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Rolling Piston Type Rotary Compressors with Special Attention to Friction and Leakage

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ROLLING PISTON TYPE ROTARY COMPRESSORS WITH SPECIAL ATTENTION TO FRICTION AND LEAKAGE

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INTRODUCTION

Rolling piston type rotary compressors, also known as rollator compressors, are similar to rotary-vane type compressors in many respects; for instance, the compression process in both the machines is accomplished by a piston and cylinder arrangement that employs circular or rotary motion instead of the usual reciprocating motion. However, the significant difference between the two designs lies in the kinematics of the mechanism of operation. While in a rotary-vane compressor, the rotor, mounted eccentrically, rotates within the cylinder; in case of rolling piston compressor the rotor (or piston) "rolls" on the inside surface of the cylinder -- hence the name. A cross section of one such compressor is shown in figure 1.

While rotary-vane compressors have been analysed by investigators in the past [1,2, 3], practically nothing is available in the open literature about the rolling piston type compressors, although no one knows what is hidden behind the vaults of the industry. Perhaps the similarity in the general principle of operation and somewhat similar geometry has prompted the investigators to side-track the essential differences between the two designs.

This paper attempts to identify the loss mechanisms in a rolling piston type compressor. Kinematics of the compressor will be analysed and volume-angle and pressure-angle relationships developed. Various losses will be identified and those that depend inherently upon the particular design of this compressor and have no similarity to other designs will be investigated. Suitable approximations will be made so as to make the various loss-formulas convenient for everyday use either by a compressor designer or an application engineer or a systems analyst. Emphasis will be on the simplicity and practical usefulness of the final formulas rather than the mathematical rigor and the simulation approach.

Various losses in a hermetic compressor have been divided into two broad categories: (1) the energy losses and (2) the mass flow losses (see also reference [10]). They are:

A. Energy Losses

(1) Motor loss

(2) Friction loss in the bearings

(3) Friction loss in the cylinder

(4) Compression loss (due to the compression process not being ideal)

(5) Valve loss (includes gas pulsations, overcompression, etc.)

(6) Lubricant pumping loss

B. Mass Flow Losses

(1) Clearance volume loss

(2) Leakage loss

(3) Back-flow loss

(4) Suction-gas heating loss

(5) Loss due to lubricant flow

The analysis of some of the losses listed above is similar in approach to other compressors, for instance motor loss, loss in bearings, clearance volume loss, etc. and this will not be discussed here to save time and space. The most significant losses that are inherently different in this particular design from other designs are: (1) friction loss in the cylinder and (2) leakage loss. In the following, these two losses will be analysed and their simplified formulas developed.

KINEMATICS OF THE ROLLING PISTON TYPE COMPRESSOR

Description of the Compressor:

Figure 1 shows a schematic sketch of a typical rolling piston type rotary compressor. The roller (or the rolling piston) is mounted on a shaft having an eccentric. The shaft rotates about the center of the cylinder and the roller rolls over the inside surface of the cylinder, thereby rotating about the eccentric. If the tolerances were perfect, the roller would have a perfect rolling motion. However, in practice this does not happen and the roller has a complicated motion consisting of rolling coupled with the slipping motion relative to the cylinder.

Angular Speed of the Roller:

Thus the motion of the roller can be considered as consisting of two different types of motions superimposed over each other. (1) Pure rolling motion and (2) slipping motion. Referring to figure 2, if the motion were of pure rolling type, we would have following relationships:

$$R_c\theta = R_r\psi \tag{1}$$

$$\psi = \frac{R_c}{R_r}\,\theta = \frac{1}{a}\,\theta \tag{2}$$

Then, rotation of roller about its center because of pure rolling motion in the positive θ-direction will be given by $(\theta-\psi)$, or

$$\theta-\psi = \theta(1 - \frac{1}{a}) \tag{3}$$

where a is the radius ratio (R_r/R_c). Therefore, roller speed due to rolling motion,

$$\omega_2 = (\dot{\theta}-\dot{\psi}) \tag{4}$$

or $$\omega_2 = \dot{\theta}(1 - \frac{1}{a}) = \omega_1(1 - \frac{1}{a}) \tag{5}$$

where ω_1 is the motor speed [rad/sec]. Now suppose ω_3 [rad/sec] is the angular speed of slipping only in the positive θ-direction. Then the net angular speed (ω_4) of the roller will be given by:

$$\omega_4 = \omega_2 + \omega_3 = \omega_1(1 - \frac{1}{a}) + \omega_3 \tag{6}$$

This, then, is the absolute angular speed of the roller. However, the speed of roller about its center or its speed relative to the eccentric (ω_r) will be $(\omega_4-\omega_1)$, or,

$$\omega_r = \omega_4 - \omega_1 = \omega_3 - \frac{\omega_1}{a} \tag{7}$$

Since ω_1 is already known (speed of the motor), the only unknown in equations (6) and (7) is ω_3, which will be obtained as shown later from the free-body analysis of the roller once the friction forces have been analysed.

