1988

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ABSTRACT

This paper has pointed out that there is an high-frequency excitation for the gas flow in the reciprocating compressor due to the vibration of the valve plate. This high-frequency excitation may not be neglected, so a revisionary method for the excitation function $U_p(t)$ in the reference(1)has been proposed. The computed waveshapes of the pulsating pressure seems to agree with the experiments fairly well. The problem of the relation between the working state of valve and the gas pulsation in plenum chamber has been solved.

INTRODUCTION

It is well-known that the gas pulsation in plenum chamber and piping of the reciprocating compressor is caused by the intermitent suction or discharge of the valve. It is clear that the working state of the valve has an effect on the characteristics of gas pulsation. Many researchers have studied the gas pulsation on the basis of the perturbation theory and consider d that the excitative function $U_p(t)$ is proportional to the piston speed.

One of the authors used the formula of excitative function $U_p(t)$ proposed by reference(1)to simulate the gas pulsation, when the working processes were studied in a reciprocating compressor. The computed results indicated that there was an obvious difference in the waveshape of pulsating pressure between the practical measurement and theoretical solution based on the previous excitative function $U_p(t)$, when the vibration occurred in the valve plate(2). As shown in Fig.1, the difference may be relevant with that the high-frequency excitation due to the vibration of the valve plate was neglected. If the motion of the valve plate was normal, the computed results may approximately describe the pulsating pressure, as shown in Fig.2. Therefore, the vibration of the valve plate was supposed to be an additional excitation for the gas flow in the plenum chamber. It should be mentioned that the influence of the vibration of the valve plate is not important in the discharge chamber due to the short open period of the discharge valve. In this work, the calculation of the pulsating pressure in a plenum chamber is taken as an example to show the influence of the valve on the gas pulsation.

ESTABLISHING MODELS

In general, a combined system of the cylinder and the suction or discharge pipe in a reciprocating compressor is shown in Fig.3. A physical model shown in Fig.4 is used to analyse the pulsating pressure in the plenum chamber. In this figure, the point 1 is the joint position connected with atmosphere or large container, and the point 4 is the extreme of the valve plate connected with the cylinder. ℓ_1 is the length of the pipe, and ℓ_2 is the equivalent length of the valve passage. V_3 is the volume of the plenum chamber.

The mathematical model established in this work is based on the perturbation theory. The relation between the pulsating pressure p^* and pulsating mass flow ξ^* at the point 1 and 4 can be described by the following transfer matrix formulas,

$$\begin{bmatrix} p_4^* \\ \xi_4^* \end{bmatrix} = M_{4-3} \cdot M_{3-2} \cdot M_{2-1} \begin{bmatrix} p_1^* \\ \xi_1^* \end{bmatrix} \qquad (1)$$

or rewritten as

$$\begin{bmatrix} p_4^* \\ \xi_4^* \end{bmatrix} = M_{4-1} \begin{bmatrix} p_1^* \\ \xi_1^* \end{bmatrix} \quad (2)$$

where M_{4-1} – the transfer matrix between the pulsating parameters at the point 1 and 4.

$$M_{4-1} = M_{4-3} \cdot M_{3-2} \cdot M_{2-1} = \begin{bmatrix} M_{11} & M_{12} \\ M_{21} & M_{22} \end{bmatrix} \quad (3)$$

M_{4-3}, M_{2-1} – the transfer matrix of the equi-section pipe ℓ_1 and ℓ_2, as expressed by

$$M = \begin{bmatrix} ch(\alpha + jk)\ell & -\frac{a}{F} sh(\alpha + jk)\ell \\ -\frac{F}{a} sh(\alpha + jk)\ell & ch(\alpha + jk)\ell \end{bmatrix} \quad (4)$$

M_{3-2} – the transfer matrix of the plenum chamber, as expressed by

$$M = \begin{bmatrix} 1 & 0 \\ \frac{V_s \omega}{a^2} j & 1 \end{bmatrix} \quad (5)$$

where α – damping factor;
a – speed of sound;
ω – excitative frequency,
k – wave number, and $k = \omega / a$;
F – flow area of the pipe.

The numberical values of the matrix elements such as M_{11}, M_{12}, M_{21}, and M_{22} in the formula (3) can solved, if the geometric parameters of the plenum volume and pipe, the thermodynamic parameters, and the moving state are known.

As shown in Fig.4, the point 1 is the opening end in acoustics. In this case, $p_1^* = 0$. The point 4 is the excitative end, and the excitative mass flow can be expressed by the following formula,

$$\xi_4^* = \rho F U_p^*(t) \quad (6)$$

where ρ – mass density of the gas at the point 4;
F – equivalent flow area of the valve;
$U_p^*(t)$ – excitative speed.

