View metadata, citation and similar papers at core.ac.uk

brought to you by CORE

provided by Purdue E-Pubs

Purdue University
Purdue e-Pubs

International Compressor Engineering Conference

School of Mechanical Engineering

1976

On the Dynamic Analysis and Evaluation of Compressor Mufflers

R. Singh

T. Katra

Follow this and additional works at: https://docs.lib.purdue.edu/icec

Singh, R. and Katra, T., "On the Dynamic Analysis and Evaluation of Compressor Mufflers" (1976). *International Compressor Engineering Conference.* Paper 217.
https://docs.lib.purdue.edu/icec/217

This document has been made available through Purdue e-Pubs, a service of the Purdue University Libraries. Please contact epubs@purdue.edu for additional information.
Complete proceedings may be acquired in print and on CD-ROM directly from the Ray W. Herrick Laboratories at https://engineering.purdue.edu/Herrick/Events/orderlit.html

ON THE DYNAMIC ANALYSIS AND EVALUATION OF COMPRESSOR MUFFLERS

Rajendra Singh

Thomas Katra

Acoustics/Dynamics Engineering
Carlyle Compressor Company
Carrier Corporation
Syracuse, N. Y. 13201

INTRODUCTION

Suction and discharge gas pulsations in positive displacement compressors can be major contributors to noise radiation and structural vibrations. Also, the fluid-borne noise can be transmitted to other components of the system (e.g. evaporator, condenser, and interconnecting pipelines in an airconditioning system). Therefore, mufflers are often needed to control the gas oscillations by attenuating sound energy over the frequency range of interest.

This paper is not an exhaustive and comprehensive study of the design and evaluation of muffler components. For this, a number of excellent references are available [1 - 6]. Rather it attempts to point out some of the limitations of existing linear one dimensional acoustic filter theories. The dynamic effects associated with fluid flows are briefly discussed, in order to point out the necessity for incorporating the study of these effects in an experimental program.

To evaluate the acoustical characteristics of muffler elements, a measurement technique using impulse excitation is proposed and explained. The discussion of the necessity, concept, procedures, instrumentation and results of the impulse technique form the core of the paper.

EVALUATION OF MUFFLER CHARACTERISTICS

Nonlinear Effects

Generally linear acoustical formulations are used to analyze mufflers [2 - 5]. These formulations involve a number of simplifications. They are applicable for small wave amplitudes and do not include detailed fluid flow effects and interactions. But in real compressor environments, apart from steady flows, one can witness turbulence and finite wave amplitudes. These are nonlinear phenomena. For some muffler components, the linear theory may still be adequate as nonlinear effects may not be dominant. Even for components exhibiting strong nonlinearities, linear acoustic theory can be used as the basis and can be modified to include nonlinear phenomena [1, 2, 5 - 15]. Before analyzing a muffler element, it is important to understand the effects of flow-sound interactions and finite wave amplitudes.

A number of researchers, over the last few years, have investigated the effects of fluid flows on sound propagation in acoustic resonators [7, 8, 10-15]. They have concentrated on simple configurations, and from their studies, one can draw a coherent set of conclusions. However, much more work needs to be done before comprehensive design guidelines and complete and general mathematical models are available. Based on the existing literature [1 - 16], the two main nonlinear effects can be explained as follows.

(1) Flow-sound interactions: These can be categorized into three broad effects as: (a) convective effect (b) fluid-induced damping and (c) fluid-induced sound amplification.

Convective effect results in wave propagation with an increased sonic velocity in the direction of flow and a decreased velocity opposite to the fluid flows. Some investigators [2, 12] have demonstrated that the pressure reflection coefficient can exceed unity with flow. However, the sound reflection defined on the basis of energy can never exceed unity, even in the case of superimposed flow. With flow, it generally decreases. Because of the convective effect, the resonant frequencies are lowered.

Steady flows increase the system damping. These losses can occur as a result of the interactions between sound and vorticity modes. Sound fields can trigger turbulence which represents a loss of energy. These frictional effects are important at area changes and in acoustic inertial elements.

Reactive mufflers, in some cases, may amplify the sound. It has been proposed by several investigators [2] that eddies caused by the flow are synchronized by the sound and are converted into sound energy. Thus there can be a continuous energy transfer from the fluid mode to the acoustic mode which can result in amplification of the sound wave.

(2) Finite-wave amplitudes: Certain components like orifices show a nonlinear behavior [14, 15]. At higher sound pressure levels, the acoustic pressure is not proportional to the particle velocity, rather it is quadratic function of velocity.

At small amplitudes, viscous resistance is the only resistance and is much smaller than the reactance. But at finite amplitudes, nonlinear resistance may dominate the total impedance.

Perhaps the most important implication of these effects is their influence on a reliable prediction of realistic damping. A literature survey [1 - 16] indicates that at present damping formulations are available only for a few specific cases. Sound dissipation results from the following effects: viscous and heat conduction boundary layer dissipation, radiation resistance, finite amplitudes, fluid-induced damping and porous material dissipation in the case of soft walls. It can be safely stated that acoustic damping is a complicated phenomena, and in refrigeration compressors, it is a very significant factor. Refrigerants mixed with lubricating oils are much more dense and viscous than air. Also, one finds narrow connecting passages exhibiting strong friction effects. Figure 1 illustrates a typical example of this. Figure 1 demonstrates a comparison between computer simulated and experimentally measured results for a typical small compressor. The spectra are at the exit of the discharge valve. A mathematical simulation with only a linear viscous model predicts all the harmonics closely except the one associated with a resonance. With an empirically determined damping model, the third harmonic prediction is satisfactory. But, the empirical model was valid for that particular system only and only at a particular frequency. Therefore it is important to have a continuous frequency spectra of acoustic characteristics of components. If one has a complete picture of natural frequencies and associated dampings, then the design processes can be simplified.

Performance Indices

Various indices are used for performance evaluation including: noise reduction, attenuation, insertion loss, and transmission loss [3]. It should be noted that out of these indices, transmission loss* is the only one which is a characteristic of a muffler element alone. The others include source and load impedance effects and thus are overall system properties. For measurement purposes, it has always been difficult to isolate a muffler acoustically, and that is why transmission loss index is measured only under some restricted conditions. Since the total sound pressure levels are easy to measure, sometimes insertion loss and noise reduction indices are taken as muffler performances. These are valid for the system under study. With a change in the system, insertion loss and/or noise reduction are no longer indications of the muffler characteristics. Therefore it is imperative that only transmission loss be considered an inherent muffler performance index. (Acoustic impedances are also inherent muffler characteristics, but are not convenient for performance evaluation.)

