COS(D1*(X1+D(5)))
      RETURN
      END
```

```
51  RELEASE. 2.0                REFP                DATE = 76335          12/55/53

      SUBROUTINE REFP(PIN,RE,PNS,REF,VARD,X1,D,FR,SON,Z2,PMAX,PMIN,PR,
     $U,SN,XN)
      DIMENSION X(10),D(10),RE(10),PS(10),Z2(10),PC(10)
      DIMENSION XN(10)
      DIMENSION PR(10),U(10),SN(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                  +
C     +       WITHIN THIS SUBROUTINE, THE INCIDENT PRESSURE AND          +
C     +       THE REFLECTION FACOR ARE VARIED , ONE AT A TIME, IN        +
C     +       ORDER TO ARRIVE AT THE VALUES OF THESE RESPECTED           +
C     +       PARAMETERS THAT BEST FIT THE DATA.                         +
C     +                                                                  +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      CALL COR(X,FR,SON,D,X1)
      NMAX=10
      DO 10 J=1,NMAX
      CALL PCAL(REF,X,PNS,PC)
      CALL CMPRE(PC,Z2,VR1)
      RF2=REF
      IF(VR1.LE.VARD) GO TO 30
      GO TO 6
   30 CALL REGV(VR1,U,PNS,PR,RF2,RE,SN,SON,X1,XN)
    6 NMX=10
      Z5=0.70
      RIN=(1-REF)/NMX
      DO 50 K=1,NMX
      IF(RF2.LT.Z5) GO TO 40
      RF2=(1+RF2)/2
      GO TO 3
   40 RF2=REF+K*RIN
    3 CALL PCAL(RF2,X,PNS,PC)
      CALL CMPRE(PC,Z2,VR2)
      IF(VR2.LE.VR1) GO TO 4
      GO TO 55
    4 VR1=VR2
      IF(VR1.LE.VARD) GO TO 31
      GO TO 7
   31 CALL REGV(VR1,U,PNS,PR,RF2,RE,SN,SON,X1,XN)
    7 CONTINUE
   50 CONTINUE
   55 CONTINUE
      V1=((PMAX+PMIN)/2-PNS)*0.40
      PNS=PNS+V1
   10 CONTINUE
      RETURN
      END
```

```
      SUBROUTINE MAXP(P,PMAX,DMAX,D)
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +     THIS SUBROUTINE LOCATES THE MEASURED MAXIMUM OF THE            +
C     +     STANDING IN THE CONDUIT                                        +
C     +     PMAX=MAXIMUM MEASURED PRESSURE                                 +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
      DIMENSION P(10),D(10)
      IF(P(1).GE.P(2)) GO TO 1
      PMAX=P(2)
      DMAX=D(2)
      GO TO 2
    1 PMAX=P(1)
      DMAX=D(1)
    2 IF(PMAX.GE.P(3)) GO TO 3
      PMAX=P(3)
      DMAX=D(3)
    3 IF(PMAX.GE.P(4)) GO TO 4
      PMAX=P(4)
      DMAX=0.0
    4 IF(PMAX.GE.P(5)) GO TO 5
      PMAX=P(5)
      DMAX=-D(4)
    5 IF(PMAX.GE.P(6)) GO TO 6
      PMAX=P(6)
      DMAX=-D(5)
    6 RETURN
      END
```

```
      SUBROUTINE SUMD(Z2,VARD)
      DIMENSION Z2(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +        THIS SUBROUTINE CALCULATES THE SUM OF THE VARIANCES IF      +
C     +        THERE WERE A 25 PER CENT ERROR IN THE FIT OF A CURVE TO     +
C     +        THE DATA.  THIS GIVES THE MAXIMUM ALLOWABLE ERROR IN        +
C     +        ATTEMPTING TO FIT A CURVE TO THE DATA                       +
C     +        VARD=SUM OF THE VARIANCES FOR A 25 PER CENT ERROR           +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      W2=0.0
      DO 1 I=1,6
      W=Z2(I)*0.1
      W1=W**2
      W2=W1+W2
    1 CONTINUE
      VARD=W2*6.25
      RETURN
      END
```

