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An Approach for a High-Speed Double Acting Compressor Using a Single Elastic Piston Ring to Reduce the Specific Power

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AN APPROACH FOR A HIGH SPEED DOUBLE ACTING COMPRESSOR USING A SINGLE ELASTIC PISTON RING TO REDUCE THE SPECIFIC POWER

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ABSTRACT

This paper, through analysis to the specific power of the compressor, indicates that to reduce the power loss caused by friction is an important way to reduce the specific power of the compressor, investigates the realizability employing a single piston ring or a piston ring with small elasticity in the high speed double acting compressor, and raises a standpoint for a high speed double acting compressor to use a single piston ring with small elasticity to reduce the specific power. The experiment to verify this standpoint has been done at the compressor of the series L2-10/8-I, the efficient result, haring reduced the specific power by 3 - 3.5 %, has been shown.

SYMBOLS

b thickness of piston ring

D piston diameter

f area for lear off gas

g acceler ation of qravety

K C_p/C_v

m polytropic exponent of the process

N shaft power

N_i indicated power

N_m mechanical power

N_r piston ring frictional power

N_{rg} piston ring frictional power from gas pressure

N_{rs} Piston ring frictional power from elastic force

n r.p.m

p pressure of piston ring to cylinder

p_a pressure of piston ring inertia

p_b gas pressure after piston ring

p_d discharge pressure

p_g gas perssure before piston ring

p_i indicated perssure

p_m frictional pressure

p_{rg} gas pressure in the piston ring trough

p_{rs} pressure of elastic to cylinder

p_s suction pressure

Q capacity

q specific power

q_i specific indicated power

(q_i) theroretical specific indicated power

q_m specific mechanical power

R gas constant

r crank radius

s piston strok

T temperature of lear off gas

T_s suction temperature

t width of piston ring

V piston velocity

X piston displacement

α relative clearance volume

δ_d relative discharge pressure loss

δ_s relative suction pressure loss

ε tetal pressure ratio

ε_i stage pressure ratio

η mechanical efficiency of piston ring

λ volumetric efficiency

μ coefficient of frection

φ runoff coefficient

ω angular velocity compressor

INTRODUCTION

For the compressor to reduce the specific power is very important. At present, the studies for reduct the specific power are more coucentrated on the intercooler, valves, gasflow pulse, however for this paper, is concentrated on the reduction of frection. The compressor power loss caused by frection are 5 - 15 %[1] of the compressor shaft, it equals to the power loss caused by the valves. Among the friction power loss, the loss caused by the piston rings are 43 - 53 %[2]. So reducing the number of the piston rings or abating the piston ring elasticity is a way to decrease the friction power loss. When this way is adopted, the leakage increase, consequently the capacity decrease. To provent this, two ways may be employed:

a) Turn the harmful suspension to a beneficial suspension, i. e. during the suspension process, for a double acting compressor, let the high pressure gas via the suspension clearance entre the other side cylinder, being counted in the capacity.

b) Increase the cylinder diameter by a small differem to make up the leakage. The experiment on the high speed double acting compressor L2-10/8-I has shown the a tangible result to reduce the specific power by 3 - 3.5 %.

THE WAY TO REDUCE THE SPECIFIC POWER

The formula for specific power is $q = N/Q$. The N consists of the indicated power N_i ard the mechanical power N_m (mainly friction power). Reforning the formula as

$$q = \frac{N_i}{Q} + \frac{N_m}{Q}$$

Let

$$q_i = \frac{N_i}{Q} \tag{1}$$

$$q_m = \frac{N_m}{Q} \tag{2}$$

q_i, q_m represent the specific indicated power and the specific mechamical power separately so

$$q = q_i + q_m \tag{3}$$

The formula of indicated power of multistage compressor is derived by considering the material[2] (with the simplification $\varepsilon_i = \varepsilon^{1/i}$, $T_{si}/T_{s1} = 1$)

$$N_i = 1.634 \frac{m}{m-1} p_{s1} Q \sum_{i=1}^{N} \left\{ \left[\varepsilon^{1/i} \left(\frac{1+\delta_d}{1-\delta_s} \right) \right]^{\frac{m-1}{m}} - 1 \right\}_i$$

Substituting it into (1)

$$q_i = 1.634 p_{s1} \frac{m}{m-1} \sum_{i=1}^{N} \left\{ \left[\varepsilon^{1/i} \left(\frac{1+\delta_d}{1-\delta_s} \right) \right]^{\frac{m-1}{m}} - 1 \right\}_i \qquad (4)$$

From (4) we can see that for a given ambient the q_i depents on only the relative resictance pressure loss δ_s, δ_d. hat is the current way decreasing δ_s, δ_d to reduce q_i. However it is limited to decrease δ_s and δ_d. If δ_s=0, δ_d= 0, the ideal value of the q_i could be represented as

$$q_i = 1.634 p_{s1} \frac{m}{m-1} \sum_{i=1}^{N} \left(\varepsilon^{\frac{m-1}{im}} - 1 \right)_i$$

For a two stage compressor with a discharge pressure 9 ata the $\{q_i\}$= 4.22 $KW/m^3/min$. It is obviously when δ_s, δ_d is approaching to zeroes a fangible reduction of q_i is too difficult to do.