Classical Measurement Techniques

Gatley [17, 18] concluded that only a few investigators directed their efforts at evaluating small mufflers, as encountered in refrigeration systems. He compared the standing wave tube, three-pressure method and pulse method for determining reflection and transmission factors, and recommended the standing wave tube method. According to Gatley [17], the pulse or transient method is "relatively undesirable" for the direct measurement of reflection and transmission factors. The two-pressure [19] and three-pressure [17] methods are useful only when the acoustic characteristics at one point are known.

*Transmission loss is defined as the ratio of muffler incident acoustic intensity to transmitted acoustic intensity, in decibels.

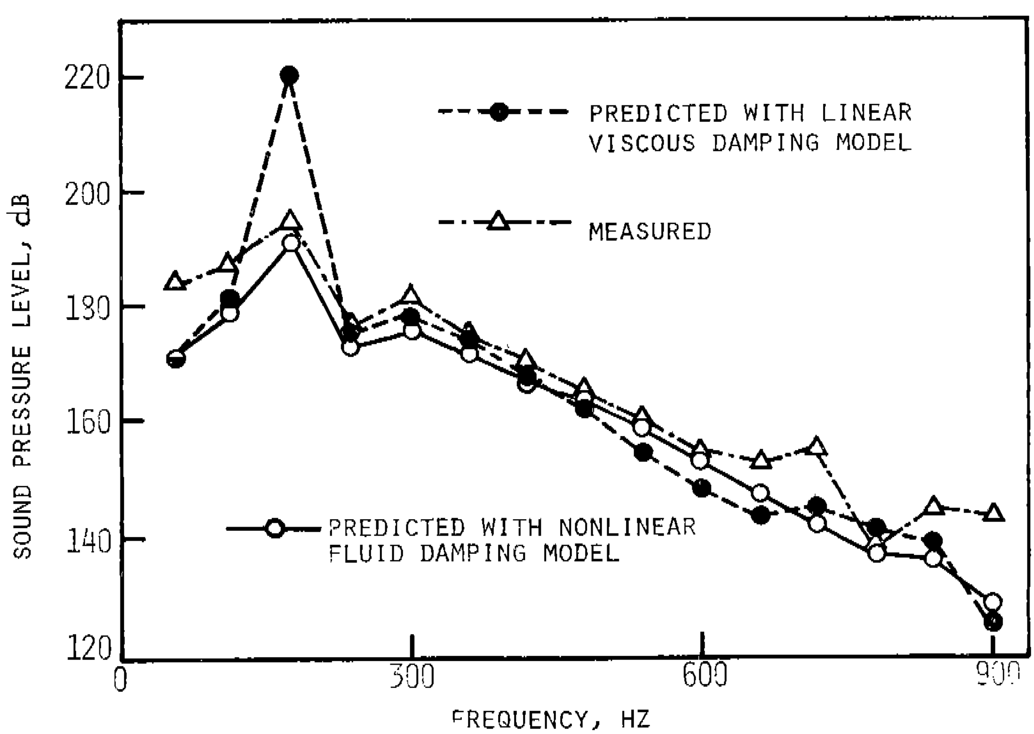


Figure 1

Compressor Discharge Pressure Spectra Prediction With Linear and Nonlinear Models

Perhaps the most attractive and easy to use method is the standing wave tube. It has been used by Gatley [17], Alfredson [7], Melling [19], Schaffart [14] and Sullivan [16] etc. The method mainly determines input impedance, reflection and absorption coefficients. With the use of an anechoic termination after the muffler, the standing wave tube method can be extended to transmission loss measurements. The standing wave tube is limited towards the lower frequencies because of the microphone traverse. The process is tedious and time consuming as only one frequency measurement can be conducted at a time. Furthermore, it requires continuous pressure measurement along the standing wave tube, which is especially cumbersome and inaccurate at low frequencies. It is a rather difficult process for measurements with moving fluids especially in small tubes. Alfredson [7], Schaffart [14] and Sullivan [16] allowed for superimposed fluid flows and measured acoustical characteristics.

Singh [20] demonstrated an efficient and direct measurement procedure for determining acoustical impedances. It utilized a known volume velocity input, and harmonic, random and transient excitations were attempted. The method was tried for a composite acoustic system under steady state conditions. However, it is relatively difficult to extend this technique to measurements with flows.

AN IMPULSE TECHNIQUE: PHYSICAL CONSIDERATIONS

Concept

Because of the inherent limitations of existing classical methods, an "impulse technique" has been developed. It differs fundamentally from the "pulse" or "transient" method, as commonly referred to in the literature [12, 17, 20], which is really a quasi-steady state method. This pulse method utilizes a gated burst of a sinusoidal signal, long enough to approximate steady state response but short enough to avoid reflection problems. Thus, it is inherently a single frequency measurement. Conversely, the present method utilizes a rough approximation of an impulse or delta function. An ideal impulse function contains equal energy at all frequencies, has zero length in time, and an infinite amplitude [21, 22]. Therefore, it is possible to determine the response of a system at all frequencies with a single impulse excitation. This concept has long been used for mathematical analyses [21], although, of course, an ideal impulse is not physically realizable. This is not necessarily a serious limitation, however, as impulse-like physically realizable functions can be generated. Such functions are adequate over typical frequency ranges of interest, and pose little or no theoretical compromises in measurement accuracy or completeness if proper techniques are used.

In structural dynamics, an impulse technique is commonly used [23]. System characteristics are determined by striking the structure with a force gauge equipped hammer and measuring vibration response to the measured force input. Its direct analog is impractical in acoustic systems, as only one physical quantity, pressure, can be reliably measured over a broad frequency range. However, compressor applications often involve small diameter muffling systems, thus insuring a plane wave propagation over the frequency range of interest. Therefore, an acoustic pressure impulse can be used as an excitation, with the resulting transmitted and/or reflected signals being treated as the system response. Since these are traveling waves in an essentially non-dispersive system, they can be sorted out in time and analyzed separately. All the information necessary to determine muffler characteristics is contained in these two responses to the known input excitation. Hence, the technique can be used efficiently to determine muffler transmission loss over the plane wave frequency range under a variety of conditions, especially with flow. This was the immediate objective of the current work. Also, all necessary information is readily available to compute such complex acoustic characteristics as impedances.

