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AN INVESTIGATION ON THE CALCULATING FOR OPTIMAL LIFT OF THE RING VALVE

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ABSTRACT

This paper analyses the loss of energy in ring valves of the compressor, investigates the calculating method for the main structure parameter—optimal lift, and derives the calculating formula. The measument result on the compressor L2-10/8-I shows that the valves designed through this calculating formula has more tangible result for economizing energy than others.

SYMBOLS

A	valve area
Ae	valve seat area
Av	valve lift area
Ap	piston area
b	passage width of the valve seat
Cd	speed of the dischage valve
Cs	speed of the suction valve
Cp	speed of the piston
Cm	average speed of the piston while the valve works
Cmd	average speed of the piston while the dischage valve works
Cms	average speed of the piston while the suction valve works

D	piston diameter
Di	middle diameter of valve plate
g	acceler ation of gravety
H	valve lift
Hd	discharge valve lift
Hs	suction valve lift
h	beforehand compressional deformation of the valve spring
i	valve number of the same name
K	valve spring rigidity
l	deformation of the valve spring
m	mass of the valve plate and spring; polytropic exponent of the process
n	r.p.m. of the compressor
p	pressure in the cylinder
p_d	actual discharge pressure
p_s	actual suction pressure
Δp_d	pressure loss of the discharge valve
Δp_s	pressure loss of the suction valve
Δp_{sH}	spring pressure loss from the valve lift
Δp_{sh}	spring pressure loss from beforehand compressional deformation of the valve spring
p_f	passage pressure loss of the valve
p_{fe}	passage pressure loss of the valve seat
p_{ev}	passage pressure loss of the valve lift
p_i	inertia pressure loss if the valve plate and spring
Q	volume flow of the valve

r	crank radius
s	piston stroke
W	work loss of the valve
W_d	work loss of the discharge valve
W_s	work loss of the suction valve
W_{sh}	work loss from the spring beforehand compressional deformation
W_{sH}	work loss of the spring lift compressional deformation
Z	number of the spring
α	opening angle of the valve
α_o	valve seat coefficient of the flow
β	valve lift area coefficient on the flow
λ	ratio of crankthrow to length of connecting rod
φ	utilization coefficient of the valve seat passage area, i.e. ratio valve seat passage area in concist of stiffening ribs to the valve seat passage area concest of stiffening ribs
ε	pressure ratio
ρ_d	discharge density
ρ_s	suction density
ω	angular velocity of compressor

INTRODUCTION

The ring valve in the piston compressor is a kind of automatic valve widely used. The ring valve directly affects the economic indexes and running reliability of the compressor, that is why designers attach great importance to the design of the compressor valve.

The lift is the mainstructural parameter of the ring valve and is vital important to the energy loss and service life of the valve. However, up to now, no calculating method may be emploied to design the optimal lift of the ring valve except by experence, so that designers have given up the optimization, liminate the ring lift by only the impact speed of the valve plate.

This paper investigates the calculating method and gives the calculating formula for optimal lift, and cites the results tested on the compressor L2-10/8-I by using several valves which have different lifts each other. The tests indicate that the valve designed through this calculating formula has more tangible effect than others.

PRESSURE LOSS OF THE VALVE

The gas flow system of the compressor consists of series throttlings. The gas valve among them is the main throttling. Whith the throttling, there must be pressure loss, in addition, there are springs and valve plates in the valve, so that the pressure loss includes three parts

a) The pressure loss of the gas flow led up to by the gas flow

$$\Delta p_f = \frac{\rho}{2g} \left(\frac{A_p C_p}{A_v} \right)^2$$

b) The pressure loss led up to by the spring

$$\Delta p_s = \frac{KZil}{A_e}$$

c) The pressure loss led up to by inertia of the valve plate and spring

$$\Delta p_i = \frac{m}{A_e} \cdot \frac{d^2H}{dt^2}$$

Analyse the pressure loss through Fig. 1. The Fig. 1a is of p-θ pressure indicator, Fig. 1b is of H_d-θ

discharge valve plate displacement, Fig. 1c is of $H_s-\theta$ suction valve plate displacement, Fig. 1d is of $C_p-\theta$ instantaneous piston speed, Fig. 1e is of $C-\theta$ instantaneous gas flow lpeed, Fig. 1f is of p-L spring pressure loss.