Volume-Angle and Pressure-Angle Relationships:

Referring to Fig. 3, following geometrical relationships can be easily established:

$$R_r^2 = e^2 + R^2 - 2eR\cos(\theta-\phi) \tag{8}$$

or

$$R = R_c(1-a)\cos(\theta-\phi) \pm \sqrt{R_c^2(1-a)^2\cos^2(\theta-\phi) + R_c^2(2a-1)} \tag{9}$$

Since the radius ratio a is slightly less than 1 in all practical designs, the quantity (2a-1) will always be a positive number. Therefore, the quantity under the radical sign will always be greater than the term outside the radical sign. Hence, we must neglect the negative sign between the two terms as physically meaningless. Thus, we can write:

$$R = R_c[(1-a)\cos(\theta-\phi) + \sqrt{(1-a)^2\cos^2(\theta-\phi) + (2a-1)}\,] \tag{10}$$

To find the total area trapped between the roller and the cylinder, we will integrate an elemental area dA as shown, from $\phi = 0$ to $\phi = \theta$. Thus,

$$A(\theta) = \int_0^\theta dA = \int_0^\theta \frac{1}{2}(R_c^2 - R^2)\,d\theta \tag{11}$$

Substituting R from Eq. 10 into Eq. 11 and integrating, we finally get the following relationship:

$$A(\theta) = \frac{R_c^2}{2}\Big[(1-a^2)\theta - \frac{(1-a)^2}{2}\sin 2\theta - a^2\sin^{-1}[(\frac{1}{a} - 1)\sin\theta] - a(1-a)\sin\theta\sqrt{1 - (\frac{1}{a} - 1)^2\sin^2\theta}\,\Big]$$

or

$$A(\theta) = \frac{1}{2}R_c^2 f(\theta) \tag{12}$$

where,

$$f(\theta) = \Big[(1-a^2)\theta - \frac{1}{2}(1-a)^2 \sin 2\theta - a^2 \sin^{-1}\big[(\frac{1}{a} - 1)\sin\theta\big] - a(1-a)\sin\theta\sqrt{1-(\frac{1}{a} - 1)^2 \sin^2\theta}\Big] \quad (13)$$

The volume $V(\theta)$ will then be given as follows:

$$V(\theta) = hA(\theta) = \frac{1}{2} hR_c^2 f(\theta) \quad (14)$$

So far we have neglected the effect of blade thickness in the above discussion. However, for accurate analysis, the blade effect can be considered as follows:

True value of $V(\theta)$

$$= \text{apparent value of } V(\theta) - \frac{1}{2} th\delta_e$$

where δ_e is the blade extension and t is the blade thickness. But

$$\delta_e = [R_c - (R)_{\phi=0}]$$

or

$$\delta_e = R_c\Big[1 - (1-a)\cos\theta - \sqrt{(1-a)^2\cos^2\theta + 2a - 1}\Big] \quad (15)$$

Then

$$V'(\theta) = V(\theta) - \frac{1}{2} th\delta_e \quad (16)$$

where $V(\theta)$ is given by Eq. (14) and δ_e by Eq. (15). Since the compression process can be represented by the equation PV^n = constant, where n is the coefficient of polytyropic compression, we can express pressure corresponding to any angle θ as follows:

$$P(\theta) = P_s\left[\frac{V'(2\pi)}{V'(\theta)}\right]^n \quad (17)$$

FRICTION LOSSES IN CYLINDER

Referring again to Fig. 1, it can be easily seen that the possible friction loss mechanisms in the cylinder are: (1) friction between the blade tip and the roller, (2) friction between the cylinder-head and the two faces of the roller, (3) friction between the cylinder-head and the two faces of the eccentric, (4) friction at the point of contact between the roller and the cylinder, (5) friction between the roller and the eccentric, and (6) blade-slot friction. The last one can be conveniently disregarded since there will not be any appreciable side pressure between the blade and the slot, thereby rendering the friction force between the blade and the slot negligible. Also, since relative motion between cylinder and roller is rolling plus sliding without any appreciable contact force (the only force is due to centrifugal effect of roller mass which is negligible), it is assumed that energy loss due to friction between roller and cylinder at the contact point will be negligible. The remaining four losses can be treated as follows.

