The Estimation of the Frictional Losses of the Rolling Piston Type Refrigerant Compressors

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TBSTRACT

This paper is concerned with the frictional losses in rolling piston type refrigerant compressors. The forces balance equations, the frictional forces and the frictional losses of each part of the compressor have been studied. The comparisons of the computation results with the exprimental results show good correlation.

NOMENCLATURE

R_r	radius of roller m
P_d	compressed gas pressure P_a
V_o	displacemont volume m
R_c	radius of cylinder m
L_s	length of vane m
θ	crank angle rad
μ, μ_1	the coefficient of boundary lubrication
C_3	the clearance between the roller and the eccentric shaft m
P_γ	the oil film pressure in the point that arbitrary angle in infinite bearing.
Z	the coordinate along the bearing width
ψ	relative clearance
B	width of bearing m
η	the viscosity of lubricant in the mean temperature of lubricant inlet and outlet Pa.s
ρ	the density of lubricant, for mineral oil, = 850 950, kg/m
W_s	the weight of eccentric shaft and rotor kg
R_{s2}	the outer radius of thrust bearing m
W_r	the weight of roller kg
R_{r1}	the inner radius of roller end m
w_p	the absolute angular velocity of roller 1/s
At	total area of the balancer ends normal to the tangential line of the rotor m
n_s, n_1	the rotating speed of the motor R.P.S.
P_3	the percentage of the frictional losses between shaft and main bearing in total frictional losses
P_4	the percentage of the frictional losses between roller and eccentric shaft in total frictional losses
P_6	the percentage of the frictional losses between lower roller end and cylinder plate end in total frictional losses
P_8	the percentage of frictional losses between vane and cylinder slot in total frictional losses
R_o	vane nose radius m
P_s	suction pressure P_a
H	height of cylinder, width of vane m
w	thickness of vane m
e	shaft eccentricity
k_s	vane spring constant
I_p	roller mass moment of inertia
P_f'	the oil film pressure in the point that distance from middle line is Z, arbitrary angle , (in finite bearing)
k_B	the coefficient of oil film pressure drop when considering end leakage, it is a function of eccentricity and the ratio of width-diameter.
F	the summary of oil film pressure (vertical direction), the load that bearing can bear.
V	the tangential velocity of journal m/s
C_p	the specific heat capacity, for mineral oil, C_p = 168 2100, J/kg C
k_s	the coefficient of heat transfer, for the light load or there are difficuty in heat trarsfer, k_s=54 J/m s C
R_{s1}	the inner radius of thrust bearing m
R_{r2}	the outer radius of roller end m
V_1	the sliding velocity of vane in the m/s cylinder slot m/s
V_∞	velocity of refrigerant m/s
C_D	drag force coefficient
D_R	average diameter of balancer
P_1	the percentage of the frictional losses between vane tip and roller in the total frictional losses
P_2	the percentage of the frictional lesses betwcen shaft and sub bearing in total frictional losses
P_5	the percentage of the frictional losses between upper roller end and cylinder plate end in total frictional losses
P_7	the percentage of frictional losses between roller and cylinder wall in total fractional losses
P_9	the percentage of windage losses between balancer end and refrigerant gas in total frictional loss
P_{10}	the percentage of frictional losses between sheft and cylinder plate end (thrust dearing) in total frictional losses

INTRODUCTION

Rolling piston type refrigenant compressors have been used in air conditioning units and household refrigerators.

Since energy consumption is an important feature of those compressors, strenuous efforts have been made to estimate the frictional losses of these type compressors. In this paper, special attention was paid to the frictional losses of main bearing and sub bearing. The theory of finite length bearings were applied and two assumptions were made:

1. The compression process begins from $\theta = 0$.
2. The eccentricity between the main bearing and the sub bearing is neglected.

