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Dynamic Stability Criterion for Reed Valves in Refrigerant Compressors

by

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ABSTRACT

One of the most significant vibration sources of higher-frequency noises in refrigerant compressors used for air conditioners or refrigerators is the reed valve which controls the refrigerant gas flow from the discharge port. In order to examine the higher-frequency vibrations of the reed valve exposed to compressed gas flow, we conducted a vibration test for a simple axisymmetrical model consisting of a valve, valve plate, cylinder and piston. First, compressed air was conducted into the flow chamber through a capillary tube at the piston center, and the valve was excited to induce a forced vibration over as wide a frequency range as possible. Then, the fluid pressure in the flow chamber was measured for different cylinder volumes to determine the amplitude and the phase-lag relative to the valve vibration, thus making it possible to obtain a dynamic stability criterion for self-excited vibration of the reed valve. Secondly, a free vibration test was performed on the reed valve to verify that the reed valve actually experiences a self-excited vibration when the dynamic stability criterion obtained from the forced vibration test is not satisfied. In addition, the vibration frequency and self-excitation level of the reed valve were examined.

1. INTRODUCTION

If the rotatory behavior of the crankshaft in refrigerant compressors used for air conditioners or refrigerators is carefully analyzed with regard to the fundamental dynamics of machinery, it is possible to calculate the fundamental vibrations of the whole compressor, vibrations which are induced by unbalanced inertia forces of the moving elements⁽¹⁾⁻⁽⁹⁾. Such an analysis can reveal vibrations with relatively low frequencies (up to frequencies corresponding to at most 10th order of crankshaft revolutions), but cannot evaluate correctly the higher frequency vibrations such as those causing the noise. It is frequently stated that the major sources of the noise in refrigerant compressors⁽¹⁰⁾ are the electromagnetic vibrations of the motor inducing motor noises⁽¹¹⁾, the elastic vibrations of the reed valve inducing valve noises, the pulsating flow of refrigerant gas inducing gas impulse noises, the slap motion of the piston inducing piston-slap noises⁽¹²⁾, and the elastic vibrations of the crankshaft inducing crankshaft noises⁽¹³⁾⁻⁽¹⁴⁾. There have been many studies especially on the vibrations of the reed valve controlling the refrigerant gas flow from the discharge port. Among them, a study by Trella & Soedel⁽¹⁵⁾ is particularly significant from the viewpoint of clarifying the self-excited vibration mechanism of the reed valve. Trella studied the mutual interaction between the compressed refrigerant gas flow and the valve vibration, and theoretically analyzed the dynamic behavior of the reed valve during discharge. It was assumed that the pressure of compressed gas in the cylinder is uniform throughout the cylinder. Recently Ziada et al.⁽¹⁶⁾ however, reported an experimental study which concluded that the pressure distribution in the cylinder cannot necessarily be assumed to be uniform, and the formation of standing waves should be taken into account. In other words, the vibration frequency of the reed valve is independent of the valve stiffness and determined by the frequency of standing waves. It then follows that a theoretical treatment which considers the

standing waves in the cylinder space is desirable. Such exact treatment is considered fairly difficult and there exists a strong suspicion that it may not be absolutely necessary for calculating reed valve vibrations. Elimination of this suspicion will significantly affect the direction of future research. Such exact treatment is also important for defining the mechanism of self-excited vibration and the dynamic stability criterion of the reed valve.

The reed valve is clearly excited by the fluctuating pressure of discharged gas, induced by the valve vibration itself. If the gas pressure responded with no phase-lag relative to the valve vibration, it is a matter of course that the excitation energy per one vibration cycle, supplied by the discharged gas to the vibrating valve, is zero, thus resulting in no self-excited vibration. However, if the response of discharged gas pressure has some phase-lag relative to the valve vibration, for example, due to an inertia effect of the discharged gas flow or some acoustic interaction between the discharge port and the cylinder, the excitation energy is not necessarily zero, and thus under certain conditions a self-excited vibration will definitely occur. Clearly from these considerations, it is important to determine the characteristics of the gas pressure induced in the flow chamber by the valve vibration. For this purpose, the present study presents forced vibration test results for an axisymmetrical sound-vibration model composed of a valve, valve plate, cylinder and piston. The flow chamber of a rolling-piston type rotary compressor has a complicated shape with regard to the discharge port and the half crescent-shaped cylinder, but in this study the flow chamber was represented by an extremely simple model. The validity of such a treatment will be carefully examined in the present text. Since it is clear that the gas pressure is greatly affected by the vibration frequency of the valve, a forced vibration test was adopted to make it possible to vary the vibration frequency across a wide range. Compressed air was conducted into the flow chamber through a capillary tube at the piston center, and a periodic disturbance was applied to the discharged air flow by forcing the valve to vibrate over as wide a frequency range as possible, from 50 to 500 Hz. The responding air pressures in the flow chamber were measured at various locations along the flow axis to reveal the characteristics of the amplitude and phase-lag relative to the valve vibration. Furthermore, by changing the cylinder volume, the effects of the Helmholtz resonance frequency on the pressure characteristics could be examined, thus making it possible to obtain a dynamic stability criterion for self-excited vibration of the reed valve.