This paper takes the suction valve as an example to analyse. The running process has four stages (Fig. 1a): The first stage is a preparatory stage ($\theta_a-\theta_b$), the second stage is an opening stage ($\theta_b-\theta_c$), the third is a completely stage ($\theta_c-\theta_d$), the fourth is a closing stage ($\theta_d-\theta_e$).

The pressure loss of the preparatory stage ($\theta_a-\theta_b$) is produced by spring deformation precompressed. If there is no the preceformation on the spring, the valve would open at point a. With the predeformation, the valve do not open at point a as the clearance volume has expanded to point a. When the clearance volume continue expanding to point b, having overcome the spring force coused by pre-compression, the valve opens. The pressure loss in thes stage (O_a-O_b) is

$$\frac{KZih}{A_e}$$

The pressure loss in the opening stage ($\theta_b-\theta_c$) constitutes three parts

a) The pressure loss produced by the spring deformation from point b where the valve plate goes from to point c where the valve plate meets the valve guard c (Fig. 1a)

b) The pressure loss caused by accelarating the valve plate and spring:

$$\frac{m}{A_e} \cdot \frac{d^2H}{dt^2}$$

c) The pressureloss from gas flow:

$$\frac{\rho}{2g} \left(\frac{A_p C_n}{A_v} \right)^2$$

The pressure loss in the completly open stage (θ_c-θ_d) contains two:

a) The pressure loss under the completly compressional deformation (h + H) of the spring, expressed by

$$\frac{KZi\ (\ h + H\ )}{A_e}$$

b) The pressure loss caused by gas flow

$$\frac{\rho}{2g}\left(\frac{A_p C_p}{A_v}\right)^2$$

The pressure loss during the process of closing stage (θ_d-θ_e) contains three parts:

a) Started from the valve guard and ended with dashing against the valve seat, the pressure loss consumed by the spring deformation, expressed by

$$\frac{KZil}{A_e} \qquad (\ h < l < H\ )$$

b) The pressure loss consumed by accelarating the valve plate and spring $\frac{m}{A_e} \cdot \frac{d^2H}{dt^2}$

c) The pressure loss caused by the gas flow

$$\frac{\rho}{2g}\left(\frac{A_p C_p}{A_v}\right)^2$$

WORK LOSS OF THE VALVE

Because of the pressure loss during the valve-running process, the suction pressure and discharge pressure in the cylinder are not p_s and p_d separately, instead, $p_s - \Delta p_s$ and $p_d + \Delta p_d$ are they. With the piston area A_p, the piston force increases by $p_s \cdot A_p$ and $p_d \cdot A_p$ separately during the suction process and discharge process. If the piston displacement $S_e - S_a$ is occurres during suction and $S_{e'} - S_{a'}$ during discharge, the work consumed for overcoming the suction and discharge resistance

$$W_s = \Delta p_s A_p (S_e - S_a) = \Delta p_s (V_e - V_a)$$

$$W_d = \Delta p_d p_A (S_{e'} - S_{a'}) = \Delta P_d (V_{e'} - V_{a'})$$

The total work loss

$$W = W_s + W_d$$

Fig. 2 the compressor PV indicator diagram shows the work loss. The area of abc and the area of a'b'c' are the workloss for suction and discharge separately.

in order to simplify the concoution a few simplifications are assumed:

a) Neglect the influence to the work loss caused by damping and oil stick.

b) Neglect the influence caused by the mass of the valve and spring.

c) The valve lift area is constant while the valve leaves from the valve seat.

d) The ratio of the effective valve lift area to the calculated area is a constant, i.e. $\beta =$ const.

Fig. 3b shows the gas flow pressure loss diagram under the simplifications. The area $V_b f V_e$ is the work loss caused by gas flow of the suction valve.

$$W_f = \frac{\rho}{2g} \left(\frac{A_p C_m}{A_v} \right)^2 (V_e - V_b)$$

Fig. 3c is the spring pressure loss diagram of the pre-deformation of the simplified spring. The area VghVe represents the pre-defomation work loss.

$$W_{sh} = \frac{KZih}{A_e} (V_e - V_b) - \frac{KZih^2}{2}$$

Fig. 3d expresses the pressure loss caused by the compressional deformation of the spring. The area VijV represents the pressure loss.

