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SELF-EXCITATION MECHANISM OF REED VALVE IN REFRIGERANT COMPRESSORS

by

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ABSTRACT
This study presents a basic idea for the self-excitation mechanism of reed valves in refrigerant compressors used for air conditioners or refrigerators. The detailed experimental data on the phase lag of fluid pressure, which were obtained from a forced vibration test on a simple axisymmetrical model, namely a resonator, composed of the valve, the valve plate, the cylinder and the piston, were presented, first. Secondly, the Helmholtz resonance frequency for the resonator with flow was empirically determined from the experimental data on pressure phase-lag, and therewith the end correction factor in calculating the Helmholtz resonance frequency was calculated backwards, thus figuring out the contraction of flow in the discharge port. Conclusively, the possibility of dynamic behavior of the contraction of flow in the discharge port was suggested, and thus the basic self-excitation mechanism of reed valves was clarified.

INTRODUCTION
From a basic standpoint that the self-excitation mechanism of reed valves in refrigerant compressors used for air conditioners or refrigerators should be, first of all, clearly identified, a forced vibration test on a simple axisymmetrical model, namely a resonator, composed of the valve, the valve plate, the cylinder and the piston was performed in the previous study [1]: the valve was exposed to compressed gas flow and forced to vibrate in the frequency range from 50 to 500 Hz, and the fluid pressure was measured to reveal its frequency characteristics and spatial distribution in amplitude and phase lag relative to the valve vibration; the similar measurement was performed for various cylinder volumes. As a result, first, a dynamic stability criterion for self-excited vibration of the reed valve was presented, where the forced vibration frequency was divided by the Helmholtz resonance frequency to reduce it to a nondimensional form. In the previous study, furthermore, the validity of its dynamic stability criterion was ensured by performing a free vibration test on the reed valve, and in addition, the vibration frequency of reed valve in flow, which is significantly higher than its natural frequency in still air, and the significant works of the fluid force inducing self-excited vibration also, were carefully examined. However, a basic self-excitation mechanism of the reed valve has not been still addressed at all.

Among theoretical studies especially on the vibrations of the reed valve controlling the refrigerant gas flow from the discharge port, there has been a study by Trella [2], in which the mutual interaction between the compressed refrigerant gas flow and the valve vibration is treated: the flow region was divided into 4 sections, and the continuity equation and the momentum equation were applied to each flow section, thus deriving basic
equations of motion. To obtain the solutions, however, a fairly complicated procedure of numerical calculations is needed, since the solutions in 4-divided sections have to be continuously chained. Nevertheless, when his analytical method was applied to derive periodic solutions for our forced vibration tests, the analytical results could not show a good agreement with the experimental results, after all of our efforts. In his analysis, a fixed flow region that the shear layer flow separated from the leading edge at the discharge port never changes in time was basically assumed. It comes into a serious question if such an assumption is descriptive for the real flow oscillating in the discharge port. In order to reveal this question also, the dynamic behavior of the compressed gas flow causing the reed valve to vibrate has to be necessarily examined.

In this study, first, the detailed experimental data on pressure phase-lag of the compressed air flow in the discharge port are presented for various diameter ratios of the discharge port to the cylinder, and it is shown that if the forced vibration frequency is reduced to a nondimensional form based on the Helmholtz resonance frequency for the resonator without any flow, all experimental data cannot be reduced to a universal form: any dynamic similarity cannot be addressed. Then, secondly, the Helmholtz resonance frequency for the resonator with flow is empirically determined from the experimental data on pressure phase-lag, and therewith the end correction factor in calculating the Helmholtz resonance frequency is calculated backwards, and thereby the contraction of flow in the discharge port is figured out. Finally, the possibility for the contraction of flow in the discharge port to change dynamically during vibrations of the reed valve is suggested, and thereby the phase-lag characteristics of the fluid pressure in the discharge port is well understood, and thus the self-excitation mechanism of the reed valve is addressed.