The other two parameters, ξ_1^*, and p_4^* at the both ends can be obtained from the formulas (2) and (3), that is,

$$\xi_1^* = (\xi_4^* - M_{21} p_1^*) / M_{22}$$
$$p_4^* = M_{11} p_1^* + M_{12} \xi_1^* \quad (7)$$

According to the transfer matrix formula (4) of the equi-section pipes, the pulsating pressure p_3^* at the point 3 can be expressed as

$$p_3^* = p_4^* ch(\alpha + jk)\ell_2 + \frac{a}{F} sh(\alpha + jk)\ell_2 \quad (8)$$

As the dimensions in all direction of the volume V_s are far shorter than the wavelength, the pulsating pressure was supposed to be the same in all position of the volume. In that way the pulsating pressure p_2^* at the point 2 of the inlet of the plenum chamber is equal to p_3^*, that is, $p_2^* = p_3^*$. If the high-frequency resonance of gas column for the system shown in Fig.4 occurs, the method of single harmonic resonant amplitude can be used to correct the calculation.

DESCRIPTION OF THE EXCITATION

The connection between the gas in the cylinder of the reciprocating compressor and the gas flow in the pipe is presented only when the valve plate is opened. As the gas pulsation is caused by the intermitent suction or discharge of the valve, it is clear that the periodic velocity change of the gas flow at the side of the valve should be the excitation for the gas flow in the pipe.

For the normal working condition of the valve, the excitative function can be described as the follows: The opening or closure of the valve is completed in a twinkling, and the velocity of the gas flow at the side of the valve is proportional to the piston speed. An air compressor type 2ZA-1.5/8 with vertical double-cylinder single-action was used in the experiments and calculation in this work. The excitative function $U_p(t)$ at the point 4 of the suction valve may be expressed as follows,

$$U_p(t) = \begin{cases} 0 & 0 < \theta < \theta_s \\ br\omega(\sin\theta + \frac{\bar{\lambda}}{2}\sin 2\theta) & \theta_s \leq \theta \leq \pi \end{cases} \tag{9}$$

where θ - rotative angle of the crankshaft;
θ_s - open angle of the valve plate;
r - rotative radius of the crankshaft;
b - proportional coefficient, that is, the ratio of the piston area to the equivalent flow area of the valve;
$\bar{\lambda}$ - ratio of the rotative radius of the crankshaft to the length of the connecting rod.

The excitative speed $U_p(t)$ is the function with the period of . The curve of the excitative speed versus the rotative angle of the crankshaft is indicated in Fig.5. When the pulsating response is calculated on the basis of the perturbation theory, an harmonic function is regarded as the excitation. In this case, the excitative function $U_p(t)$ should be analysed by Fourier method, then the harmonic values $U_p^*(t)$ at each step are obtained. The $U_p(t)$ is expanded to the Fourier's series,

$$U_p(t) = \frac{a_0}{2} + \sum_{n=1}^{\infty} (a_n \cos n\theta + b_n \sin n\theta) \tag{10}$$

$$a_n = \frac{1}{\pi}\int_0^{\pi} U_p(t)\cos n\theta \, d\theta \qquad (n = 0, 1, 2, \ldots\ldots)$$

$$b_n = \frac{1}{\pi}\int_0^{\pi} U_p(t)\sin n\theta \, d\theta \qquad (n = 1, 2, 3, \ldots\ldots)$$

The harmonic values $U_p^*(t)$ at each step calculated from the formula (10) is substituted into the formula (6), then the pulsating pressure $p_3^*(t)$ at the point 3 can be obtained from the formula (7) and (8).

When the vibration occurred in the valve plate, as shown in Fig.1, the high-frequency vibration of the valve plate led on to the change of the excitative speed, then the pulsating pressure. On the basis of the various calculations and analyses, a simple and convenient method to consider the influence of the valve vibration was proposed in this work. By using the previous perturbation theory, the moving speed of the valve plate was supposed to be an additional high-frequency excitation. Fig.6 shows the calculated curve of the valve speed. This high-frequency excitative speed was analysed by Fourier method, and then the harmonic values $U_v^*(t)$ at each step were obtained. A twenty-step calculation was carried out in this work, becase the excitation was in the high-frequency range. $U_v^*(t)$ was substituted into the formular (6)-(8), then the pulsating pressure $\tilde{p}_3^*(t)$ caused by the high-frequency excitation $U_v^*(t)$ at the point 3 can be obtained. Adding the high-frequency pulsating pressure $\tilde{p}_3^*(t)$ to the pulsating pressure $p_3^*(t)$ calculated from the excitative speed $U_p^*(t)$, the revisory results on the pulsating pressure were obtained.