$$W_{sH} = \frac{KZiH}{A_e}(V_e - V_b)$$

Summing the four diagrams Fig. 4b,c,d, the Fig. 4e which shows the pressure loss of the suction valve under the simplefications is attained. The area VgfijhV represents the work loss of the suction valve, the sum is

$$W_s = W_f + W_{sh} + W_H = \frac{\rho}{2g} \left(\frac{A_p C_m}{A_v} \right)^2 (V_e - V_b) + \frac{KZih}{A_e} (V_e - V_a) - \frac{KZih^2}{2} + \frac{KZiH}{A_e} (V_e - V_b) \qquad (1)$$

$$\text{Where: } A_v = 2\pi iH\Sigma D_i \qquad (2)$$

$$A = \pi ib\varphi\Sigma D_i \qquad (3)$$

The formula (1) is used for calculating the work loss of suction valve. In the same way, the formula for calculating the work loss of cischarge valve is

$$W_d = (V_{e'} - V_{b'}) \left[\frac{\rho}{2g} \left(\frac{A_p C_m}{A_v} \right)^2 + \frac{KZiH}{A_e} \right] + \frac{KZih}{A_e} (V_{e''} - V_{a'}) - \frac{KZih^2}{2} \qquad (4)$$

CALCULATING FOR THE OPTIMAL LIFT OF THE RING VALVE

According to the formulas (1), (4), we can see that the lift H and the pre-deformation h of the spring are main decisive parts of the work loss of the valve, i.e. $W = W (h, H)$.

From (1), (4), we can get the second partral derivative with respect to h,

$$\frac{\partial^2 W}{\partial h^2} = -KZi$$

With $K>0$, $Z>0$, $i>0$, it is evidently the $\partial^2 W/\partial h^2 < 0$, so W (h) has not minimum value, but only maximum. The smaller the spring pre-deformation is, the less the work loss of the valve. The second partial derivative of the formula (1) or (4) with respect to H is

$$\frac{\partial^2 W}{\partial H^2} = \frac{3\rho}{g}\left(\frac{A_p C_m}{2\pi\beta i\Sigma D_i}\right)^2 \frac{1}{H^4}\left(V_e - V_b\right)$$

Where $\partial^2 W/\partial H^2 > 0$, because of $V_e > V_b$. So that W(H) has minimum value. The first partial derivative:

$$\frac{\partial W}{\partial H} = \left[\frac{KZi}{A_e} - \frac{\rho}{g}\left(\frac{A_p C_m}{2\pi\beta i\Sigma D_i}\right)^2 \frac{1}{H^3}\right]\left(V_e - V_b\right) \qquad (5)$$

When (5) takes zero, the W(H) has minimum value. In this case,

$$H = \left[\frac{b\varphi\rho}{4gKZ\pi\Sigma D_i}\left(\frac{A_p C_m}{i\beta}\right)^2\right]^{1/3}$$

$$= \left[\frac{\pi b\varphi\rho}{64gKZ\Sigma D_i}\left(\frac{D^2 C_m}{i\beta}\right)^2\right]^{1/3} \qquad (6)$$

The formula (6) is suitable for both suction valve and discharge value. Where C_m is the average speed of the piston. If the opening angle of the valve is α and the angle of the poston strode end at the valve closure end is π, then the sustained opening angle of both suction valve and discharge valve are all $\pi-\alpha$, the average speed of the piston with in the limits of $\pi-\alpha$

$$C_m = \frac{1}{\pi-\alpha}\int_\alpha^\pi C_p d\theta = \frac{1}{\pi-\alpha}\int_\alpha^\pi r\omega\left(\text{Sin}\theta + \frac{\lambda}{2}\text{Sin}2\theta\right)d\theta$$

Where $r = S/2$, $\omega = \frac{\pi n}{30}$, so

$$C_m = \frac{Sn}{60} \frac{1}{1-\frac{\alpha}{\pi}} \left[\left(1 - \frac{\lambda}{2} \right) + \mathrm{Cos}\alpha \left(1 + \frac{\lambda}{2} \mathrm{Cos}\,\alpha \right) \right] \qquad (7)$$

Where the opening angle of the suction valve

$$\alpha = \mathrm{Cos}^{-1} \left[1 - 2\alpha_0 (\varepsilon^{1/m} - 1) \right] \qquad (8)$$

and of the discharge valve

$$\alpha = \mathrm{Cos}^{-1} \left[1 - 2 (1 + \alpha_0) \varepsilon^{-1/m} + 2\alpha_0 \right] \qquad (9)$$