Set-up Design

Fig. 2 conceptually illustrates this impulse technique. An excitation is provided by the sound source, and the acoustic signal propagates toward the muffler, in the presence of steady flow if desired. The incident wave excites the muffler and the resulting reflected and transmitted waves contain information regarding the muffler characteristics. Several general considerations are important in the set-up design and data parameter selection. The excitation should resemble an impulse function in order to obtain adequately uniform excitation over the desired frequency range. But the pulse should be wide enough to have adequate power spectral density, and most importantly must be a physically realizable signal. For transmission loss, one has to capture isolated incident and transmitted waves. Furthermore, for muffler impedance measurements, the isolated reflected wave is also needed.

The tubing lengths and the location of transducers, as shown in Fig. 2, are therefore very critical. Transducer #1 should capture the incident wave before the reflection from the muffler arrives. Similarly, the transmitted wave should be picked up by transducer #2 before the reflection of transmitted wave from the open end starts coming back. With reference to the definitions of Fig. 2, these criteria dictate the following design guidelines.

$$(x_m - x_1) > \delta_{in} \, [(c^2 - V^2)/2c] \qquad (1)$$

$$(x_e - x_2) > \delta_{tr} \, [(c^2 - V^2)/2c] \qquad (2)$$

In addition, the source and transducer #1 must be arranged so that any extraneous reflections of the input excitation from the flow supply end arrive after data acquisition is complete.

Data Acquisition Considerations

The following data acquisition parameters require proper and careful selection: (1) time window (or record length) for data acquisition (2) time resolution (or its reciprocal, sampling frequency) (3) maximum frequency of interest (4) frequency resolution and (5) number of sampling points. However, the mathematical inter-relationships between these parameters dictate that only two are independent· i.e., selection of any two determines the other three [22]. Thus depending upon the evaluation

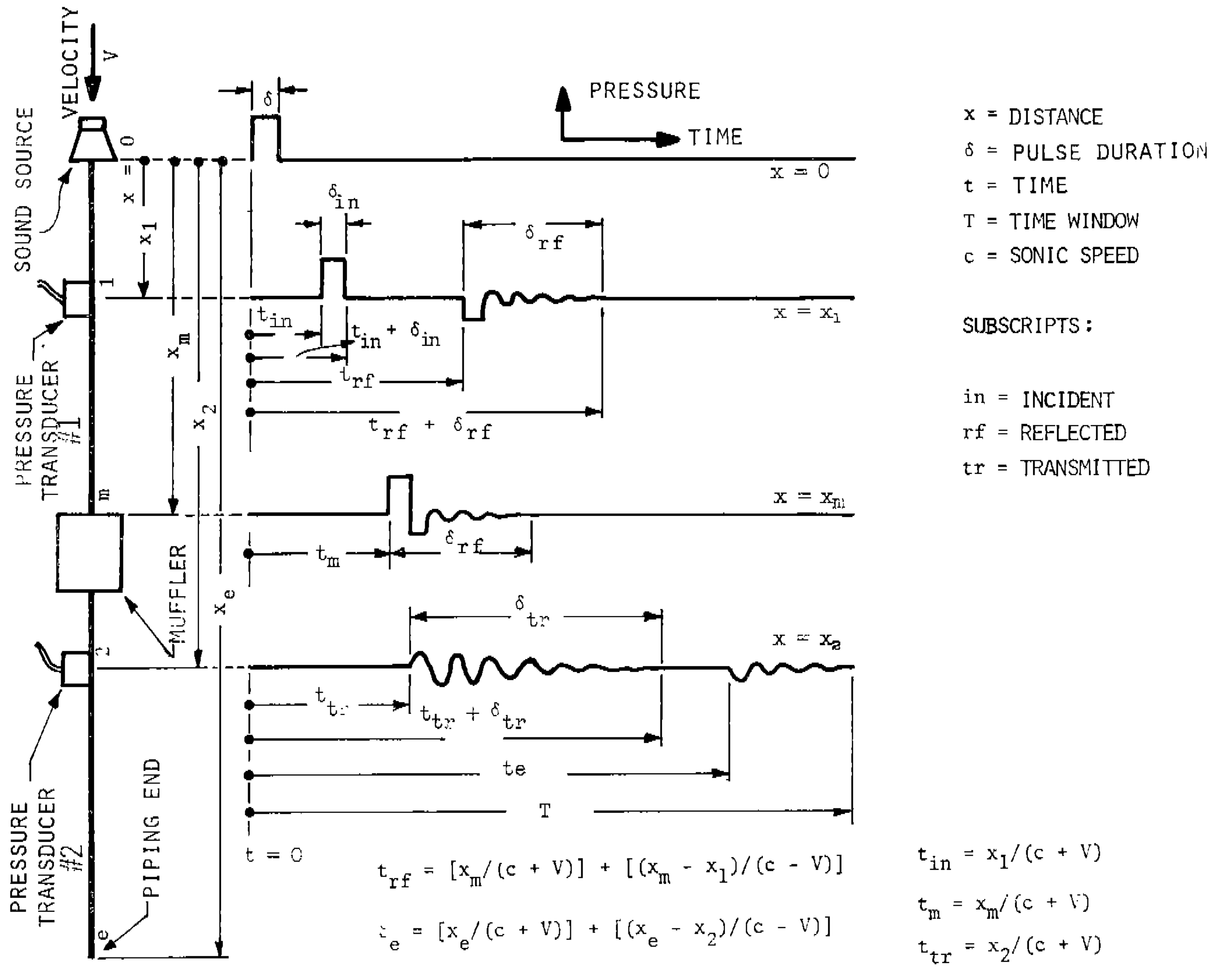


Figure 2. Illustration of Impulse Technique

criteria and available instrumentation, an optimum compromise set must be selected. The time window selection is further complicated from the tubing length viewpoint. It is very critical as the time window must be long enough to capture the entire pressure wave signal during its first appearance at the transducer location, and should be short enough so that no undesired reflections are captured. Otherwise these will have to be eliminated as part of the data processing. Again, with reference to Fig. 2, these criteria can be stated as,

$$T \geqslant [x_m/(c + V)] + [(x_m - x_1)/(c - V)] + \delta_{rf} \quad (3)$$

$$T \geqslant [x_2/(c + V)] + \delta_{tr} \quad (4)$$

IMPULSE TECHNIQUE: MEASUREMENT CONSIDERATIONS

Measurement Stand and Instrumentation

The measurement stand used by the authors to apply the impulse technique to evaluation of mufflers in air and refrigerants is depicted schematically in Fig. 3. In the special case of R-22, note that at normal room temperature and atmospheric pressure, the sonic speed (approximately 600 ft/sec or 180 m/sec), is very near to R-22 sonic speed at typical air conditioning compressor discharge conditions. Therefore, the flow conditioning equipment has been selected such that it is capable of bringing the R-22 vapor temperature to within 3°F of the room temperature, at a flow velocity of 60 ft/sec (18 m/sec) for the pipe sizes used, and the system is operated within 7% of atmospheric pressure. Since the flow velocities in compressor suction and discharge lines rarely exceed Mach o.1, the set-up is designed to allow a maximum velocity of approximately 130 ft/sec (40 m/sec) for measurements with air as the working fluid. If the test is to be conducted without flow, then flow is maintained only long enough to purge the system.

