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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, having reduced the specific power by 3 - 3.5 %, has been shown.

SYMBOLS

\( b \) thickness of piston ring

\( D \) piston diameter

\( f \) area for leak off gas

\( g \) acceleration of gravity

\( K \) \( \frac{C_p}{C_v} \)

\( m \) polytropic exponent of the process
N  shaft power
Ni  indicated power
Nm  mechanical power
Nr  piston ring frictional power
Nrg piston ring frictional power from gas pressure
Nrs Piston ring frictional power from elastic force
n  r.p.m
p  pressure of piston ring to cylinder
Pa  pressure of piston ring inertia
Pb  gas pressure after piston ring
Pd  discharge pressure
Pg  gas pressure before piston ring
Pi  indicated pressure
Pm  frictional pressure
Prg gas pressure in the piston ring trough
Prs pressure of elastic to cylinder
Ps  suction pressure
Q  capacity
q  specific power
qi  specific indicated power
(qi) theoretical specific indicated power
qm  specific mechanical power
R  gas constant
r  crank radius
s  piston strok
Ts  temperature of lean off gas
Ts  suction temperature
For the compressor to reduce the specific power is very important. At present, the studies for reducing the specific power are more concentrated on the intercooler, valves, gasflow pulse, however for this paper, is concentrated on the reduction of friction. The compressor power loss caused by friction are 5 - 15% 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%. 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 increases, consequently the capacity decreases. To prevent 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 \( q = \frac{N}{Q} \). The \( N \) consists of the indicated power \( N_i \) and the mechanical power \( N_m \) (mainly friction power). Reforming the formula as

\[
q = \frac{N_i}{Q} + \frac{N_m}{Q}
\]

Let

\[
q_i = \frac{N_i}{Q} \quad (1)
\]

\[
q_m = \frac{N_m}{Q} \quad (2)
\]

\( q_i, q_m \) represent the specific indicated power and the specific mechanical power separately so

\[
q = q_i + q_m \quad (3)
\]

The formula of indicated power of multistage compressor is derived by considering the material \( \xi \) (with the simplification \( \xi = \xi^{1/i} \), \( T_{s1}/T_{s1'} = 1 \))

\[
N_i = 1.634 \frac{m}{m-1} \frac{p_s}{Q} \xi \left[ \left( \frac{1}{1+\frac{\delta_d}{\delta_s}} \right) \frac{m-1}{m} - 1 \right]
\]

Substituting it into (1)
\[ q_i = 1.634 p_s \frac{m}{1 \text{-} m-1} \left\{ \left( \frac{1+\delta_d}{1-\delta_s} \right)^{m-1} - 1 \right\} \]  

(4)

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

\[ q_i = 1.634 p_s \frac{m}{1 \text{-} m-1} \frac{\kappa}{\lambda} \left( \frac{1}{1 \text{-} m} - 1 \right) \]

For a two stage compressor with a discharge pressure \( 9 \text{ atm} \) the \( \{ q_i \} = 4.22 \text{ KW/} \text{m}^3/\text{min} \). It is obviously when \( \delta_s \), \( \delta_d \) is approaching to zeroes a fangible reduction of \( q_i \) is too difficult to do.

Differetiate (2) and considering \( Q = \pi D^2 v \lambda / 4 \):

\[ dq_m = \frac{N_m}{Q} \left( \frac{dN_m}{N_m} - \frac{d\lambda}{\lambda} - \frac{2dD}{D} \right) \]

(5)

Based on (5), three approaches are available:

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

THE METHOD 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 compres-
singing and discharging processes and the sealing is no need for seeking 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 introdused bellow

\[ \eta = \frac{N_{rg}}{N_r} \]  

Where, \( N_{rg} \) — the friction power loss caused by friction sealing.

\( N_r \) — the total friction 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 against the cylinder wall, we name it as the first sealing, the \( p_g \) functions as sealing also. However it acts after establishing 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 \( \theta \), 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 \( \Delta p/2 = (p_g - p_b)/2 \). But 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 \( \theta \) 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 calculated through

\[ N_{rs} = 1DB\mu p_{rs} \]  

and that caused by \( p_g \)

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\[ N_{rg} = (D-2t)(p_{g1} + p_{g2} + \ldots + p_{gi})bV\mu \approx \frac{DbV\mu p}{2} \]  
(8)

The total
\[ N_r = N_{rs} + N_{rg} = DbV\mu (ip_{rs} + \Delta p/2). \]  
(9)

Substituting (8), (9) into (6)
\[ \eta = \frac{1}{1 + \frac{2ip_{rs}}{\Delta p}} \]  
(10)

The \( \Delta p \) in formula (10) is given, so the only way to raise \( \eta \) is to decrease \( i \) or \( p_{rs} \). From formulas (7), (9)
\[ dN_r = \frac{\partial N_r}{\partial i} di + \frac{\partial N_r}{\partial p_{rs}} = DbV\mu (p_{rs} di + ip_{rs}) \]  
(11)

