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The Development and Experimental Verification of an Acoustic Damping Model for a Frequency Domain Digital Pulsation Simulation

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THE DEVELOPMENT AND EXPERIMENTAL VERIFICATION OF AN ACOUSTIC DAMPING MODEL FOR A FREQUENCY DOMAIN DIGITAL PULSATION SIMULATION

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ABSTRACT

This paper summarizes the development of an acoustic damping model used in a frequency domain digital pulsation simulation program. The theoretical basis for the model is presented as are laboratory data verifying the accuracy of the pulsation amplitude predictions for both resonant and nonresonant conditions.

INTRODUCTION

Transfer matrix acoustic pulsation simulations are receiving increased usage for pressure pulsation analyses in reciprocating compressor piping systems. The transfer matrix approach requires certain simplifying assumptions in order to solve the basic differential equations. While these assumptions enhance the speed of the calculations, the accuracy of the results is adversely affected. One frequently simplified calculation is that of acoustic damping. Since the pressure pulsation amplitude at a resonance is critically dependent upon the damping, simplification of this calculation is one of the major sources of error in a pulsation simulation.

The work outlined in this paper was accomplished during the period from January, 1987 to October, 1987 and started with a transfer matrix pulsation simulation program which was not sufficiently accurate in its pulsation amplitude calculations. The method of modeling the acoustic damping was improved until acceptable agreement between calculated results and laboratory data was achieved.

NOMENCLATURE

a	Pipe radius
A	Cross-sectional pipe area
c	Sonic velocity
C	Pipeline capacitance
Cp	Specific heat at constant pressure
g	Acceleration of gravity
Hu	Upstream pressure
Hd	Downstream pressure
k	Thermal conductivity
ℓ	Pipe length
L	Pipeline inertance
Qu	Upstream flow
Qd	Downstream flow
R	Resistance (pressure drop per unit length)
Z	Acoustic impedance
β	Complex frequency
γ	Ratio of specific heats
η	Shear coefficient of viscosity
ν	Kinematic viscosity
ρ	Fluid density
σ	Acoustic damping (real part of complex frequency)
σ_f	Frictional damping
σ_{vt}	Viscous and thermal conduction damping
ω	Angular frequency

THEORETICAL BASIS

Most transfer matrix pulsation simulations are derived from the same differential equation. The steps to the solution are well documented and will not be repeated here. The matrix form of the solution to the differential equation[1] is:

$$\begin{bmatrix} Hd \\ Qd \end{bmatrix} = \begin{bmatrix} \cosh\beta\ell & -Z\sinh\beta\ell \\ -\frac{\sinh\beta\ell}{Z} & \cosh\beta\ell \end{bmatrix} \begin{bmatrix} Hu \\ Qu \end{bmatrix} \qquad (1)$$

Where $\beta = \sigma + i\omega/c$

When an acoustic resonance is present, the accuracy of the pulsation amplitude calculation is critically dependent upon the value of σ, the acoustic damping. One method of calculating the damping utilizes a special case of the classical acoustic attenuation factor. The general form of the equation for sound attenuation in a large volume due to the viscous damping of the fluid and thermal conduction[2] is:

$$\sigma_v = \frac{\omega^2}{2\rho c^3}\left(\frac{4\eta}{3} + \frac{k(\gamma-1)}{Cp}\right) \qquad (2)$$

For acoustic wave propagation in air through a cylindrical pipe, equation (2) can be reduced[3] to:

$$\sigma_v \cong \frac{1.39}{ac}\sqrt{\frac{\nu\omega}{2}} \qquad (3)$$

The factor of 1.39 accounts for the thermal conduction of air[2].

When this model was used in a transfer matrix pulsation simulation, accurate results could be obtained for a given gas composition by multiplying the damping by a constant. This constant was different for different gases and was approximately proportional to the molecular weight. Through laboratory tests, these multipliers were determined to be 1.0 for carbon dioxide, 0.5 for nitrogen and 0.1 for helium (Fig. 1). However, cases were found where this simple damping model yielded inconsistent results (Fig. 2). The derivation[3] of equation (3) required the assumption that the pipe diameter was small (< 1 inch) to insure the flow was laminar and that the wall friction would be large enough to be significant. Due to these inconsistencies and the assumption of small pipe in the derivation, this approach to modeling acoustic damping was determined to be inadequate.