Differetiate (2) and cousidering $Q=\pi D^2 vn\lambda/4$:

$$dq_m = \frac{N_m}{Q} \left(\frac{dN_m}{N_m} - \frac{d\lambda}{\lambda} - \frac{2dD}{D} \right) \qquad (5)$$

Based on (5), three approaches are available:

a) Decrease friction power loss N_m. b) Increase discharge coefficient λ . c) Increase the cylinder diameter D. Only decreasing N_m will be discussed in this paper.

THE METBOD TO DECREASE THE FRICTION POWER LOSS OF THE PISTON

The friction power loss is vital important for a high speed compressor. There are two parts of the frichion power loss — the useful and useless. The function of the piston ring is of the dynamic sealing. That is useful. However because of the chopping of the suction and discharge the sealing is necessary only during the compressing and discharging processes and the sealing is no need for sueking and the compressor starting, stopping, unloading so that is the useless part . To analyses the utility rate , the concept on mechanical efficiency of the piston ring is introdued bellow

$$\eta = \frac{N_{rg}}{N_r} \tag{6}$$

Wher, N_{rg} — the friction power loss caused by frection sealing.

N_r — the total frection power loss of the piston ring.

The N_r depends on its pV value, V represents (Fig. 1) velocity relative to the cylinder wall, p represents the contact pressure by which the piston ring acts on the cylinder wall. There are two parts within p. One is p_{rs} caused by the piston ring elastic force, the other is gas pressure p_g. p_{rs} gives a pre-pressure agains the cylinder wall, we name it as the first sealing, the p_g functions as sealing also . However it acts after esfablishing the first sealing, subsequently the first sealing is useless. The Fig. 2a illustrates the ideal relation between an ideal piston ring elastic p_{rs} and θ, i.e. when necessary to establish the first sealing the p_{rs} has a certain value, when unnecessary the p_{rs} is nulled, consequently only p_g is left, its average is $\Delta p/2 = (p_g - p_b)/2$. Bat we have no way to approach that, the only way we can do is to reduce p_{rs} to the minimum — the installating stress $[p_{rs}]$, the curve of contat pressure p and θ is shown in Fig. 2b, it is piled from $[p_{rs}]$ and $\Delta p/2$. The friction power loss caused by piston ring elastic force is calculeted throngh

$$N_{rs} = iDbV\mu p_{rs} \tag{7}$$

and that caused by p_g

$$N_{rg} = (D-2t)(p_{g1} + p_{g2} + \dots + p_{gi})bV\mu \approx \frac{DbV\mu\Delta p}{2} \tag{8}$$

The total

$$N_r = N_{rs} + N_{rg} = DbV\mu\,(ip_{rs} + \Delta p/2) \tag{9}$$

Substituting (8), (9) into (6)

$$\eta = \frac{1}{1 + \dfrac{2ip_{rs}}{\Delta p}} \tag{10}$$

The Δp in formula (10) is given, so the anly way to raise η is to decrease i or p_{rs} . From formulas (7), (9)

$$dN_r = \frac{\partial N_r}{\partial i}\,di + \frac{\partial N_r}{\partial p_{rs}} = DbV\mu\,(p_{rs}di + ip_{rs}) \tag{11}$$

Form (11) we can see that if the number of the piston rings is subtracted by Δi, the compressor shaft power will decrcase by $DbV\mu p_{rs}\Delta i$; if reduce the elastic force by Δp_{rs} , the shaft power will decrease by $DbV\mu i\Delta p_{rs}$; the third point is that the reduction of the shaft power is propotional to its rotating valocity. So it is an important way to decrease the elasticity of the piston rings or the number of the piston rings for reduce the friction power loss of the piston rings.

ANALYSIS OF THE POSSIBILITY

Usually there two piston rings in the high speed double acting compressor.