**FORCED VIBRATION MODEL**

A simplified model for examining the fluid pressure change induced by the forced vibration of the reed valve used for air conditioners is shown in Fig. 1. The reed valve is simulated by a column having the same diameter $d_v$ as the reed valve, fixed at 12 mm. The column is directly connected to a magnetic exciter and installed immediately in front of the discharge port having a diameter $d_r$ and a length $W$, fixed at 8.7 mm and 14.5 mm, respectively. The mean opening height of the valve is expressed by $\delta$. The compressed air fed into the cylinder through a small hole (3.4 mm) at the center of the piston blows out through the gap between the valve and the outer surface of the valve plate. The cylinder diameter $D$ and the entire flow path length $L$ were varied at different values, respectively.

Here represent the vibratory displacement toward the discharge port from the mean opening height by the following form:

$$X_e = X_0 \cos 2\pi Ft,$$

where $F$ is the forcing frequency and $X_0$ the amplitude. This forced vibration causes the compressed air pressure $p(X_p; t)$ in the discharge port and cylinder to fluctuate. The air pressure does not necessarily fluctuate without a phase lag, however, caused by an inertial effect of the mass flow of air, for example. The air pressure can thus be expressed in the following form:

$$p(X_p; t) = p_0 \cos (2\pi Ft - \phi),$$

where $\phi$ represents the phase lag relative to the valve vibration defined in Eq.(1). The fluctuating air pressure was measured using a pressure transducer (TOYODA PD-104...
attached to the middle of the discharge port wall. The valve vibration was measured using a noncontact-type vibration transducer (IMV PB-0310).

**HELMHOLTZ RESONANCE FREQUENCY FOR A RESONATOR WITHOUT FLOW**

The acoustic vibration model shown in Fig. 1 is a Helmholtz type resonator itself. As for the resonance frequency for the resonator without any flow, Izumi et al. [3] has presented the following expression giving a precise value of Helmholtz resonance frequency \( F_h \):

\[
F_h = \frac{c}{2\pi} \sqrt{\frac{3}{S_W L_r W_e}} \left[ 1 - \left( \sqrt{1 + S_1 \frac{L_r}{W_e}} + \sqrt{1 + S_2 \frac{L_r}{W_e}} \right) + \sqrt{1 + S_1 \left( \frac{L_r}{W_e} + \frac{W_e}{L_r} \right) + S_2^2} \right],
\]

in which the end correction is taken into consideration, by introducing the effective length of the discharge port, \( W_e \):

\[
W_e = W (1 + \beta_c), \quad \beta_c = \frac{e_c d_r}{2W}.
\]

\( \beta_c \) is the end correction factor for the discharge port length, and \( e_c \) a sub-end correction factor (see Eq. (3) and Fig. 6 in Reference [3]). In Eq. (3), \( c \) is the sound velocity and \( L_r \) the cylinder length. \( S_1 \) is given by the following expression:

\[
S_1 = \frac{12 \pi^2}{9} \xi^2,
\]

as a function of the diameter ratio of the discharge port to the cylinder, \( \xi = d_r / D \).

**PHASE-LAG CHARACTERISTICS OF FLUID PRESSURE IN DISCHARGE PORT**

The mean pressure in the cylinder was fixed at about 4.5 kPa, the mean opening height of the valve, \( \delta \), at about 0.2 mm, and the forced vibration amplitude of the valve, \( X_0 \), at about 25 \( \mu \)m. The forced vibration frequency was varied from 50 to 500 Hz. The fluctuating fluid pressure in the discharge port was measured and the phase lag relative to the valve vibration, \( \phi \), was precisely calculated. The phase lag is very significant to know the energy transfer from the air to the valve.

