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DISCHARGE GAS PULSATIONS IN A VARIABLE SPEED COMPRESSOR

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Abstract
Gas pulsations in the discharge side of a variable speed compressor are studied by two different Helmholtz resonator models. The first model, which treats the space under a discharge valve cover as one volume, is compared to the second one, which considers the restrictions imposed by assembly bolts to the space under the valve cover. Results from both models are compared to experimental data. The effect of compressor speed on discharge gas pulsations is discussed.

Introduction
During each cycle of compressor operation, refrigerant gas is pumped into the cylinder, compressed to a certain pressure, and discharged out of the cylinder into the discharge manifolds. Since the discharge ports open only for a small fraction of the duration of each revolution, the mass flow through the discharge ports is discontinuous; since the discharge process repeats itself as a cyclic operation, the mass flow through the discharge port is periodic.

To the fluid in the manifolds or cavities on the discharge side, this intermittently dumped gas is an excitation source. Consequently, the pressure in those cavities oscillates as long as the compressor is running. This kind of gas pulsations influences the thermodynamic process inside the cylinder, and introduces noise problems. Whether the purpose of study is to investigate the performance and efficiency or to describe the noise source, a discharge gas pulsation model is very necessary. Among several phenomena inherent to the operation of compressors, gas pulsations play an important role.

Past experiences and studies show that the linear acoustic theory has been very successful when applied to small compressors [1-2]. The linear acoustic theory is based on the assumption that sound waves have an infinitesimally small pressure magnitude. This assumption is satisfied in most of the compressors used for refrigeration and air-conditioning. The linear acoustic models have been proven to give satisfactory results when the acoustic pressure is smaller than 20% of the mean pressure [1].

A simple linear model, called the lumped parameter approach, has been widely used to study gas pulsations in small volumes and short passages of compressors. In this approach, an acoustical system is considered as a combination of two lumped elements: a small volume and a short neck. This simple linear model is also called the Helmholtz resonator approach. Soedel et al. [3] applied the Helmholtz resonator approach to model gas pulsations in compressor manifolds. Mutyla and Soedel [4] also used this model to analyze the gas pulsations in a two-stroke cycle engine manifold. Lee et al. [5] applied this approach to model gas pulsations in a pulse combustion device.

The model based on this approach can either be solved in the frequency or time domain. The time domain Helmholtz resonator model is used in this study.

Helmholtz Resonator Approach
Volumes and necks are commonly used elements in compressor discharge sides, whose dimensions are usually much smaller than the wavelength of sound at frequencies of interest. Under such circumstances, the volume and neck pair can be approximated as a linear vibration system by treating the gas in the neck as an incompressible plug and the
gas in the volume as inertialess and compressible; the incompressible gas plug oscillates like the mass element in a vibration system with the elastic spring stiffness provided by the elasticity of the compressible gas in the volume, as shown in Figure 1.

The dynamic equation of the gas plug can be obtained by applying Newton’s second law to the plug, referring to the free body diagram shown in Figure 1:

\[ L A \rho_0 \ddot{\xi} = \bar{p} A - D \dot{\xi} \]  

(1)

where \( \dot{\xi} \) is the displacement of the gas plug, \( A \) is the cross-sectional area of the neck, \( \rho_0 \) is the mean density, \( \bar{p} \) is the perturbation pressure, \( D \) is the equivalent damping coefficient, and \( L \) is the effective length of the gas plug, which is the geometrical length of the neck plus an end correction term

\[ L = L_s + \frac{\pi A}{2} \]  

(2)

When the acoustic pressure is small relative to the mean pressure, it is proportional to the density change of the gas, and can be expressed as

\[ \bar{p} = K_0 \frac{\Delta \rho}{\rho_0} = -K_0 \frac{\Delta V}{V} \]  

(3)

where \( V \) is the volume of the cavity, and \( K_0 \) is the bulk modulus which is related to the mean density and speed of sound in the refrigerant gas by

\[ K_0 = \rho_0 a_0^2 \]  

(4)

The volume change \( \Delta V \), referring to Figure 1, can be expressed as

\[ \Delta V = \frac{1}{\rho_0} \int_0^t m\,dt + A \dot{\xi} \]  

