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Friction Power in Sliding Vane Type Rotary Compressors

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1. Abstract

The aim of this paper is to calculate the lost power for blades into refrigerating rotary type compressor with R134a through friction stresses along one rotation of the compressor rotor. Friction stresses are calculated using their angle of rotation dependence for done compressor working regime. The compressor friction work value for vanes type rotary compressor along one rotation is get considering constant value for the friction coefficient and normal compression process.

2. Introduction

Conventional sliding vane types rotary compressor have free motion vanes built in rotor channels (Figure 1). Due to the eccentric assembly of the rotor into the stator cylinder and to the centrifugal forces action, vanes are moving in their channels and they are sticking to the stator cylinder sharing in cells the left space between the rotor, the stator cylinder and the vanes. The volume of one cell is changing continuously during one rotation of the rotor, which makes possible to reach stages of the compression process (aspiration, compression, discharge and expansion).

The volume of a cell as function of the rotation angle \( \phi \) is defined by [1,2] (Figure 2):

\[
V_\phi = R \cdot e \cdot L \cdot \left( \beta + 2 \sin \frac{\beta}{2} \cos \phi + \frac{1}{2} e_R \sin \beta \cos 2\phi - \frac{1}{2} e_R \beta \right)
\]

where: \( R \) - radius of the stator cylinder; \( e \) - eccentricity, \( e = R - r \); \( r \) - radius of the rotor; \( L \) - length of the rotor; \( \beta = 2\pi/Z \) -the angle between two consecutive cells; \( Z \) - the number of the vanes; \( e_R = e/R \) -relative eccentricity.

The maximum and the minimum values of the cells volume is achieved for \( \phi = 0 \) and \( \phi = \pi \), which means:

\[
V_{max} = R \cdot e \cdot L \cdot \left( \beta + 2 \sin \frac{\beta}{2} + \frac{1}{2} e_R \sin \beta - \frac{1}{2} e_R \beta \right)
\]

\[
V_{min} = R \cdot e \cdot L \cdot \left( \beta - 2 \sin \frac{\beta}{2} + \frac{1}{2} e_R \sin \beta - \frac{1}{2} e_R \beta \right)
\]

A simplified equation for the cells volume calculus as function of the rotation angle of the rotor is:

\[
V_\phi = R \cdot e \cdot L \cdot \beta \cdot \left( 1 + \cos \phi - e_R \sin^2 \phi \right)
\]

which goes to the equation:

\[
V_{max} = 2R \cdot e \cdot L, \quad \text{and} \quad V_{min} = 0
\]

Taking into account the vane's thickness the equation 4 becomes [1,3]:

\[
V_\phi = R^2 \cdot L \cdot \left( A + B \cos \phi + C \cos 2\phi \right)
\]

where: \( b_R = b/R \) - relative thickness of the vane, and the coefficients \( A, B, C \) are:
\[
A = e_R \left[ \beta (1-e_R/2) - b_R \left(1-e_R/4\right) \right]; \quad B = e_R \left(2 \sin(\beta/2) - b_R \cos(\beta/2) \right);
\]
\[
C = \frac{e_R^2}{2} \left(2 \sin \beta - \frac{b_R}{2} \cos \beta \right).
\]  

(7)

For this case the maximum and minimum values of the cells volume are:
\[
V_{\text{max}} = R^2 L \left(A+B+C\right), \quad \text{and} \quad V_{\text{min}} = R^2 L \left(A-B+C\right)
\]

(8)

To benefit of the main advantages of this rotary type compressor the problem is to evaluate the power loss due to the vanes friction, which has its importance on the global compressor power consumption. The way to get this power is presented below.

3. Loss power due to vanes friction

The vane has a complex motion which is resulted from the composition of the transport motion (motion of rotation) round about rotor axis and the relative motion (motion of translation) reported to the rotor. To acquire friction forces \(F_1, F_2\) and \(F_3\) (see Figure 3.) or normal reactive forces \(N_1, N_2\) and \(N_3\), it is necessary to establish which forces are operating on the vane as a function of the rotor rotation angle. The forces are presented as below:

- the pressure force \(P\); is estimated using the pressure difference \(\Delta p\) between two consecutive cells.

