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DESIGN DEVELOPMENT AND TRIBOLOGY OF RECIPROCATING HYDRAULIC SEALS[1]

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Abstract

The development of reciprocating hydraulic seals is the simultaneous progress of seal designs and their tribo-mechanical system. The progress is resulted by the improvements in seal material characteristics, technology and design (cross section form, seal edge shape, friction surface quality, etc.), lubrication conditions, accuracy, reliability and life. Anyhow by time the increasing importance of environmental protection gave particular emphasis to leakage control.

In these developing processes emphasized aims are to economize by reducing losses – both leakage and friction – and also reducing the required sealing space (housing volume), while adequate and reliable sealing life is obtained.

The present paper provides contribution in revealing tribological – mostly friction – characteristics, design and efficiency of some characteristic reciprocating hydraulic seals.

Keywords: tribology of reciprocating hydraulic seals, piston and piston rod seals, efficiency of seals, design of seals.

1. Introduction

For different application fields reciprocating hydraulic seals are friction seals and they show a big variety by design and used materials as well. Except some particular cases these friction seals work on lubricated surfaces.

For varying operating parameters both the lubrication conditions and the lubricating film formations are subject to change between the seal contact and the friction surfaces of the (hydraulic cylinder) hole or rod. In case of alternating and reciprocating motions the lubrication conditions always make a change when there is any change in motion even if the rest of the working parameters – pressure (p_w) and temperature (t_w) – are unchanged. The major change in the lubrication condition is caused significantly by the rebuilding process of the lubricating film at the transition periods at each stroke stops and restarts. However stabilized lubrication conditions may be sustained – characteristically – in the middle of the stroke at constant speed of the motion.

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In principle the worse lubrication conditions may be developed at the transition periods of reciprocating seals in hydraulic equipments. Here the effect of high operating (system) pressure ($p_{w,\max} \leq 25, 40$ MPa or more) makes harder conditions for the lubricating film rebuilding process. Therefore it is of outstanding importance to reveal these disadvantageous lubrication conditions for hydraulic piston and piston rod seals.

Surveying the history of the sealing application the simple compression packing rings were the 'first' friction seals used in almost all motion cases in the early times. These types of seals were packed in stuffing boxes (glands) for reciprocating, rotary and alternating motions as well.

In these times there were practically no environmental protection requirements and leakage moderation served only the economy and safety of the applied technological processes. Also very modest requirements existed for hydraulic machineries regarding the main working parameters (speed, temperature and pressure).

The advantages of compression packing rings were the simplicity of the stuffing box design, manufacture, mounting and operation, while the disadvantages were the great losses (leakage and friction as well). There are the consequences of the unfavourable sealing mechanism principle of compression packing ring seals. As a result of the great losses the use of conventional packing rings reduced very much by now and they are applied in some particular fields only. (However, the newer generation of compression packing rings provides reasonable and popular sealing solution again – due to the advanced friction characteristics of applied PTFE and other advanced materials – for special applications of rotary and alternating shafts.)

2. Seal Design and Housing Developments

For reciprocating hydraulic applications the next step of seal design development was to use impregnated leather 'V' and 'U' rings in the conventional stuffing boxes (housings) instead of the compression packing rings. The difference in seal design entirely changed the sealing mechanism principle by introducing the 'automatic sealing mechanism' where bigger working pressure automatically produces bigger sealing pressure. However, due to the mechanical characteristics and the big leakage values of impregnated leather materials, neither 'V', nor 'U' rings could be considered as really sufficient sealing solutions. Consequently, they were practically dropped out from use by now.

By all means, the principle of the automatic sealing mechanism idea was a great innovation of the time and since it is applied for many different types of seals, including all later developments of reciprocating hydraulic seals having elastomeric sealing element.

During the last decades the reciprocating seals went through reasonable and conscious developments in the applied materials, technology, design – form and seal edge shape –, accuracy and reliability as well. In this development process the major aims were double: obtaining good operating characteristics and economizing, such as: reducing losses (both leakage and friction) and also to reduce the required room for seals (housing volume), while adequate sealing life is sustained.

The improvements in reciprocating hydraulic seal design apply different shapes, elements and material combinations (*Fig. 1*). The seal classification presented is far from being a complete one. It contains only some very typical groups of seals, starting with the basic forms in the top row and followed by some of their developments in the columns concerned.

