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TRIBOLOGY OF ELASTOMERIC AND COMPOSITE RECIPROCATING HYDRAULIC SEALS

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Abstract

The present paper provides a short historical review on the attempts done to reveal the tribological behaviour of the reciprocating hydraulic seals. The discussed topics are regarding to the development trends in hydraulic piston and piston rod seals, the concerned tribological models, their utilizations and critical analysis. Furthermore it includes the evaluation of main operating characteristics – leakage and friction losses – and the (concerned) endurance tests and friction characteristics tests as well, based on the author's research results.[1]

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

1. Introduction

The significance of the tribological behaviour – friction, lubrication and wear – of the different groups of friction seals may be assessed by analysing the friction and leakage performances obtained from endurance and friction characteristic tests regarding to the operating parameters and conditions concerned.

The kind of motion between the lubricated frictioning surfaces principally effects the lubrication conditions. According to the motion the friction seals may be classified as rotary shaft seals, alternating rotary and reciprocating valve seals and reciprocating, or piston and piston rod seals.

Regarding the tribological behaviour of seal, the importance of it depends mostly on the magnitude of the losses (friction force, leakage and wear) on the operating parameters on one hand and on the demands of the application on the other hand. Big and not permitted losses should be avoided to answer efficiency and environmental protection and other requirements.

Elastomeric and composite materials are extensively used for radial lip seals of shafts, valve shaft (rod) seals and reciprocating piston and piston rod seals. Comparing to the most typical operating conditions of the radial lip seal to the piston

[1] Friction force determination and recommended calculation method was published in an earlier issue of this periodical [11]

and piston rod seals some substantial differences are found not only in the kind of motion but in the applied operating pressure range, too. The reciprocating motion has changes in lubrication conditions at each stroke end and also the magnitude of the operating pressure range is much higher, especially for hydraulic piston and piston rod seals.

These circumstances, the relatively big losses, give special emphasis to the significance of the tribological behaviour of the reciprocating elastomeric and composite hydraulic seals.

The present paper attempts to provide a survey on the reciprocating sealing development trends in the seal design with special emphasis on the tribological models and behaviours concerned.

2. Development of Reciprocating Seals

The demands for higher reliability and efficiency rates for the hydraulic systems necessitated the extensive research and development programs in sealing technology, too.

In the field of reciprocating hydraulic seals the first well documented methodical research program was closely connected to the development projects of hydraulic actuation and control, made to air force order in the second world war [1].

During the next decades the industrial boom generated fast developments in almost all fields of the technology including material processing technologies and hydraulics, too. Higher operating parameter (pressure, speed and temperature ...) limits, efficiency and reliability were demanded for machineries. All these requirements and possibilities made great impacts on the development of sealing technology in general and on reciprocating seals in particular. The reciprocating, or piston and piston rod seals went through great changes – regarding material, design, form, sealing edge shape and accuracy as well – from the early types up to the present elastomeric and composite seals (*Fig. 1*). Regardless the stuffing box gland packings the development of reciprocating seals followed three different major lines. These development lines were started with impregnated leather *U*- and *V*-rings and later on with simple elastomeric (rubber) *O*-rings, consecutively. During the developing processes all three lines of seals showed the same tendency. They left the unnecessary elements, the sealing edge was produced more accurately on one hand and the seal became more sophisticated, to answer the requirements by applying the method of function distribution on the other hand. So, different element – or incorporated part – of the seal provides the sealing effect, the thrust, the guiding and the surface wiping or cleaning. Consequently, the different function requires different design, shape and material characteristics and in this regard most of the highly loaded seals become ‘composite’ seals some way or other.