THE ANALYSIS OF FORCES

1. Vane

The magnitude of forces are as follows (Fig. 1):

$$F = P_\theta H\{(e + R_o + R_r) - [e\cos\theta + (R_o + R_r)\cos\alpha]\} \quad (1)$$

where $\alpha = \arc\sin\left(\frac{\varepsilon\sin\theta}{R_r + R_o}\right)$

$$V_p = \left\{\frac{R_c^2(2\pi-\theta)}{2} - \frac{R_r^2(2\pi-\psi)}{2} + \left(\sqrt{R_r^2 - e^2\sin^2\theta} + e\cos\theta\right)\frac{e\sin\theta}{2}\right\}H \quad (2)$$

$$V_o = \pi H(R_c^2 - R_r^2) \quad (3)$$

$$F_4 = P_c W H \quad (4)$$

$$F_3 = P_s\left[\frac{W}{2} - \frac{R_o e\sin\theta}{(R_o + R_r)}\right]H \quad (5)$$

$$F_9 = P_\theta\left[\frac{W}{2} + \frac{R_o e\sin\theta}{(R_o + R_r)}\right]H \quad (6)$$

$$F_{10} = m_v[e\omega^2\cos\theta + (R_o + R_r)\dot{\alpha}^2\cos\alpha] \quad (7)$$

$$F_{17} = F_s L_7 H \quad (8)$$

$$F_{18} = P_D L_7 H \quad (9)$$

$$F_{19} = \frac{(P_D - P_s)}{2} L_7 H \quad (10)$$

$$F_{20} = \frac{(P_c - P_D)}{2} L_7 H \quad (11)$$

$$F_7 = k_s[2e - (e + R_o + R_r) + (e\cos\theta + (R_o + R_r)\cos\alpha)] \quad (12)$$

Combinating with equations (1),(2),(3) and (4) yields the following equations

$$B(1) = F_4 + F_7 - F_{10} - F_9 - F_8$$

$$B(2) = F_{17} - F_{18} + F_{19} - F_{20} - F_1$$

$$B(3) = F_{18}\frac{L_7}{2} - F_{17}\frac{L_7}{2} + F_1 L_6 + F_{20}\frac{L_7}{3} - F_{19}\frac{L_7}{3}$$

$$F_5 = \frac{B(1)/\mu_1 - B(2) - 2B(3)/L_7}{(2L_5/L_7 - 1)\cos\alpha - \sin\alpha/\mu_1 - \frac{(2L_5/L_7 - 1)\sin\alpha + \cos\alpha/\mu_1}{\mu_1}} \quad (\alpha > 0^\circ) \quad (13)$$

$$F_5 = \frac{-B(1)/\mu_1 - B(2) - 2B(3)/L_7}{(2L_5/L_7 - 1)\cos\alpha - \sin\alpha/\mu_1 - \frac{(2L_5/L_7 - 1)\sin\alpha + \cos\alpha/\mu_1}{\mu_1}} \quad (\alpha < 0^\circ) \quad (14)$$

$$F_6 = \frac{|F_5|}{\mu_1} \quad (15)$$

$$F_2 = \frac{B(3)}{L_7} + \frac{L_5}{L_7}(F_5\cos\alpha - F_6\sin\alpha) \quad (16)$$

$$F_3 = B(2) + B(3)/L_5 + F_2 - F_2 L_7/L_5 \tag{17}$$

$$F_4 = 2(P_\theta - P_S) H R_r \sin\frac{\psi}{2} \tag{18}$$

$$F_{13} = \sqrt{[F_6 \cos(\alpha + \pi/2) + F_{11} \cos(3\pi/2 + \psi/2 - \theta]^2 + [F_{11} \sin(\alpha + \pi/2) + F_{11} \sin(\frac{3\pi}{2} + \frac{\psi}{2} - \theta)]^2} \tag{19}$$

The azimuth angle is as follows

$$\sigma = tg^{-1} \frac{F_6 \sin(2 + \pi/2) + F_{11} \sin(3\pi/2 + \psi/2 - \theta)}{F_6 \cos(2 + \pi/2) + F_{11} \cos(3\pi/2 + \psi/2 - \theta)} \tag{20}$$

2. Rolling piston

Fig. 2 shows the forces applied to the rolling piston. The momental balance equation is as follows

$$I_p \dot{\omega}_r = M_c - R_r F_5 - M_b \tag{21}$$

where M_b is the frictional moment from cylinder wall to rolling piston surface.