For acoustical wave propagation in a round pipe, the damping due to wall friction can be added to the damping due to viscous effects and thermal conduction to give the total sound attenuation. If the frictional damping is expressed in terms of pressure drop, the effects of pipe elbows, tees, valves, diameter changes, etc. on acoustic attenuation can be quantified and added to the total. The propagation constant[1] can be expressed by:

$$\beta = \sqrt{C\omega(L\omega+R)} \qquad (4)$$

Equation (4) can be rearranged[1] to equal:

$$\beta = \frac{AR}{2\rho c} + i\frac{\omega}{c} \qquad (5)$$

The piping network can be divided into elements that represent straight lengths of uniform diameter pipe with reducers, elbows, valves, etc. specified as end conditions of each element. The pressure drops due to these end conditions as well as the pressure drops due to wall friction can be easily calculated if the velocity is known. The total acoustic damping can be expressed as the sum of the results from equation (2) and equation (5):

$$\sigma = \frac{\omega^2}{2\rho c^3}\left(\frac{4\eta}{3} + \frac{k(\gamma-1)}{Cp}\right) + \frac{AR}{2\rho c} \qquad (6)$$

Since the pressure drop calculations are nonlinear, the gas flow velocity cannot be used to calculate the acoustic damping due to friction. The dynamic fluid velocity can be calculated from equation (1). Separating the flow portion of equation (1) gives:

$$Qd = -\frac{Hu\ \sinh\beta l}{Z} + Qu\ \cosh\beta l \qquad (7)$$

Equation (7) can be used to calculate the dynamic fluid velocity at any point in the piping and the result can then be used to calculate the damping due to friction.

IMPLEMENTATION

This method of using fluid velocity to calculate acoustic damping was incorporated into a transfer matrix pulsation simulation program. The program used an ideal cylinder and ideal valve model to provide the flow input data to the pulsation simulation which was iterated twice. The first iteration used damping which was calculated by equation (3) with a damping multiplier chosen to yield a good approximation of the fluid velocity. The maximum rms fluid velocity was calculated in each piping element and at each harmonic. Standard pressure drop equations were then used to determine the pressure drop in each pipe. The total acoustic damping was calculated according to equation (6) and the pulsation calculation was then rerun.

VERIFICATION

A series of tests were run to verify the accuracy of the pulsation amplitude calculations when using this damping model. A reciprocating compressor was operated in a closed test loop under laboratory conditions. A variable length resonator was assembled by connecting a series of gate valves at one foot intervals. This valve manifold was connected between the suction and discharge lines (Fig. 3). To maintain uniform temperatures in the resonant length during discharge tests, a small amount of hot gas was leaked back to the suction line. A total of seventeen test cases were run with helium, nitrogen and carbon dioxide at a variety of compressor speeds, gas temperatures, line pressures and resonant lengths. Pulsation measurements were made at eleven points in the system and all of the results were compared to the pulsation levels calculated by the simulation.

REQUIRED ACCURACY

The primary use of a pulsation simulation is as a design tool which can be used to predict pulsation levels in a piping system. Electrical analog simulations have been in use for several years with acceptable results. It was felt that in order for a digital simulation to be a useful tool for piping analyses, the accuracy of the calculated pulsation amplitudes had to be superior to the accuracy of analog. According to comparisons of analog results to the data from the closed loop testing, the approximate accuracy of analog was:

For nonresonant conditions:	-50% to +50%
For resonant conditions:	-50% to +400%

It was established that the required accuracy for the digital pulsation simulation was:

For nonresonant conditions:	-25% to +25%
For resonant conditions:	-50% to +100%

RESULTS

Initial attempts at using this damping model revealed apparent inconsistencies in the pulsation amplitudes at resonances. The calculated pulsation amplitudes at resonances were within the accuracy requirements at some points and were significantly higher at other points (100%-300%). The points where the amplitudes were within the accuracy requirements were in straight, continuous pipes (such as Fig. 3 Test Point 7).

The points where the calculated pulsation amplitudes were much higher were near discontinuities (valves, elbows, etc.) in the piping (such as Fig. 3 Test Point 9) and in the cylinder gas passages. It was determined that in order to match the laboratory data, it was necessary to model all pressure losses throughout the system. All elbows, tees, valves, diameter changes, etc. were modeled as pressure losses at the ends of straight lengths of pipe. The irregular shapes of the cylinder passages caused a much higher pressure loss than would have been calculated if the passages were modeled as straight, round pipes. To account for this difference, flow test data were used to calculate an effective wall roughness for the cylinder passages which was then used in the pressure drop calculation. Once all pressure losses were accurately modeled, the established accuracy criteria was met (Fig. 4).

CONCLUSION

The accuracy of Dresser-Rand's frequency domain pressure pulsation simulation was improved to a level superior to that of an electrical analog simulation through the development of an improved acoustic damping model which accounted for viscous damping, thermal conductivity and friction effects. This program is currently being used commercially to perform acoustic analyses of reciprocating compressor piping systems.

