1982

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ANALYSIS OF THE MOTION OF PLATE TYPE SUCTION OR DISCHARGE VALVES MOUNTED ON RECIPROCATING PISTONS

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ABSTRACT

The equations of motion of plate type suction and discharge valves mounted on reciprocating pistons are given. Their solutions by modal series expansion are discussed and finally given in a form that conforms to the notation and approach used in previous work. Using the definition of an equivalent pressure, it is shown that both types of valves are assisted in closing, which may result in premature closing. During opening, suction valves are assisted by the piston acceleration, but discharge valves are in general not, depending on the compression ratio. It is concluded that the piston acceleration is not a negligible effect and should be included in simulation programs.

INTRODUCTION

Piston mounted valves are perhaps mainly useful in the refrigeration industry, where suction valves are and have been mounted on reciprocating pistons in low side compressors, where the suction gas fills the crank housing. While the author is not aware of an actual design, the same is possible in principle for discharge valves, provided the discharge gas fills the crank housing which is the case in so called high side compressors. The air and gas compressor industry faces the difficulty to keep air or gas from being contaminated by lubricant, and is therefore less likely to utilize such a design.

Piston mounted valves are attractive because of the continuing quest of designers to find enough space in the cylinder head to accommodate valve porting and plates without increasing the clearance volume unreasonably. As the compressor volume becomes larger, space restrictions become more severe. Utilizing the piston surface area almost doubles the available porting space. The potential penalty one pays is undesirable valve behavior, unless one compensates for the effect of the piston motion properly.

A piston mounted suction valve was first simulated by one of the author's students, J.M. Baum [1], as part of a Master's thesis. Unfortunately, the thrust of this simulation was directed toward the understanding of the gas pulsations in the discharge piping of a four cylinder automatic air-conditioning compressor. The suction valve had a negligible influence on it and was modeled only for completeness sake. Thus, no conclusions about the essential behavior of a piston mounted valve were drawn at the time. This paper attempts to remedy this to some extent.

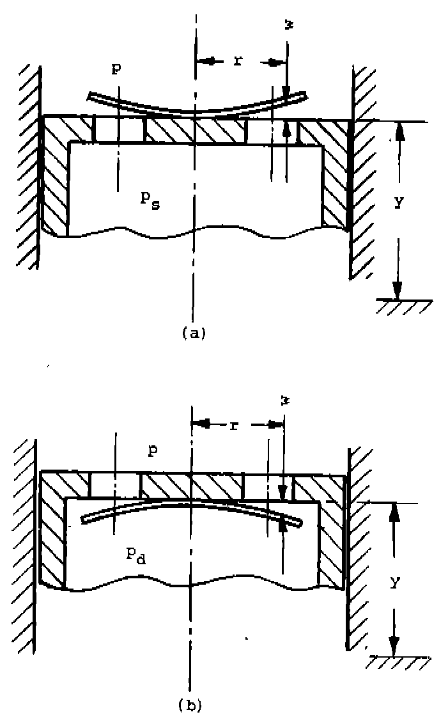

Fig. 1 Suction and Discharge Valve on Piston

EQUATIONS OF MOTION

We have to consider two cases. In the first case, shown in Fig. 1a, a suction valve is mounted on the piston, as it might be in a low side refrigeration compressor, where the suction gas fills the crankcase. As reference position y for the piston motion, we use the bottom dead center position (BDC). Positive displacements of the suction valve are in the same direction as the positive direction of the reciprocating piston displacement. Thus, one may write the kinetic energy of the suction valve as

$$K_s = \frac{\rho_s h_s}{2} \int_\theta \int_r [\dot{w}_s(r,\theta,t)+\dot{y}(t)]^2 \, r dr d\theta \qquad (1)$$

The dot indicates differentiation with respect to time.

For the case of a high side refrigeration compressor, where the discharge gas fills the crankcase, the direction of positive piston displacement is opposite to the direction of positive valve displacement as shown in Fig. 1b. The kinetic energy of the discharge valve is, therefore,

$$K_d = \frac{\rho_d h_d}{2} \int_\theta \int_r [\dot{w}_d(r,\theta,t) - \dot{y}(t)]^2 \, r dr d\theta \qquad (2)$$

Subscripts s and d distinguish between suction and discharge.