In addition, a free vibration test was performed on the reed valve to verify the dynamic stability criterion obtained from the forced vibration test. Results showed that the reed valve actually undergoes a self-excited vibration when the dynamic stability criterion is not satisfied. Finally, the role of the discharged air pressure in determining the vibration frequency and self-excitation level of the reed valve was examined.

2. FORCED VIBRATION MODEL

A simplified model for examining the forced vibration of the reed valve used for air conditioners is shown in Fig. 1 and its major dimensions are shown in Table 1. The reed valve is simulated by a column having the same diameter d_v as the reed valve has. The column is directly connected to a magnetic exciter and installed just in front of the discharge port having a diameter d_r and a width W . The mean opening height of the valve is expressed by δ . The dimensions d_v , d_r and W , shown in Table 1, represent the dimensions of a real compressor. The vibratory displacement toward the discharge port is given in the following form:

$$X_e = X_0 \cos 2\pi F t \quad (1)$$

where F is the forcing frequency and X_0 is the amplitude.

The upstream side of the discharge port consists of the cylinder and the piston. This is primarily a model of a reciprocating compressor. The cylinder bore D was fixed at a value of 26 mm in the present study. In practice, when the piston moves in recipro-

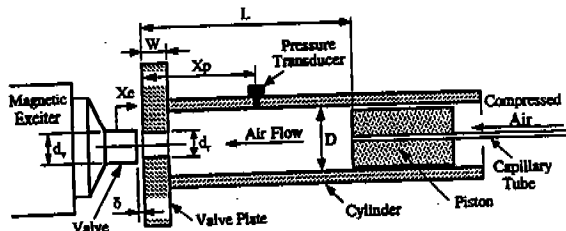


Figure 1. A simplified model to examine the forced vibration of the reed valve

Table 1. Major dimensions of the forced vibration model

d ₁ (mm)	d ₂ (mm)	D (mm)	L (mm)	W (mm)
8.7	12.0	26.0	u p t o 190	14.5

cating compressors, the refrigerant gas in the cylinder is compressed and pushed out through the discharge port. In this case, the volume of the cylinder space changes continuously and hence it is hard to examine the effect of the cylinder volume on the valve vibration. In the present study, therefore, the piston was fixed at different distances from the discharge port, up to 175.5 mm, and compressed air was fed into the cylinder through a small hole at the center of the piston. The bore of the hole was 3.4 mm, far smaller than the cylinder bore.

The compressed air blows out through the gap between the valve and the outer surface of the valve plate. If the valve vibrates at this time, the discharge rate of the air alternately increases and decreases. This 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 inertia effect of the mass flow of air, for example. The air pressure can thus be expressed in the following form:

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

where ϕ 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 S0.1F-1340) attached to the wall of the cylinder. The location of the pressure transducer is given by the horizontal distance from the outer surface of the valve plate, X_p . In addition, the valve vibration was measured using a non-contact type vibration transducer (IMV PB-0310).

3. EXPERIMENTAL RESULTS OF THE FORCED VIBRATION TEST

3.1 Fluid Pressure Acting on the Valve

The valve undergoing forced vibration of the form given by Eq.(1) induces the fluid pressure given by Eq.(2). The fluid pressure acts on the valve. The energy per one vibration cycle, E , supplied by the fluid to the valve forced to vibrate can be calculated as follows:

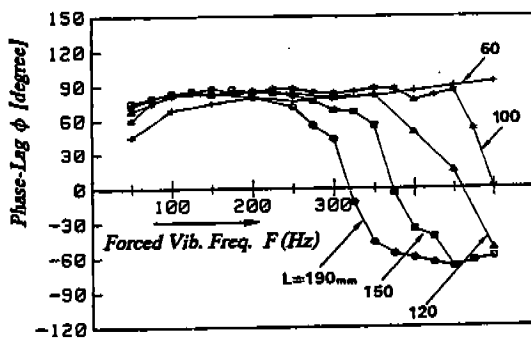
$$E = \int_{1 \text{ cycle}} -A p(0; t) dX_e$$

$$= \pi A P_0 X_0 \sin \phi. \quad (3)$$

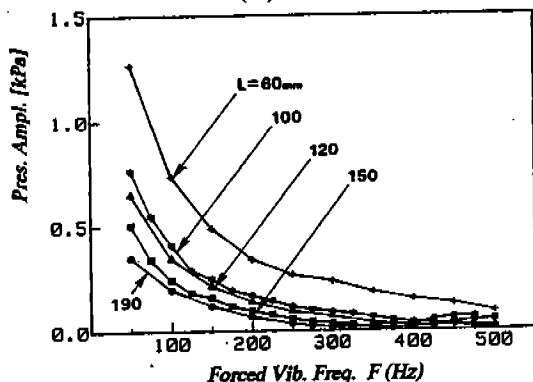
If this energy E is positive, it may be said that the reed valve undergoing free vibration at the same frequency as that at which it was forced to vibrate can lead to self-excited vibration. If the energy consumption due to mechanical damping is larger than the energy E , the reed valve naturally cannot induce a self-excited vibration even if the energy E is positive. If the mechanical damping effect is assumed to be negligibly small, however, one may conclude that the reed valve induces a self-excited vibration corresponding to the positive values of the energy E . Consequently, if the energy E is carefully examined by the forced vibration test in which the forcing frequency is varied over a wide range, a dynamic stability criterion for the reed valve with no damping could be obtained. From this viewpoint, it is important to calculate the energy supplied by the fluid to the valve forced to vibrate. As known from Eq.(3), moreover, the sign of the energy E is definitely determined by the phase-lag ϕ . Therefore, it is clearly important to reveal the characteristics of the phase-lag ϕ .

In the forced vibration test, first, the pressure was measured at $X_p = 5$ mm in the discharge port nearest the valve, whereby the forcing frequency F was varied over as wide a range as possible, from 50 to 500 Hz, and the distance from the outer surface of the valve plate to the piston, namely, the flow chamber length, L , was varied from 60 to 190 mm. The mean pressure in the cylinder was set at 4.5 kPa, the valve vibration amplitude X_0 at 25 μ m and the mean valve opening height δ at 85 μ m. The phase-lag ϕ calculated from the measured pressures is shown in Fig.2a, whereby the variable is the length, L . The phase-lag ϕ takes on a positive constant value of about 80° for any value of L when the forcing frequency F is comparatively small, but ϕ gradually decreases as F increases and approaches a negative value of about -80°. As shown in Fig.2b, the pressure amplitude decreases as F increases. Moreover, as the length L decreases, the pressure amplitude takes on larger values in general.

It is worthwhile emphasizing that the phase-lag ϕ takes on the values of plus and minus depending on the forcing frequency F , and the forcing frequency at which the phase-lag ϕ becomes zero increases as the flow chamber length L decreases. This prop-



(a)



(b)

Figure 2. Fluid pressure measured at $X_0 = 5$ mm (discharge port) at a mean pressure in the cylinder of 4.5 kPa, mean height of the valve opening, $\delta = 85 \mu\text{m}$, and vibration amplitude of the valve, $X_0 = 25 \mu\text{m}$: (a) Phase-lag relative to the valve vibration toward the discharge port; (b) Amplitude as a function of forced vibration frequency.

Table 2. Helmholtz resonance frequency F_h

L (mm)	F_h (Hz)
60	603
100	467
120	426
150	381
190	339

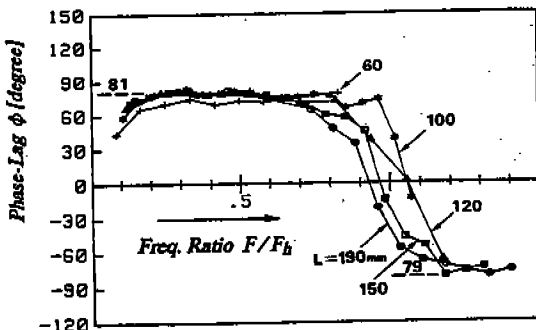


Figure 3. Phase-lag data plotted over the frequency ratio of forcing frequency F to Helmholtz resonance frequency F_h .