FUTURE DEVELOPMENT

With the development of an improved acoustic damping model, the accuracy of this pulsation simulation has been improved to a level considered adequate for the program's intended purpose. There were, however, a few test cases where resonant peaks in the test data were missed by the simulation (Fig. 4a). There is room for further improvement in the damping model as well as in the cylinder simulation. The interaction of compressor valves and pressure pulsations is presently being investigated.

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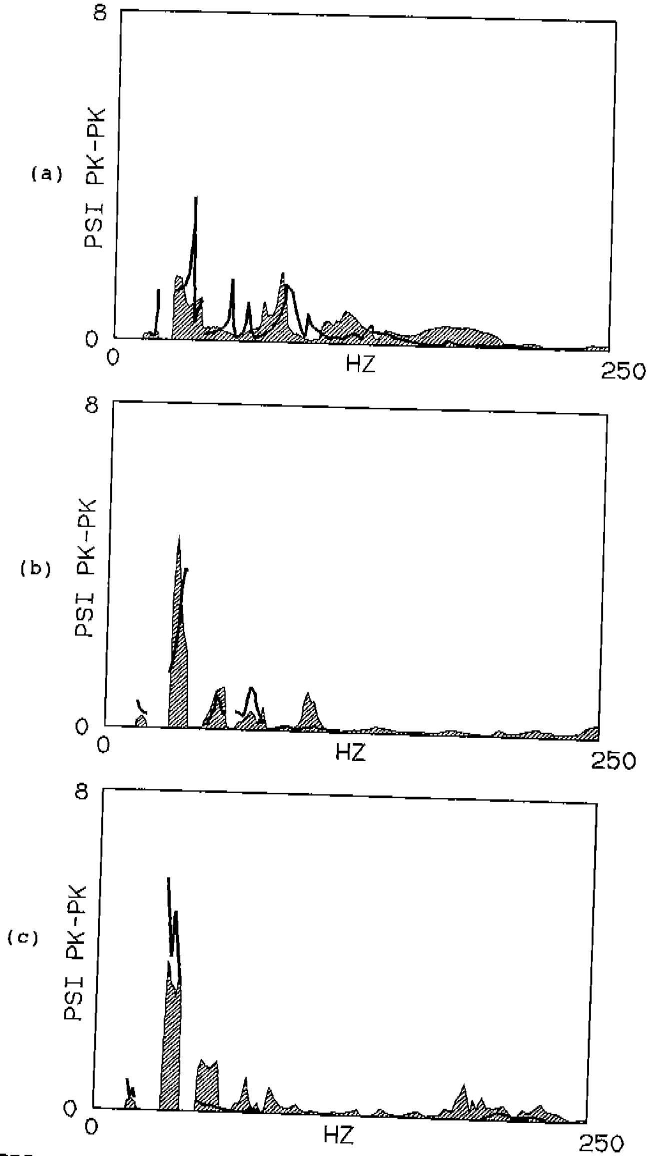

FIG. 1 Comparison of laboratory data to results from a pulsation simulation using a simple damping model (Eq. 3). Lab data is shown by the shaded areas.
(a) Helium test point 7; damping multiplier=0.1
(b) Nitrogen test point 9; damping multiplier=0.5
(c) CO2 test point 7; damping multiplier=1.0

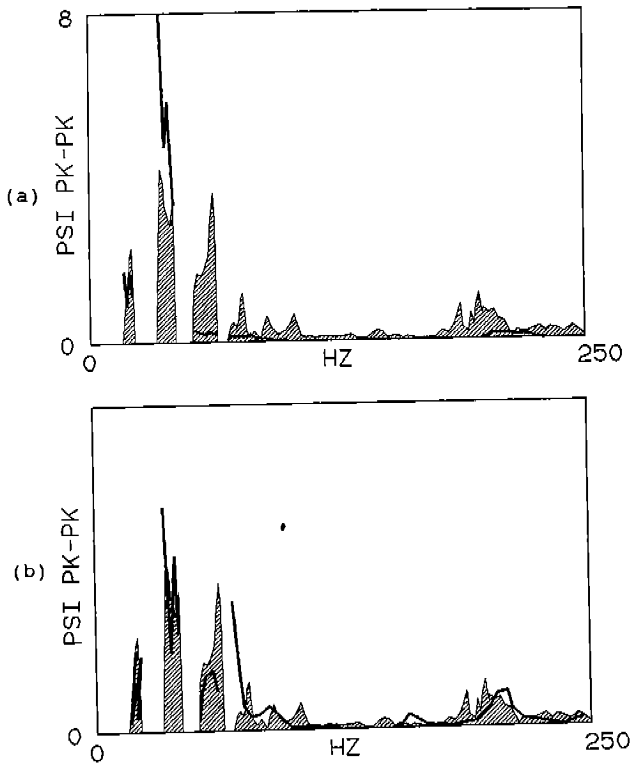

FIG. 2 Comparison of laboratory data to results from pulsation simulations at test point 9 of a carbon dioxide test. The laboratory data is shown by the shaded areas.
(a) Inconsistent results with the damping calculated by equation (3).
(b) Improved accuracy with the damping calculated according to equation (6).