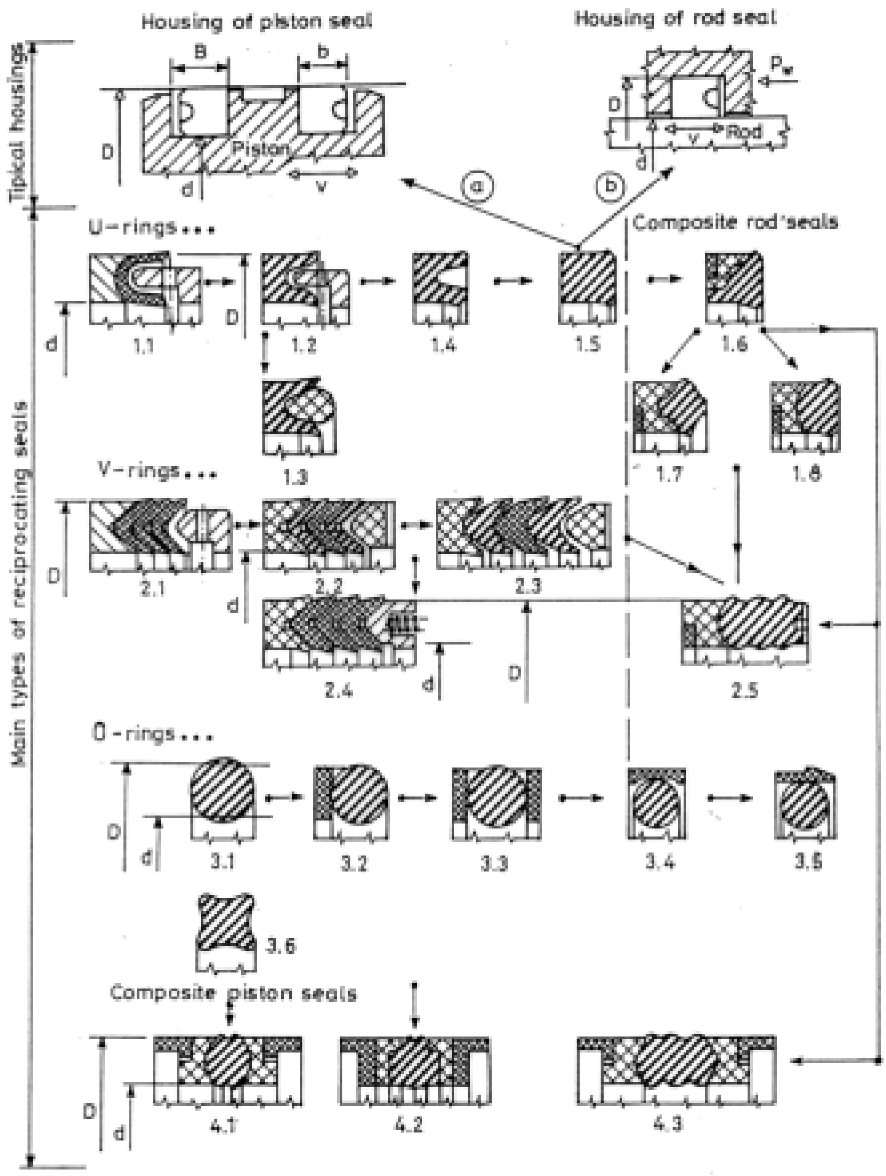

Fig. 1. Reciprocating seals, main types and development

3. Principles of Sealing Mechanism

In general, the long accepted basic principles for proper sealing effect and sealing mechanism include the following proved statements:

- The starting sealing effect (p_{to}), before operating pressure is applied, is produced by the overlapping – or interference – of the sealing (edges) on the cylindrical contact surfaces at any type of elastomeric or composite seals.
- By elevating the operating pressure value, the sealing pressure is automatically and proportionally increased ($p_t = p_{to} + p_w$) as elastomeric materials behave (on high pressure) like high viscosity fluids. The effect of this phenomenon results proper 'automatic' sealing effect all along the operating pressure range.

4. Tribological Behaviour, First Model Attempted

Above these principal statements the tribological – friction lubrication and wear – behaviour of the seals should be revealed. Besides the measurements of the main operating characteristics (friction and leakage) during endurance tests of seals, which resulted a number of friction and leakage curves, an early attempt was made to describe the lubrication phenomena by a 'rigid wedge-shape' model.

Unfortunately the rigid model was found unsuitable to describe the lubrication and friction phenomena in the fluid film existing between the elastomeric seal and the alternating surface [1]. Consequently the measured and calculated values did not show reasonable relationship neither for leakage nor for friction. Therefore the friction force and leakage endurance curves and friction characteristic curves, obtained from tests, provided the only essential sources of information on the tribological behaviour of the seals.

5. Endurance Test Curves Lasting Behaviour of Friction and Leakage

Due to the lack of useful model and formulae describing any of the tribological behaviours, the development of seals was based only on the results of the endurance and friction characteristic tests for several decades.

However – in spite of the later developments – the endurance (long lasting) test and friction characteristic test curves still remained to have deciding roles in the development and research works for seals.

Characteristics of friction force and leakage curves are shown in *Fig. 2* and *Fig. 4*. Furthermore, typical friction force and leakage endurance curves are presented by *Fig. 3* and *Fig. 5*.

To 'generalise' the friction force and leakage endurance curves the coefficient of friction ($\mu_c = F_c/F_t$) and some specific leakage (q[mm^3/cycl]) substitute them

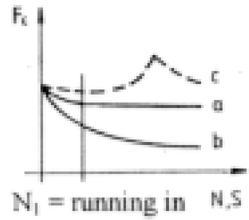

Fig. 2. Characteristics of friction force curves of elastomeric seals endurance tests

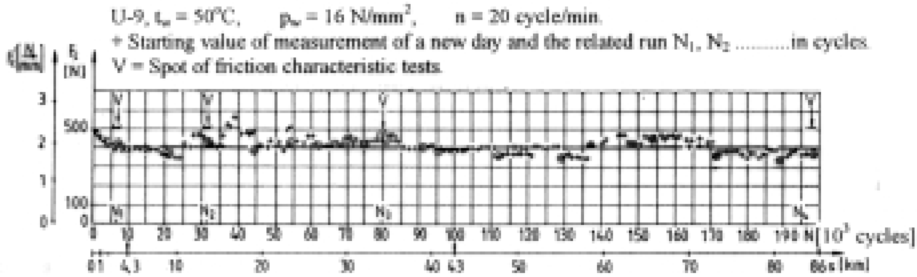

Fig. 3. Friction force curve of endurance tests of Polyurethan *U*-rings [9]

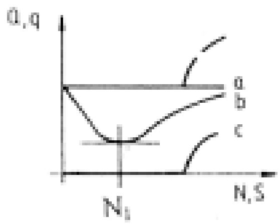

Fig. 4. Characteristics of leakage curves of elastomeric seals endurance tests

in the vertical axes of friction force and leakage curves, consecutively (from *Fig. 2* up to *Fig. 7*).