A horn driver is used to produce an acoustic impulse in the pipe. High resolution, low noise dynamic pressure transducers are installed to measure the traveling sound waves. The following two types of instrumentation systems have been used to provide excitation and analyze the transducer signals.

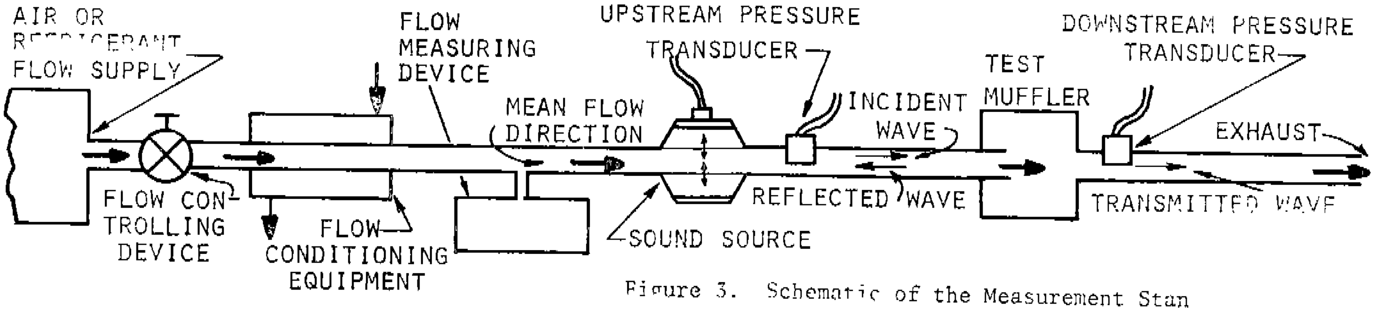


Figure 3. Schematic of the Measurement Stan

1. Analog system: A block diagram of the analog instrumentation system is shown in Fig. 4. It is only capable of single channel processing. Therefore a switching device is utilized to select one transducer at a time. To assure the validity of obtaining input and output data from separate runs, the test conditions are maintained such that repeatable pressure signals are obtained. A signal averager and real time analyzer are used for data acquisition and processing.

2. Digital system: A two-channel processing facility is depicted in Fig. 5. Both auxiliary hardware and software processing operations are shown, in a block diagram format. Note that both analog and digital system procedures are discussed jointly in the next section.

Measurement Steps

Means for exciting the system acoustically, acquisition of data, and data processing to obtain muffler characteristics are the essential measurement steps. Details are as follows.

a. Pulse generation: A wide variety of impulse like transients can be used, depending primarily on the frequency range of interest. However, one requires a relatively high power spectral density for a good signal to noise ratio, and relatively short time span to avoid excessive pipe lengths, necessitating a compromise. For the analog system, an audio ascillator and a tone-burst generator have been used to create the required pulse. One cycle of 1000 Hz sine wave was found adequate for measurements up to 2000 Hz. Although the signal is somewhat distorted by the imperfect response of the driver, the resulting transients are repeatable and have the required spectral content. For the digital system, the shape of the pulse is generated mathematically. Conceptually it is an impulse function, but has been modified to make it physically realizable. A digital to analog converter is used to change this mathematical function to a voltage suitable for driving the power amplifier and hence the horn driver.

b. Data acquisition: To allow time domain averaging and preserve phase information (see following sections), it is important to have time synchronization between the acquisition of both pressure transducer signals. In both systems this was done by locking acquisition to the pulse generation. In the analog system, the tone-burst generator triggered the signal averager. For the digital system, a common timing generator controls and synchronizes the operations of the digital to analog (D/A) and analog to digital (A/D) converters. It establishes reference time (initial point of the time window) at the moment of source excitation. This also has the advantage of "pre-triggering" the data acquisition, so that leading edge information is not lost. Since other data acquisition considerations are widely reported in the literature [22], they shall not be dealt with here.

c. Time domain averaging: The sound pressure signal is contaminated with electrical noise, and also with flow noise in the case of superimposed flow. While the desired signal is deterministic, the noise signals are random in nature. From random theory [21, 22], one knows that if the time of occurrence of a deter-