```
51  RELEASE. 2.0                FUNC                DATE = 76335          12/55/53

      SUBROUTINE FUNC(J1,FR,SON,D,C,S)
      DIMENSION C(10),D(10),S(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                      +
C     +       THIS SUBROUTINE CALCULATES THE VALUES OF THE SINES AND         +
C     +       COSINES FOR THE DISTANCE A PARTICULAR TRANSDUCER IS            +
C     +       FROM TRANSDUCER 4.                                             +
C     +       D(1),...D(5)= DISTANCES T1,T2,T3,T5,AND T6 ARE FROM T4         +
C     +       FR=FREQUENCY                                                   +
C     +       SON=SONIC VELOCITY                                             +
C     +                                                                      +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      D1=2*(2*FR/SON)*D(J1)*3.14159
      C(J1)=COS(D1)
      S(J1)=SIN(D1)
      RETURN
      END

      SUBROUTINE PSI(P,J,PS,Z2)
      DIMENSION P(10),PS(10),Z2(10)
C
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                      +
C     +       THIS SUBROUTINE CONVERTS THE PRESSURE READING FROM THE         +
C     +       TRANSDUCERS, READ IN DECIBELS ON THE CHART, TO PSI.            +
C     +       THE FOLLOWING PARAMETERS ARE USED---                           +
C     +       P(J)=THE PRESSURE IN DECIBELS                                  +
C     +       PS(J)=THE PRESSURE IN PSI                                      +
C     +                                                                      +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
C
      Z=(P(J)-42.7)/20
      Z1=10**Z
      Z2(J)=2.9*Z1
      PS(J)=Z2(J)*Z2(J)
      RETURN
      END
```

```
$1  RELEASE.2.0              MAIN              DATE = 76335          12/55/53

      DIMENSION D(10),RE(10),Z2(10),PS(10),P(10),C(10),S(10)
      DIMENSION U(10),SN(10),PR(10)
      DIMENSION X(10),XN(10),PNR(10)
C
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C     +                                                                    +
C     +     THIS PROGRAM ESTIMATES THE REFLECTION FACTOR, INCIDENT         +
C     +     PRESSURE, POSITION OF THE STANDING WAVE MAXIMA, AND THE        +
C     +     SONIC VELOCITY FOR THE STANDING WAVE. IT IS DESIGNED TO BE     +
C     +     USED WITH DATA FROM SIX STATIONARY PRESSURE TRANSDUCERS.       +
C     +     THE NUMBER OF SETS OF DATA MUST BE READ IN, PLUS THE DIS-      +
C     +     TANCES BETWEEN TRANSDUCERS 1 AND 4, 2 AND 4, 3 AND 4, 5        +
C     +     AND 4, AND 6 AND 4. THE PRESSURE DATA FROM THE TRANS-          +
C     +     DUCERS IS THEN READ IN AS WELL AS THE FREQUENCY FOR EACH       +
C     +     DATA SET.                                                      +
C     +                                                                    +
C     +     THE MAJOR PARAMETERS USED ARE:                                 +
C     +        D(1)=DISTANCE FROM TRANSDUCER 1 TO 4                        +
C     +        D(2)=DISTANCE FROM TRANSDUCER 2 TO 4                        +
C     +        D(3)=DISTANCE FROM TRANSDUCER 3 TO 4                        +
C     +        D(4)=DISTANCE FROM TRANSDUCER 4 TO 5                        +
C     +        D(5)=DISTANCE FROM TRANSDUCER 4 TO 6                        +
C     +        X1=RELATIVE POSITION OF TRANSDUCER 4 FROM THE               +
C     +           STANDING WAVE MAXIMA                                     +
C     +        REF=REFLECTION FACTOR                                       +
C     +        SON=SONIC VELOCITY(IN M/SEC)                                +
C     +        PMAX=MEASURED STANDING WAVE MAXIMA                          +
C     +        PMIN=MEASURED STANDING WAVE MINIMA                          +
C     +        PNS=INCIDENT PRESSURE                                       +
C     +        U=ERROR SQUARED(VARIANCE) FOR ESTIMATED CURVE FIT           +
C     +                                                                    +
C     ++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++++
C
      READ(5,1) N
    1 FORMAT(I2)
      WRITE(6,33)
   33 FORMAT(1H1)
      READ(5,6) (D(I1),I1=1,5)
    6 FORMAT(5(F8.2,2X))
      DO 3 N1=1,N
      READ(5,2) (P(I),I=1,6),FR
    2 FORMAT(7(F8.2,2X))
      DO 60 M3=1,5
      XN(M3)=0.0
      RE(M3)=0.0
      PR(M3)=0.0
      U(M3)=1000000
      SN(M3)=0.0
   60 CONTINUE
      DO 4 J=1,6
      CALL PSI(P,J,PS,Z2)
    4 CONTINUE
      CALL MAXP(Z2,PMAX,CMAX,D)
      CALL MINP(Z2,PMIN,DMIN,D)
      SWR=PMAX/PMIN
      CALL SUMD(Z2,VARD)
      DO 10 M1=1,15
      SON=645+M1*45
```