The energy input expression to the valves due to the total gas pressures q'_i is for both cases (i=s,d),

$$E_{Li} = \int_\theta \int_r q'_i(r,\theta,t) \, w_i(r,\theta,t) r dr d\theta \qquad (3)$$

except that for the suction valve, keeping with assumptions that are normally employed [2],

$$q'_s(r,\theta,t) = (p_s-p) f_s(r,\theta,t) \qquad (4)$$

and for the discharge valve

$$q'_d(r,\theta,t) = (p-p_d) \, f_d(r,\theta,t) \qquad (5)$$

The strain energy of the valve is also the same for both cases.

$$U_i = \frac{D_i}{2} \int_\theta \int_r \left[\left\{ \frac{\partial^2 w_i}{\partial r^2} + \frac{1}{r}\frac{\partial w_i}{\partial r} + \frac{1}{r^2}\frac{\partial^2 w_i}{\partial \theta^2} \right\}^2 - 2(1-\mu_i) \frac{\partial^2 w_i}{\partial r^2} \left\{ \frac{1}{r}\frac{\partial w_i}{\partial r} + \frac{1}{r^2}\frac{\partial^2 w_i}{\partial \theta^2} \right\} + 2(1-\mu) \left\{ \frac{1}{r}\frac{\partial^2 w_i}{\partial r \partial \theta} - \frac{1}{r^2}\frac{\partial w_i}{\partial \theta} \right\}^2 \right] r dr d\theta \qquad (6)$$

where $D_1 = E_i h_i^3 / 12(1-\mu_i^2)$.

Substituting these expressions into Hamilton's principle, which states that [3],

$$\int_{t_o}^{t_1} \left[\delta U_i - \delta E_{Li} + \delta K_i\right] dt = 0 \qquad (7)$$

we obtain by the usual operations [3] for the suction valve

$$D\nabla^4 w_s + \lambda_s \dot{w}_s + \rho_s h_s \ddot{w}_s = q_s - \rho_s h_s \ddot{y} \qquad (8)$$

and for the discharge valve

$$D\nabla^4 w_d + \lambda_d \dot{w}_d + \zeta_d h_d \ddot{w}_d = q_d + \rho_d h_d \ddot{y} \qquad (9)$$

Note that through the pressure load, an equivalent viscous damping pressure was also introduced. As expected, the piston acceleration acts as an additional forcing function on the valves.

SOLUTIONS

We approach the solution by modal series in the usual way [2,3]. Introducing

$$w_i = \sum_{m=1}^{\infty} q_{mi} \Phi_{mi} \qquad (10)$$

into the equations of motion results in an infinite set of ordinary differential equations

$$\ddot{q}_{mi} + 2\zeta_{mi}\omega_{mi}\dot{q}_{mi} + \omega_{mi}^2 q_{mi} = F_{mi} \qquad (11)$$

where, for the suction valve

$$F_{ms} = \frac{1}{\rho_s h_s N_{ms}} \int_\theta \int_r (q'_s - \rho_s h_s \ddot{y}) \Phi_{ms} r dr d\theta \qquad (12)$$

and for the discharge valve

$$F_{kd} = \frac{1}{\rho_d h_d N_{md}} \int_\theta \int_r (q'_d + \rho_d h_d \ddot{y}) \Phi_{md} r dr d\theta \qquad (13)$$

Other definitions are that

$$N_{mi} = \int_\theta \int_r \Phi_{mi}^2 \, r dr d\theta \qquad (14)$$

$$\rho_{mi} = \frac{\lambda_i}{2\rho_i h_i \omega_{mi}} \qquad (15)$$

To program the solutions, we follow reference [2]. Eq. (6.4) in reference [1] becomes

$$\ddot{q}_{ms} + 2\zeta_{ms}\omega_{ms}\dot{q}_{ms} + \omega_{ms}^2 q_{ms} = \frac{(p_s - p - \rho_s h_s \ddot{y}) \sum_{i=1}^{k_s} \Phi_{ms}(r_i,\theta_i) B_s(w_s(r_i,\theta_i))\Delta A_{is}}{A_s \rho_s h_s \sum_{j=1}^{L_s} \Phi_{ms}^2(r_j,\theta_j)\Delta A_{js}} \tag{16}$$

The integrations of Eqs. (12) and (13) are replaced by summations. A deviation of this form, with definitions of terms is found in reference [2]. Similarly, for the discharge valve, Eq. (6.1) of reference [2] becomes

$$\ddot{q}_{md} + 2\zeta_{md}\omega_{md}\dot{q}_{md} + \omega_{md}^2 q_{md} = \frac{(p - p_d + \rho_d h_d \ddot{y}) \sum_{i=1}^{k_d} \Phi_{md}(r_i,\theta_i) B_d(w_d(r_i,\theta_i))\Delta A_{id}}{A_d \rho_d h_d \sum_{j=1}^{L_d} \Phi_{md}^2(r_j,\theta_j)\Delta A_{jd}} \tag{17}$$

The first one to actually program a suction valve case was Baum [1]. He treated the case of suction valves on scotch-yoke actuated pistons. However, the main thrust of his investigation was the prediction of gas pulsations in the discharge system of a four-cylinder automotive compressor and he did not pursue the exploration of the influence of the moving piston. A typical result, at a speed of 3000 rpm, a suction pressure of 50 psia, and a discharge pressure of 250 psia, is shown in Fig. 2. No measurements of the suction valve motion were made. No suction valve stops exists, which accounts for the typical superimposed decaying oscillations at approximately the valves natural frequency.