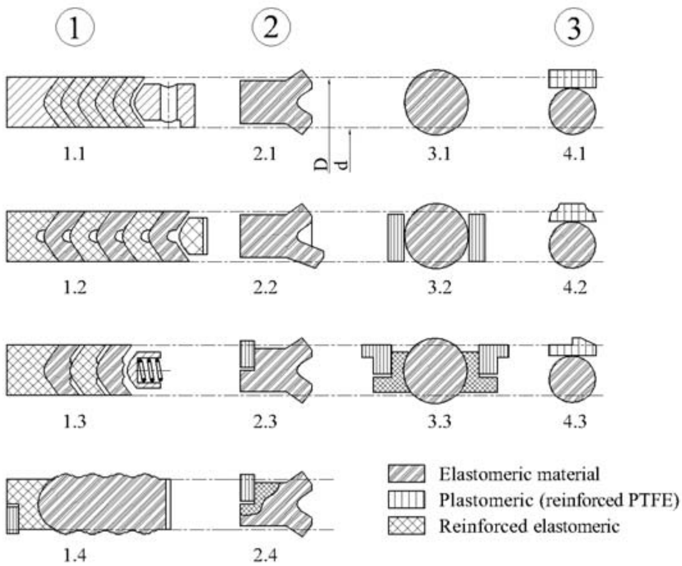

Fig. 1. A typical classification of reciprocating hydraulic seals

Together with the development of seals and concerned technologies the actual cross section of seals showed a continuous reduction from V-ring packing (1) up to the modern compact seals (4). As a consequence of seal cross section reduction the required housing volumes of seals were reduced too, both for piston and piston rod seals (*Fig. 2*).

The housing volume reduction has a great effect in economizing the product, i.e. the hydraulic cylinders, machineries and all other equipments concerned. The housing volume reduction decreases the material and space requirement and also decreases the required machining time of the products. As a result the efficiency of the product is improved to a great extent.

Considering the complex technical and economic improvement of hydraulic machines and equipments the major components of the improvements were the seal housing volume reduction and the increased system (working) pressure applied. These components brought quite big reduction in the cross section and the volume of the hydraulic equipment as a whole.

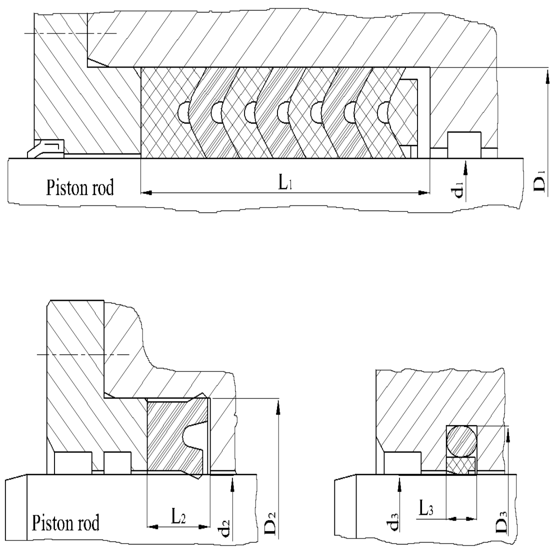

Fig. 2. Seal housings for hydraulic piston rods

3. Tribological Characteristics

From application point of view the operating behaviours and particularly the main operating characteristics of seals – leakage, friction loss and service life – have always been in the focus of users' interest.

In order to determine the main operating characteristics the long lasting (endurance) tests were carried on first. Later, friction characteristic tests were accomplished as well to get more information on the friction behaviour of seals [3], [4].

The next steps of escalating seal investigations on tribological features were provided by the tests for obtaining sealing pressure, lubricating film and temperature distribution diagrams (profiles) along the axis in the lubricating film between the seals and the friction surfaces [2, 4, 5, 6].

Anyhow, due to the great number of variants in tribological characteristics and the time-dependent features of seal materials, the predetermination of the main operating characteristic values are always based on some related test results and diagrams.

3.1. Main Operating Characteristics (Main Technical Requirements)

The main operating characteristics of seals may be determined – indirectly by estimation and – directly by endurance tests in order to obtain the formation of friction force and leakage values changes during the operation period. Here the operation life may be limited either by a complete failure effect (e.g. when the leakage can not be controlled any more) or by a permitted value of either the leakage or the friction force.

Anyhow, the main operating characteristics should meet some requirements formed by the need of the particular application field. These requirements may prescribe limitations for any or all of the main operating characteristics like:

- Clean appearance of the equipments, safety and environmental protection.
- Limited or even zero leakage may be required as it is considered the best outcome for operation and also for environmental protection. In practice zero leakage may be defined as ‘dry piston rod’ surface, which is resulted by the balanced leakage between the outstroke and in stroke working media transport.
- Good mechanical efficiency, economic operation, operation safety, reliable starting, operation and restarting conditions of the hydraulic equipments. They all need perhaps small, controlled and most of all predictable friction force formation during the complete operation conditions expected.
- Economic operation, maintenance and reliability of hydraulic equipments need controlled, low wear and adequately long life.

3.2. Leakage Control, Environmental Loading

Out of the seals’ main operating characteristics the leakage control is of utmost relevance to reduce or eliminate unnecessary environmental loading.