erty of the phase-lag suggests that it may depend on the Helmholtz resonance frequency, F_h . The Helmholtz resonance frequency was calculated for each value of the length L , as shown in Table 2. Here the values along the abscissa F of Fig. 2a were divided by the corresponding value of F_h and the phase-lag ϕ was plotted again in Fig. 3 using this nondimensional abscissa F/F_h . Interestingly, it can be seen from this figure that the phase-lag at different values of the length L roughly lies on one curve, and near the frequency ratio $F/F_h = 1.0$, the phase-lag changes sign.

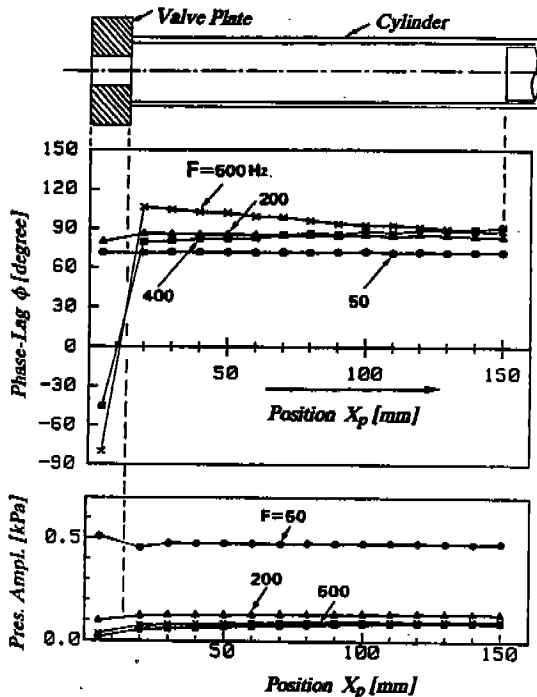


Figure 4. Phase-lag and amplitude of fluid pressure along the flow path.

3.2 Fluid Pressure Distribution along the Flow Path

The second parameter measured in the forced vibration test was the fluid pressure distribution along the flow path for the flow chamber length $L = 150$ mm. The calculated data of the phase-lag ϕ and the amplitude are plotted in Fig.4, in which the variable is the forcing frequency F . The Helmholtz resonance frequency F_h is 381 Hz (see Table 2). When the forcing frequency F is smaller than this resonance frequency F_h , that is when $F = 50$ and 200 Hz, the phase-lag ϕ shows a constant and positive value over the entire flow path: $\phi = +70^\circ$ for $F = 50$ Hz and $\phi = +80^\circ$ for $F = 200$ Hz. However, when the forcing frequency F is larger than the resonance frequency F_h , that is when $F = 400$ and 500 Hz, the phase-lag at $X_p = 5$ mm in the discharge port takes on a negative value, while the phase-lag in the cylinder is positive and roughly constant over its entire length. The amplitude of fluid pressure is uniform along the entire flow path for any value of the forcing frequency.

3.3 Study of Fluid Dynamic Behavior and Fundamental Conditions for Self-Excited Vibration

In Fig.4, the axial length of the cylinder volume, $L-W$, was set at 135.5 mm. Therefore, even if it is assumed that a lowest-order standing wave (1/4-wave length) is formed in the cylinder, its frequency is about 633 Hz. This frequency is higher than the maximum forcing frequency in the present forced vibration test (500 Hz). As seen from Fig.4, moreover, the amplitude of the pressure in the cylinder hardly changes along the flow path in the flow chamber. From these experimental results, one may conclude that no standing waves are formed in the cylinder in the present forced vibration test.

As seen from Fig. 3, moreover, the phase-lag characteristics of the pressure in the discharge port are closely related to the Helmholtz resonance frequency. Thus one may conclude that the fluid in the flow chamber undergoes a volumetric Helmholtz resonator type vibration. Furthermore, Figs.2 and 3 suggest that when the vibration frequency of the valve is smaller than the Helmholtz resonance frequency, the phase-lag of the fluid pressure is uniform across the entire region in the flow chamber and is about 70° to 80° -lag from the instant that the valve opening is smallest. When the vibration frequency of the valve is larger than the Helmholtz resonance frequency, the fluid pressure in the discharge port takes on a phase-lag of about -80° , which is opposite in sign to the phase-lag in the cylinder ($+80^\circ$). These results suggest the following dynamic stability crite-

tion for self-excited vibration of reed valves: if the vibration frequency of the reed valve is smaller than the Helmholtz resonance frequency, the reed valve never experiences self-excited vibration.