Because the sustained opening angle of the discharge valve is less than that of the suction valve, so the piston average speed of the discharge valve is less than that of the suction valve. If H_d, H_s are the optimal lifts of the discharge valve and suction valve, the ratio of H_d to H_s is

$$\frac{H_d}{H_s} = \left[\frac{\rho_d}{\rho_s} \left(\frac{C_{md}}{C_{ms}} \right)^2 \right]^{1/3} = \left[\frac{p_d}{p_s} \frac{T_s}{T_d} \left(\frac{C_{md}}{C_{ms}} \right)^2 \right]^{1/3}$$

$$= \left[\varepsilon^{1/m} \left(\frac{C_{md}}{C_{ms}} \right)^2 \right]^{1/3} \qquad (10)$$

THE LIMITS OF OPVIMAL LIFT VALVE OF THE RING VALVE

At present, designing the valve, $H < b/2$ is taken as the limits of the ring valve lift. However in the point of view of this paper, $H < b\varphi\alpha/2\beta$ should be taken. The valve lift is a series system of the valve seat area and valve lift area. The valve pressure loss caused by gas flow is

$$\Delta p_f = \Delta p_{fe} + \Delta p_{fv}$$

i.e.

$$\frac{\rho}{2g} \left(\frac{Q}{4} \right)^2 = \frac{\rho}{2g} \left(\frac{Q}{A_e} \right)^2 + \frac{\rho}{2g} \left(\frac{Q}{A_v} \right)^2$$

According to this formula we derive

$$\frac{1}{A^2} = \frac{1}{A_e^2} + \frac{1}{A_v^2}$$

From this formula we can see that among A_e and A_v the A depends on the smmaller. The valve lift area A_v is the smallest in the valves, so designers accept the valve lift area as the valve area, and accept the pressure loss of gas flow of the valve at the valve lift area as the pressure loss of the gas flow at the valve area. Evidently the area of the valve lift calculated by formula (6) must be less thav the valve seat area, i. e. $A_v < A_e$.

Substituting the formula (2), (3) into the above, then

$$H < b\varphi\alpha/2\beta \qquad (11)$$

The formula (11) tell us the limits of the ring valve lift. If the optimal lift calculated by formula (6) were out of accord with formula (11), the spring rigidity K or number of the spring would be changed in order to accord with formula (11).

CONCLUDING REMARK

We have tested and verified the above mentioned optimal lift formula on the compressor of L2-10/8-I. We take some valves (which accord with H. Davis' criterion) which have different lifts each other to test one by one at the compressor of L2-10/8-I. The tests indicate that it has tangible effeciency for economizing energy to use the valve designed through the method stated in this paper.