(5)

where \( m \) is the mass flow rate into the cavity. Substituting the above two equations into equation (3) gives

\[ \bar{p} = \frac{a_0^2}{V} \int_0^t m\,dt - \frac{a_0^2}{V} \rho_0 A \dot{\xi} \]  

(6)

Equations (1) and (6) can be rearranged as

\[ \frac{du}{dt} = \frac{1}{L A \rho_0} (\bar{p} A - D u) \]  

(7)

\[ \frac{d\bar{p}}{dt} = \frac{a_0^2}{V} m - \frac{a_0^2}{V} \rho_0 A \dot{\xi} = \frac{\rho_0 a_0^2}{V} \left( \frac{m}{\rho_0} - A u \right) \]  

(8)

where \( u \) is the velocity of the gas in the neck.

The above equation governs the gas oscillations in a Helmholtz resonator. This Helmholtz resonator approach is simple and can be applied to any kind of irregular and complicated configuration. The only restriction is that its dimensions should be small relative to the wavelength of the highest frequencies of interest. A generally accepted rule
is that the largest dimension of the resonator should be smaller than a quarter of the wavelength \([1]\). Most of the volumes and necks in small refrigeration compressors satisfy this condition.

**The Discharge System**

The discharge system of the compressor in this study is depicted in Figure 2. In order to reduce the noise caused by the impulsively discharged gas, discharge mufflers are commonly utilized in compressors. In this application, the two discharge valve covers comprise the discharge muffler. The two cavities formed by the two discharge valve covers are connected by two identical circular passages (only one is drawn in Figure 2). During each cycle, high pressure and high temperature gas is dumped into these two manifolds. The gas then flows out of the upper valve cover into a much larger shell space, and finally leaves the compressor shell into the next component, the condenser. Since the volume of the shell cavity is large compared with those of the cavities under the valve covers, we can consider the pressure in the shell space as constant. This simplification is reasonably accurate if the major interest is the effect of discharge gas pulsations on the performance of the compressor.

**A Simple Lumped Model**

As a first attempt, each of the two irregular cavities under the valve covers is lumped as one volume; the two circular passages are combined as one neck of cross-section area equal to the total cross-section area of the two passages. The schematic diagram is shown in Figure 3. Applying equations (7) and (8) to this system gives

\[
\frac{du_1}{dt} = \frac{1}{L_1 A_1 \rho_0} \left[ (\bar{P}_{01d} - \bar{P}_{02d}) A_1 - D_1 u_1 \right] \tag{9}
\]

\[
\frac{d\bar{P}_{01d}}{dt} = \frac{\rho_0 a_0^2}{V_{01d}} \left( \frac{\dot{m}_1}{\rho_0} - A_1 u_1 \right) \tag{10}
\]

\[
\frac{du_2}{dt} = \frac{1}{L_2 A_2 \rho_0} \left[ (\bar{P}_{02d} - 0) A_2 - D_2 u_2 \right] \tag{11}
\]

\[
\frac{d\bar{P}_{02d}}{dt} = \frac{\rho_0 a_0^2}{V_{02d}} \left( \frac{\dot{m}_2}{\rho_0} + A_1 u_1 - A_2 u_2 \right) \tag{12}
\]

The above equations govern the gas pulsations in the discharge manifolds, which affect the dynamic motion of the discharge valves and in turn influence the cylinder thermodynamic process. The gas pulsation model is coupled with both the valve dynamic model and the cylinder thermodynamic model \([6]\), and all the differential equations of those models have to be solved simultaneously.

The numerical simulation result predicted under the condition \(P_s=65.5\) psia (nominal suction pressure), \(T_s=44\) F (nominal suction temperature), \(P_d=242.3\) psia (nominal discharge pressure), and \(\Omega=4647.5\) rpm (compressor speed) is compared with the experimental data in Figure 4. Except for the high frequency wave which appears on the measured pressure curve, the above simple model is able to predict the major behavior of the gas pulsation.