The first successful theory for leakage calculation was the inverse hydrodynamic theory. It was based on a flexible model, where the elastic (elastomeric) seal was sliding on lubricated metal surface [1]. Here the sealing gap i.e. the lubricating film profile – between the seal and the sliding surface – can be determined from the concerned sealing pressure distribution diagram [2]. Furthermore the expected value of leakage (Q) may be obtained from the outstroke and instroke sealing gaps (h_{out} and h_{in}) by the help of the maximum gradients $(\mathrm{d}p_t/\mathrm{d}x)_{\max}$ of the sealing pressure distribution diagram, taken from the concerned direction of the motion ($\mathrm{tg}\,\alpha$ and $\mathrm{tg}\,\beta$) [1, 4]:

$$h = C\sqrt{\eta v/(\mathrm{d}p_t/\mathrm{d}x)}$$
$$Q = \pi D s(h^*_{\text{out}} - h^*_{\text{in}})/2$$

As a consequence of the outstanding importance of sealing pressure gradients the leakage can be well influenced by modifying the sealing pressure distribution diagrams. In principle the sealing pressure diagram can be well modified by seal design changes (*Fig. 3*) [10].

Static and dynamic sealing pressure: Theoretically correct information on the operating sealing pressure distribution is obtained by tests based on dynamic

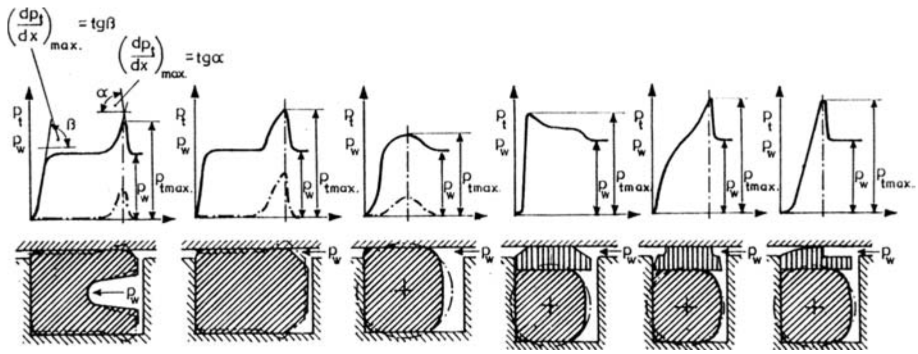

Fig. 3. Static sealing pressure distribution profiles of some reciprocating hydraulic seals having single friction edges. (From left to right: Elastomeric U-ring, □-ring, O-ring and compact composite seals with O-ring and different reinforced PTFE piston rod seal.)

sealing pressure measurements. Here, in principle, the time-dependent material behaviours of seal materials and the effect of reciprocating motion (and speed) are reflected in the sealing pressure distribution obtained. Nevertheless, the static pressure distribution diagrams can be used for research up to now due to some remarkable conclusions and considerations, such as:

- The sealing edge is rather shape-keeping for modern high-pressure reciprocating hydraulic seals. Therefore, the critical pressure gradient directions of the static and dynamic pressure distribution diagrams do not show reasonable differences. At least the values of calculated leakage – by the help of measured sealing pressure gradients (tangents) – do not give significant difference comparing to the leakage values obtained by (long lasting) tests. However, this correlation between the calculated and measured values was proved mostly for high-pressure reciprocating elastomeric seals [7].
- Regarding the in- and outgoing static sealing pressure distribution diagrams the maximum sealing pressure gradients did not show sensible changes for hard elastomeric seals in general and for U-ring seals in particular [9]. This observation may be considered as a proof to use static sealing pressure distribution diagrams for high pressure hydraulic seal developments instead of the dynamic ones.

3.3. Friction Characteristics, Friction Losses

While the long lasting or endurance tests provide the friction value changes along the operation time, the friction characteristic tests give information on the friction force change in the function of working pressure or speed at a selected point of the operation time (or cycles done). Anyway, the friction characteristic tests usually follow a certain operation time – running in periods – in order to obtain balanced friction force and leakage values.

In case of these friction characteristic tests one working parameter – mostly either the pressure or the speed – is the subject to be changed while all other parameters and circumstances are kept unchanged. Anyhow, all forms of the elaborated diagrams of friction characteristic tests provide useful (indirect) information on friction-lubrication conditions of the tested seal.

From historical point of view the friction force was measured first at different working pressure levels at constant reciprocating speed [3]. Finally, the test results obtained were expressed in the form of friction force – working pressure ($F_c - p_w$) and friction coefficient – working pressure ($\mu - p_w$) diagrams (*Fig. 4*). Both kinds of these diagrams demonstrate the friction and lubrication features of the tested seal types and may be used to find the most appropriate one for the concerned application. Furthermore these diagrams display the recommendable working pressure limits of the seal at the tested speed level (*Fig. 4*).

In the shown diagrams – at start – the friction force increases fast and then increases more and more moderate for consecutive working pressure value levels. By further working pressure elevation there is a certain zone where the friction force shows a fast increasing tendency again. This phenomenon in the force change indicates fast deterioration in the lubrication conditions in the lubricating film between the seal and the moving friction surface. So this zone indicates the permitted maximum working pressure value ($p_{w,\max}$) recommended for the tested seal [7]. These friction characteristic – working pressure diagrams can be used also for estimating the expectable friction loss values for different size seals of the same type working in the same conditions.