1. Blade tip friction:

The relative motion between the blade tip and the roller is of pure sliding nature, with the speed of rotation of the roller given by Eq. (6) and lying somewhere between zero and the eccentric velocity ω_1. In practice it is only a small fraction of ω_1, as we shall see later. Moreover, the spring force together with the under-blade oil pressure acting behind the blade (see Fig. 4) give rise to considerably high unit pressure between the blade tip and the rotor. These considerations suggest that the lubrication between the blade-tip and the roller surface is possibly of mixed nature and is somewhere between the boundary lubrication with a very thin film of lubrication between the two surfaces and the hydrodynamic lubrication with full fluid film between the two surfaces. Therefore, the friction loss can be calculated by assuming a suitable coefficient of friction (f). For this type of application f can be taken from .008 to .02 [4]. An average value of .015 seems to be reasonable for most cases. From the free-body diagram of the blade (Fig. 4) we notice that the blade will exert a force on the roller that will vary continuously from a maximum of ($\Delta Pth + k\, x_{max}$) when the spring is in fully compressed position to a possible minimum of (ΔPth) when the spring is in its free position, where x is the spring compression. Moreover, since ΔP and x vary continuously, the computations will go out of hand if the most rigorous approach were adopted. The analysis will, therefore, be approximated by taking the average values for both ΔP and x. Thus

Average friction force F

$$= f\Big((\Delta P)_{av}th + k\, x_{av}\Big) \quad (18)$$

where

$$(\Delta P)_{av} = \frac{1}{2}\Big[P_s + \frac{1}{2}(P_s + P_d)\Big] = \frac{3\, P_s + P_d}{4} \quad (19)$$

$$x_{av} = \frac{1}{2}\, x_{max} \quad (20)$$

Then loss of power due to blade-tip friction is given by:

$$\dot{E}_{FL_1} = R_r \omega_4 f \left[\frac{1}{4}(3P_s+P_d)\,th + \frac{1}{2}k\, x_{max}\right] \qquad (21)$$

2. Roller-to-Cylinder Head Friction:

The clearance space between the roller faces and the cylinder head is assumed to be filled with lubricant which gives rise to viscous drag acting upon the roller as it rotates. The radial velocity of the fluid in this clearance space will be assumed negligible as compared to the tangential velocity. Then using the well known viscous drag formula between two rotating discs [6] we can write the following relationships:

Friction torque on roller

$$= \frac{\pi\mu\omega_4}{\varepsilon_1}\left(R_r^4 - R_e^4\right) \qquad (22)$$

Corresponding friction loss, $\dot{E}_{FL_2}$

$$= \frac{\pi\mu\omega_4^2}{\varepsilon_1}\left(R_r^4 - R_e^4\right) \qquad (23)$$

3. Eccentric-to-Cylinder Head Friction

This case is exactly similar to the previous case in as much as the same "kind" of relative motion exists between the eccentric and the cylinder head. The only difference here is that the angular speed of the eccentric is different from that of roller and in some cases it is possible that the clearance between the eccentric and the cylinder head may also be different from that between the roller and the cylinder head. The friction torque and friction loss relationships can therefore be similarly written as follows:

Friction torque on eccentric

$$= \frac{\pi\mu\omega_1}{2\varepsilon_2}\left(2R_e^4 - R_s^4\right) \qquad (24)$$

Corresponding friction loss, $\dot{E}_{FL_3}$

$$= \frac{\pi\mu\omega_1^2}{2\varepsilon_2}\left(2R_e^4 - R_s^4\right) \qquad (25)$$

Note that the above two relationships are valid only if the motor shaft extends only on one side of the eccentric as shown in Fig. 1. However, if the shaft has to be extended on both sides of the eccentric for some design reason, the above equations can be easily modified.

4. Friction Between the Roller and the Eccentric

As mentioned earlier, there is a relative motion between the roller and the eccentric due to the fact that both rotate with different absolute angular velocities, which are ω_1 and ω_4 for the eccentric and the roller respectively. The relative angular velocity between the two (ω_r) is given by Eq. 7 as ($\omega_3 - \omega_1/a$). Thus, the eccentric and the roller will behave somewhat similar to a journal and a bearing and a similar analysis should, therefore, be applicable. However, since the only bearing load acting in this case is the centrifugal force of the roller due to the eccentric mounting, which is insignificant, we can approximate our analysis by assuming that the hydrodynamic film of lubricant is not created and instead the friction loss is only as a result of viscous drag between the two concentric cylinders (i.e. roller and eccentric) with a concentric film of lubricant being developed between the two. Then, assuming straight line tangential velocity distributions between the roller and the cylinder and negligible radial velocity, following relationships can easily be established [6].