This pressure difference is applied on the free vane surface: 
\[
L \cdot (p - r) = L \cdot e \cdot \left(1 + \cos \phi - \frac{e_R}{2} \sin^2 \phi \right),
\]

and after that the pressure force will be:
\[
P = L \cdot e \cdot \left(1 + \cos \phi - \frac{e_R}{2} \sin^2 \phi \right) \cdot \Delta p
\]

(9)

The pressure difference for the rotation angle \(\phi \in \left[0, \pi \right]\) is calculated as \([1,2,4]\): 
\(\"\%\)

if one cell is at the end of the aspiration stage and the consecutive cell is at the beginning of the compression stage, that means the rotation angle is \(\phi \in \left[0, \beta/2 \right]\) and the pressure difference will be:
\[
\Delta p = p_1 \cdot \left\{ \frac{2}{1 + \cos(\phi + \beta/2) - e_R \cdot \sin^2 (\phi + \beta/2)} \right\}^k - 1 \}
\]

(10)

\(\"\%\)

if both cells are on the compression stage, the rotation angle is \(\phi \in \left[\beta/2, (\varphi_c - \beta/2) \right]\) and the pressure difference will be:
\[
\Delta p = p_1 \cdot \left\{ \frac{2}{1 + \cos(\phi + \beta/2) - e_R \cdot \sin^2 (\phi + \beta/2)} \right\}^k - \left\{ \frac{2}{1 + \cos(\phi - \beta/2) - e_R \cdot \sin^2 (\phi - \beta/2)} \right\}^k \}
\]

(11)

or:
\[
\Delta p = \beta \cdot \frac{dp}{d\phi} = k \cdot \beta \cdot p_1 \cdot \sin \phi \cdot (1 + 2 \cdot e_R \cdot \cos \phi) \frac{2^k}{\left(1 + \cos \phi - e_R \cdot \sin^2 \phi \right)^k} \}
\]

(12)

\(\"\%\)

if one cell is on the end of the compression stage and the consecutive cell is on the discharge stage, the rotation angle is \(\phi \in \left[(\varphi_c - \beta/2), (\varphi_c + \beta/2) \right]\) and the pressure difference will be:
\[
\Delta p = p_2 - p_1 \cdot \left\{ \frac{2}{1 + \cos(\phi - \beta/2) - e_R \cdot \sin^2 (\phi - \beta/2)} \right\}^k
\]

(13)
if the cell is on the discharge stage, the rotation angle is \( \varphi \in [ (\varphi_C + \beta/2), \pi ] \) and the pressure difference will be:

\[
\Delta p = 0
\]

where \( p_1 \) and \( p_2 \) are the aspiration and discharge pressures; \( k \) is the adiabatic exponent of the compression curve; \( \varphi_C \) is the rotation angle corresponding to the end of the compression process, which is defined by equation:

\[
\left( \frac{p_2}{p_1} \right)^{\frac{1}{k}} = \frac{2}{1 + \cos \varphi_C - e_R \cdot \sin^2 \varphi_C}
\]

- **the centrifugal force** \( F \); results from the rotation motion with angular speed \( \omega = \) constant, considered as a concentrated force and with the application point in the mass centre of the vane:

\[
F = m \cdot \omega^2 \cdot \left( \varphi - \frac{h}{2} \right) = m \cdot \omega^2 \cdot R \cdot \left( 1 - \frac{h_R}{2} + e_R \cdot \cos \varphi - \frac{e_R^2}{2} \sin^2 \varphi \right)
\]

where: \( m \) - mass of the vane [kg]; \( h_R \) - \( h/R \) relative height of the vane [m/m]; \( h \) - height of the vane [m]

- **the relative force** \( K \); results from the relative acceleration:

\[
K = m \cdot \omega^2 \cdot \left( \cos \varphi + e_R \cdot \frac{2 \cos^2 \varphi - 1}{T} + e_R^2 \sin^2 \varphi \cdot \cos^2 \varphi \right)
\]

\[
T = \sqrt{1 - e_R^2 \cdot \sin^2 \varphi}
\]

- **the Coriolis force** \( C \); results from the Coriolis acceleration:

\[
C = 2 \cdot m \cdot \omega^2 \cdot \left( \sin \varphi + \frac{e_R}{T} \cdot \sin \varphi \cdot \cos \varphi \right)
\]