The frictional moment M_c from eccentric to roller is as follows:

$$M_c = \frac{2\pi\eta(\omega - \omega_p) H_3 R_e^3}{c_3 (1 - \varepsilon^2)^{1/2}} - \frac{1}{2} c_3 \varepsilon F_{13} \sin\varphi \tag{22}$$

According to references [1] [2], we take the ε and $tg\varphi$ as follows:

$$\varepsilon = 0.65 + 0.33 \cos\theta \tag{23}$$

$$tg\varphi = \frac{\pi\sqrt{1 - \varepsilon^2}}{2\varepsilon} \tag{24}$$

THE FRICTIONAL LOSSES IN THE JOURNAL BEARINGS

The follow assumptions were made:

1. Lubricant seems as Newtonian fluid.
2. Film is so thin that the fluid flow is laminar.
3. The viscosity of oil depends on the temperatur and pressure and will keep constant in the whole oil film.
4. The Reynolds equation of the journal bearing is simplifyed to be the one order ordinary differential equation and Gumbel's boundary condition is available.

Fig. 3 shows the geometrical relationship of the journal bearing. The follow parameters are used: The clearance $\Delta = D - d$, the radius clearance $\delta = \Delta/2$, the relative clearance $\psi = \delta/r$, the attitute of bearing $\varepsilon = e/d$. In this Figure, the attitude angle θ is the included angle between the action line of load Fr and the connecting line $\overline{oo'}$. ϕ is the included angle between the connecting line $\overline{Ao'}$ and $\overline{oo'}$. The oil film thickness h at point A equals $\delta(1 + \varepsilon\cos\varphi)$, and the oil film at the maximum pressure point ($\varphi = \varphi_0$) is $h_0 = \delta + (1 + \varepsilon\cos\varphi_0)$. The Reynolds equation in the polar coordinate system is as follows:

$$dp = \frac{6\eta V}{r\psi^2} \frac{\varepsilon(\cos\varphi - \cos\varphi_0)}{(1 + \varepsilon\cos\varphi)^3} d\varphi \tag{25}$$

By intergrating equation (25), the oil film pressure at arbitrary angle φ can be expressed as follows

$$p_\varphi = \int_{\varphi_1}^{\varphi} dp = \int_{\varphi_1}^{\varphi} \frac{6\eta V}{r\psi^2} \frac{\varepsilon(\cos\varphi - \cos\varphi_0)}{(1 + \varepsilon\cos\varphi)^3} d\varphi \tag{26}$$

For a finite length bearing, the pressure distribution of the oil film is as shown in Fig. 4.

The oil film pressure $P_{\varphi'}$ (at the position z, arbitrary angle φ) and its vertical component $P_{\varphi' y}$ (along the direction of load action line) are as follows:

$$p_{\varphi'} = p_\varphi K_B [1 - (\frac{2z}{B})^2] \tag{27}$$

$$p_{\varphi' y} = p_{\varphi}' \cos[\pi - (\varphi + \theta)] \tag{28}$$

where $1-(\frac{2Z}{B})$ is the modification of the oil film pressure for the finite length bearing and for the parabola distriution pressure.

The sum pressure F can be given by integrating φ and Z.

$$F=\int_{-\frac{B}{2}}^{\frac{B}{2}}\int_{\varphi_1}^{\varphi_2}P'_{yy}\,r\,d\varphi\,dz=\frac{2\eta VB}{\psi^2}\left\{-2\int_{\varphi_1}^{\varphi_2}\left[\int_{\varphi_1}^{\varphi_2}\frac{\varepsilon(\cos\varphi-\cos\varphi_0)}{(1-\varepsilon\cos\varphi)^2}d\varphi\right]k_B\left[\cos(\varphi+\theta)d\varphi\right]\right\} \tag{29}$$

Let $C_F=\left\{-2\int_{\varphi_1}^{\varphi_2}\left[\int_{\varphi_1}^{\varphi_2}\frac{\varepsilon(\cos\varphi-\cos\varphi_0)}{(1-\varepsilon\cos\varphi)^2}d\varphi\right]k_B\left[\cos(\varphi+\theta)d\varphi\right]\right\}$

then

$$C_F=\frac{F_r\,\psi^2}{2\eta VB} \tag{30}$$

C_F is defined as the coefficient of load-supporting, or the Summerfeld number. In this paper, C_F is taken from reference [3] since it is difficult to integrate.