Friction torque on roller in positive θ

$$\text{direction} = \frac{\mu(2\pi R_e h)R_e(-\omega_r)R_e}{\delta} = -\frac{2\pi\mu R_e^3 h\omega_r}{\delta} \qquad (26)$$

Loss of power due to friction, $\dot{E}_{FL_4}$

$$= \frac{2\pi\mu R_e^3 h\omega_r^2}{\delta} \qquad (27)$$

Total Friction Loss

In the expressions for the four friction loss terms developed above, there is one inherent term ω_3 that is not known yet. This is the angular speed of slipping of the roller relative to the cylinder. To obtain average value of ω_3, let us analyse the forces acting upon the roller under steady conditions, i.e. under zero angular acceleration. From the free-body diagram of the roller (Fig. 5) we can easily write following relationships, since sum total of the restoring torques due to blade-tip friction and roller-cylinder head friction is equal and opposite to the applied torque due to roller-eccentric friction.

$$T_1 + T_2 = T_3 \qquad (28)$$

or

$$R_r f\left[\frac{1}{4}(3P_s + P_d)th + \frac{1}{2}kx_{max}\right]$$

$$+ \frac{\pi\mu}{\varepsilon_1}\left[\omega_1(1 - \frac{1}{a}) + \omega_3\right](R_r^4 - R_e^4)$$

$$= \frac{2\pi\mu R_e^3 h}{\delta}\left(\frac{\omega_1}{a} - \omega_3\right) \qquad (29)$$

Solving for ω_3, we get:

$$\omega_3 = \frac{\frac{\pi\mu\omega_1}{a}\left[\frac{2R_e^3 h}{\delta} - \frac{(a-1)}{\varepsilon_1}R_r^4\right] - R_r f\left[\frac{(3P_s+P_d)}{4}th + \frac{kx_{max}}{2}\right]}{\pi\mu\left[\frac{2R_e^3 h}{\delta} + \frac{R_r^4 - R_e^4}{\varepsilon_1}\right]} \qquad (30)$$

Substitution of ω_3 from Eq. (30) into Eqs. (21), (23), (25), and (27) will give the individual losses and their sum will then give the total friction loss.

LEAKAGE LOSS

Figure 6 shows the various leakage paths occurring in a rolling piston compressor. They are as follows:

(i) Leakage past the contact point between the roller and the cylinder

(ii) Leakage past the blade edges

(iii) Leakage across the roller faces

(iv) Leakage past the blade tip

Based on the analogy with the rotary vane compressor, we realize that the latter two flowpaths give a negligible amount of leakage as compared to the first two and hence for an approximate analysis we will not consider them. The other two leakage loss terms can be analysed as follows.

Leakage Past The Contact Point

Assuming that the refrigerant behaves as an ideal compressible gas and critical flow conditions exists as an upper bound, and neglecting the Couette flow effect due to the rotation of the rotor (i.e., assuming that the leakage is only due to the pressure differential) we can model the leakage past the contact point as flow through convergent-divergent nozzle. Then the classical formula for maximum flow through a convergent-divergent nozzle from any standard textbook [11] will give us the maximum value of the leakage loss ($\dot{m}_{LL_1}$) in this case as an upper bound. Thus,

$$\dot{m}_{LL_1} = \delta_c h P_u \sqrt{\frac{\gamma}{RT_u}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \qquad (31)$$

where P_u and T_u are upstream pressure and temperature respectively. Since P_u and T_u are changing continuously throughout the cycle, we will have to take a differential element approach for exact analysis. However, we feel that a reasonable approximation can be made by averaging out P_u and T_u. Thus, we may write:

$$P_u = \frac{1}{2}(P_s + P_d) \qquad (32)$$

$$T_u = \frac{1}{2}(T_s + T_d) \qquad (33)$$

Substituting (32) and (33) back into (31), we get:

$$\dot{m}_{LL_1} = \frac{\delta_c h}{2}(P_s + P_d)\sqrt{\frac{2\gamma}{R(T_s + T_d)}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \qquad (34)$$

It must be mentioned here that we do realize that this equation is an approximation since we have not considered the Couette flow effect. However, since the direction of rotation of the roller is from the suction side towards the discharge side, this effect will try to reduce the leakage and hence the formula in Eq. (34) will be an upper bound.