Form (11) we can see that if the number of the piston rings is subtracted by \( \Delta i \), the compressor shaft power will decrease by \( DbV\mu p_{rs} \Delta i \); if reduce the elastic force by \( \Delta p_{rs} \), the shaft power will decrease by \( DbV\mu i \Delta p_{rs} \); the third point is that the reduction of the shaft power is proportional to its rotating velocity. 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.\(^{[1]}\)
   b) The leakage is small in a double acting compressor, the leakage is expressed as
   \[
   G = \rho f \sqrt{\frac{p_g}{RT}} \left[ 2gk \left( \frac{p_b}{p_g} \right)^{2/k} - \left( \frac{p_b}{p_g} \right)^{k+1} \right]^{k-1} \]  
(12)
The maximum of the leakage appears when

$$\frac{P_b}{P_g} \leq \left( \frac{2}{k+1} \right)^{k-1}$$

(13)

During compressing process the pressure ratio

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

Substituting them into (13), the displacement at which the maximum leakage speed will occur

$$x \geq (1+\alpha) \left[ 1 - \left( \frac{2}{k-1} \right)^{k-1} \right] s$$

(14)

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

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

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

From $G-\theta$ 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 abode in Fig. 4c. Fig. 5a, b, c show the $p_1-\theta$, $\Delta p-\theta$, $G-\theta$ of the double acting compressor. There are three situations: $\Delta p > 0$, $\Delta p = 0$, $\Delta p < 0$. The leakage occur 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 clearance expanding at the shaft side.
Generally considering the clearance volumes equal, so the area abc and efg are counteracted, i.e. the total leakage is cdeg during a circle. 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 exists during running. See Fig. 7. And single acting different to double, Fig. 8.

c) For a double acting compressor the lubricating 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 simulated according to the cylinder diameter, it is suitable 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 clearance volume reduces the volumetric efficiency of the compressor so it is pernicious. The suspension of the piston ring in the ring groove distroys the second sealing surface and increasing the leakage so it is termed pernicious 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 through the second sealing
surface and entering the compressing space to make the capacity increase. To realize this, the following relation must be satisfied

\[ 0^0 < \theta < \theta_a \quad \text{or} \quad 180^0 < \theta < \theta_b \]  

(15)

Where the \( \theta \) represents the crank shaft. \( \theta_a, \theta_b \) represent the crank shaft angles at the points where the pressure on the both sides are equal. See Fig. 9a. According to the expanding equation and compressing equation there are

\[ p_b = \left( \frac{a_s}{a_{s+x}} \right)^m \varepsilon^i p_{s1} \]  

(16)

\[ p_g = \left( \frac{a_{s+x}}{a_{s+x-x}} \right)^m \varepsilon^{i-1} p_{s1} \]  

(17)

Considering the piston displacement

\[ x = \frac{s}{2} (1 - \cos \theta) \]  

(18)

So

\[ \theta_a = \cos^{-1} \left[ 1 - \frac{2(1+\alpha)(\varepsilon^{1/m}-1)}{1+\alpha(1+\varepsilon^{1/m})} \right] \]  

\[ \theta_b = 180^0 + \theta_a \]  

(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 positive, when \( x^p = p_b - (p_g - p_m - p_a) \) is negative, the piston ring suspension will occur. where

\[ n_m = \frac{(D-2t)b^2 p_b + Db^2 p_{rs}}{\pi(D-t)t} \]

\[ p_a = \frac{mr^2 \cos \theta}{g\pi(D-t)t} \]
Considering (16), (17), (18), the relation to occur the suspension is

\[
\left(1 - \frac{1 - \cos\theta}{2(1 + \alpha)}\right) - \frac{1}{\varepsilon} p_{s1} - \frac{m\omega^2 \cos\theta}{g \pi (1 - e)} + \frac{D_b \mu p_{rs}}{\pi (D - e)}
\]

\[
> \left(1 - \frac{\theta}{\alpha}\right) (1 + \frac{1 - \cos\theta}{2\alpha}) - \frac{1}{\varepsilon} p_{s1}
\]

From Fig. 9b, the $z_p$ 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, $Q = 10^3 L/min$, $p_d = 9$ ata ). The methods stated above have been employed and the specific power reduced by 3 - 3.5 %.

CONSULTING DOCUMENTS


Fig. 1

Fig. 2a

Fig. 2b

Fig. 3
Fig. 4a
$P = \theta$

Fig. 4b
$\Delta P = \theta$

Fig. 4c
$G = \theta$

Fig. 5a
$P_1 = \theta$

Fig. 5b
$\Delta P = \theta$

Fig. 5c
$G = \theta$