4. FREE VIBRATION TEST FOR REED VALVES

A free vibration test for the reed valve was carried out to confirm the above-mentioned criterion. Instead of the magnetic exciter in the forced vibration model shown in Fig. 1, a reed valve with a thickness of 0.381 mm, shown in Fig. 5, was mounted on the valve plate. The left-hand side of the reed valve has the same diameter (12 mm) as the valve used for the forced vibration test. The right-hand side was mounted on the valve plate. The natural vibration frequency of the reed valve was 200 Hz. The mean pressure in the cylinder was set at the same value (9.8 kPa) as that in the forced vibration test, and the vibratory behavior of the reed valve was observed. Initially the reed valve was constrained by a holder so as not to vibrate. The transient vibration of the reed valve,

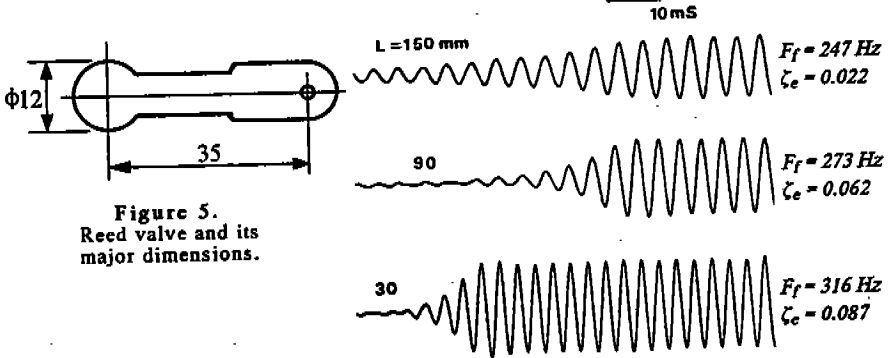


Figure 5. Reed valve and its major dimensions.

Figure 6. Measured transient vibration waveforms for $L = 150, 90$ and 30 mm.

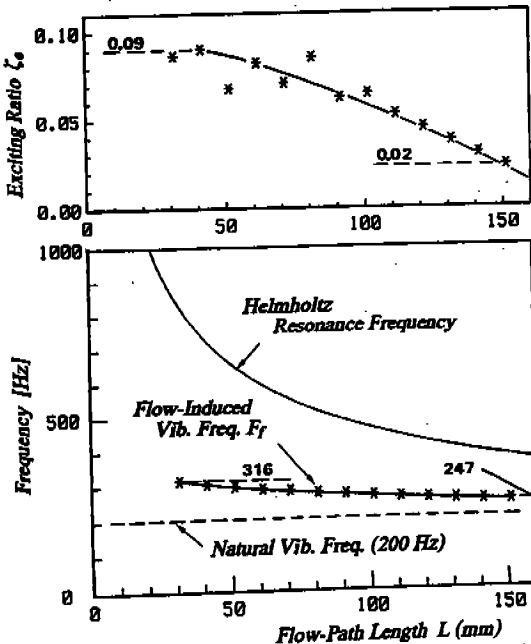


Figure 7. Vibration frequency and excitation ratio of the reed valve (data obtained from a free vibration test).

which occurs after the holder is released, was measured using a non-contact type displacement transducer, so as not to disturb the valve.

The shape of the reed valve vibration waveform changes with the flow chamber length L . Several examples are shown in Fig. 6. It is interesting to note that the vibration frequency of the reed valve is larger than the natural frequency of 200 Hz for any value of the length L . Furthermore, as the length L decreases from 150 to 90 and then to 30 mm, the vibration frequency of the reed valve increases and the excitation ratio (namely the negative damping ratio) of the self-excited vibration, ζ_e , also increases.

The flow chamber length L was changed from 150 mm to 30 mm at 10 mm intervals, and the free vibrations of the reed valve were measured at each value of the length L . Using the measured analog data for free vibrations, the vibration frequency F_f and the excitation ratio ζ_e were calculated and the results were plotted in Fig. 7. As the flow chamber length L decreases, the vibration frequency F_f increases from 247 to 316 Hz, and the excitation ratio ζ_e also increases from 0.02 to 0.09. Here it is interesting to note that:

- (1) The vibration frequency of the reed valve is far lower than the lowest-order standing wave frequency (633 Hz);
- (2) The vibration frequency of the reed valve is lower than the Helmholtz resonance frequency; and
- (3) this means that the dynamic stability criterion suggested by the forced vibration test is never satisfied.