The frictional losses for a dynamic pressure liquid lubricated bearing consist two parts, one is in the region of load-supporting. The ratio of the coefficient of friction f in the region of load-supporting to the relative clearance ψ is known as the natural parameter of friction

$$C_f=\frac{f}{\psi} \tag{31}$$

for this bearing which angle is 360 , the value of C_f' can be got from reference [3].

The natural parameter of friction C_f' for the coefficient of friction f' (in the nonload-supporting region) is

$$C_f'=\frac{f'}{\psi}=k_n\frac{\pi}{2C_F} \tag{32}$$

where k_n is the coefficient and we take it as me.

Finally, the frictional loss of the bearing can be expressed as follows:

$$N=\left(\frac{f}{\psi}+\frac{f'}{\psi'}\right)\psi F_r V \tag{33}$$

Fig. 5 shows the forces applied to the main bearing and the sub-bearing.

$$F_{13}=F_{mc}+F_{mf} \tag{34}$$

let $$C_1=\frac{l_{mc}}{l_{mf}} \tag{35}$$

then $$F_{mf}=F_{13}\cdot\frac{C_1}{1+C_1} \tag{36}$$

$$F_{mc}=\frac{F_{13}}{1+C_1} \tag{37}$$

The constructional parameters of the bearings are shown in table 1.

table 1

	shaft diameter (mm)	bearing length (mm)	Maximum radital clearance (mm)	Width-length ratio B/d
main bearing	20	44	0.01E-3	2.2
sub bearing	20	22	0.015E-3	1.1
eccentric bearing	31	17.5	0.011E-3	0.565

The frictional losses in the main bearing, sub-bearing and eccentric bearing are as follows:

$$L_{bf} = \left(\frac{f_1}{\psi_1} + \frac{f_1'}{\psi_1}\right)\psi_1 \omega R_s F_{m1} \quad (38)$$

$$L_{bc} = \left(\frac{f_2}{\psi_2} + \frac{f_2'}{\psi_2}\right)\psi_2 \omega R_s F_{mc} \quad (39)$$

$$L_{br} = \left(\frac{f_3}{\psi_3} + \frac{f_3'}{\psi_3}\right)\psi_3 (\omega - \omega_p) R_r F_{13} \quad (40)$$

In the above Analysis, the temperature rise of lubricant oil Δt can be determined by the follow heat balance equation:

$$N = c \rho Q \Delta t + A k_s (t_2 - t_1) \quad (41)$$

THE MECHANICAL LOSSES IN THE OTHER PARTS OF THE ROLLING PISTON TYPE REFRIGERANT COMPRESSOR

The mechanical losses are mainly caused by rubbing surfaces with oil films, besides this, there is a aerodynamic losses between the balance weight to the refrigerant gas.

The following assumptions are made in order to simplify the calculation.

(1) Only the fluid or boundary friction are considered.

(2) Coefficient of friction for boundary friction is always constant.