RESULTS

Fig.7 gives the calculated and experimental results of the pulsating pressure in the suction chamber of the compressor. If only the excitative function $U_p^*(t)$ is regarded as the calculating waveshape, the general trend of the calculated waveshape looks to be similar with the measurements, but the calculated results can not reflect the high-frequency waveshape in the practical measurement. The results indicate that the pulsating amplitude of the pressure calculated is 8.68% of the nominal suction pressure. However, the experimental value is 13.6% of the nominal pressure. Moreover, in the practical measurement, the high-frequency pulsating waveshape is obviously corresponding to the vibrational waveshape in the valve. The opening and the closure of the valve correspond to the trough and the crest of the pulsating pressure.

After the revision, the waveshape calculated has reflected some high-frequency produced by the vibration of the valve, and seems to agree with the measurements well. The pulsating amplitude of the pressure calculated is 14.45% of the nominal suction pressure, and approaches the practical value of 13.6%. For the revisory calculation of the pulsating pressure, the moving speed of the valve plate calculated is regarded as the excitation. The difference of the phase between the calculation and measurement is because the difference of the phase in the valve moving law calculated and practical measurement. At the stage of the closure of the valve plate, the difference in the vibrational amplitudes of the pulsating pressure is caused by the difference in the vibrational amplitudes of the valve plate calculated and the practical measurement.

CONCLUSIONS

1. Anomalous working state of the valve gives an influence on the characteristics of the gas pulsation. It is indicated that the calculating method proposed by this work is identified to be feasible.
2. It is necessary to study the relation between the working state of the valve and the gas pulsation in the plenum chamber. This is a prerequisite for the excellent design of the valve and piping.

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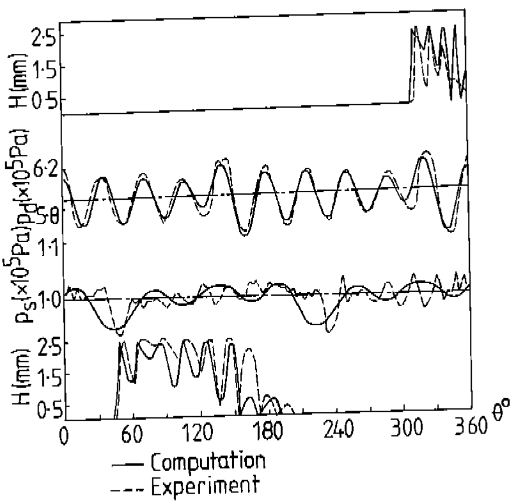

Fig. 1 The difference in the waveshape between the experiment and the calculation (p_s=98000Pa, p_d=588000Pa, T_s=285K, N=500rpm)

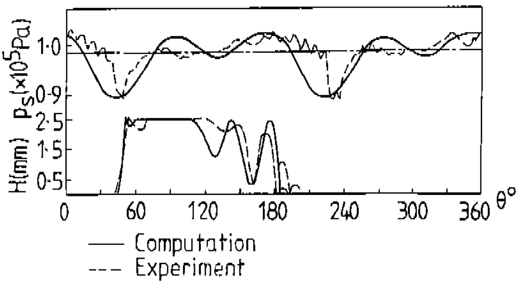

Fig. 2 The difference in the waveshape between the experiment and the calculation (p_s=98000Pa, p_d=58000Pa, T_s=285K, N=680rpm)

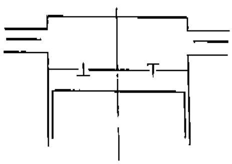

Fig. 3 A combined system of the cylinder and pipe.

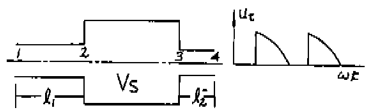

Fig. 4 Physical model.

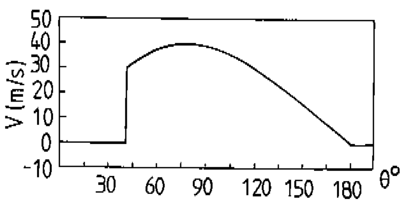

Fig. 5 Curve of the excitative speed.

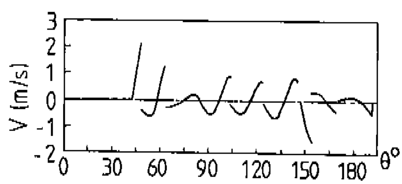

Fig. 6 Calculated curve of the valve speed.

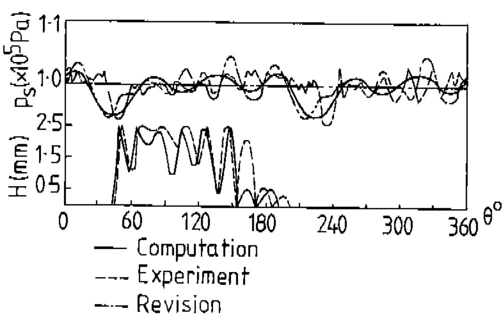

Fig. 7 The calculated and experimental results (p_s=98000Pa, p =588000Pa, T_s=285K, N=500rpm)