The normal reactive forces \( N_1, N_2 \) and \( N_3 \) are in fact the normal reactive forces of the rotor on the vanes, and their values are explicit unknown. Using the equilibrium of forces on the vane direction and on the perpendicular vane direction, and also using the equilibrium of torque’s applied on the mass centre of the vane, the reactive forces will be:

\[
N_1 = \frac{-\mu_f C + (1 - f)(F + K) + \mu P}{E}
\]

\[
N_2 = \frac{-(1 - \mu^2)C + 2\mu(F + K) + [2 - f(1 + \mu^2)]P}{2E}
\]

\[
N_3 = \frac{(1 - \mu^2 - 2f)C + 2\mu F + K + f(1 + \mu^2)P}{2E}
\]

\[
E = (1 - f) - \mu^2(1 + f)
\]

where:

\[
f = \left( \frac{p - r}{h} \right) = \frac{\varphi \cdot \left( 1 + \cos \varphi - \frac{e_R^2}{2} \sin^2 \varphi \right)}{h} = \frac{e_R}{h_R} \cdot \left( 1 + \cos \varphi - \frac{e_R^2}{2} \sin^2 \varphi \right)
\]

It is pointed that equations at 9 to 22 were established using equation 4 for the cell volume, that leads to the remark that the normal reactive force \( N_1 \) is on the radial direction of the vane and the compressor works without a dead-space \((V_m = V_{min} = 0)\). So, for the rotation angle \( \varphi \in [\pi...2\pi] \) the pressure force doesn’t exist. It is assumed also that the friction coefficient \( \mu_1 \) between the vane and the stator cylinder, and the friction coefficient \( \mu_2 \) between the vane and the rotor are same, at a constant value, and the compressor working regime is that the calculus regime.

Lost mechanical work due to the vane friction at a complete rotation is:
\[ L_{j1} = \mu \int_{0}^{t} N_1 \rho d\phi, \]

\[ L_{j2,3} = \mu \left( -\int_{0}^{t} N_{2,3} d\phi + \int_{x}^{t} N_{2,3} d\rho \right) \quad (23) \]

where: \( \rho \) is the variable radius at the top of the vane, and its equation is:

\[ \rho = R \left( 1 + e_R \cos \phi - \frac{e_R^2}{2} \sin^2 \phi \right), \]

and \( d\rho = -e \sin \phi (1 + e_R \cos \phi) d\phi \), is the infinitesimal variation of the radius.

Total lost mechanical work due to vane friction, for all \( Z \) vanes, is:

\[ P_F = \frac{Z \cdot n}{60 \cdot 10^3} \sum_{j=1}^{Z} L_{j1} \quad \text{[kW]} \quad (24) \]

where \( n \) is the rotational speed [rot min\(^{-1}\)].

To show up the influence of the friction coefficient on the compressor power this study has performed on the refrigeration compressor, with following technical characteristics:

- Theoretical volumic flow \( V_t = 38.8 \text{[m}^3\text{h}^{-1}] \); rotational speed \( n = 2940 \text{[rot min}^{-1}] \); stator cylinder radius \( R = 0.026 \text{[m]} \); relative eccentricity \( e_R = 0.11 \); relative length \( L_R = L/R = 3 \); relative height \( h_R = 0.44 \); number of vanes \( Z = 10 \); vane's weight \( m = 0.007 \text{[kg]} \).

For 4 compression ratios \((p_2/p_1 = 2, 3, 4, 5 \text{ and 6})\) the results of the study, like the relative compressor power loss \((P_f/P_{ad})\) as function of the friction coefficient \( \mu \), are shown in Figure 4., using that the adiabatic power of the theoretical compressor, which is:

\[ P_{ad} = \frac{k}{k-1} p_t V_t \left[ \left( \frac{p_2}{p_1} \right)^{k-1} - 1 \right] \cdot \frac{1}{36} \cdot 10^{-5} \text{[kW]} \quad (25) \]

4. Conclusions

The relative loss compressor power due to vanes friction, as shown in Figure 4, is strongly affected by the consumed power of the compressor, and this influence increases with the friction coefficient rise and decreases with the compression ratio rise. In some circumstances the consumed power of the compressor can be bigger than 50% of the adiabatic power. To reduce the influence of the friction coefficient on consumed power of the compressor it is imposed a plentiful lubrication of the compressor. Indirectly, this influence can be decreased by using the proper materials and the compressor constructive scheme with inclined vanes.

5. References

Figure 1. The construction scheme and the compression process for vane type rotary compressor.

Figure 2. The main constructive dimensions.

Figure 3. The forces which are acting on the single vane.

Figure 4. The relative power loss of the vane type rotary compressor.