8th International Tribology Conference
Osma internacionalna konferencija o tribologiji
Beograd, 8. - 10. oktobra 2003.

STUDY OF THE PRESSURE DISTRIBUTION AND HYDRODYNAMIC FRICTION FORCE IN A COAXIAL SEALING TRIBO-SYSTEM

Prof. dr Eng. Tudor DEACONESCU, Lecturer Eng. M.A. **Andrea DEACONESCU**
"Transilvania " University of Brasov, Romania

Summary

The paper presents a determination method of the pressure distribution in the gap of magnitude ***g****, generated at the occurrence of a relative motion, the imposed condition being a zero flow rate of fluid loss (perfect sealing). Ensuring a perfect tightness is obtained when the oil loss due to the pressure difference between the two sealed-off spaces is balanced by the fluid transport.*

A second objective of the paper is the computational determining of the hydrodynamic friction force magnitude generated at the interface of the seal and its adjacent surface.

Keywords: *hydrodynamic friction force, coaxial sealing tribo-system*

Lately a significant development was registered in relation to sealing systems with coaxial rings, consisting of a seal of various possible forms of its cross-section and an O-ring that ensures the pre-stressing of the entire assembly. The coaxial rings are used for the packing of both pistons and piston rods, the main constructive forms being presented in the figure below:

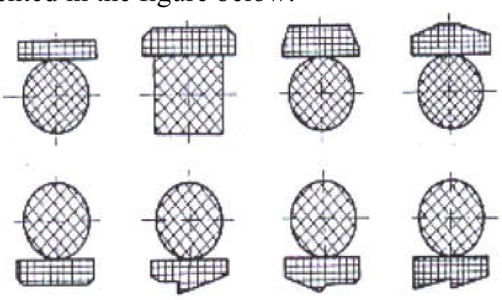

Fig.1

Figures 2 and 3 show these systems for the packing of pistons and piston rods, respectively:

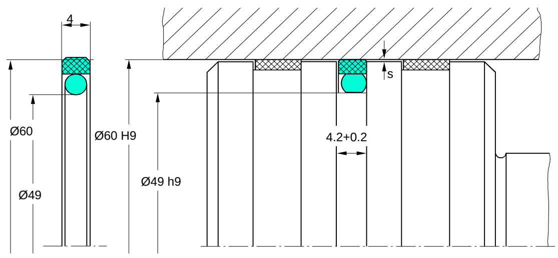

Fig. 2

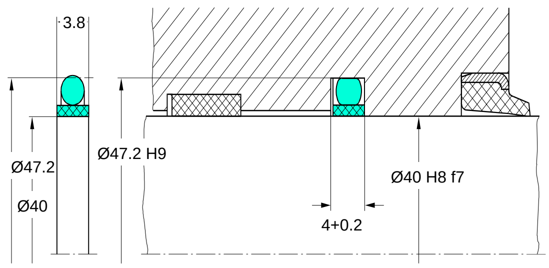

Fig. 3

The O-rings have no contact with the moving surfaces, hence are not subject to wear. Their role is merely to generate the pressure required for stressing the PTFE-made seal.

For linear hydraulic motors, at occurrence of motion of the mobile assembly, a gap of magnitude g is generated between the seal and the sealed-off surface. Whilst the existence of this gap creates a significant advantage– reduced friction forces, it also bears the disadvantage of untightness. The lack of tightness causes a certain loss of fluid, which can be a leakage (the Poiseuille component of the flow) or transport of fluid (the Couette component of the flow).

The leakage represents a loss of fluid even in an immobile state, due to the pressure difference between the two sealed-off chambers, while when mobile the volume of lost fluid increases. Fluid transport is determined by the existence of a lubricant film on the mobile part, required for ensuring minimum friction forces.

Further on we shall present a method for the determination of the pressure distribution in the gap of size g, generated during relative motion, the imposed condition being that of zero fluid losses (perfect sealing).

In order to ensure perfect sealing, the loss of oil due to the pressure difference between the two sealed-off chambers must be balanced by the fluid transport. This can be expressed by equation (1):

$$Q = \int_0^g \dot{x} \cdot dy = \mathbf{0} \qquad (\mathbf{1})$$

where the fluid velocity $\dot{x}$ is computed by equation (2):

$$\dot{x} = \frac{\mathbf{1}}{\mathbf{2} \cdot \eta} \cdot \frac{dp}{dx} \cdot y \cdot (y - h) - \frac{v}{g} \cdot y + v \qquad (\mathbf{2})$$

The variation mode of velocity in the gap is shown in figure 4:

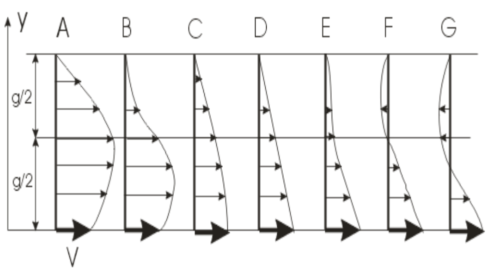

Fig. 4

The seven cases shown by the figure correspond to the following situations:

A: $\dot{x}_{max} = 2v$

B: $\dot{x} = v$ when y = g/2

C: d$\dot{x}$/dy = 0 when y = 0

D: d$\dot{x}$/dy = const. when dp/dx = 0

E: d$\dot{x}$/dy = 0 when y = g

F: $\dot{x}$ = 0 when y = g/2

G: $\int \dot{x} \cdot dy$ = Q = 0

Further on the last case will be analysed, for a flow rate of Q = 0.