**A Refined Lumped Model**

The contour of the space under each of the two discharge valve covers is drawn in Figure 5. A more careful examination of this configuration indicates that inside each valve cover are five Helmholtz resonators in series, due to the restrictions imposed by assembly bolts. Therefore, the discharge system is modeled as a combination of 10 volumes and 14 necks, in the way shown in Figure 6. This can be considered as an oscillatory system with 14 degrees
of freedom. The dynamic equation for the Helmholtz resonator discussed in the previous section can be extended to each of those elements. Applying equation 7 to the 10 volumes of discharge system gives the following equations:

\[
\frac{d\tilde{p}_{1d}}{dt} = \frac{\rho_0 a_0^2}{V_1} \left( \frac{\dot{m}_1}{\rho_0} - A_1 u_{1d} + A_5 u_{5d} \right)
\]

\[
\frac{d\tilde{p}_{2d}}{dt} = \frac{\rho_0 a_0^2}{V_2} (A_1 u_{1d} - A_2 u_{2d})
\]

\[
\frac{d\tilde{p}_{3d}}{dt} = \frac{\rho_0 a_0^2}{V_3} (A_2 u_{2d} - A_3 u_{3d} - A_{14} u_{14d})
\]

\[
\frac{d\tilde{p}_{4d}}{dt} = \frac{\rho_0 a_0^2}{V_4} (A_3 u_{3d} - A_4 u_{4d})
\]

\[
\frac{d\tilde{p}_{5d}}{dt} = \frac{\rho_0 a_0^2}{V_5} (A_4 u_{4d} - A_5 u_{5d} - A_{13} u_{13d})
\]

\[
\frac{d\tilde{p}_{6d}}{dt} = \frac{\rho_0 a_0^2}{V_6} (A_{13} u_{13d} + A_{10} u_{10d} - A_6 u_{6d} - A_{11} u_{11d})
\]

\[
\frac{d\tilde{p}_{7d}}{dt} = \frac{\rho_0 a_0^2}{V_7} (A_6 u_{6d} - A_7 u_{7d})
\]

\[
\frac{d\tilde{p}_{8d}}{dt} = \frac{\rho_0 a_0^2}{V_8} (A_7 u_{7d} + A_{14} u_{14d} - A_8 u_{8d})
\]

\[
\frac{d\tilde{p}_{9d}}{dt} = \frac{\rho_0 a_0^2}{V_9} (A_8 u_{8d} - A_9 u_{9d} - A_{12} u_{12d})
\]

\[
\frac{d\tilde{p}_{10d}}{dt} = \frac{\rho_0 a_0^2}{V_{10}} \left( \frac{\dot{m}_2}{\rho_0} + A_9 u_{9d} - A_{10} u_{10d} \right)
\]

Applying Newton's second law to each of the 14 necks gives

\[
\frac{du_{1d}}{dt} = \frac{1}{L_1 A_1 \rho_0} \left[ (\tilde{p}_{1d} - \tilde{p}_{2d}) A_1 - D_1 u_{1d} \right]
\]
\[
\frac{du_{2d}}{dt} = \frac{1}{L_2 A_2 \rho_0} \left[ (\bar{p}_{2d} - \bar{p}_{3d}) A_2 - D_2 u_{2d} \right]
\]
(24)

\[
\frac{du_{3d}}{dt} = \frac{1}{L_3 A_3 \rho_0} \left[ (\bar{p}_{3d} - \bar{p}_{4d}) A_3 - D_3 u_{3d} \right]
\]
(25)

\[
\frac{du_{4d}}{dt} = \frac{1}{L_4 A_4 \rho_0} \left[ (\bar{p}_{4d} - \bar{p}_{5d}) A_4 - D_4 u_{4d} \right]
\]
(26)

\[
\frac{du_{5d}}{dt} = \frac{1}{L_5 A_5 \rho_0} \left[ (\bar{p}_{5d} - \bar{p}_{6d}) A_5 - D_5 u_{5d} \right]
\]
(27)

\[
\frac{du_{6d}}{dt} = \frac{1}{L_6 A_6 \rho_0} \left[ (\bar{p}_{6d} - \bar{p}_{7d}) A_6 - D_6 u_{6d} \right]
\]
(28)