DISCUSSION

It is possible to draw some general conclusions about the influence of a valve being located on a reciprocating piston. Let us take as an example the scotch-yoke mechanism, sketched in Fig. 3. Selecting as reference position the bottom dead center position (BDC), we obtain

$$y = R(1 - \cos \Omega t) \tag{18}$$

The acceleration of the piston is, therefore,

$$\ddot{y} = R\Omega^2 \cos\Omega t \tag{19}$$

From Eqs. (16), the equivalent pressure that would at a first glance seem to prevent opening of the suction valve is

$$p_e = \rho h \ddot{y} \tag{20}$$

since

$$p_s - p - \rho_s h_s \ddot{y} > 0 \tag{21}$$

is the condition for the opening of the suction valve. The equivalent pressure may also be written

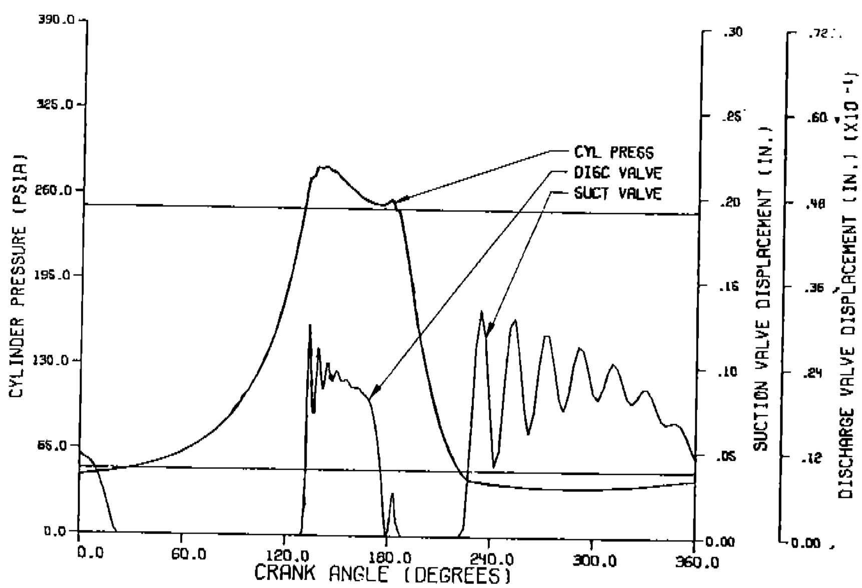

Fig. 2 Typical Simulation Result (1)

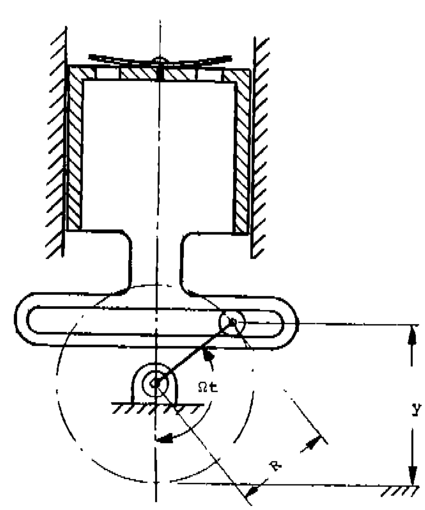

Fig. 3 Scotch-Yoke Drive

$$p_e = P_e \cos \Omega t \qquad (22)$$

where the amplitude is

$$P_e = \rho h R \Omega^2 \qquad (23)$$

The effective pressure is plotted in Fig. 4. We see from this figure, that far from preventing opening of the suction valve, the equivalent pressure actually assists it since it is negative in the region between π and $\frac{3\pi}{2}$, which is also the region where a suction valve usually opens, unless the clearance volume and the compression ratios are excessive.

