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ABSTRACT

This paper investigated the causes of the noise at around 250Hz due to the discharge pressure pulsation and proposed the methods to reduce the low frequency noise of a reciprocating compressor in a refrigerator. The compression cycle, valve dynamics and piston kinematics equations were first modeled and solved to obtain the mass flow rate. And, then the 1-D wave equation was modeled in order to estimate the pressure pulsation in the discharge line with the mass flow rate calculated. The roles of the noise source and discharge line characteristics for the low frequency noise were totally understood using the simulation model. The simulation results were in good agreement with the experiments and the improved design for the discharge line was proposed. It is shown that the path line between the cylinder head and discharge chamber is a key component to determine the amplitude and frequency of the discharge pulsation.

1. INTRODUCTION

In the operation of the reciprocating compressor, a refrigerant is suctioned into the cylinder and compressed and then discharged into the discharge path as the piston reciprocates. During this process, it cannot continuously discharge the refrigerant and causes a discharge pulsation. The suction/discharge pulsations of the refrigerant result in a vibration and noise of the compressor. In particular, a discharge pulsation creates the low frequency band noise. In order to attenuate a noise generated during the discharge process of a compressor, the volume of the discharge muffler, the length and cross-sectional area of a tube, and others can be considered. Although it is simple to attenuate a discharge pulsation by increasing the volume of the discharge muffler, a limited space of the compressor makes it impossible to increase the volume of the discharge muffler. Pressure pulsations in a discharge line are generated by the reciprocating action of the pistons as the automatic valves open and close. These pressure pulsations are one of the primary sources of the vibration and noise emitted from the compressor. Therefore, the low frequency noise created by the discharge pulsation makes a person uncomfortable. In the paper, mathematical modeling and simulation of the dynamics of compressors are performed so that the effects of various design parameters. The first law of thermodynamics and a simplified 1-DOF equation of the valves (Soedel, 1972, Hamilton, 1974) are used to calculate the pressure in a cylinder and mass flow rates through the valves. Then the pressure pulsation in the discharge line of a reciprocating compressor is predicted using the four pole parameter approach developed by Lai and Soedel(1996). These results are then used to reduce the level of the gas pulsations in the discharge line of a reciprocating compressor.

2. Governing equations

The equations governing the refrigerant flow are briefly described in this section. In order to predict the pressure pulsation in the discharge line of a reciprocating compressor, mathematical models are developed for the re-expansion, suction, compression, and discharge processes and components such as suction/discharge valves. First, the equation based on the first law of thermodynamics was derived to calculate the instantaneous pressure and temperature in the cylinder. Second, the valves were modeled as a simplified 1-DOF and then the mass flow rates through the suction/discharge valves were calculated by the assumption of 1-D compressible flow (Hamilton, 1974). And, then the discharge path after discharge valve is modeled using the four pole parameter approach and the mass flow rate is used as the source to predict the discharge pressure pulsation.

2.1 1St law of thermodynamics

In order to obtain the instantaneous pressure and temperature in the cylinder, the first law of thermodynamics was used. As shown in Equation (1), the thermodynamic characteristics are influenced by the piston kinematics, valve dynamics, and pressure and temperature. The rate of change of temperature in the cylinder during re-expansion, suction, compression, and discharge processes is given by

$$\left[m_{cv}\left(\frac{\partial h_{cv}}{\partial T_{cv}}\right)_v - V_{cv}\left(\frac{\partial P_{cv}}{\partial T_{cv}}\right)_v\right]\frac{\partial T_{cv}}{\partial t} = \left\{\begin{array}{l} -\dfrac{dm_{cv}}{dt}h_{cv} + \dfrac{dQ}{dt} + \dfrac{dm_{in}}{dt}h_{in} - \dfrac{dm_{out}}{dt}h_{out} \\ -\left[m_{cv}\left(\dfrac{\partial h_{cv}}{\partial v_{cv}}\right)_T - V_{cv}\left(\dfrac{\partial P_{cv}}{\partial v_{cv}}\right)_T\right]\left[-\dfrac{V_{cv}}{m_{cv}^2}\dfrac{dm_{cv}}{dt} + \dfrac{1}{m_{cv}}\dfrac{dV_{cv}}{dt}\right] \end{array}\right\} \quad (1)$$

2.2 Equation of state for R600a

Refrigerant in cylinder is R-600a and is assumed to be pure without oil. For this paper, the modified Benedict-Webb-Rubin equation (MBWR) (Younglove and Ely, 1987) is used as follows:

$$P = \left\{\sum_{n=1}^{9} a_n \rho^n + \exp(-\delta^2)\sum_{n=10}^{15} a_n \rho^{2n-17}\right\} \times 100 \quad [\text{kPa}] \quad (2)$$

2.3 Piston kinematics

The volume of the cylinder is determined by the piston kinematics. In order to get the total control volume in the cylinder, the clearance volume has to be added to the cylinder volume.

$$X(t) = r(1-\cos\omega t) + \lambda r\left(1 - \sqrt{1-\left(\frac{\sin\omega t}{\lambda}\right)^2}\right) \quad (3)$$

where r, l, and λ are crank radius, connecting rod length, and l/r, respectively.