Leakage Past The Blade Edges

We can model the leakage through the clearance between the blade edges and the cylinder head too as a flow through converging-diverging nozzle since the physical situation remains pretty much the same. Then using similar assumptions as in previous case we can write this loss also as an upper bound,

$$\dot{m}_{LL_2} = 2\delta_e \varepsilon_1 P_u \sqrt{\frac{\gamma}{RT_u}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)(\gamma-1)}} \qquad (35)$$

where ε_1 is the clearance between the blade and the cylinder head, which is normally the same as the clearance between roller faces and the cylinder head. Again, to simplify, we can average out P_u and T_u. Also, from the geometry of the compressor we can find δ_e as a function of θ. Thus,

$$\delta_e = R_c\left[1 - (1-a)\cos\theta - \sqrt{(1-a)^2\cos^2\theta + 2a-1}\right] \qquad (36)$$

However, equation (36) can be used only if we want to use the Eq. (35) as the instantaneous leakage rate and then integrate it over the cycle to get the average leakage rate. We feel that for an approximate analysis, the average value of δ_e could be used and the process of numerical integration avoided without much loss of accuracy. Thus:

$$(\delta_e)_{min} = 0 \tag{37}$$

$$(\delta_e)_{max} = R_c - [R_r - (R_c - R_r)]$$

$$= 2(R_c - R_r) \tag{38}$$

Therefore

$$(\delta_e)_{average} = \frac{1}{2}\left(\delta_{e_{min}} + \delta_{e_{max}}\right)$$

$$= (R_c - R_r) \tag{39}$$

Finally, the approximate formula becomes:

$$\dot{m}_{LL_2} = \varepsilon_1 (R_c - R_r)(P_s + P_d)\sqrt{\frac{2\gamma}{R(T_s + T_d)}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \tag{40}$$

where R is the gas constant. The total leakage loss will then be the sum of equations (34) and (40).

AN EXAMPLE CASE

The applicability and usefulness of the above analysis can best be demonstrated by using it to analyse a typical example case. Following is the friction loss and leakage analysis of a typical 1/4 hp rolling piston refrigerating compressor with following measured dimensions and assumed operating conditions:

RPM of the motor = 3600

$R_r = 23.9 \times 10^{-3}$[m]

$R_c = 26.972 \times 10^{-3}$[m]

$R_e = 9.895 \times 10^{-3}$[m]

$R_s = 6.35 \times 10^{-3}$[m]

$h = 11.98 \times 10^{-3}$[m]

$\delta = .007 \times 10^{-3}$[m]

$\delta_c = .0025 \times 10^{-3}$[m]

$\varepsilon_1 = .010 \times 10^{-3}$[m]

$\varepsilon_2 = .010 \times 10^{-3}$[m]

$t = 4.6 \times 10^{-3}$[m]

$x_{max} = 6.15 \times 10^{-3}$[m]

$k = 5000$[N/m] (28.5 lbf/in)

$\rho_s = 6.96$[kg/m^3]

$P_s = 137895$[N/m^2] (20 psi)

$T_s = 294.11$°K (70°F)

$P_d = 1378950$[N/m^2] (200 psi)

$T_d = 355.22$°K (180°F)

$f = 0.015$

$\mu = 99974 \times 10^{-7}\left[\frac{\text{N-sec}}{\text{m}^2}\right]$

$\left(14.5 \times 10^{-7} \frac{\text{lb sec}}{\text{in}^2}\right)$

$\gamma = 1.14$

$R = 69\left[\frac{\text{N-m}}{\text{Kg-°K}}\right]$

$a = R_r/R_c = 0.886$

$\omega_1 = 2\pi N/60 = 120\pi = 377$[rad/sec]

Calculations

Angular velocity of sliding of roller (ω_3) = 72.5 [rad/sec]

Sliding RPM of roller = 692

Angular velocity of roller relative to its center (ω_r) = −353.0[rad/sec] (Note that negative sign indicates that the roller appears to rotate in a direction opposite to that of the eccentric.)

Absolute angular velocity of roller (ω_4) = 24.0[rad/sec]

Blade tip friction loss = 0.345 watts

Eccentric-to-cylinder head friction loss = 3.916 watts

Roller faces-to-cylinder head faces friction loss = 0.573 watts

Roller-eccentric friction loss=12.978 watts

Total friction loss in cylinder=17.8 watts

Leakage past the contact point, $\dot{m}_{LL_1}$ = 96.6 × 10^{-6}[kg/sec]

Leakage past the blade edges, $\dot{m}_{LL_2}$ = 198.2 × 10^{-6}[kg/sec]

Total leakage loss, $\dot{m}_{LL}$ = 294.8×10^{-6}[kg/sec] = 1.06[kg/hr]

The ideal mass flow for this compressor, as per its geometry and given dimensions, should have been 8.84 kg/hr. Thus we have a leakage loss of approximately 12%. We also notice that the cylinder friction loss as obtained above is approximately 9% of the power input. Both these values seem to be reasonable.