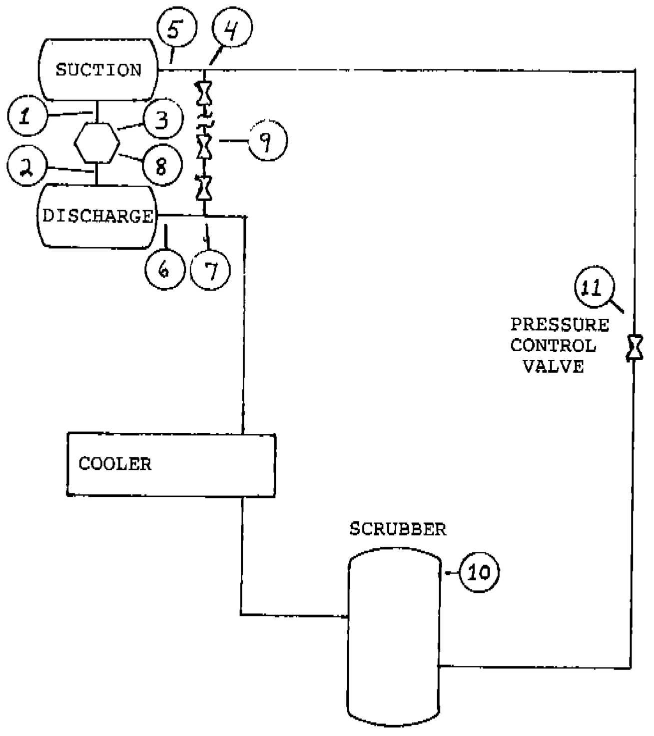

FIG. 3 Configuration of the closed loop test setup used to verify the accuracy of the acoustic damping model. Pulsation test points are indicated by the circled numbers. Test point number 9 was moveable and was always mounted at the end of the resonant length.

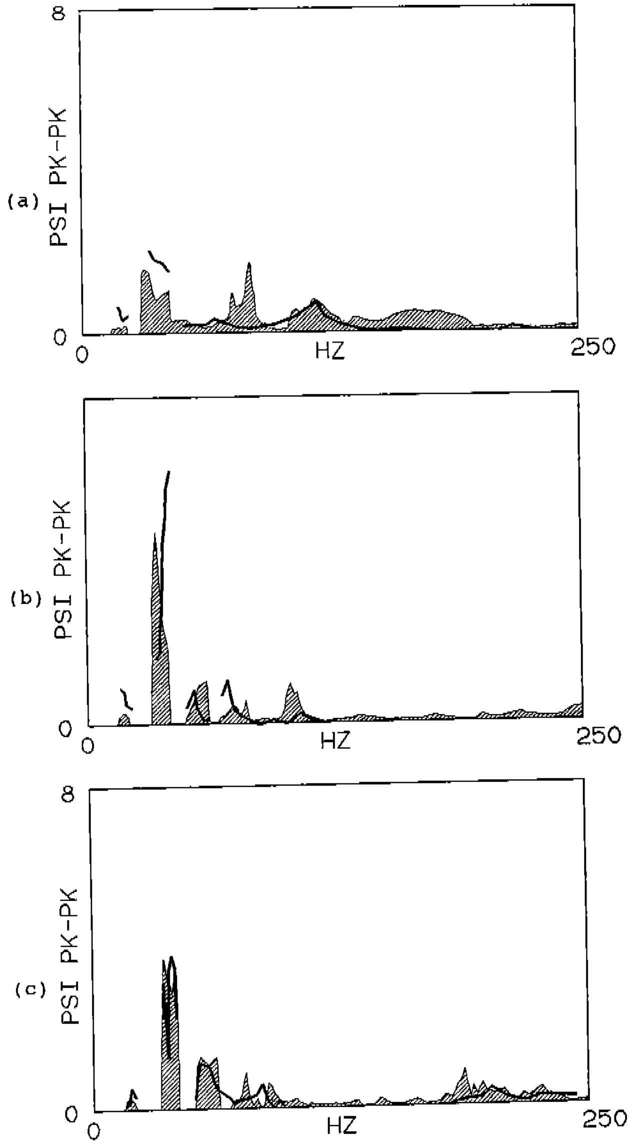

FIG. 4 Comparison of laboratory data to results from a pulsation simulation using an improved damping model which included the effects of friction. (Eq. 6). Lab data is shown by the shaded areas.
(a) Helium test point 7
(b) Nitrogen test point 9
(c) Carbon dioxide test point 7