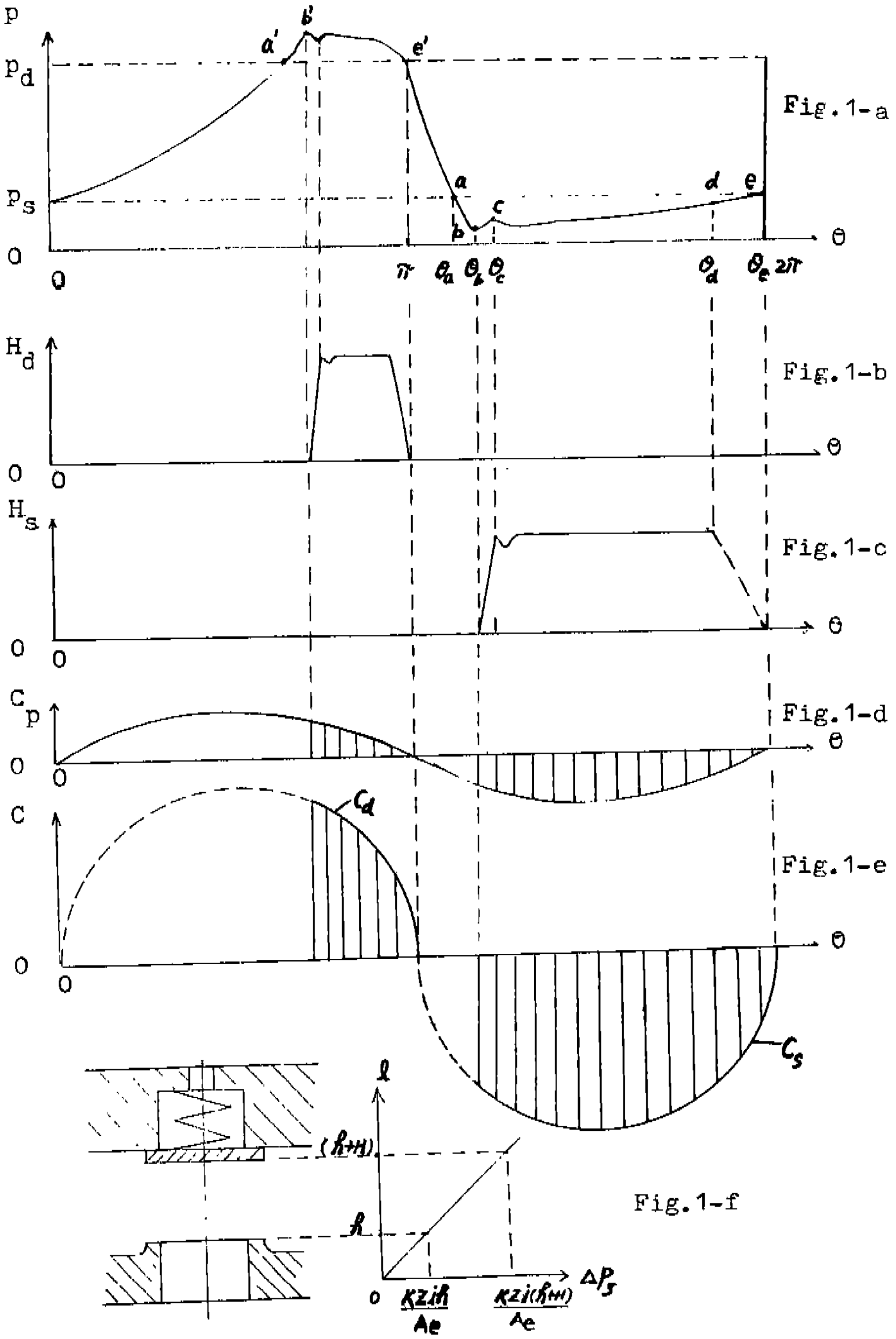

Fig.1-a

Fig.1-b

Fig.1-c

Fig.1-d

Fig.1-e

Fig.1-f

Fig. 2

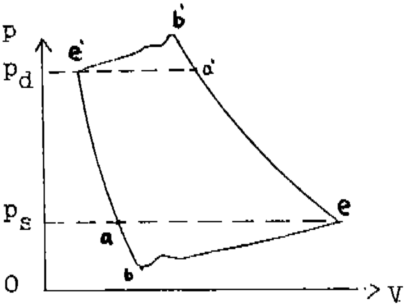

Fig. 3-a

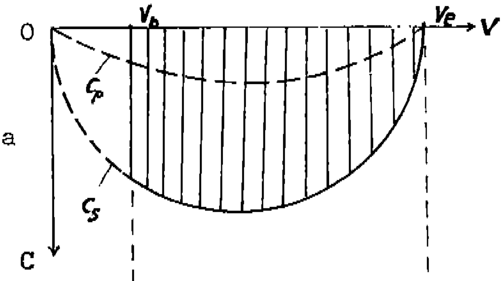

Fig. 3-b

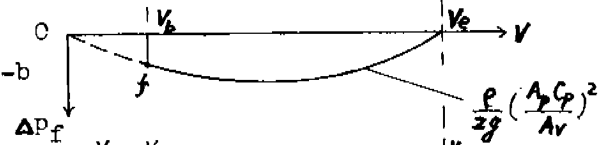

Fig. 3-c

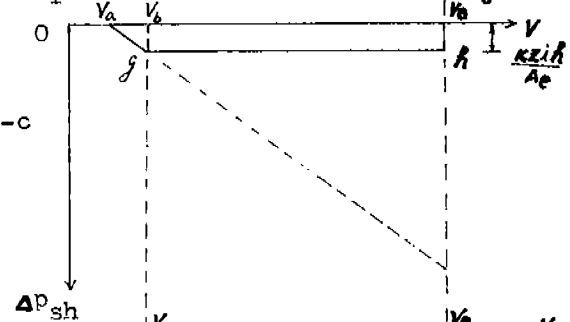

Fig. 3-d

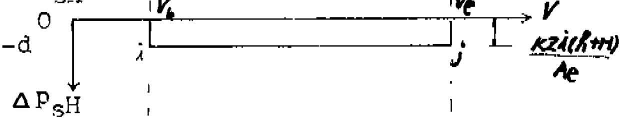

Fig. 3-e