CLOSURE

The typical compressor geometry of a rolling piston type rotary compressor has been analysed. Pressure-angle and volume-angle relationships were developed. The existence of various energy losses and mass flow losses was explained and two most important losses typical to the special geometry of this type of compressor, i.e. the friction losses and the leakage losses, were analysed. Simplified formulas were developed with the idea that they may serve as a quick checking tool in the design process rather than a lengthy simulation model that could be used for ultimate optimization, although the fundamental relationships were pointed out that could be used, if so desired, in such a program. An example case was analysed that demonstrates the basic simplicity and applicability of the approach. The results seem to be reasonable, given the fact that the data were only approximately measured and/or arbitrarily assumed. Furthermore, emphasis was more on demonstration of the method than on the accuracy of the results. The method can be a great tool in the hands of a designer who is interested in studying the effect of various design parameters on the losses so that the design could be modified at the initial stage itself to give better performance. However, we also realize that total simulation and optimization will of course be the ultimate step in the complete analysis of this compressor as any other.

NOMENCLATURE

a	Radius ratio $\frac{R_r}{R_c}$ [dimensionless]
$A(\theta)$	Area of the compression chamber at angle θ [m^2]
e	Eccentricity [m]
$\dot{E}_{FL}$	Friction loss [Nm/sec]
f	Coefficient of friction [dimensionless]
$f(\theta)$	A function of angle of rotation θ
F	Friction force [N]
h	Cylinder height [m]; blade height [m]
k	Spring constant [N/m]
$\dot{m}_{LL}$	Loss in mass flow rate due to leakage [kg/sec]
N	RPM of the motor
$P(\theta)$	Pressure in the compression chamber at angle θ [N/m^2]
P_s	Desired suction pressure [N/m^2]
P_d	Desired discharge pressure [N/m^2]
P_u	Upstream pressure [N/m^2]
ΔP	Underblade oil pressure [N/m^2]
R	Radius vector; gas constant of refrigerant [Nm/kg°K]
R_c	Cylinder radius [m]
R_r	Roller radius [m]
t	Blade thickness [m]
R_e	Radius of eccentric [m]
R_s	Radius of shaft [m]
T_1	Friction torque of blade on roller [N-m]
T_2	Viscous drag torque on roller from cylinder [N-m]
T_3	Viscous drag torque on roller from eccentric [N-m]
T_s	Ideal suction gas temperature [°K]
T_d	Ideal discharge gas temperature [°K]
T_u	Upstream temperature [°K]
$V(\theta)$	Volume corresponding to position θ [m^3]
$V'(\theta)$	Corrected volume corresponding to position θ [m^3]
x	Spring compression [m]
x_{max}	Maximum spring compression [m]
δ_e	Blade extension [m]
δ_c	Minimum clearance between roller and cylinder [m]
δ	Radial clearance between roller and eccentric [m]
ε_1	Clearance between roller and cylinder head faces [m]; clearance between blade and cylinder head [m]
ε_2	Clearance between eccentric and cylinder head faces
ϕ	Arbitrary angle [radians]
γ	Specific heat ratio of refrigerant gas [dimensionless]
μ	Coefficient of viscosity [Nsec/m^2]
θ	Angular position of rotor [radians]
ω_1	Angular velocity of eccentric [radians/sec]
ψ	Rolling angle of roller [radians]
ω_2	Rolling angular velocity of roller [rad/sec]
ω_3	Slipping angular velocity of roller [rad/sec]
ω_4	Net angular velocity of roller [rad/sec]
ω_r	Angular velocity of the roller relative to the eccentric [rad/sec]