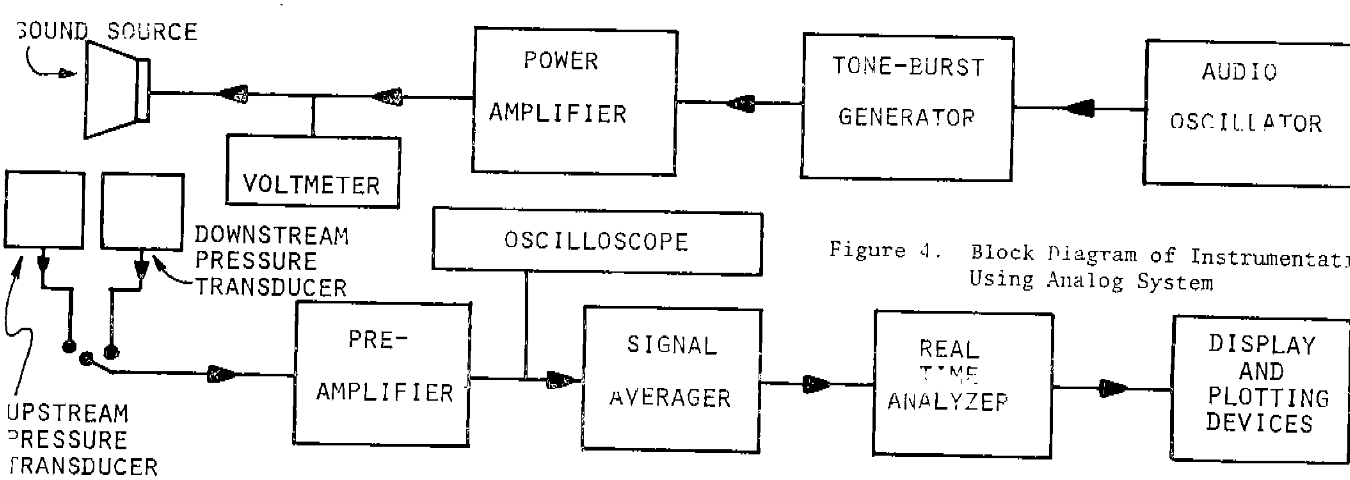


Figure 4. Block Diagram of Instrumentation Using Analog System

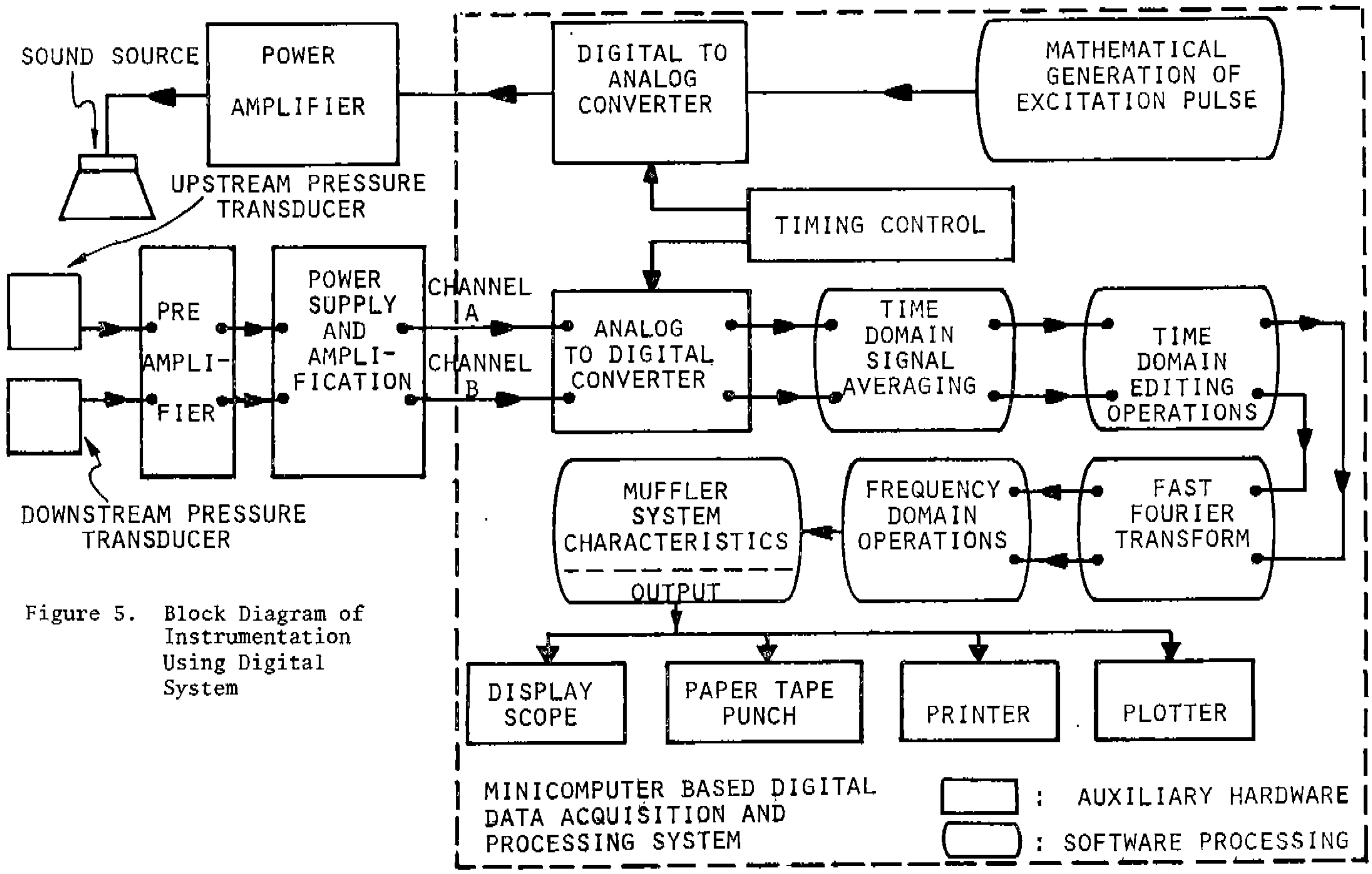


Figure 5. Block Diagram of Instrumentation Using Digital System

ministic signal is known (true for the present case), signals can be unearthed from random noise by using time domain everaging. Signal components, being in phase with each other, add linearly with each successive accumulation. The noise components, being ramdom, will tend to cancel each other. The signal to noise ratio of accumulated data thus improves with the number of averages. Thus the spurious contents can be eliminated from the pressure signals. In the analog system, averaging is done by a signal averager; in the digital system, it is done by direct computation. Note that adequate settling time must be allowed between successive averaging cycles to let spurious wave reflections decay to ambient levels.

d. Time domain operations: If the time window is such that it contains unwanted wave reflections, they must be edited out prior to any frequency domain operation. In the digital system, incident, reflected and transmitted waves are separated simply by mathematically setting the unwanted data values equal to zero. For the analog system, the time window must be judiciously chosen in such a way that the upstream transducer captures only the indicent wave and the downstream transducer picks up only the transmitted wave. Thus, only isolated waves are acquired and no editing operations are required.

e. Frequency domain operations: For analog system instrumentation, the real time analyzer in transient mode is really measuring the magnitude of the Fourier transform of its input signal. Thus, both incident and transmitted auto-power spectra are measured separately and plotted. The transmission loss can be calculated from these, as per the definition given earlier. On the other hand in the digital system, a Fast Fourier Transform (FFT) capability allows true Fourier analysis. Auto and cross power spectra of incident and transmitted waves are computed, and both the magnitude and phase of the transmission loss can be derived from them. (For impedance computations, the reflected pressure signal must also be processed.)

RESULTS AND DISCUSSION

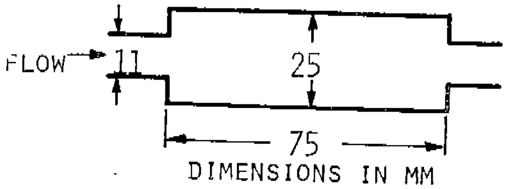
An Illustrative Case

In order to demonstrate the steps involved in the impulse technique, a data set has been taken for a simple expansion chamber muffler (shown in Fig. 6) with some intermediate steps included. The digital system was used (Fig. 5), with the following test parameters:

Time window = 100 msec
Time resolution = .097 msec
Maximum analysis frequency = 5120 Hz
Sampling frequency = 10,240 Hz
Frequency resolution = 10 Hz
No. of sampling points = 1024
Number of averages = 100
Medium: Air

The results are presented in Fig. 7. The isolated indicent wave is shown in Fig. 7(a), and the resulting transmitted wave in 7(b); their corresponding autospectra are given in 7(d) and 7(e). It is interesting to note that while the incident wave differs

Figure 6 Muffler Configuration



substantially from an ideal impulse primarily due to imperfections of the driver, its spectrum is relatively smooth and contains substantial energy over the entire frequency range of interest. Thus, the filtered effect of the muffler can be clearly seen in 7(e). The resultant transmission loss amplitude curve is shown in 7(i). Also, for illustrative purposes, the reflected wave (needed for impedance computations) is shown in Fig. 7(c). It was obtained from the same time record as that of the incident wave by computational manipulations.