2.4 Mass flow rate

Hamilton(1974) derived the mass flow equation for the calculation of the mass flow rate through the valve based on one dimensional compressible flow through an orifice. In this paper, the equation derived by Hamilton is used and then the mass flow rate is later used as a source to calculate the pressure pulsation in the discharge line.

$$\dot{m} = \sqrt{\frac{2\rho\Delta P}{\left(\dfrac{1}{(KA)_1^2} + \dfrac{1}{(KA)_2^2}\right)}} = (KA)_e\sqrt{2\rho\Delta P} \quad (4)$$

2.4 Valve dynamics

The valve is simplified and modeled as 1-DOF mass/spring system. The discharge valve and stoppers are modeled as mass/spring system and piecewise linear springs as shown in Figure 1.
The valve dynamics is assumed to be governed by the following equation of motion.

$$M_{ef}\ddot{x} + C\dot{x} + Kx = F \qquad (5)$$

At the point of contact, $x = x^*$, stiffness of the valve, K_{total}, is a function of displacement and can be expressed as follows:

$$K_{total} = \left\{ \begin{array}{ll} k_1 & w(x^*,t) \le d_1 \\ k_1 + k_2 & d_1 \le w(x^*,t) \le d_2 \\ k_1 + k_2 + k_3 & d_2 \le w(x^*,t) \end{array} \right\} \qquad (6)$$

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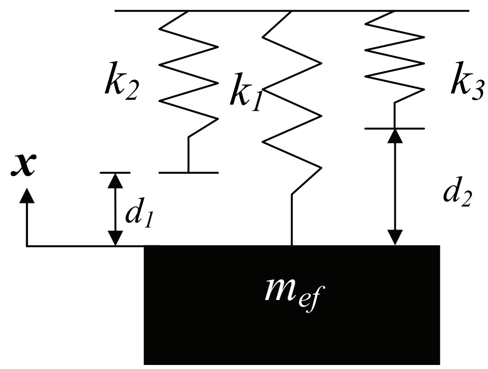

Figure 1. Simplified valve model

2.5 Effective force area

The effective force on valve is estimated by the following equation proposed by Hamilton (1974).

$$F = (P_{up} - P_{down})\left\{ A_2' + \frac{A_1'(KA)_e^2}{(KA)_1^2} - \frac{A_1'(KA)_e^2}{(KA)_2^2} + \frac{A_2'(KA)_e^2}{(KA)_2^2} \right\} \qquad (7)$$

2.6 Acoustic model for the discharge path

The complicated discharge path is simplified as shown in Figure 2 to investigate the discharge pressure pulsations. The four pole parameter approaches and transfer functions are used to calculate the pressure pulsation in the discharge path with the assumption of linear and plane wave theories.

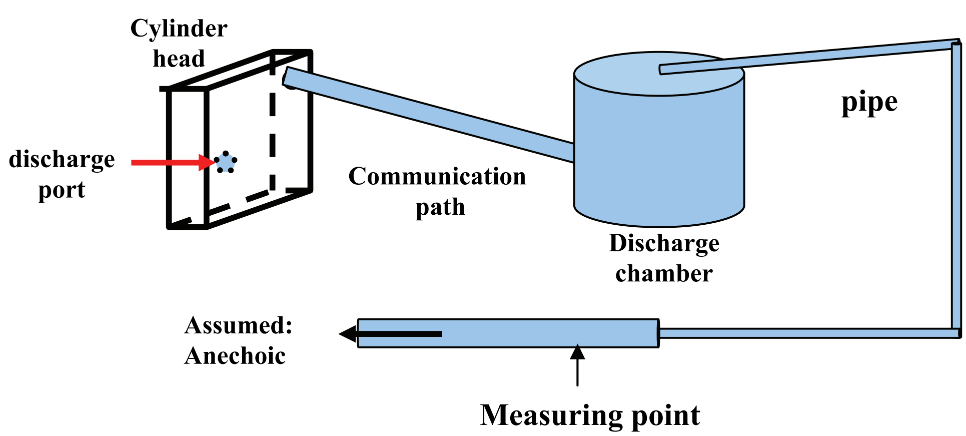

Figure 2. Simplified discharge path

2.6.1 Cylinder head
The complicated cylinder head is simplified as the rectangular box. Because the height of the rectangular box cavity is small enough, the variation of pressure along the height direction is not considered.