Anyhow, the diagrams of friction force – constant (stabilized) reciprocating speed ($F_c - v_c$) or friction coefficient – constant reciprocating speed ($\mu_w - v_c$) give better and more expressive information on lubrication conditions than the friction coefficient – working pressure diagrams. These diagrams – in the function of reciprocating speed – may by called the ‘Stribeck diagrams’ of reciprocating hydraulic seals. Examining the friction coefficient values of different test pressure level, it is found when higher working pressure is used. Any of these diagrams can be used to determine the friction coefficient (lubrication condition) change in terms of speed change, to estimate the expected friction loss for different seal diameter of the same type (operating in the same conditions) and also to find the optimum speed (v_{opt}) where the friction coefficient diagram shows a minimum value (*Fig. 5* and *6*).

However, the earlier introduced elastic model was not suitable for friction force estimation, it would require a properly elaborated ‘true’ tribological model [9]. A useful true tribological model is not yet available. It needs further clarifications and investigations of the concerned tribological features. Consequently, the expected friction force may be obtained from calculations based on diagrams of friction characteristic tests.

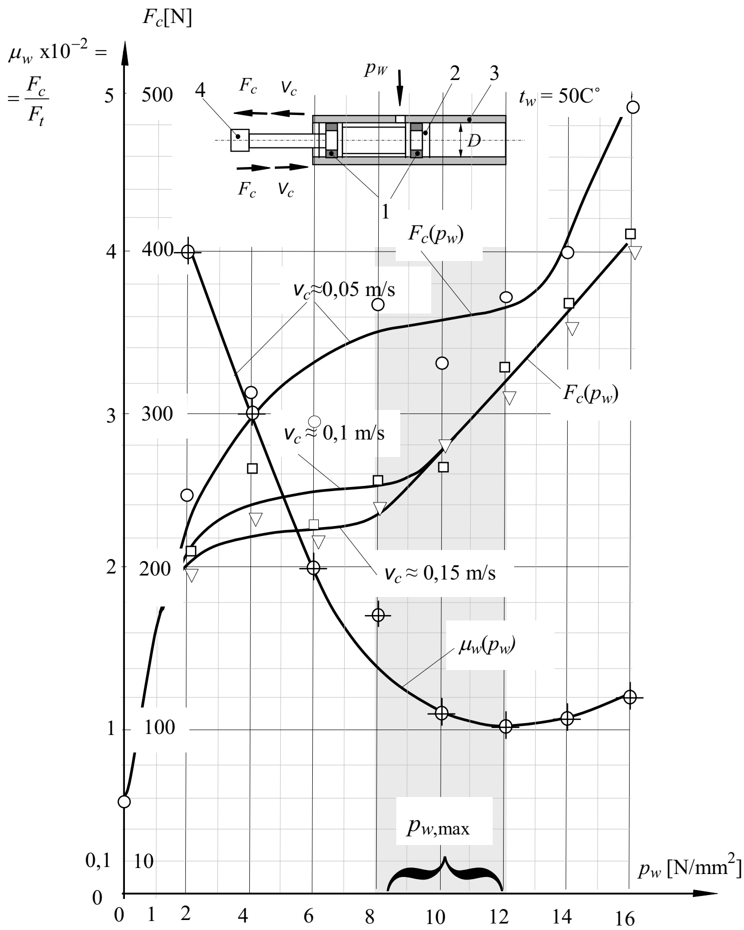

Fig. 4. Friction force – working pressure and friction coefficient – working pressure diagrams of Silicon U-rings$^{(1)}$

Furthermore these friction coefficient – constant reciprocating speed ($\mu - v_c$) diagrams can be developed and generalized, to be the basis of determining the expected friction force for seal diameters and working conditions different from the tested one (*Fig. 7* and *8*) [4], [7].

In these improved 'Stribeck' diagrams the friction coefficient is expressed in relation with a unit less number ($Z = \eta \cdot v_{c/}\bar{p}_t \cdot b$, or $Z \approx v_c \cdot \eta / p_w \cdot b$). After factoring out the speed (v_c) the expression remained ($\eta / p_w b$) is constant for a certain operation (test) condition and working pressure. Consequently, the friction coefficient is subject to the change of reciprocating speed (v_c) only.