1. Upper roller end and cylinder plate end

$$L = \frac{\pi \eta \omega_r^2 e^3}{h}(R_{r2}^2 - R_{r1}^2) \quad (42)$$

2. Shaft and cylinder plate end

$$L = \frac{2\mu W_s}{3} \frac{(R_{s2}^3 - R_{s1}^3)\omega}{R_{s2}^2 - R_{s1}^2} \quad (43)$$

3. Lower roller and cylinder plate end

$$L = \frac{2\mu W_r}{3} \frac{(R_{r2}^3 - R_{r2}^2)\omega}{R_{r2}^2 - R_{r1}^2} \quad (44)$$

4. Vane and cylinder slot

$$L = (|F_2| + |F_3|) V \quad (45)$$

5. Vane tip and roller

$$L = F_6 V \quad (46)$$

$$V = \omega_p R_r + e\omega \cos(\theta + \alpha) \quad (47)$$

6. Balance weight and refrigerant gas

$$L = \frac{\rho}{2} C_D (\pi D_R n)^3 A_r \quad (48)$$

For each rotating angle, the frictional forces and relative velocities between two contact surfaces are not constant, so, the frictional losses vary according to the rotating angle. If we defind L_s as a losses during each degree of rotating angle

$$L_s = \frac{\vec{F}\,\vec{V}}{360\, n_s}$$

then the total frictional loss during one rotation is

$$L_s = n_s \sum_{0}^{360} \frac{(\vec{F}\,\vec{V})}{360\, n_s}$$

Applying the above equations to a rolling piston type refrigerant compressor under different operating condition, the results are shown in table 2.

Table 3 lists the different mechanical losses in the same operating condition when the clearance of the main bearing changes. The operating conditions are listed in table 4.

table 2

operating condition	mechanical losses W	pressure ratio	P %	P %	P %	P %	P %	P %	P %	P %	P %	P %
No. 1	20.47	2.784	4 7768	5.2967	3.0324	3.8672	0.0026	0.0185	4.8476	74.999	0.2501	2.917
No. 2	19.06	2.39	4.3999	6.9587	3.9738	4.6069	0.0038	0.0206	6.6928	69.9345	0.2686	3.140
No. 3	18.24	1.957	4.0104	8.4004	4.6582	5.4167	0.0047	0.0218	8.137	69.7803	2.3807	
No. 4	22.4	3	4.954	4.233	2.4357	3.577	0.0018	0.0167	3.7639	78.1354	0.2285	

table 3

	clearance mm	total mechanical loss W	P %	P %	P %	P %	P %	P %	P %	P %	P %
main	0.01 E-3	22.42	5.031	4.2295	2.4338	3.5741	0.0018	0.0166	3.7609	78.072	0.2284
bearing	0.008 E-3	22.79	4.9497	5.7774	2.3944	3.5164	0.0018	0.0164	3.7001	78.8102	0.2247
sub	0.015 E-3	22.42	5.031	4.2295	2.4338	3.5741	0.0018	0.0166	3.7609	78.072	0.2284
bearing	0.01 E-3	22.59	4.9931	4.1977	3.1687	3.5472	0.0018	0.0165	3.7325	17.4857	
eccentric	0.011 E-3	22.42	5.031	4.2295	2.4338	3.5741	0.0018	0.0166	3.7609	78.072	
bearing	0.008 E-3	22.41	5.1279	4.231	2.4346	3.4194	0.002	0.0173	3.7075	78.099	

table 4

rating	suction pressure bar	discharge pressure bar	pressure ratio	suction temperature C	input power W	refrigerating capacity	E.E.R	volumetic efficiency %
No. 1	3.538	9.849	2.784	11.7	49308	1775.0	3.60	0.855
No. 2	4.018	9.604	2.39	12.3	507.6	2037.91	4.05	0.862
No. 3	5.008	9.8	1.957	11.4	49826	2592.9	6.06	0.874
No. 4	3.979	11.936	3	12.2	58913	1776.4	3.02	0.792

table 5

operating condition	pressure ratio		indicated power W		frictional power W	shaft power W		motor input power W	motor efficiency %		mechanical coso %	
	compu.	experi.	compu.	experi.	compu.	compu.	experi.	experi	compu.	experi.	compu.	experi.
No. 1	2.784	2.784	352.88	348.44	23.47	373.35	368.91	493.08	75.7	74.8	5.5	5.55
No. 2	2.39	2.39	335.57	323.77	19.06	354.63	341.83	507.6	70.0	67.3	5.4	5.58
No. 3	1.957	1.957	321.14	316.53	18.24	339.38	334.77	478.26	71	70	5.37	5.45
No. 4	3	3	425.59	420.53	22.4	447.99	442.93	589.13	76	25.2	5.0	5.06

Table 5 lists the comparisons of the simulative computation with the experimental results.