In most cases of vibration of a bluff body in the fluid flow, the vibration frequency becomes lower than the natural frequency naturally due to the effect of fluid added mass. Examples of this are the flow-induced vibrations of hydraulic gates (radial gate and long-span gate)⁽¹⁷⁾⁻⁽³⁰⁾. As shown in Fig. 7, however, the self-excited vibration frequency of the reed valve becomes higher than the natural frequency (200 Hz). For $L=30$ mm, the vibration frequency (316 Hz) is about 1.58 times higher than the natural frequency. This can be explained as follows: the fluid pressure acting on the reed valve, $p(0;t)$, given by Eq. (3), can be reduced to the following form:

$$p(0;t) = p_{0X} \cos \phi \cdot X_e - \frac{p_{0X}}{2\pi F} \sin \phi \cdot \dot{X}_e, \quad (4)$$

where p_{0X} ($= p_0/X_0$) represents the pressure amplitude per unit vibration amplitude of the valve. The first term on the right-hand side is proportional to the vibratory displacement of the reed valve and the second term is proportional to the velocity. A simple vibration model shown in Fig. 8 can be used for clarification, where the equivalent mass of the reed valve is represented by m_v , the equivalent damping coefficient by C and the equivalent spring coefficient by K . The vibratory displacement of the reed valve toward the discharge port is represented by X_e . The fluid force given by $Ap(0;t)$ acts against the following form:

$$m_v \ddot{X}_e + (C - \frac{p_{0X}}{2\pi F} \sin \phi) \dot{X}_e + (K + p_{0X} \cos \phi) X_e = 0. \quad (5)$$

One can see that the displacement term of the fluid pressure plays a role in increasing the valve stiffness, since $\cos \phi$ always takes on a positive value. Secondly, the velocity term plays a role in increasing the negative resistance for $\phi > 0$ or the positive resistance for $\phi < 0$. The function of the fluid pressure in increasing the valve stiffness may be termed a fluid spring effect. Due to this effect of fluid pressure, the vibration of the reed valve, coupled with the discharge flow, has a frequency naturally higher than its natural frequency. We know from Fig. 2a that the phase-lag ϕ takes on a value of about $+80^\circ$ for the frequency range shown in Fig. 7. Figure 2b showed, however, that the pressure amplitude increases as the length L decreases. Therefore, as the length L decreases, the fluid spring constant given by $p_{0X} \cos \phi$ increases, thus resulting in a higher valve vibration frequency.

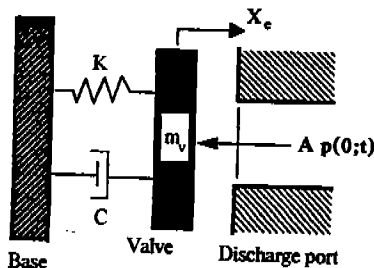


Figure 8. Simple vibration model representing reed valve vibration.

5. CONCLUSION

In this paper, results of a forced vibration test were presented for the reed valve frequently used in refrigerant compressors. A dynamic stability criterion for self-excited vibration of the reed valve was derived in the form that depends upon the magnitude of the ratio of the vibration frequency of the reed valve to the Helmholtz resonance frequency. Moreover, this dynamic stability criterion was confirmed by a free vibration test for the reed valve.

The present tests were conducted on a simple model similar to the flow chamber of reciprocating compressors. Since the fluctuating pressure in the flow chamber was definitely caused by a volumetric vibration of gas, however, one may conclude that the results obtained herein are basically applicable to the rolling-piston type rotary compressors, which also possess a compression chamber with a fairly complicated shape.

In this study, air at room temperature was used as the working fluid, and the mean cylinder pressure was relatively low. To extend these results to operating compressors, forced and free vibration tests for the reed valve need to be examined for highly compressed and heated refrigerant gas.