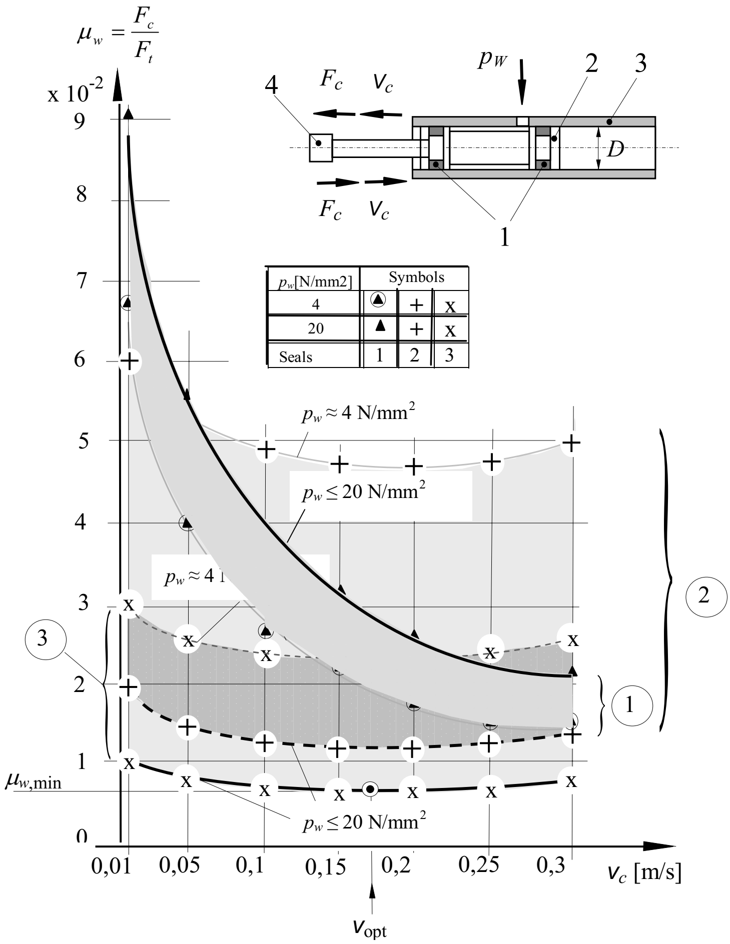

Fig. 5. Friction coefficient – reciprocating speeds diagrams of impregnated leather V- ring (1) Polyurethane U- ring (2) and NBR O-ring (3) seals [(2)(3)]

For calculation (estimation) purpose the expected stabilized friction force can be determined by the following relationships [7]:

$$F_c = \mu \cdot F_t = \mu \cdot (A_t \cdot \bar{p}_t) = \bar{p}_t \cdot b \cdot D \cdot \pi \cdot \mu_w, \text{ or } F_c = \bar{p}_t \cdot b \cdot d \cdot \pi \cdot \mu_w$$

and

$$\mu_w = c_1 + c_2/z,$$

where $\mu_w = c_1 + c_2/z$ is a suitable upper covering enveloping curve of the test curves. Furthermore both parameters $c_1(p_w)$ and $c_2(p_w)$ can be received from the concerned diagrams (for Polyurethane U-rings and NBR O-rings) [7].

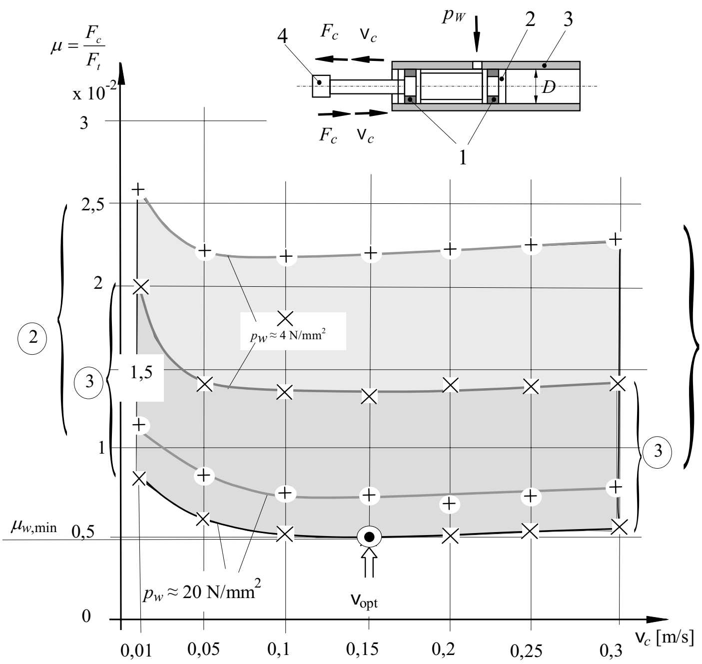

Fig. 6. Friction coefficient – reciprocating speed diagrams of Polyurethane U- rings (2) and NBR O-rings (3)$^{(2)}$

Here the average sealing pressure – obtained from the static sealing pressure distribution diagram – is about the value of the working pressure value (if $p_w <$ 4 MPa) and it is expressed by the formula of

$$\bar{p}_t = \frac{1}{b}\int_{x=0}^{b} p_t(t)\mathrm{d}x \approx p_w \text{ [4]},$$

while the maximum friction force value is determined by the stabilized friction force (F_c) and the direction change and operation factors (c_3 and c_4) [7]:

$$F_{\max} = c_3 \cdot c_4 \cdot F_c.$$

In this formula the direction change factor is subject to the seal applied (e.g. it is in the range of $1 \leq c_3 \leq 1.5$ for Polyurethane U-rings) and the operation factor is referring to the different lubrication conditions of outstroke ($c_4 \approx 0.5$ 'pumping operation') and instroke ($c_4 \approx 1.5$ 'motoring operation').