The obtained results for the phase lag are shown in Fig. 2 in which the cylinder diameter \( D \) changes from 18 mm to 26, 36, 47 and 65 mm. Since the discharge port diameter \( d_r \) is fixed at 8.7 mm, the diameter ratio \( \xi \) varies from 0.483 to 0.134. The abscissa is the ratio of the forced vibration frequency \( F \) to the Helmholtz resonance frequency for the resonator without any flow, \( F_h \), calculated from Eq. (3). Interestingly, it can be seen from each diagram that the phase lag at different values of the entire flow path length \( L \) roughly lies on one curve. However, there exists a problem that the frequency ratio at which the phase lag changes sign is not necessarily 1.0, and as the diameter ratio \( \xi \) decreases, its frequency ratio significantly increases from 1.08 to 1.36.

Since the phase lag turns from \( +80^\circ \) to \( -80^\circ \), by about \( 180^\circ \), as shown in each diagram in Fig. 2, it must be reasonable to consider that any resonance phenomena was related to the present subject. It was first suspected that its resonance phenomena is the Helmholtz type acoustic resonance, and the forced vibration frequency was made in nondimensional form on the basis of the Helmholtz resonance frequency for the resonator without flow, given by Eq. (3). However, the results shown in Fig. 2 suggests that some correction for the calculating method of Helmholtz resonance frequency is needed.

**HELMHOLTZ RESONANCE FREQUENCY FOR THE RESONATOR WITH FLOW AND CONTRACTION OF FLOW IN THE DISCHARGE PORT**

The resonator shown in Fig. 1 has a compressed air flow from the piston toward the valve. Thus, it would be reasonable to suspect that the Helmholtz resonance frequency for the present case does not necessarily coincide with that for the resonator without any flow. Then the nondimensional abscissa based on the Helmholtz resonance frequency for the resonator with flow should have been introduced. However, it seems that there have been few studies which present a method of calculation of the Helmholtz resonance frequency for the resonator with flow.

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Here introduce a rough assumption that the frequency at which the pressure phase-lag takes on a value of zero will coincide with the Helmholtz resonance frequency for the resonator with flow. The frequency data deduced from the experimental data shown in Fig. 2 are presented in Fig. 3, where the abscissa is the entire flow path length \( L \). The curves represent the theoretical values calculated from Eq. (3), where the end correction factor \( \beta_c \) was so adjusted that the frequency data for each diameter ratio \( \xi \) of the discharge port to the cylinder lie on a theoretical curve. Interestingly, it is seen from this figure that the theoretical curves well simulate the empirical frequency data, respectively.

The values of end correction factors \( \beta_c \), deduced backwards from the empirical values of Helmholtz resonance frequency for the resonator with flow, are shown in Fig. 4a, where the abscissa is the diameter ratio \( \xi \). The solid line shows the theoretical values of end correction factor for the resonator without any flow. Interestingly, the deduced values of end correction factor are far smaller than the theoretical curve, and take minus values for the diameter ratio smaller that about 0.4. The minus values significantly mean that as easily seen from Eq. (4), the effective length of the discharge port is smaller than its real length, that is, the volume performing piston motion in the discharge port is smaller than the real volume of the discharge port. It would be reasonable to figure out that such a result was caused by the contraction of flow: the flow separated from the leading edge at the front corner of the discharge port builds up a contraction, thus reducing the effective volume in the discharge port.

The mean value of Reynolds number at the discharge port is about 6400, and thereby it must be natural that the contraction of flow occurs at the discharge port [4]. Thereupon, from the deduced values of end correction factor, furthermore, the coefficient of contraction, \( C_c \) was deduced. The obtained results are shown by a small black circle in Fig. 4b. As the diameter ratio \( \xi \) decreases, the coefficient of contraction, \( C_c \), decreases. Such a tendency well agrees with that for the water flow, shown by the solid line. Moreover, the values of the coefficient of contraction well agree with the measured values from 0.6 to 0.8 for the air flow [4].

**DYNAMIC BEHAVIOR OF CONTRACTION OF FLOW IN THE DISCHARGE PORT AND ITS SIGNIFICANCE TO PRESSURE PHASE-LAG**

The pressure phase-lag data shown in Fig. 2 were re-plotted, as shown in Fig. 5, where the abscissa is the nondimensional frequency based on the Helmholtz resonance frequency for the resonator without any flow.