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REFERENCES

- (1) Imaichi, K., Ishii, N., Saito, S. & Imasu, K., Leakage Effects on Indicator Diagram at Stopping of Reciprocating Compressors, Proc. Purdue Compressor Technology Conference, July, 1978, pp.283-288.
- (2) Imaichi, K., Ishii, N., Imasu, K. & Murai, M., A Device for Stopping Single-Cylinder Reciprocating Compressors silently by Greatly Reducing Vibrations, Proc. 15th Int. Congress of Refrigeration, IIR, Sept., 1979, pp.727-733.
- (3) Imaichi, K., Ishii, N., Imasu, K., Muramatsu, S. & Fukushima, M., Higher Frequency Vibratory Components Caused by Elastic Vibrations of the Crank Shaft in Reciprocating Compressors, Proc. Purdue Int. Compressor Engineering Conference, July, 1980, pp.90-96.
- (4) Imaichi, K., Fukushima, M., Muramatsu, S. & Ishii, N., "Vibration Analysis of Rotary Compressors, Proc. Purdue Compressor Technology Conference, July, 1982, pp.275-282.
- (5) Ishii, N., Imaichi, K., Muramatsu, S., Fukushima, M. & Matsunaga, H., The Study of Rolling-Piston Rotary Compressor Dynamic Behavior When Stopping to Reduce Noise and Vibration Level, Proc. Purdue Int. Compressor Engineering Conference, July, 1984, pp.259-266.
- (6) Ishii, N., Fukushima, M., Sano, K. & Sawai, K., A Study on Dynamic Behavior of a Scroll Compressor, Proc. Purdue Int. Compressor Engineering Conference, Vol.3, Aug., 1986, pp.901-916.
- (7) Ishii, N., Yamamura, M., Morokoshi, H., Muramatsu, S., Sawai, K., Osaka, M. & Yamamuka, M., Mechanical Efficiency of Various Large Capacity Scroll Compressors, Proc. 17th Int. Congress of Refrigeration, Wien, Aug., 1987, pp.468-474.
- (8) Ishii, N., Fukushima, M., Sawai, K., Sano, K. & Imaichi, K., Dynamic Behavior of a Scroll Compressor (Dynamic Analysis), JSME International Journal, Series 3, Vol.31, No.1, March, 1988, pp.58-67.
- (9) Ishii, N., Fukushima, M., Yamamura, M., Muramatsu, S., Sano, K. & Sakai, M., The High Mechanical Efficiency of Rolling-Piston Rotary Compressors, Proc. 2nd World Congress on Heating, Ventilating, Refrigerating and Air Conditioning (CLIMA2000) at Sarajevo, Yugoslavia, 1989, p.91.
- (10) Neubauer, E. T., High Speed Two-Pole Hermetic Compressor, Proc. of ASHRAE Semiannual Meeting, Feb. 11-14, 1963, New York.
- (11) Morill, W. J., Harmonic Theory of Noise in Induction Motors, Transactions of AIEE, 1940, p.189.
- (12) Ungar, E. E. & Ross, D., Vibrations and Noise due to Piston-Slap in Reciprocating Machinery, J. Sound Vibration, 2-2, 1965, pp.132-146.
- (13) Ishii, N., Imaichi, K., Muramatsu, S. & Fukushima, M., A Computer Simulation of Higher Frequency Vibrations of a Reciprocating Compressor, Proc. 16th Int. Congress of Refrigeration, IIR, Paris, Sept., 1983, pp. 418-423.
- (14) Imaichi, K., Ishii, N., Imasu, K., Muramatsu, S. & Fukushima, M., A Vibration Source in Refrigerant Compressors, Trans. ASME: J. Vibration, Acoustics, Stress and