1. The possibility to employ a single piston ring:

a) The leakage is small in a high speed compressor see Fig. 3.[2]

b) The leakage is small in a double acting compressor, the leakage is expressed as

$$G = \varphi f \frac{p_g}{\sqrt{RT}} \sqrt{\frac{2gk}{k-1}\left[\left(\frac{p_g}{p_b}\right)^{2/k} - \left(\frac{p_b}{p_g}\right)^{\frac{k+1}{k}}\right]} \tag{12}$$

The maximum of the leakage appearces when

$$\frac{p_b}{p_g} \leqslant \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \qquad (13)$$

During compressing process the pressure ratio

$$\frac{p_b}{p_g} = \frac{p_{si}}{\left(\frac{\alpha s + s}{\alpha s+s-x}\right)^k p_{si}} = \left[1 - \frac{x}{(1+\alpha)s}\right]^k$$

Substituting them into (13), the displaciment at which the maximum leakage speed will oceur

$$X \geqslant (1+\alpha) \left[1 - \left(\frac{2}{k-1}\right)^{\frac{1}{k-1}}\right] s \qquad (14)$$

For an ordinary compressor X$\geqslant$0.4s (stroke). For a single acting compressor it occurs all the way except the first stage.

c) There are counter-flow during compressing, the leakage

$$W = \int_{0^\circ}^{360^\circ} G d\theta$$

From G-θ curve, the leakage is represented by the area under the curve, the positive represents gas leakage and the negative represents the counter-flow. For a single acting compressor, its leakage is represented by the area abcde in Fig. 4c. Fig. 5a,b,c show the p_i-θ, Δp-θ, G-θ of the double acting compressor. There are three situations:$\Delta p > 0$, $\Delta p = 0$, $\Delta p < 0$. The leakage oceur when $\Delta p > 0$, no leakage when $\Delta p = 0$, the counter-flow when $\Delta p < 0$, The area cdeg represents the leakage during compressing and discharging at the cover side, the area efg represents leakage during the clearance expanding at the cover side, the area abc represents the counter flow from the shaft side to the cover side during clearence expanding at the shaft side.

Generally considering the clearance volumes equel, so the area abc and efg are counteracted, i.e. the total leakage is cdeg during a cirele. Comparing with Fig. 4c, the leakage of a double acting compressor is much smaller than that of a single acting one.

The analysis above shows it is useful to employ a single piston ring in an high speed double acting compressor.

2. The possibility to employ a piston ring with small elasticity:

a) For a high speed compressor the average pressure in the piston ring groove is higher than that in the cylinder. Fig. 6[3].

b) For a double acting compressor, in the piston ring groove, the pressure exsists during running. See Fig.7. And single acting different to double, Fig. 8.

c) For a double acting compressor the labricating oil is supplied by oil pump, and is adjustable, so it is unnecessary to scrape the oil.

So it is possible to employ a piston ring with small elasticity for a high speed double acting compressor. However at present the standard is simuleted according to the cylinder diameter, it is suitoble for low speed, and high speed either, so it is not reasonable for the high speed double acting compressor.

THE UTILIZATION OF THE CLEARANCE EXPANDING GAS

It is known the clearence volume reduces the volumetric efficiency of the compressor so it is peruicious. The suspension of the piston ring in the ring groove distroys the second sealing surface and increasing the leakage so it is termed pernicions motion. But it could be utilized in the double acting compressor.If it is controlled to make suspension during one side of the cylinder is in the initial compressing stage and the other side is in the clearance expanding stage, then the high pressure gas will go throuzh the second sealing surface and entering the compressing space to make the capacity increase. To realige this, the following relation must be satisfied

$$0^{\circ} < \theta < \theta_a \quad \text{or} \quad 180^{\circ} < \theta < \theta_b \tag{15}$$

Where the θ represents the crank shaft. θ_a, θ_b represent the crank shaft angles at the points where the present on the both sides are equal. see Fig. 9a. According to the expanding equation and compressing equation there are

$$p_b = \left(\frac{\alpha s}{\alpha s + x} \right)^m \varepsilon^i p_{s1} \tag{16}$$

$$p_g = \left(\frac{\alpha s + s}{\alpha s + s - x} \right)^m \varepsilon^{i-1} p_{s1} \tag{17}$$

Considering the piston displacement

$$X \doteq \frac{s}{2} \left(1 - \mathrm{Cos}\theta \right) \tag{18}$$

So

$$\begin{cases} \theta_a = \mathrm{Cos}^{-1} \left[1 - \dfrac{2\,(1+\alpha)(\varepsilon^{1/m} - 1)}{1 + \alpha(1 + \varepsilon^{1/m})} \right] \\ \theta_b = 180^{\circ} + \theta_a \end{cases} \tag{19}$$