\[
\frac{du_{7d}}{dt} = \frac{1}{L_7 A_7 \rho_0} \left[ (\bar{p}_{7d} - \bar{p}_{8d}) A_7 - D_7 u_{7d} \right]
\]
(29)

\[
\frac{du_{8d}}{dt} = \frac{1}{L_8 A_8 \rho_0} \left[ (\bar{p}_{8d} - \bar{p}_{9d}) A_8 - D_8 u_{8d} \right]
\]
(30)

\[
\frac{du_{9d}}{dt} = \frac{1}{L_9 A_9 \rho_0} \left[ (\bar{p}_{9d} - \bar{p}_{10d}) A_9 - D_9 u_{9d} \right]
\]
(31)

\[
\frac{du_{10d}}{dt} = \frac{1}{L_{10} A_{10} \rho_0} \left[ (\bar{p}_{10d} - \bar{p}_{11d}) A_{10} - D_{10} u_{10d} \right]
\]
(32)

\[
\frac{du_{11d}}{dt} = \frac{1}{L_{11} A_{11} \rho_0} \left[ (\bar{p}_{11d} - 0) A_{11} - D_{11} u_{11d} \right]
\]
(33)

\[
\frac{du_{12d}}{dt} = \frac{1}{L_{12} A_{12} \rho_0} \left[ (\bar{p}_{12d} - 0) A_{12} - D_{12} u_{12d} \right]
\]
(34)

\[
\frac{du_{13d}}{dt} = \frac{1}{L_{13} A_{13} \rho_0} \left[ (\bar{p}_{13d} - \bar{p}_{6d}) A_{13} - D_{13} u_{13d} \right]
\]
(35)
\[
\frac{d u_{14d}}{dt} = \frac{1}{L_{14} A_{14} \rho_0} \left[ (\bar{p}_{3d} - \bar{p}_{5d}) A_{14} - D_{14} u_{14d} \right] 
\]

The above 24 first order differential equations, which govern the gas pulsations in the discharge muffler, are coupled with both the cylinder thermodynamic equations and the valve dynamic equations through the mass flow out of the cylinder. After determining the acoustic pressures in the discharge cavities, we can get the total pressures by adding the acoustic pressures to the mean discharge pressure.

The computed result shown in Figure 7 is obtained under the same condition \( P_s = 65.5 \text{ psia}, T_s = 44 \text{ F}, P_d = 242.3 \text{ psia}, \) and \( \Omega = 4647.5 \text{ rpm} \) as that shown in Figure 4. The computed result compares better to the experimental data than the result of the previous model. The differences between the experiment result and the predicted one are partly caused by the empirically determined damping coefficients, whose values are difficult to be determined exactly from first principles.

**Gas Pulsations under Different Speeds**

Figure 8 shows the pressure pulsations under the same condition as the previous cases except at a higher speed \( \Omega = 6011.7 \text{ rpm} \), simulated by the refined model. It is not difficult to see that the gas in the discharge manifolds oscillates more violently at the higher speed, as one would intuitively expect.

**Conclusions and Future Work**

Two different Helmholtz resonator models are used to study the gas pulsations in the discharge manifolds of a variable speed compressor. Both of them are compared to experimental data. The second model is more complex but more precise than the first one. Computer simulation results indicate that the increase of compressor speed enhances, in general, pressure pulsations in the discharge manifolds.

To further improve the accuracy of the model, a continuous approach may be instead used to simulate the flow in the two circular passages connecting the space under the lower valve cover to the upper valve cover. These two passages are relative long in comparison with the others. The gas dynamics model can be applied to this case so that the whole problem can still be solved in the time domain.

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**References**

Compressible and inelastic fluid

Figure 1 A Helmholtz resonator

Figure 2 The discharge system

Figure 3 A crude Helmholtz model

Figure 4 Comparison of the first model prediction with experimental data

Figure 5 Contour of the discharge cavity
Figure 4 The discharge muffler model

Figure 7 Pressure pulsations in cavity 5 of Figure 3 (4647.5 rpm)

Figure 8 Pressure pulsations in cavity 5 of Figure 3 (6011.7 rpm)