- Reliability in Design, Vol.106, 1984, pp.122-128.
- (15) Trella, T. J. & Soedel, W., Effect of Valve Port Gas Inertia on Valve Dynamics (Part I: Simulation of a Poppet Valve), Proc. Purdue Compressor Technology Conference, 1974, pp.190-207.
- (16) Ziada, S., Shine, S. J. & Buhlmann, E. T., Self-Excited Vibrations of Reciprocating Compressor Plate Valves, Proc. Purdue Int. Compressor Engineering Conference, Vol.3, Aug., 1986, pp.398-414.
- (17) Imaichi, K. & Ishii, N., Instability of an Idealized Tainter-Gate System without Damping Caused by Surface Waves on the Backwater of Dam, Bulletin of the Japan Society of Mechanical Engineers, 20-146, 1977, pp.963-970.
- (18) Ishii, N. & Imaichi, K., Water Waves and Structural Loads Caused by a Periodic Change of Discharge from a Sluice-Gate. Bulletin of the Japan Society of Mechanical Engineers, 20-146, 1977, pp.998-1007.
- (19) Ishii, N., Imaichi, K. & Hirose, A., Instability of elastically suspended Tainter-gate system caused by surface waves on the upstream channel of a dam, ASME Journal of Fluids Engineering, 99-4, 1977, pp.699-708.
- (20) Ishii, N., Imaichi, K. & Hirose, A., Dynamic Instability of Tainter-Gates. In Practical Experiences with Flow-Induced Vibrations (eds Naudascher, E. & Rockwell, D.), 1980, pp.452-460. Berlin: Springer-Verlag.
- (21) Ishii, N., Imaichi, K. & Yamasaki, M., Flow-Induced Structural Vibration of Single-Arm Tainter Gates. In Proc. 20th IAHR Congress (Moscow), 1983, pp.317-324.
- (22) Ishii, N. & Naudascher, E., Field Tests on Natural Vibration Modes of a Tainter-Gate. In Channels & Channel Control Structures (ed. Smith, K. V. H.), 1984, pp.209-222, Berlin: Springer-Verlag.
- (23) Ishii, N., Naudascher, E. & Thang, N., A Flow Induced Vibration of Long Span Gates. In Proc. Int. Conf. on Flow-Induced Vibrations (Bowness-on-Windermere), 1987, pp.293-303. London: British Hydraulic Research Association.
- (24) Ishii, N. & Knisely, C.W., Shear Layer Dynamics and Unsteady Discharge beneath a Long-span Gate undergoing Streamwise Vibrations. In Computational Modelling and Experimental Methods in Hydraulics (HYDROCOMP'89 in Dubrovnik, Yugoslavia) (eds C. Maksimovic & M. Radojkovic), 1989, pp.500-509. Amsterdam: Elsevier Applied Science.
- (25) Ishii, N., Flow-Induced Vibration of Long-Span Gates (Verification of Added Mass and Fluid Damping), JSME International Journal, Ser.II (Fluid Engineering, Heat Transfer, Power, Combustion, and Thermophysical Properties), 33-4, 1990, pp.642-648.
- (26) Ishii, N. & Knisely, C.W., Shear Layer Behavior under a Streamwise Vibrating Gate and its Significance to Gate Vibration. JSME International Journal, Ser. III (Vibration, Control Engineering, and Engineering for Industry), 33-2, 1990, pp.131-138.
- (27) Ishii, N., Knisely, C.W. & Sanagi, T. to be appeared in 1991 Flow-Induced Vibration of Long-Span Gates Caused by Shedding Vortices (Vibration Criteria, Level of Fluid Excitation and Added Mass). JSME International Journal, Ser. III (Vibration, Control Engineering, and Engineering for Industry), to appear in March, 1992.
- (28) Ishii, N. and Naudascher, E., A Design Criterion for Dynamic Stability of Tainter Gates, Journal of Fluids and Structures, 6, 1992, pp.67-84.
- (29) Ishii, N., Flow-Induced Vibration of Long-Span Gates (Part I: Model Development), Journal of Fluids and Structures, to appear in August, 1992.
- (30) Ishii, N. & Knisely, C.W., Flow-Induced Vibration of Shell-Type Long-Span Gates, Journal of Fluids and Structures, to appear in June, 1992.

APPENDIX: NOMENCLATURE

A	valve area exposed to pressure [m ²]
C	equivalent damping coefficient [Ns/m]
d _r	bore of discharge port [m]
d _v	diameter of valve [m]
D	cylinder bore [m]
E	energy per one vibration cycle, supplied from fluid to valve [Nm]
F	forcing frequency [Hz]
F _h	Helmholtz resonance frequency [Hz]
F _f	frequency of flow-induced vibration [Hz]
K	equivalent spring coefficient [N/m]
L	total length of flow chamber [m]
m _v	equivalent mass of valve [kg]
p	air pressure [Pa]
p	pressure amplitude [Pa]
p ₀	pressure amplitude per unit vibration amplitude [Pa/m]
p _{0x}	time [s]
t	thickness of valve plate [m]
W	vibratory displacement of valve, toward discharge port [m]
X _o	vibration amplitude of valve [m]
X ₀	distance from outer surface of valve plate [m]
X _p	mean height of valve opening [m]
δ	excitation ratio
ζ _e	phase-lag of fluid pressure, relative to valve vibration [°]
φ	