Now consider the effect of introducing flow. The downstream transducer signal containing transmitted wave is shown without averaging in Fig. 7(f). The masking of signal detail by the flow noise is apparent. After time domain averaging, however, the flow noise is effectively eliminated, revealing the required signal detail (see Fig. 7(g)). However, spurious reflections can still be seen at about 28 msec. This is then removed by time domain editing, as shown in Fig. 7(h) on an expanded scale. The transmission loss curve with flow, shown in Fig. 7(j), can be compared to the result without flow (Fig. 7(i)). Note that in the case of this simple muffler, magnitude differences are small.

Technique Verification

Through application of standard signal processing theory [22], this technique can be shown to be generally valid for linear systems, and to be valid with appropriate attention to excitation amplitudes for investigating non-linear effects. Perhaps more satisfying, however, is a heuristic verification by comparison with results obtained by conventional experimental and analytical methods.

In Fig. 8, some results are compared for three typical expansion chamber type muffler configurations (without flow - impulse technique data taken using analog system). In Fig. 8(a), impulse technique results are compared with the standing wave tube method results; note the excellent agreement. In Fig. 8(b) for a pipe resonator type of expansion chamber muffler, impulse technique results are compared to computed results. Again, the agreement is excellent. Finally, in Fig. 8(c), measured and computed curves are compared for a more complex muffling system, and again agreement is very good. The computations of Fig. 8(b) and 8(c) are based on Alfredson's formulations [7, 8].

In a similar manner, the results shown in Fig. 7(i) could have been compared to calculated results for verification. For the nonflow case, a similar configuration has been compared in Fig. 8(a). However, for the case with flow (Fig. 7(j)), the required analytical techniques are much less clear. Although

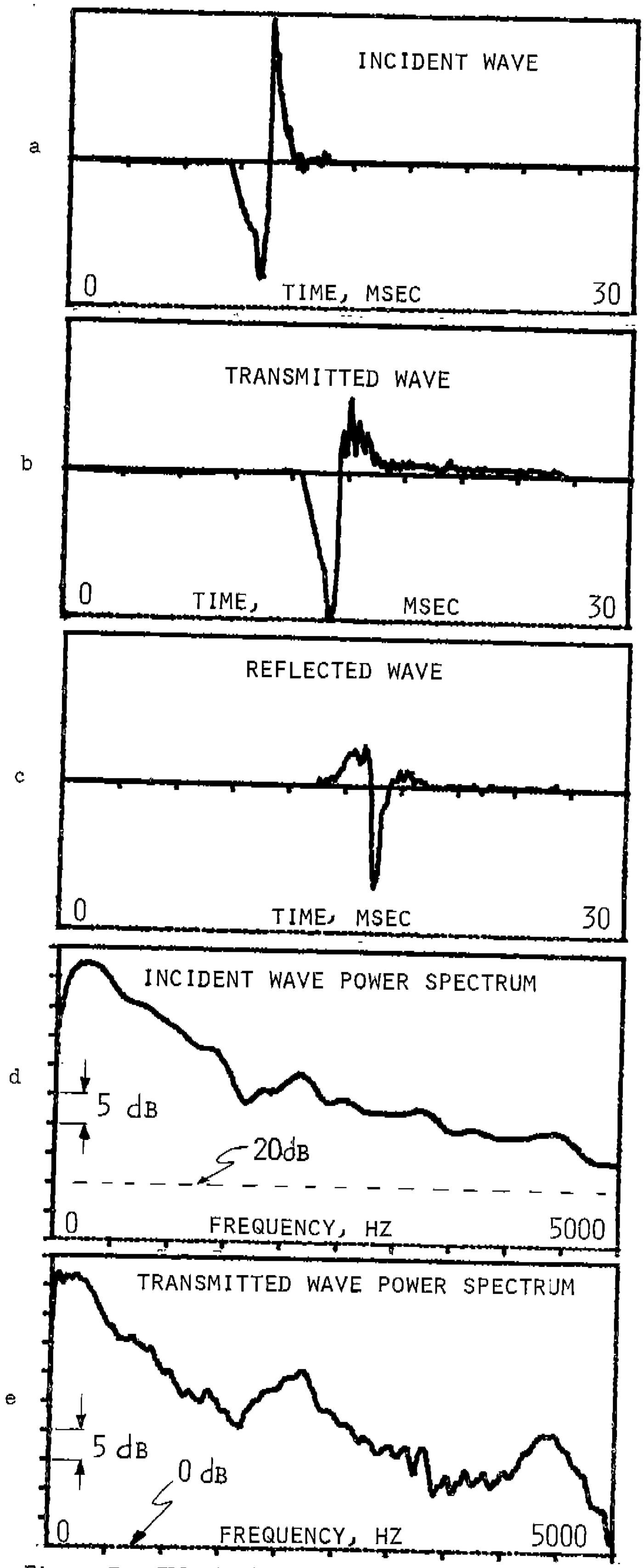


Figure 7 Illustration of Some Measurement Steps and Results

the flow effects were minimal for that simple configuration, they become much more significant for mufflers containing elements such as perforated tubes, which often find application in reciprocating compressors. For such cases, experimental evaluation is much more necessary.

CONCLUDING REMARKS

The main aim of this paper has been to propose and describe an impulse technique for evaluation of acoustic filters in general and compressor mufflers in particular. The authors feel that it has distinct advantages over some other techniques, and are not aware of any similar previous efforts. In the section on muffler characteristics, and in particular in the discussion of flow effects, the paper has not attempted to be exhaustive or innovative. Rather, the intent has been merely to point out the need for much additional work in this area, and by implication the value of an accurate, efficient experimental technique.