There are forces along the shaft on a unit area p_g, p_b, p_m, p_a showing on Fig. 10. Take the piston moving direction as the posifive, when $\Sigma p = p_b - (p_g - p_m - p_a)$ is negative, the pistonring suspansion will occur. where

$$p_m = \frac{(D-2t)b\mu p_b + Db\mu p_{rs}}{\pi (D-t)t}$$

$$p_a = \frac{mr\omega^2 \mathrm{Cos}\theta}{g\pi (D-t)t}$$

Considering (16), (17), (18), the relation to occur the suspension is

$$\left[1 - \frac{1-\cos\theta}{2(1+\alpha)}\right]^{-m} \mathcal{E}^{i-1} p_{s1} + \frac{mr\omega^2\cos\theta}{g\pi(D-t)t} + \frac{Db\mu p_{rs}}{\pi(D-t)t}$$

$$> (1 - \frac{b\mu}{\pi t})(1 + \frac{1-\cos\theta}{2\alpha})^{-m} \mathcal{E}^{i} p_{s1} \qquad (20)$$

From Fig. 9b, the Σp is negative during $0^\circ < \theta < \theta_{ao}$, $180^\circ < \theta < \theta_{bo}$ the useful suspension will occur.

CONCLUDING REMARKS

The experiment has been carried out on the two stage compressor L2-10/8-I (n = 980 r.p.m, Q = $10M^3$/min, p_d= 9 ata). The methods stated above have been employed and the specific power reduced by 3 - 3.5 % .

CONSULTING DOCUMENTS

[1] T. F. Condlajiywa, " Horizontal Opposed Compressor", U.S.S.R. 1968.

[2] M. E. Flenker, " Reciprocating Compressor ", U.S.S.R. 1960.

[3] B. Y. Jicibol, " The Theory & Calculate of the Piston Ring ", U.S.S.R.