CONCLUSION

(1) The increase the radial clearance of bearing leads to decrease the lubricant outlet temperature and to increase lubricant leakage, but only a littld change of the total power consumption.

(2) The ratio of width-diameter of main bearing in rolling piston type rotary compressor is about 2 ~ 2.2, this value is vary few to use in the other machine. The ability of load-supporting of bearing is not proportional to the width of bearing. To decrease the ratio of with-diameter will improve the load ability and the operating stability of the bearing. It is better to decrease the width-diameter ratio to 1.0 ~ 1.5.

(3) High viscosity lubricant oil is common used in rolling piston type rotary compressor, it will lead to increase the thickness of the oil film, and increase the power consumption.

(4) The magnitude and the direction of the bearing load in rolling piston type rotary compressors are changed at all time, therefore, the shaft center never stay in a fixed place. The dynamic pressure film is always at the nonstability state. It is better to use the spirality oil groove than to use the straight line type

oil groove for lubricating.

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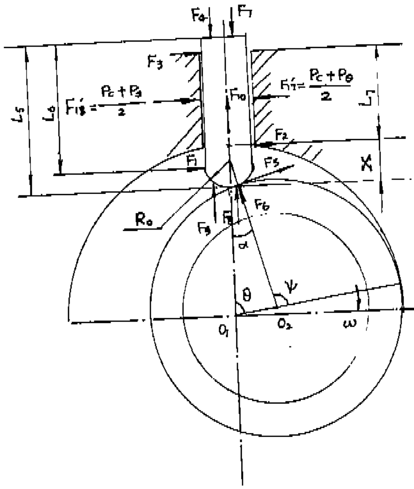

Fig.1 Forces Acting on The Vane

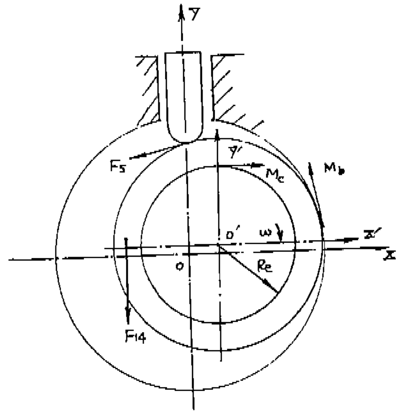

Fig.2 Forces Acting on The Roller

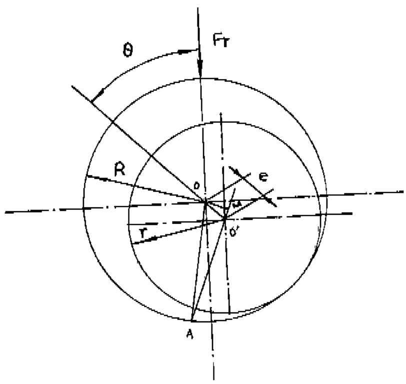

Fig.3 The Geometrical Relationshap of The Journal Bearing

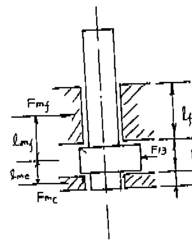

Fig.5 Force Acting on The Main Bearing And The Sub-bearing

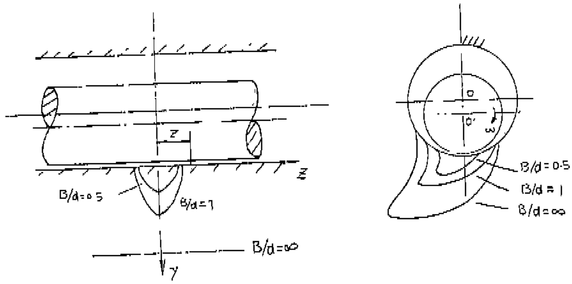

Fig. 4 The Pressure distribution of the oil film