It should be pointed out that there is much room for improvement and refinement in the experimental procedure. In particular, the authors feel that the excitation function can be improved to increase energy density, and thus dynamic range. Dynamic range is the main limitation of the impulse method as compared to some other techniques using harmonic excitation. This, together with the cost, complexity, and effort involved must be weighed against its advantages. One distinct advantage, of course, is the ease with which continuous transmission loss curves can be generated. Of more fundamental importance, however, is the fact that complex characteristics (i.e., both magnitude and phase) can be determined isolated from the effects of source and terminating impedances. This has at least three advantages: 1) fundamental properties can be better evaluated; 2) the effectiveness of given mufflers in realistic environments can be predicted if source and terminating impedances are known; and, 3) a building block approach, wherein muffler elements are evaluated experimentally and combined analytically, becomes feasible. Thus both experimental and theoretical effects can converge to a unified approach. It is in these directions that the authors feel much fruitful future work lies.

ACKNOWLEDGEMENTS

The authors wish to acknowledge numerous individuals, especially C. Ebbing, D. King, D. Neville, J. Rainey and R. Ryan of the Research Division, Carrier Corporation, for their kind assistance.

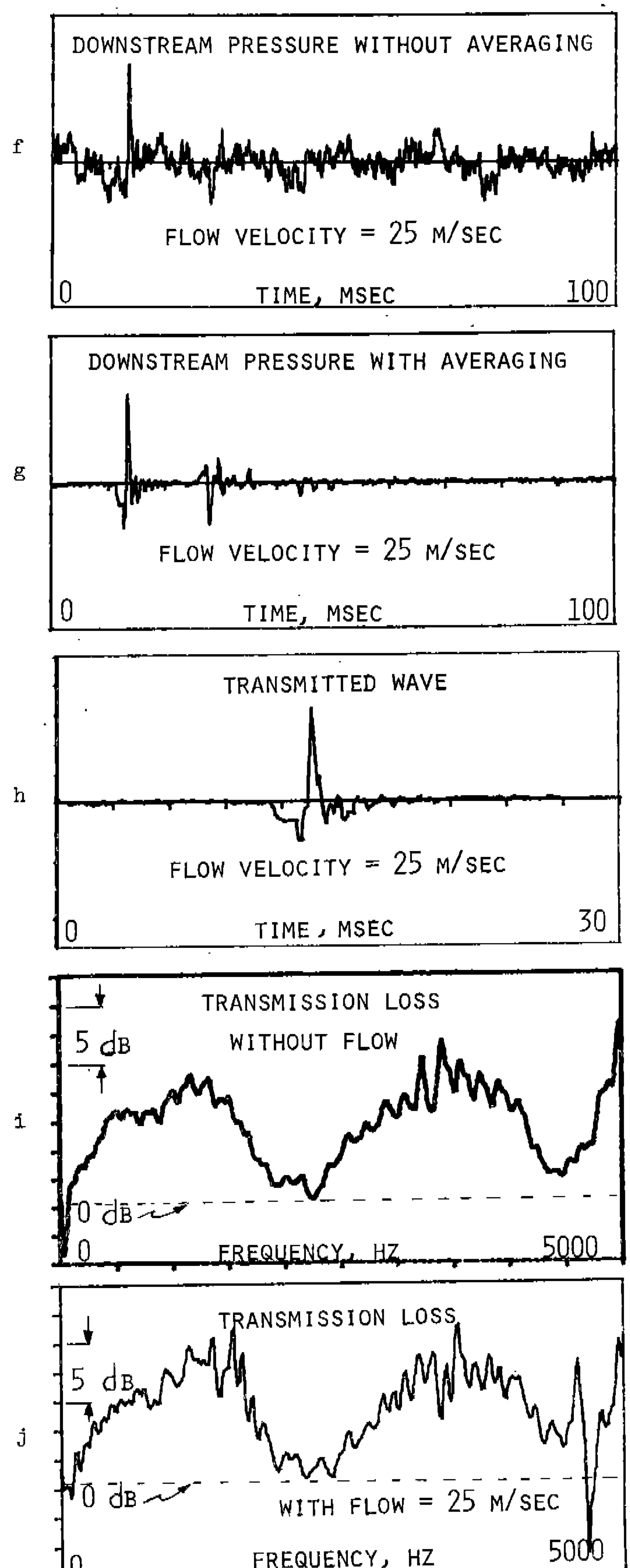
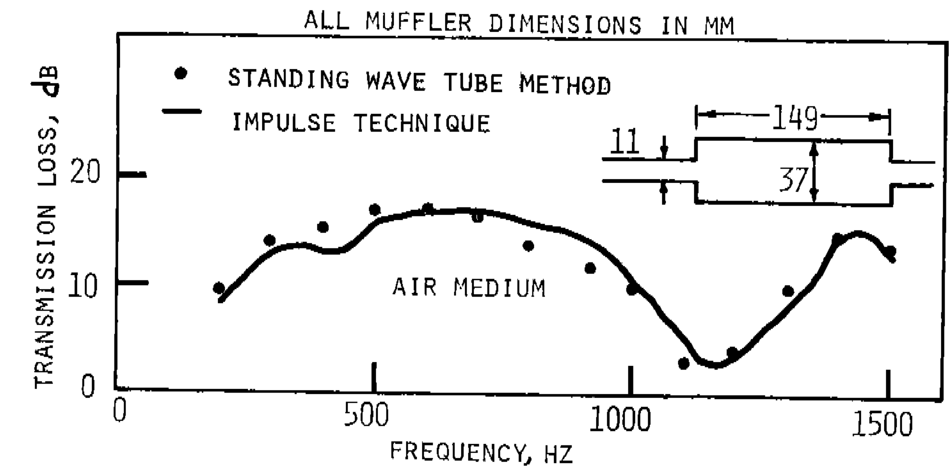
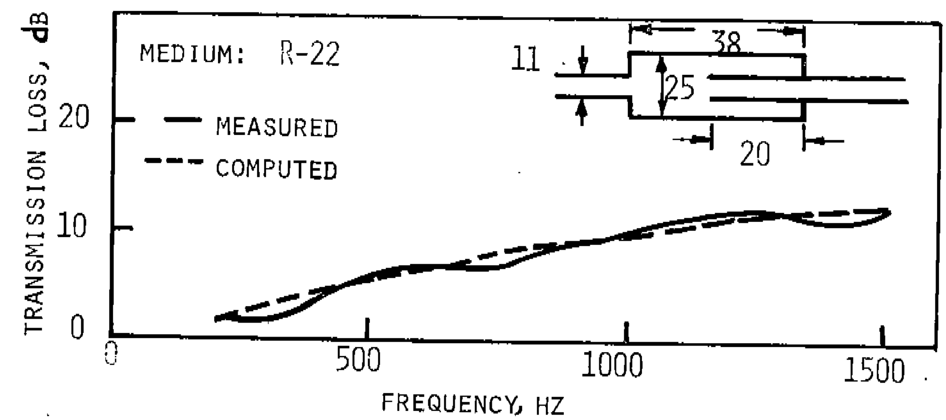