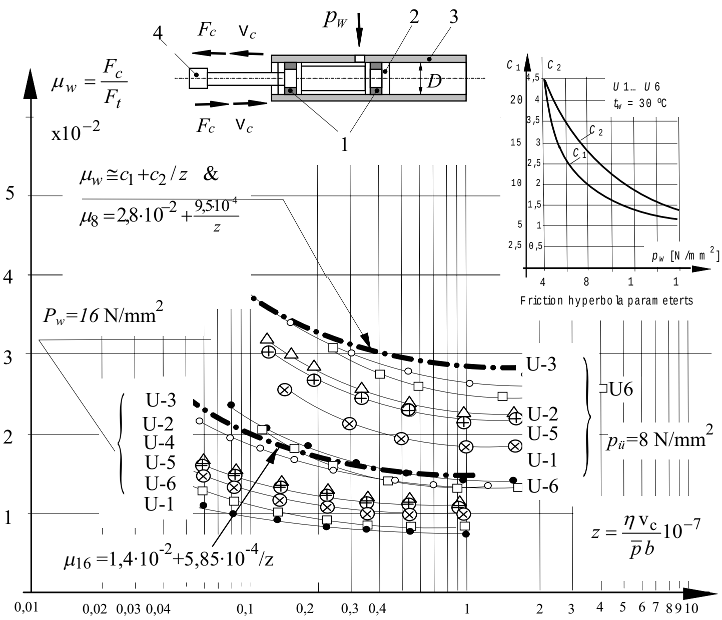

Fig. 7. Friction coefficient ('Stribeck') diagrams of Polyurethane U-rings (having different shapes in the same cross section)(2)

3.4. Comparing Friction Behaviours

Analysing and comparing the friction characteristic diagrams in case of V-ring packings (1), U-rings (2) and compact composite seals (*4* in *Fig.1*), the results show remarkable differences and improvements in friction-lubrication feature. These seals – in the order mentioned above – show remarkable reduction tendency of friction coefficient values (*Fig. 9*) [10]–[12]. (Here O-rings are not considered for comparison purpose as they are applied for reciprocating seals only in particular cases when leakage values are not limited strictly.)

The magnitude of friction coefficient of V-rings (1) – in the function of (constant) reciprocating speed – provided the greatest values all along the test speed limits. Here the friction coefficient curve had the sharpest and continuous reduction at increased speed values (*Fig. 9*). It suggested the presence of an increasing hydrodynamic lubrication effect, while the seal was still in mixed friction state within the whole test speed range.

For U-ring seals (2) the friction characteristic change was more favourable, started with a relatively big initial friction coefficient value, which was continuously reduced by the increasing test speed. The running down of the friction coefficient curve indicated the presence of the increasing hydrodynamic lubrication effect.

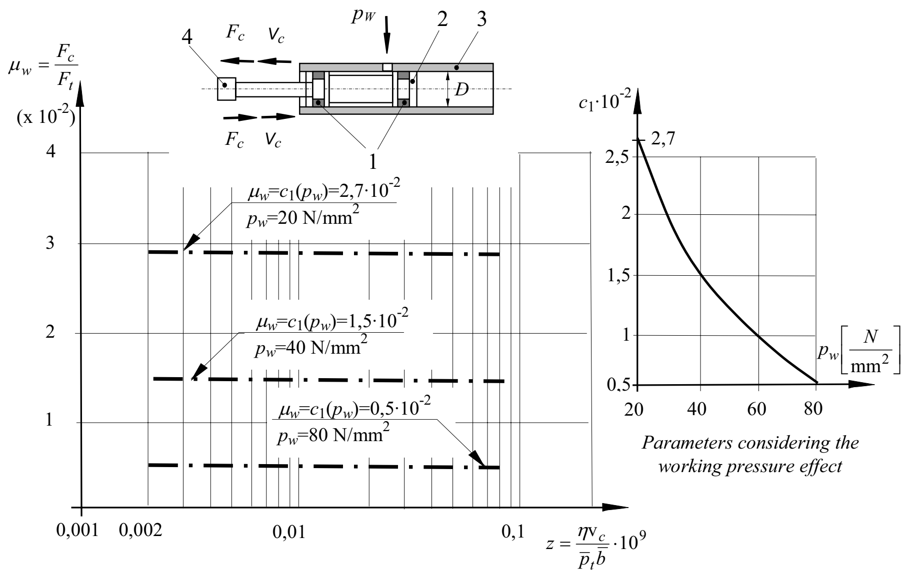

Fig. 8. Friction coefficient diagrams of a compact seal (O-rings having reinforced buck up rings)

Even lubrication optimums were found – having $\mu_{w,\min}$ – in many cases between the reciprocating speed limits (*Fig. 5* and *9*).

The type of the 'compact composite seal' (4) the starting values of friction coefficient provided smaller protruding values. The run out of the friction curves might suggest to be substituted by a constant run out (*Fig. 8* and *9*). Here the magnitude of friction coefficient values was about in the same range obtained for 'U' rings. However, the anti stick-slip characteristic of the PTFE content reduced the initial friction coefficient value reasonably.