![Fig. 2 Phase-lag of fluid pressure vs. frequency ratio based on Helmholtz resonance frequency for the resonator without any flow](image-url)
frequencies for the resonator with flow, shown in Fig. 3. It is of great significance in Fig. 5 that when the frequency ratio is smaller than 1.0, the phase lag takes on plus values. The zero phase-lag represents just when the valve closes. Therefore, the plus phase-lag represents that the air pressure becomes large when the valve moves to open. In this case, the air pressure pushes the valve in the same direction the valve moves, thus providing energy to the valve to vibrate. Thus one may imagine that if the valve forced to vibrate is elastically so suspended that its natural vibration frequency becomes to be the same as the forced vibration frequency, naturally, the valve takes in the energy from the air flow, thus inducing the self-excited vibration when the mechanical damping effect is small.

Here it is of great importance to suspect why the air pressure for the frequency ratio smaller than 1.0 becomes large when the valve moves to open. Interestingly, this dynamic phenomena is quite opposite to the static one: when the valve opens statically, naturally, the air pressure decreases. The major cause inducing such dynamic phenomena will be the dynamic behavior of the shear layer separated from the leading edge at the front corner of the discharge port, as shown in Fig. 6. It would be rather unnatural to consider that the shear layer is fixed in space, in spite of the valve vibration. If it is imaged that the shear layer dynamically moves in response to the valve vibration, the phase-lag characteristics of the fluid pressure, shown in Fig. 5, can be basically figured out. When the valve moves to open, as shown by "a", "b" and "c" in Fig. 6, the flow velocity of the compressed air in the discharge port increases and thereby the coefficient of contraction decreases, that is, the shear layer moves toward the center axis, as drawn in Fig. 6. Such movement of the shear layer definitely decelerates the main flow in the discharge port. Its deceleration rate of flow velocity becomes largest when the opening speed of the valve becomes largest, and at its moment shown by "b", naturally, the air pressure becomes largest. It can be explained in this way that when the phase-lag data for low frequency ratio approaches about 90°, as shown in Fig. 5.

Subsequently, try to examine the phase-lag when the frequency ratio is large, which approaches about -90°. The shear layer is of course a mass flow, and thereby, when the frequency of the valve becomes large, the shear layer cannot follow the movement of the valve. In such case that the shear layer cannot move, when the valve moves to open, the major flow in the cylinder is accelerated and thus the fluid pressure decreases. Therefore, the phase lag Φ for high frequency ratio approaches about -90°, as shown in Fig. 5.

**BASIC SELF-EXCITATION MECHANISM OF REED VALVE**

The discussions in the previous section result in the following summary. When the vibration frequency of the reed valve is smaller than the Helmholtz resonance fre-
frequency for the resonator with flow, the fluid pressure in the discharge port supplies energy to the reed valve, and its energy transfer causes the reed valve to induce a self-excited vibration. The basic self-excitation mechanism for lower vibration frequency of the reed valve is addressed as follows: when the valve starts to open, the flow velocity in the discharge port increases; accordingly, the coefficient of contraction of flow decreases; the decreased coefficient of contraction decelerates the gas flow in the discharge port, thus increasing the pressure in the discharge port; the increased fluid pressure pushes the valve in the same direction the valve opens, amplifying the opening motion of the valve. Analogous, but reversed, behavior occurs as the valve moves to close its opening, amplifying the original closing motion.

CONCLUSIONS

In this paper, results of a forced vibration test were presented for the reed valve frequently used in refrigerant compressors were presented, and by introducing an idea of mutual interaction of the reed valve and the shear layer in the discharge port, the basic self-excitation mechanism of the reed valve was well addresses. On the basis of the addressed mechanism, a rather simple theoretical method making it possible to calculate the self-excited vibrations of reed valves in refrigerant compressors, would be established. It should be noted here that this study suggests a significant idea for calculating the Helmholtz resonance frequency for a resonator with flow.

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