Starting from the similarity criteria of Reynolds and Euler, a new invariant can be introduced, denoted by T which is obtained by multiplying Re and Eu. Thus we obtain:

$$T = \mathrm{Re} \cdot Eu = \frac{\rho \cdot v \cdot h}{\eta} \cdot \frac{p}{\rho \cdot v^2} = \frac{p \cdot h}{\eta \cdot v} \qquad (\mathbf{3})$$

For a ring-shaped gap, the quantity h = 2g, can hence be written as:

$$T = \frac{\frac{dp}{dx} \cdot \mathbf{2} \cdot g \cdot \Delta}{\eta \cdot v} \qquad (\mathbf{4})$$

where Δ is a quantity introduced such as to render the equation above non-dimensional, and that is equal to p/(dp/dx).

Starting from equation (5):

$$Q = \pi \cdot d \cdot \left(\frac{v \cdot g}{\mathbf{2}} - \frac{g^{\mathbf{3}}}{\mathbf{12} \cdot \eta} \cdot \frac{dp}{dx}\right) = \mathbf{0} \qquad (\mathbf{5})$$

the value of the invariant T is:

$$T = \frac{\mathbf{12} \cdot \Delta}{g} \qquad (\mathbf{6})$$

It can be thus written that:

$$T = \frac{\frac{dp}{dx} \cdot \mathbf{2} \cdot g \cdot \Delta}{\eta \cdot v} = \frac{\mathbf{12} \cdot \Delta}{g} \qquad (\mathbf{7})$$

Figure 5 presents the positioning of a coaxial sealing system once a relative motion has occurred:

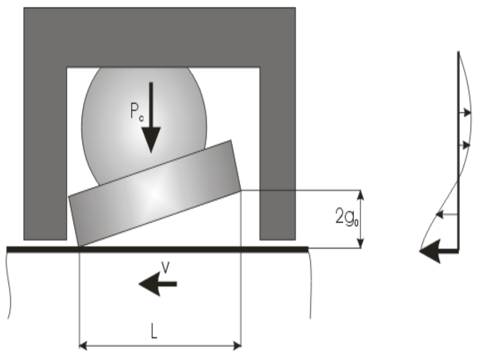

Fig. 5

Upon occurrence of a relative velocity v, the PTFE seal will rotate and be deformed, thus additionally compressing the O-ring. In this case the height of gap g becomes a function of x of the form:

$$g(x) = 2 \cdot g_0 \cdot \frac{x}{L} \qquad (8)$$

where g_0 is the mean value of the gap.

Equation (8) describes a linear evolution of the gap height until a maximum value g(L) = $2g_0$. Equation (9) is obtained by replacing equation (8) in equation (7):

$$T = \frac{\frac{dp}{dx} \cdot 2 \cdot (2 \cdot g_0 \cdot \frac{x}{L}) \cdot \Delta}{\eta \cdot v} = \frac{12 \cdot \Delta}{2 \cdot g_0 \cdot \frac{x}{L}} \qquad (9)$$

wherefrom follows:

$$\frac{dp}{dx} = \frac{3}{2} \cdot \frac{\eta \cdot v \cdot L^2}{g_o^2 \cdot x^2} \qquad (10)$$

At half the length L of the seal (x = L/2) it is admitted that pressure p(x) = p_c, where p_c is the contact pressure between the O-ring and the seal. In this case the variation law of pressure in the gap becomes:

$$p(x) = p_c + 3 \cdot \frac{\eta \cdot v \cdot L}{g_0^2} \cdot (1 - \frac{L}{2 \cdot x}) \qquad (11)$$

Equation (11) describes that evolution of pressure in the gap that ensures a zero leakage flow.

The hydrodynamic friction force generated at the interface of the seal and its adjacent surface is computed with equation (12):

$$F_{HD} = \tau_{HD} \cdot (1 - \beta) \cdot A_n \qquad (12)$$

where:

β = ration of the real and rated contact areas,

A_n = rated contact area,

τ_{HD} = hydrodynamic tangential stress, computed by equation (13):

$$\tau_{HD} = \frac{1}{2} \cdot \frac{dp}{dx} \cdot (2 \cdot y - g) - \frac{\eta \cdot v}{g} \qquad (13)$$

By replacing in the above equation formulae (4), (6) and (8), the tangential stress becomes:

$$\tau_{HD} = -\frac{2 \cdot \eta \cdot v}{g_0 \cdot x} \cdot L \qquad (14)$$

In this case the hydrodynamic friction force developed in the gap will be computed by equation (15):

$$F_{HD} = \pi \cdot d \cdot \int \tau_{HD} \cdot dx = -2 \cdot \pi \cdot d \cdot \frac{\eta \cdot v}{g_0} \cdot L \cdot \ln(\frac{L}{x}) \qquad (15)$$

The (-) sign indicates that the direction of the hydrodynamic friction force is opposite to that of the relative velocity.

REFERENCES

[1] Deaconescu, T., Sisteme de etanşare mobile, cu frecări reduse, utilizate în acționările hidraulice ale maşinilor-unelte.(Mobile Low Friction Sealing Systems Used in Machine-Tool Hydraulic Drives) Doctoral dissertation, Transilvania University of Braşov, 1997.

[2] Werneke, P.W., Untersuchung der physikalischen Vorgänge in Spalten von Hydraulikdichtungen. Doctoral dissertation, Aachen, 1983.