Worth mentioning that – similarly to the regular compact composite piston seal – the extremely high pressure ($p_w < 100$ N/mm^2) O-ring seal having reinforced PTFE back up rings showed an approximately constant run out of the friction coefficient curve in the whole range of the test speed ($v_c < 0.01$ m/s). Here the high pressure and the motion squeezed some PTFE content of the back up rings into the sealing gap and covered the O-ring friction surface with a thin layer of PTFE material. This thin layer worked like a 'quasi-PTFE piston ring' and it was the reason of the similar friction coefficient curve run out of the two seemingly different seal types (*shapes 2.2 and 4.2 in Fig. 1*).

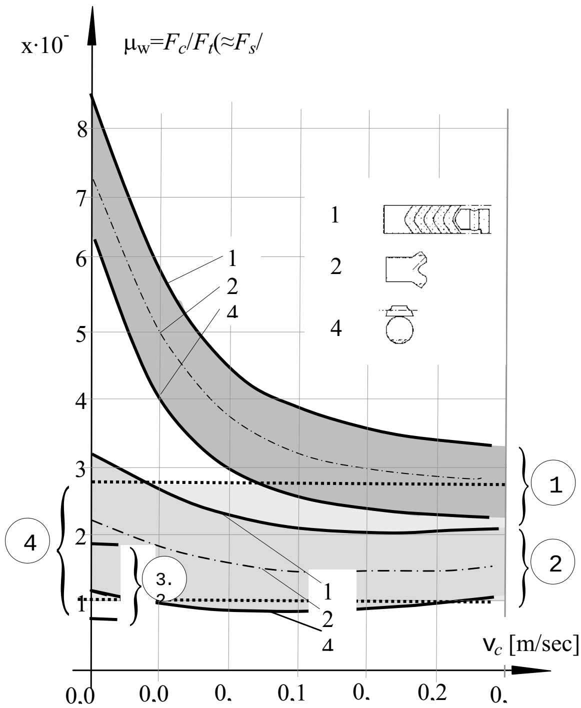

Fig. 9. Friction coefficient in the function of reciprocating speed for hydraulic piston seal[(2)(4)(5)]

4. Seal Efficiency, Housing Volume and Overall Economy

In order to compare some frequently used reciprocating hydraulic seals the selected ones are V-ring (1), U-ring (2) and a composite compact seal (*4 in Fig. 1*). The tendencies of the mechanical efficiency ($\eta_m = 1 - \mu_w$), the housing volume (V) and the length (L) changes are demonstrated well by a comparative study of these selected hydraulic seals.

The tendencies show an astonishingly fast improvement in the mechanical efficiency while the seal housing volume reduced immensely within the mentioned order of improvements. Simultaneously of these results economics showed reasonable improvements due to the smaller seal housing requirement, the smaller piston surface for higher system pressure and the better efficiency (smaller friction loss) for higher system pressure. All of these factors tend to reduce the mass of the hydraulic equipments and improve their overall economy.

Using the relationships of the diagrams the improvements of design changes and mechanical efficiency can be presented in the example of a selected and typical hydraulic equipment as follows (*Fig. 10*):

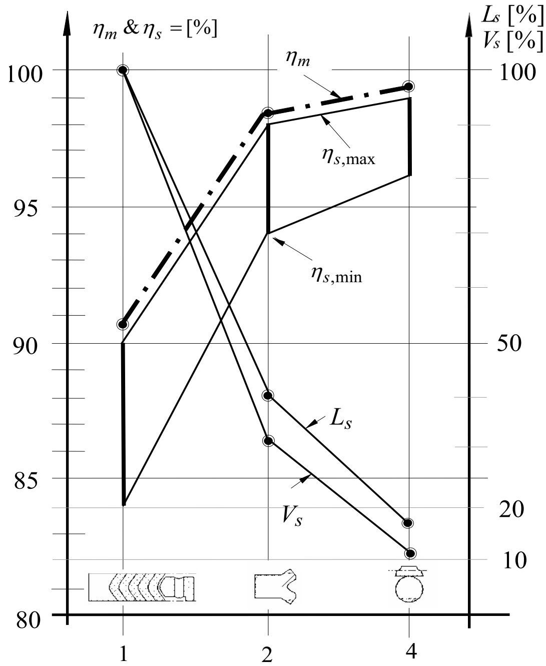

Fig. 10. Efficiency and housing volume requirement of different seal types

- The hydraulic cylinder volume (mass) is reducing reasonably – especially for short strokes – by applying U- rings instead of V- rings and the volume reduces further by applying compact composite seals instead of V-rings. Here – considering the same working pressure – the cylinder volume reduction is about 33% in the first case and 46% in the second case (for a selected stroke length of about 40 mm).
- The efficiency improvement is astonishingly fast as well and does not really depend on the stroke length. The mean value of efficiency is about 87% for 'V' rings, about 96% for 'U' rings and more than 98% for the compact composite seals (*Fig. 10*).