$$\frac{\partial^2 p}{\partial x^2}+\frac{\partial^2 p}{\partial y^2}-\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2}=-\frac{\ddot{m}}{h}\delta(x-x^*)\delta(y-y^*) \tag{8}$$

2.6.2 Discharge chamber
The discharge chamber is modeled as the cylinder and the variation of pressure along the radial and tangential directions are not considered.

$$\frac{\partial^2 p}{\partial z^2}-\frac{1}{c^2}\frac{\partial^2 p}{\partial t^2}=-\frac{\dot{m}}{\pi r_c^2}\delta(z-z^*) \tag{9}$$

2.6.3 Pipe system
The pipe and communication path can be described using four pole parameters as follows:

,

$$\begin{bmatrix} Q_1 \\ P_1 \end{bmatrix}=\begin{bmatrix} A & B \\ C & D \end{bmatrix}\begin{bmatrix} Q_2 \\ P_2 \end{bmatrix} \tag{10}$$

where
$$A=\cosh(\gamma' L);\ B=\frac{i\omega\ \ S\sinh(\gamma' L)}{\rho\ \ c_0^2\gamma'} \qquad \gamma'=\xi\cdot\left(\frac{\sqrt{\nu\cdot f}}{c_0 d}\right)+i\frac{\omega}{c_0}$$

$$C=\frac{\rho\ \ c_0^2\sinh(\gamma' L)}{i\omega\ \ S};\ D=\cosh(\gamma' L)$$

Using four pole parameters, a series of connected components as shown in Figure 2, can be described as

$$\begin{bmatrix} Q_{in} \\ P_{in} \end{bmatrix}=\begin{bmatrix} A_{ch} & B_{ch} \\ C_{ch} & D_{ch} \end{bmatrix}\begin{bmatrix} A_{cp} & B_{cp} \\ C_{cp} & D_{cp} \end{bmatrix}\begin{bmatrix} A_{dc} & B_{dc} \\ C_{dc} & D_{dc} \end{bmatrix}\begin{bmatrix} A_{pipe} & B_{pipe} \\ C_{pipe} & D_{pipe} \end{bmatrix}\begin{bmatrix} Q_{out} \\ P_{out} \end{bmatrix} \tag{11}$$

where *A, B, C,* and *D* are the four pole parameters of each component.
Assuming an anechoic condition to the condenser, the impedance of the condenser is

$$Z_{out}=\frac{P_{out}}{Q_{out}}=\frac{\rho c}{S_{out}} \tag{12}$$

where S_{out} is the cross-sectional area of the pipe in Figure 2.
All of equations derived here are solved simultaneously in Matlab (Park *et al*, 2007, 2008).

3. Experiment and simulation results

The pressure pulsation at the measuring point shown in Figure 2 is measured using piezoelectric pressure sensor and then the pressure pulsation is compared with experimental result. Using the simulation model, the parametric studies are performed in order to reduce the discharge pressure pulsation.

3.1 Comparison between the analytical results with the experimental results

In order to verify the simulation model, the simulated result is compared with experimental result as shown in Figure 4. The compressor is operated in ASHRAE condition. The discharge pressure pulsation is measured as a function of time. In order to obtain the analytical result, the calculated mass flow rate shown in Figure 3 is used as a source in the acoustic model. The pressure pulsation from the simulation model is in good agreement with the experimental result as shown in Figure 4.