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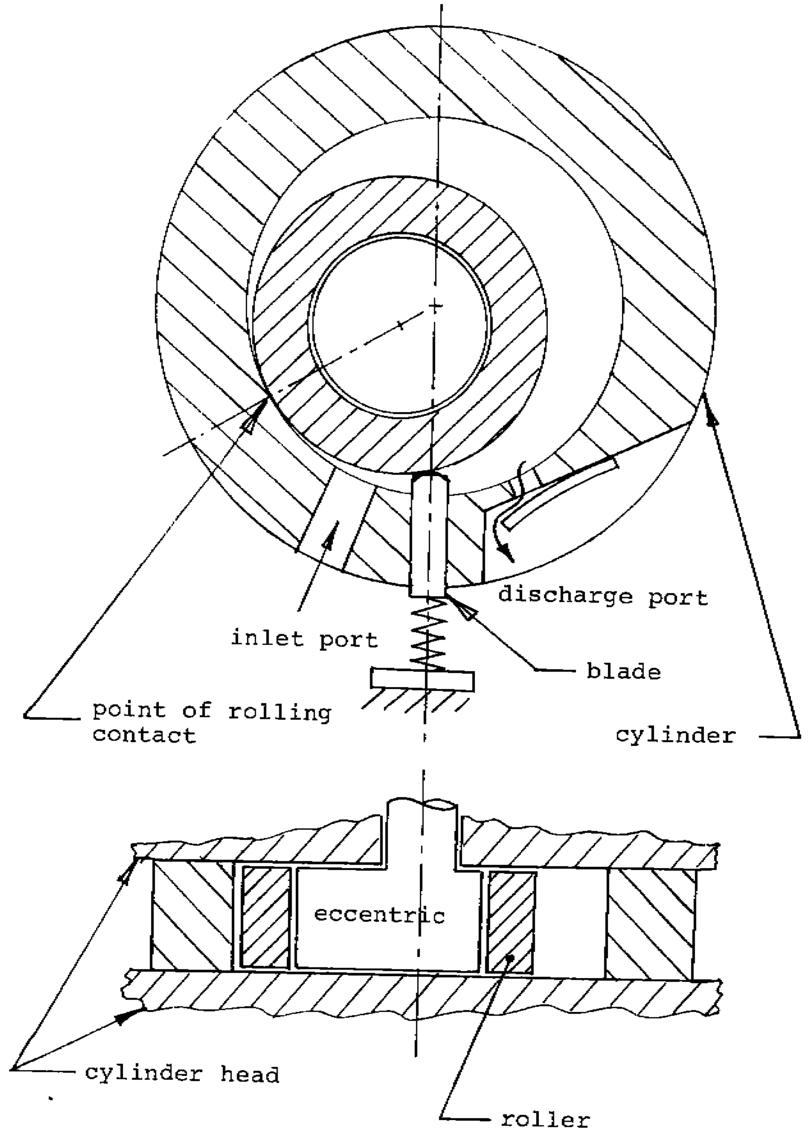


Fig. 1: Cross-section of a typical rolling piston compressor.