Symbols, Denominations and Notes

F_c [N]	Stabilized friction force value (at the middle of the stroke)

F_t [N] $= p_w \cdot A_t$	Sealing force
$F_{\max}$ [N]	Maximum friction force
$F_{\max}$ [N] $= A_D \cdot p_w$	Maximum axial force produced by the cylinder
c_3	Direction change factor
c_4	Operation factor
$\mu_w = F_c/F_t$	Friction coefficient
A_D [mm^2]	Complete piston surface area
A_t [mm^2] $= b \cdot D \cdot \pi$, or $b \cdot d \cdot \pi$	Friction surface of the seal
D [mm]	Piston hole diameter, or piston seal outer diameter
d [mm]	Piston rod diameter, or rod seal inner diameter
p_w [N/mm^2]	Working/operating/system/test pressure
p_t [N/mm^2 $= p_{t0} + p_w$	Sealing pressure
p_{t0} [N/mm^2]	Starting sealing pressure (due to the interference)
$\bar{p}_t$ [N/mm^2]	Average sealing pressure
v [m/s]	Reciprocating speed
v$_c$ [m/s]	Constant (or stabilized) reciprocating speed (at about the middle of the stroke)
$Z = \eta \cdot v_c/\bar{p}_t \cdot b \approx \eta \cdot v_c/p_w \cdot b$	Dimensionless number
η [Ns/m^2]	Dynamic viscosity
$\eta_s = 1 - m_w$	Efficiency of the seal
$\eta_m = 1 - \nu_m$	Mechanic efficiency of the cylinder
$\nu_m = Fc/F_{\max}$	Loss of the hydraulic cylinder
V [mm^3]	Volume of the sealing house
L [mm]	Length of the seal housing

(1) Seal parameters: U-ring, SI 65 IRHD, $D = 70$ mm. Test parameters: Hidro 20 oil, $t_w = 20 - 24$ °C, $v_c < 0.35$ m/s, $0 < p_w < 16$ N/mm^2 and $R_a < 0.16$ μm.

(2) Seal parameters: U-ring, $D = 70$ mm, PU 90 IRHD. Test parameters as above.

(3) Seal parameters: O-ring, $D = 70$ mm, NBR 70 IRHD. Test parameters as above.

(4) Seal parameters: Compact seal, O-ring and two glass fiber reinforced back up rings. Test parameters: oil-water emulsion, $t_w = 20 - 24$ °C, $v_c < 0.1$ m/s, $p_w < 80(100)$ N/mm^2 and $R_a < 0.08$ μm.

(5) Seal parameters: Compact composite seal (+ 2 pcs reinforced PTFE piston rings), $D = 80$ mm. Test parameters: Hidro 20 oil, $t_w = 20 - 24$ °C, $v_c < 0.35$ m/s, $p_w < 20(25)$ N/mm^2.

References

[1] BLOK, H., Inverse Problems in Hydrodynamic Lubrication and Design Directives for Lubricated Flexible Surfaces, *Symposium on Lubrication and Wear*, 1963 University of Huston, Texas, USA.

[2] MÜLLER, H. K., Leckverluste und Reibung von elastische Dichtungsringen an hin- und herbewegten Kolbenstangen, *Konstruktion* Heft 4 (1963), pp. 149–157.

[3] HOPP, H., Untersuchungen über den Reibungswert von Dichtelementen für Hubewegungen, *Hydraulik und Pneumatik Technik*, April, 1957.

[4] BISZTRAY BALKU, S., Dugattyútömítések surlódása és tömítetlensége hidraulika elemekben. (Friction and Leakage of Piston Seals.) (In Hungarian) (Diss.) BME, 1984. Budapest.

[5] WERNECKE, W. P., Analysis of Reciprocating Sealing Process, *11-th Int. Conf. On Fluid Sealing*, 1987, Cannes, France.

[6] PROKOPP, H. J. – MULLER, H. K., Film Thickness Contact Pressure on PTFE rod seals, *12-th Int. Conf. On Fluid Sealing*, 1989, Brighton.

[7] BISZTRAY BALKU, S., Tribology of Hydraulic Seals for Alternating Motion, *Periodica Polytechnika, Mechanical Engineering*, **39** No. 3–4 (1995).

[8] WEISS, H., Is it a Future Progress in Seal Development for Fluid Power Application, *Sterowanie I napped hydrauliczny*, **2** (1955), Wroclaw, Poland.

[9] BISZTRAY BALKU, S., Tribology of Elastomeric and Composite Reciprocating Hydraulic Seals, *Periodica Polytechnika, Mechanical Engineering*, **43** No. 1 (1999).

[10] BISZTRAY BALKU, S., Hidraulika dugattyú és dugattyúrúd tömítések tribologiai tulajdonságai, *Gép*, LII. Évfolyam, (2001) pp. 10–11.

[11] BISZTRAY BALKU, S., Design Developments, Tribological Characteristics and Efficiency of Reciprocating Seals, *Proceedings of the Third Conference on Mechanical Engineering*, 2002, Budapest.

[12] BISZTRAY BALKU, S., Development of Hydraulic Seal Design, Tribology and Efficiency, *XIth International Scientific-Technical Conference, Seals and Sealing Technology…*, 25–27 of May, 2004 Wroclaw, Polonica Zdroj, Poland.