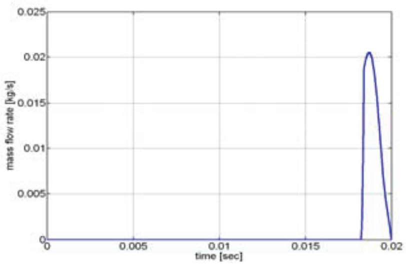

Figure 3. Estimated mass flow rate

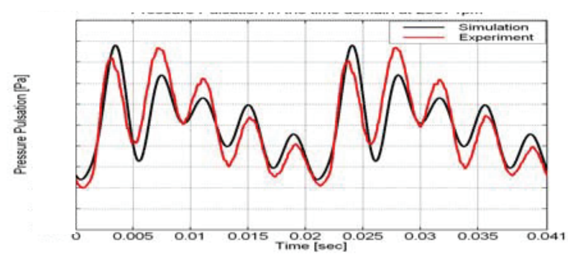

Figure 4. Comparison between experiment and simulation

3.2 Parametric study

Using the simulation model, discharge pressure pulsations are calculated with the changes of the dimensions in the discharge path. It is revealed that the communication path between cylinder head and discharge chamber in Figure 2 is one of the important components to determine the discharge pressure pulsation amplitude and the number of fluctuations in a period. According to the simulation results, the length of communication path should be increased and the diameter of the communication path should be decreased in order to reduce the discharge pressure pulsation. The configuration of the newly designed communication path is shown in Figure 5. The simulation results with the changes of the length and diameter of the communication path are shown in Figure 6. As shown in figure, the amplitude of the pressure pulsation is reduced by about 20% and the number of fluctuations of the pressure pulsation is reduced. Although the figure of the transmission loss for the discharge path is not included in this paper, the reason for the change of the pressure pulsation behavior is that the lower cut-off frequency of transmission loss is closely related to the dimensions of the communication path in the discharge path. As the diameter of communication path is decreased and the length of communication path is increased, the lower cut-off frequency of transmission loss is decreased. Therefore, the number of fluctuations of pressure pulsation is reduced to 1 as shown in Figure 6. Also, it is shown that the simulation results are in good agreement with experimental results.

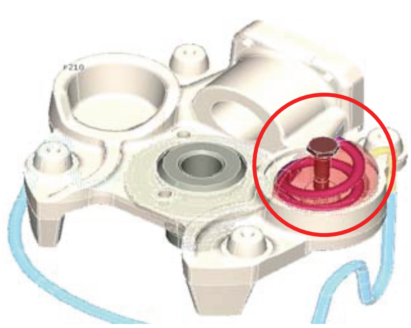

Figure 5. Newly designed communication path

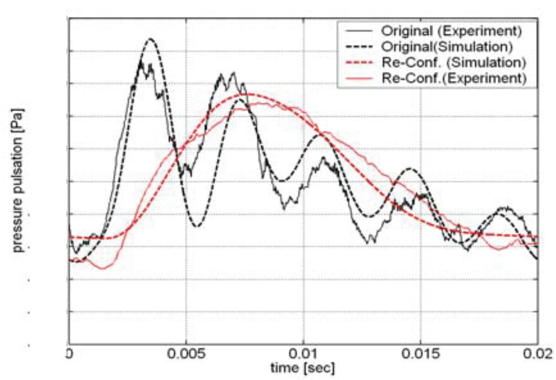

Figure 6. Discharge pressure pulsations with new design

3.3 Reduction of the low frequency noise

In order to investigate the role of pressure pulsations to the noise level, the sound pressure levels are measured as shown in Figure 7. It is shown that the fourth harmonic (around 250Hz) response is substantially reduced when the communication path is newly designed as explained in this paper. From this result, it is verified that the pressure pulsation plays an important role in sound pressure level at the specific frequency range.

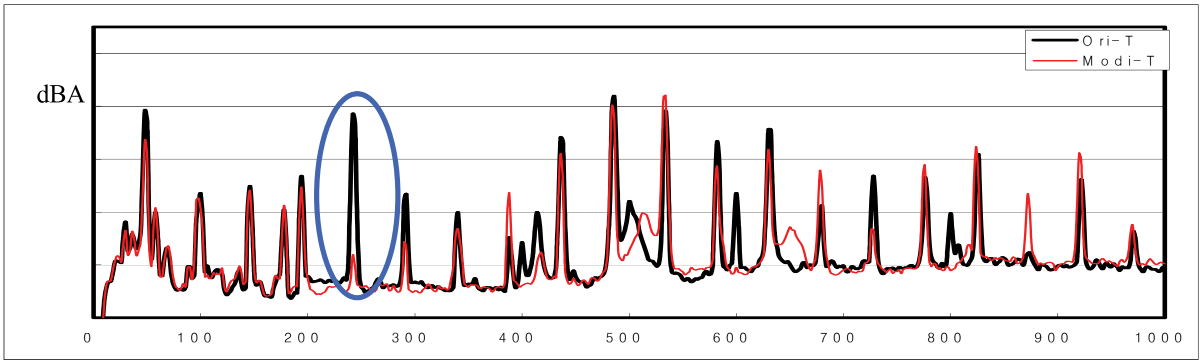

Figure 7. Comparison of sound pressure levels between original and modified discharge paths

4. Conclusions

The first law of thermodynamics and the simple 1-DOF valve equations were used for the compressor compression cycle. Also, the discharge path was modeled using the four pole parameter approach. The calculated mass flow rate is used as the input for the acoustic model. All governing equations were simultaneously solved in Matlab. The analytical results of discharge pressure pulsation were in good agreement with the experimental results.
In order to reduce the discharge pressure pulsation, the parametric studies were performed and then found that the dimensions of the communication path are the most important factors to reduce the lower cut-off frequency of the transmission loss of discharge path. Using the newly designed discharge path, a significant reduction of the discharge pulsation at about 250Hz was achieved.

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