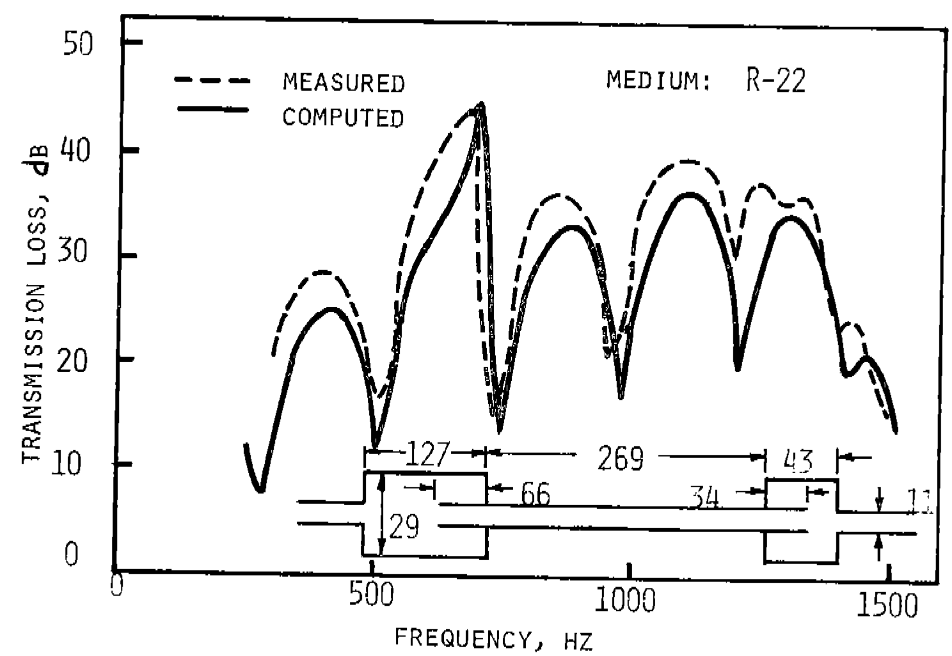
Figure 7 continued



a. Simple Expansion Chamber



b. Pipe Resonator



c. Composite Pipe Resonator

Figure 8

Comparisons Of Results

REFERENCES

1. K. U. Ingard, 1953, J. Acoust. Soc. Am., 25(6), 1037-1061, "On The Theory and Design of Acoustic Resonators".

2. E. Meyer and E. G. Neumann, 1972, "Physical and Applied Acoustics", New York: Academic Press.

3. T. F. W. Embleton, 1971, "Noise and Vibration Control", Edited by L. L. Beranek, Chapter 12- "Mufflers", New York: McGraw Hill Book Co.

4. R. Singh and W. Soedel, 1974, Purdue Compressor Tech. Conf., Proc. 102-123, "A Review of Compressor Lines Pulsation Analysis and Muffler Design Research, Part I - Pulsation Effects and Muffler Criteria, Part II - Analysis of Pulsating Flows".

5. M. L. Munjal and A. V. Sreenath, 1973, Shock and Vib. Digest, 5(11), 2 - 14, "Analysis and Design of Exhaust Mufflers - Recent Developments".

6. E. K. Bender and A. J. Brammer, 1975, J. Acoust. Soc. Am., 58(1), 22 - 30, "Internal Combustion Engine Intake and Exhaust System Noise".

7. R. J. Alfredson and P. O. A. L. Davies, 1971, J. Sound and Vib., 15(2), 176 - 196, "Performance of Exhaust Silencer Components".

8. T. L. Parrott, 1973, NASA, TN D-7309, "An Improved Method for Design of Expansion Chamber Mufflers With Application to An Operational Helicopter".

9. P. K. Baade, 1975, Purdue University Seminar, "Flow Effects in Mufflers - Aerodynamic Acoustic Interactions".

10. I. K. Gösele, 1965, VDI-Berichte, 88, 123 - 130, "The Damping Behaviour of Reactive Mufflers With Air Flow".

11. D. Ronneberger, 1972, J. Sound and Vib., 24(1), 133 - 150, "The Acoustical Impedance of Holes in the Wall of Flow Ducts".

12. K. U. Ingard and V. K. Singhal, 1973, J. Acoust. Soc. Am., 54(3), 1343 - 1346, "Upstream and Downstream Sound Radiation Into a Moving Fluid".

13. K. U. Ingard and V. K. Singhal, 1974, J. Acoust. Soc. Am., 55(3), 535 - 538, "Sound Attenuation in Turbulent Pipe Flow".

14. R. H. Schaffart, 1972, Univ. of Missouri-Rolla, Ph.D. Thesis, "An Experimental Investigation of the Effects of Ambient and Heated Steady Flow and Intense Sound Levels on the Response of Acoustic Filter Elements".

15. K. U. Ingard and H. Ising, 1967, J. Acoust. Soc. Am., 42, 6 - 17, "Acoustic Nonlinearity of An Orifice".

16. J. W. Sullivan, 1974, Purdue University, Ph.D. Thesis, "Theory and Methods for Modeling Acoustically-Long, Unpartitioned Cavity Resonators for Engine Exhaust Systems".

17. W. S. Gatley and R. Cohen, 1969, J. Acoust. Soc. Am., 46(1), 6 - 16, "Methods for Evaluating the Performance of Small Acoustic Filters".

18. W. S. Gatley and R. Cohen, 1971, ASHRAE Tr., 76, 2128, "Development and Evaluation of a General Method for Design of Small Acoustic Filters".

19. T. H. Melling, 1973, J. Sound and Vib., 28(1), 23 - 54, "An Impedance Tube for Precision Measurement of Acoustic Impedance and Insertion Loss at High Sound Pressure Levels".

20. R. Singh, 1975, Purdue University, Ph.D. Thesis, "Modeling of Multicylinder Compressor Discharge Systems".

21. D. C. Champeney, 1973, "Fourier Transforms and Their Physical Applications", New York: Academic Press.

22. J. S. Bendat and A. G. Piersol, 1971, "Random Data: Analysis and Measurement Procedures", New York: Wiley-Interscience.

23. I. E. Morse, W. R. Shapton, D. L. Brown and E. Kuljanic, 1972, 13th Int. Mach. Tool Des. Res. Conf., Birmingham, U. K. (Also Hewlett-Packard App. Note 140-3), "Applications of Pulse Testing for Determining Dynamic Characteristics of Machine Tools".