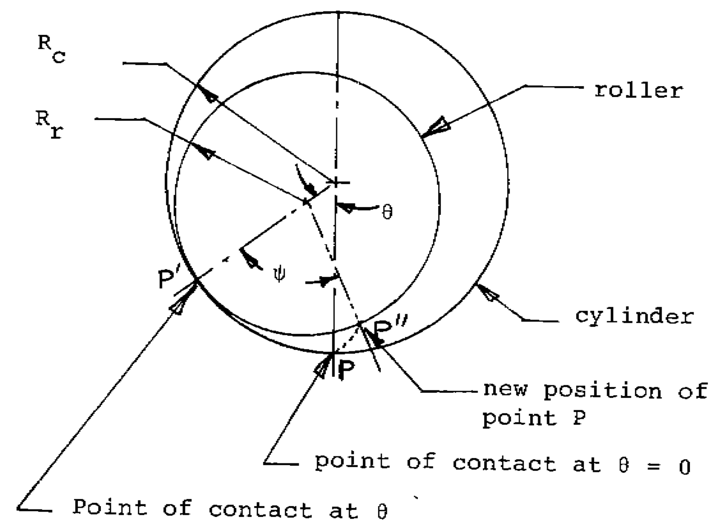


Fig. 2: Rolling motion of roller

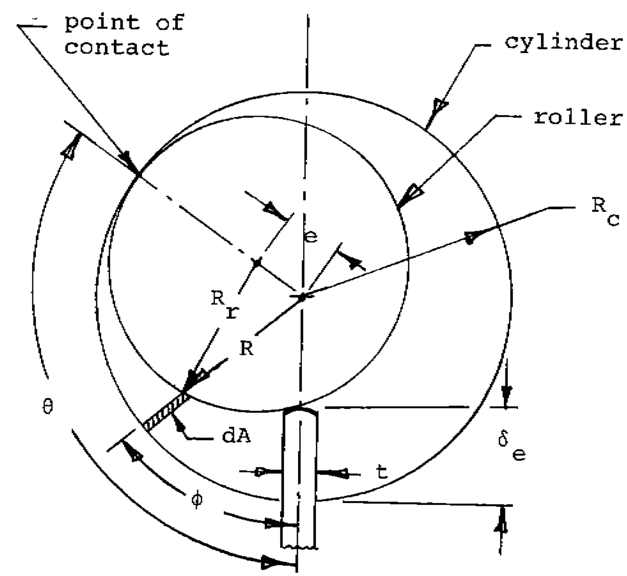


Fig. 3: Volume-angle relationship

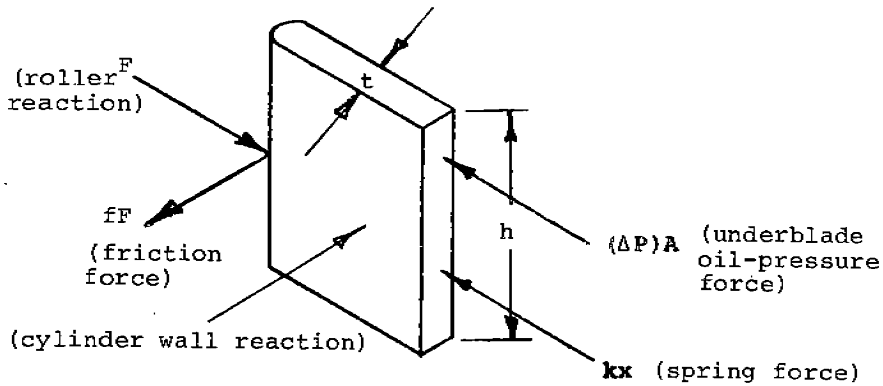


Fig. 4: Blade tip friction

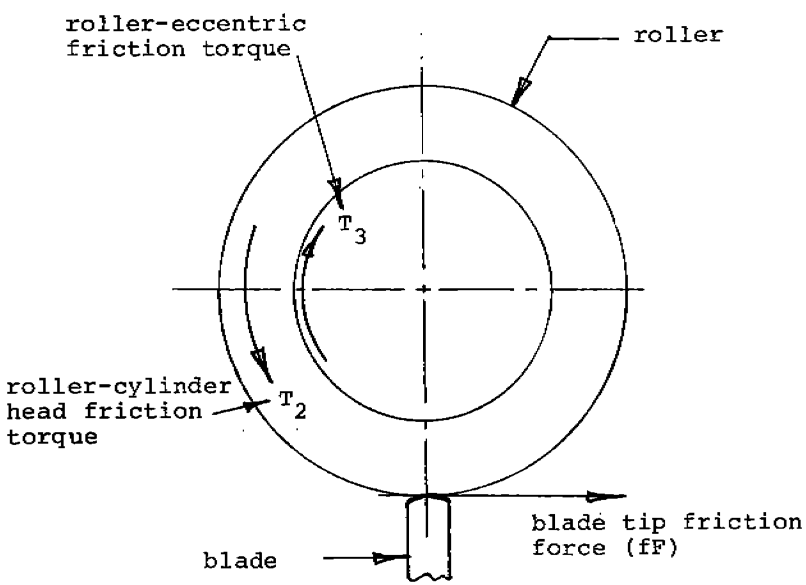
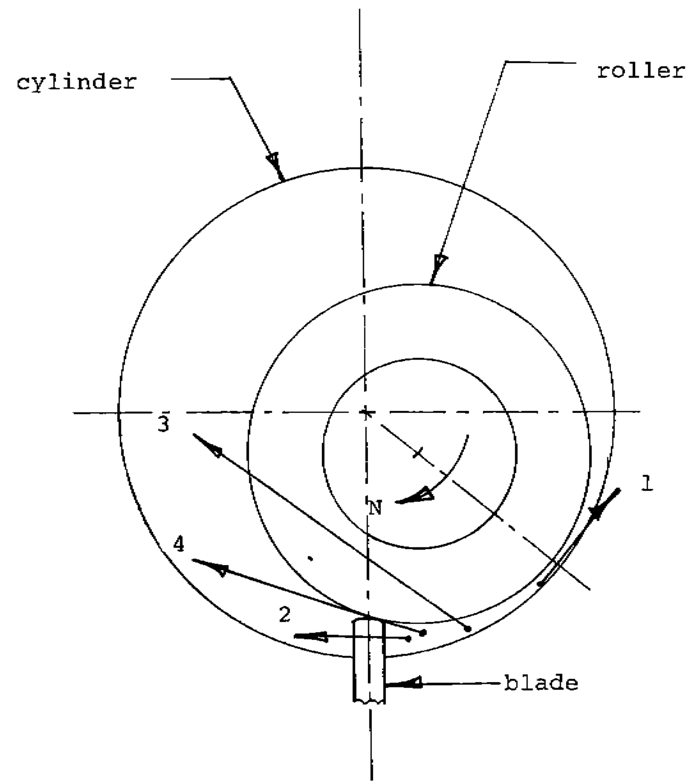


Fig. 5: Free-body diagram of roller



Legend:
1. Leakage past the contact point between the roller and the cylinder.
2. Leakage past the blade edges.
3. Leakage across the roller faces.
4. Leakage past the blade tip.

Fig. 6: Leakage flow-paths in a rolling piston compressor