```
1  RELEASE 2.0              MAIN              DATE = 76335          12/55/53

           DO 70 N5=1,25
           REF=(SWR-1)/(SWR+1)
           PNS=(PMAX+PMIN)/4
            HAL=SON/(50*FR)
           X1=DMIN+(N5-1)*HAL
           CALL REFP(PIN,RE,PNS,REF,VARD,X1,D,FR,SON,Z2,PMAX,PMIN,PR,U,SN,XN)
       70  CONTINUE
       10  CONTINUE
           WRITE(6,34)
       34  FORMAT(5X,////)
           WRITE(6,37) FR
       37  FORMAT(1HO,'FREQUENCY IS',2X,F6.0,2X,'HZ',/)
           WRITE(6,36)      VARD
        36 FORMAT(1HO,'MAXIMUM ALLOWABLE ERROR IS ',1X,F8.2)
           CALL PRED(RE,PR,SN,XN,PNR,FR,D)
           WRITE(6,22) (Z2(I6),I6=1,6)
       22  FORMAT(5X,'MEASURED PRESSURES',5X,6(F8.2,3X))
           WRITE(6,21) (PNR(I5),I5=1,6)
        21 FORMAT(5X,'BEST FIT PRESSURES',5X,6(F8.2,3X))
           WRITE(6,80)
        80 FORMAT(1X,'REFLECTION FACTOR',3X,'INCIDENT PRESSURE',2X,'ERROR SQU
          $ARED',2X,'SONIC VELOCITY',2X,'POSITION OF X',/)
           DO 65 M2=1,5
           WRITE(6,20) RE(M2),PR(M2),U(M2),SN(M2),XN(M2)
       20  FORMAT(9X,F6.4,12X,F8.2,8X,F8.2,6X,F8.2,9X,F6.2,/)
        65 CONTINUE
       3   CONTINUE
           STOP
           END
```

FREQUENCY IS 200. HZ

MAXIMUM ALLOWABLE ERROR IS 11.66

MEASURED PRESSURES	5.53	6.06	5.53	6.06	5.16	5.04
BEST FIT PRESSURES	5.26	5.83	5.96	5.73	5.08	5.45

REFLECTION FACTOR	INCIDENT PRESSURE	ERROR SQUARED	SONIC VELOCITY	POSITION OF X
0.0918	5.52	0.58	1320.00	0.63
0.0918	5.50	0.58	1320.00	0.63
0.0918	5.47	0.59	1320.00	0.63
0.0918	5.52	0.59	1320.00	0.49
0.0918	5.50	0.60	1320.00	0.49

FREQUENCY IS 150. HZ

MAXIMUM ALLOWABLE ERROR IS 222.94

MEASURED PRESSURES	25.85	29.34	24.40	25.26	20.06	20.06
BEST FIT PRESSURES	26.27	28.14	26.56	24.77	20.30	20.83

REFLECTION FACTOR	INCIDENT PRESSURE	ERROR SQUARED	SONIC VELOCITY	POSITION OF X
0.1877	23.74	7.14	1320.00	1.00
0.1877	23.10	7.45	1320.00	1.00
0.1877	23.74	8.12	1275.00	0.94
0.1877	23.10	8.49	1275.00	0.94
0.1877	24.12	9.50	1320.00	1.00