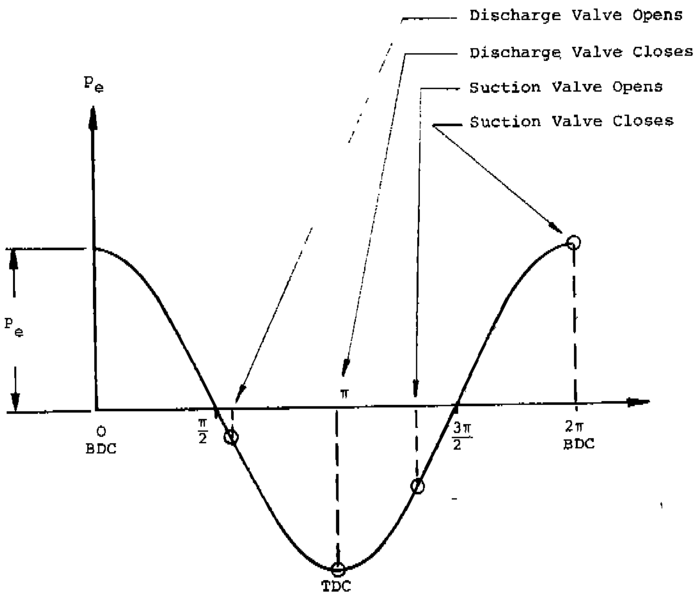

Fig. 4 Effective Pressure

When it comes to closing, the equivalent pressure tends to close a valve that is well designed under stationary conditions, prematurely. But since a common problem is to prevent suction valves from closing too late, which is done by making the valves stiffer among other measures, it would seem that the closing assistance by the equivalent pressure is potentially beneficial.

The situation is somewhat different for discharge valves, but not much. In this case, the condition for opening is

$$p - p_d + \rho_d h_d \ddot{y} > 0 \qquad (24)$$

if the compression ratio is relatively low so that the discharge valve opens ideally before $\pi/2$, opening will be premature and will be assisted by the effective pressure. For large compression ratios, when the ideal opening occurs past $\pi/2$, opening will be retarded by the effective pressure. This is a negative influence. Closing of the discharge valve, which should ideally occur at top dead center (TDC), will be premature and assisted. Again, since proper closing is sometimes a problem, this is in balance a perhaps desirable feature.

To obtain physical insight into the magnitude of this equivalent pressure, let us choose the following typical valve parameters: $\rho = 0.000726 [lb_f s^2/in^4]$ for steel, $h = 0.03$ [in.], and $R = 1.0$ [in.]. At 3600 rpm, the maximum equivalent pressure is 3.1 psi, at 5400 rpm, it is 6.9 psi. Obviously, the effect is not negligible since the pressure differential created by the valve as an orifice is of the same order of magnitude. To reduce the effect, the only measure is to reduce the mass per unit area of the valve plate. However, there is no reason satisfactory suction and discharge valves located on a reciprocating piston cannot be designed so that they perform satisfactorily. It is important, when deciding on the effective stiffness and mass, to take the equivalent pressure resulting from the piston motion into account. This is best done by way of a computer simulation.

NOTATION

K_s, K_d	= Kinetic energy
$\rho_s \rho_d$	= Density
h_s, h_d	= Thickness
r, θ	= Coordinates
w_s, w_d	= Valve displacement
y	= Piston displacement
E_{Ls}, E_{Ld}	= Load energy
q'_s, q'_d	= Pressure loads
p_s, p_d	= Suction and discharge pressures
U_i	= Strain energy
D_i	= Bending stiffness
μ_i	= Poissons ration
E_i	= Young's modulus
∇^2	= Laplacian operator
λ_s, λ_d	= Viscous damping coefficient
$\ddot{y}$	= Piston acceleration
q_{mi}	= Modal participation factor
ζ_{mi}	= Modal damping factors

ω_{mi} = Natural frequencies

Φ_{mi} = Natural modes

p = Cylinder pressure

B_s = Effective force area [2]

$\Delta A_{is}, \Delta A_{is}$ = Incremental areas [2]

A_s = Valve plate area [2]

k_s, k_d = Number of force area elments [2]

L_s, L_d = Number of valve plate elements [2]

R = Crank radius

Ω = Crank rotation speed in rad/s

p_e = Equivalent pressure

P_e = Equivalent pressure amplitude

REFERENCES

[1] J.M. Baum, "Computer Simulation of a Four Cylinder Air Conditioning Compressor with Special Attention to Discharge Cavity Interactions," MSME Thesis, Purdue University, 1975.

[2] W. Soedel, "Introduction to Computer Simulation of Positive Displacement Type Compressors," Short Course Text, Purdue University, 1972.

[3] W. Soedel, Vibrations of Shells and Plates, Marcel Dekker, Inc., New York, 1981.