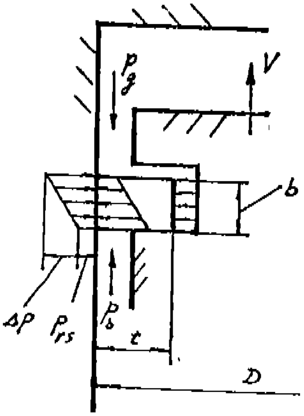

Fig. 1

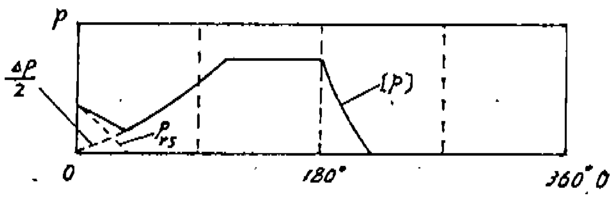

Fig. 2a
p - θ

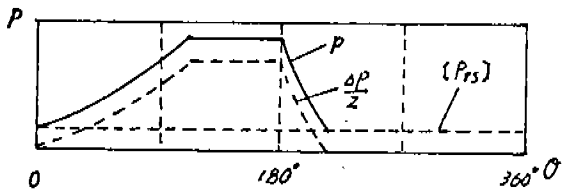

Fig. 2b
p - θ

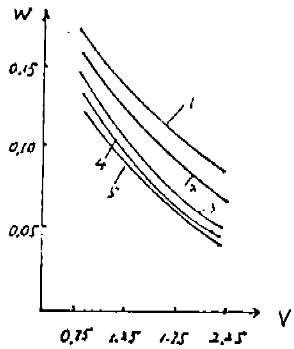

Fig. 3

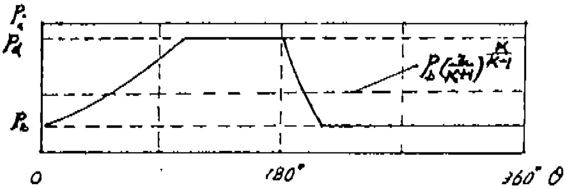

Fig. 4a
$p_i - \theta$

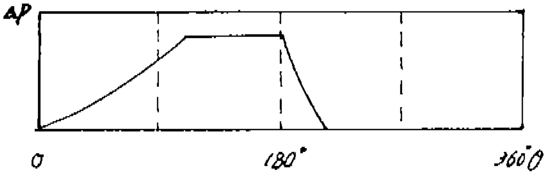

Fig. 4b
$\Delta p - \theta$

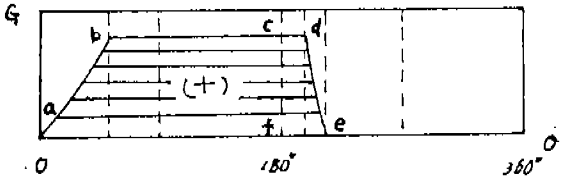

Fig. 4c
$G - \theta$

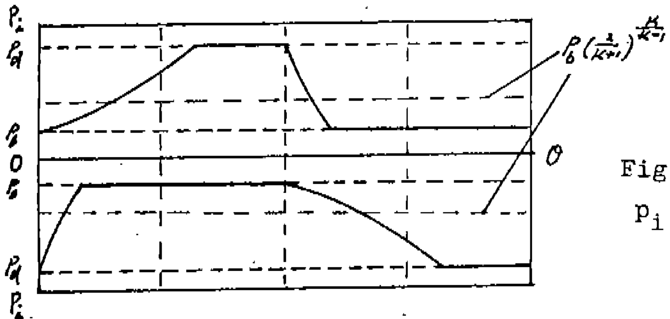

Fig. 5a
$p_i - \theta$

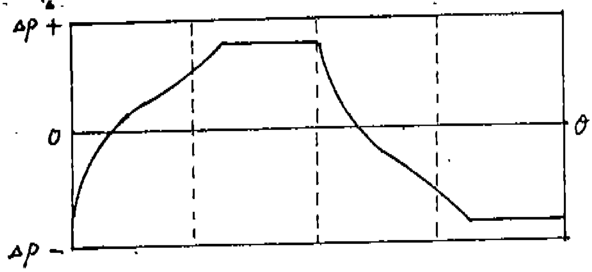

Fig. 5b
$\Delta p - \theta$

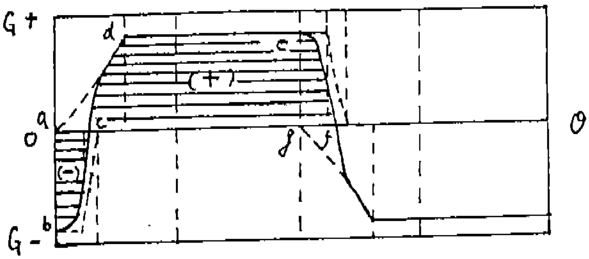

Fig. 5c
$G - \theta$

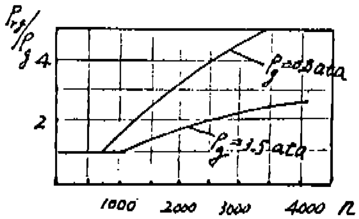

Fig. 6

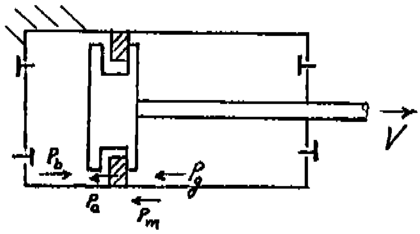

Fig. 10

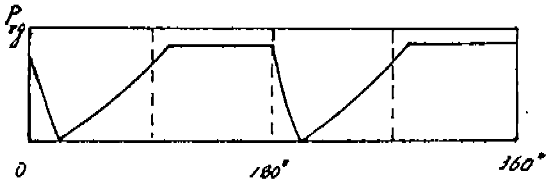

Fig. 7
p_{rg}- θ

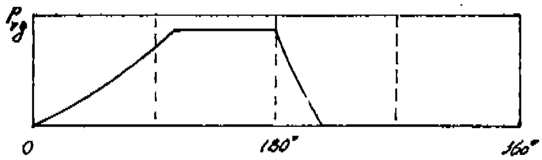

Fig. 8
p_{rg}- θ

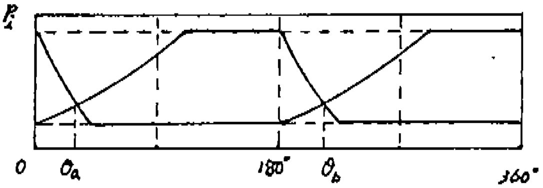

Fig. 9a
p_i - θ

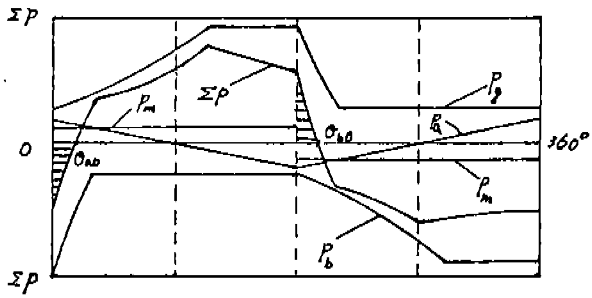

Fig. 9b
Σp - θ