The friction force and leakage curves of endurance tests provide references on the average measured friction force and leakage values during the operation time. The changes in the measured values refer always to some happenings in the fluid film, the lubrication conditions, and in the friction and wear state of the frictioning surface of the alternating seal.

For properly selected seals – when all operating parameters and conditions are

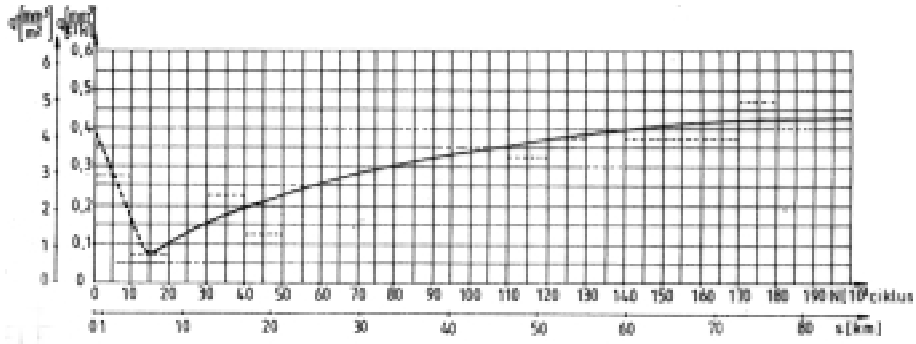

Fig. 5. Leakage curve of endurance tests of Polyurethan U-rings [9]

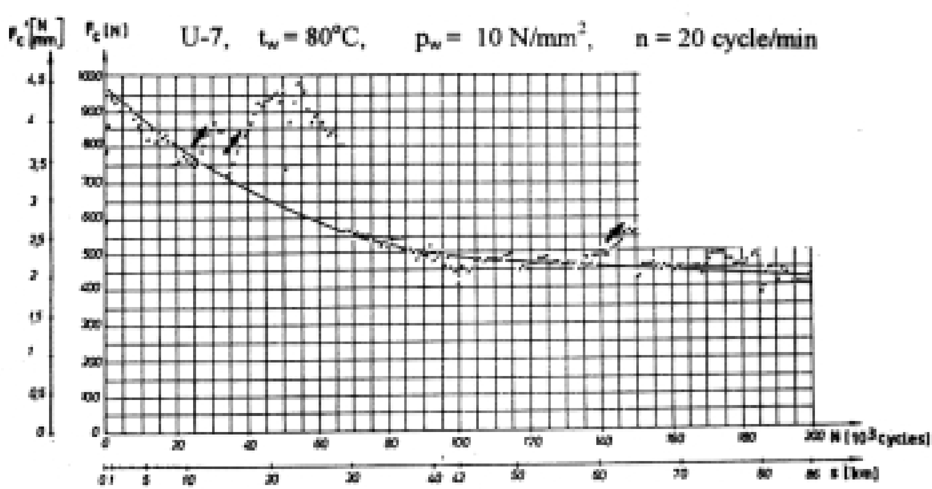

Fig. 6. Friction force curve of endurance tests of Silicon U-rings [9]

within the recommended ranges – both the friction and leakage endurance curves show a well balanced run out following the period of running in (marked by N_1 in *Fig. 2*). However, if the operating parameters are beyond the recommended range then the seal's tribological behaviour may show abrupt changes.

- Examining some typical examples from this point of view: if the operating pressure is too high for the operating sealing gap then the back-up edge of the seal penetrates into the sealing gap and some particles of the seal material are broken off. These particles damage the seal back-up and may cause sudden friction force increment (see the arrows in *Fig. 6*) while remaining in the

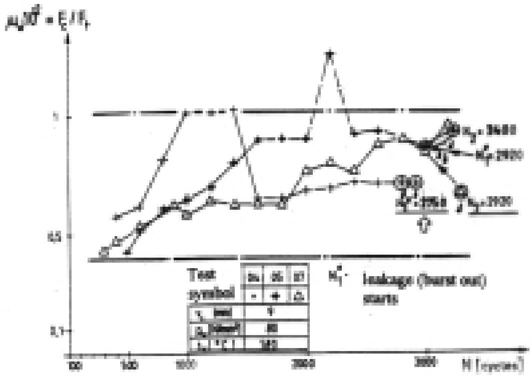

Fig. 7. Life-span of low speed piston seal at high pressure and temperature (water). $N_1^\bullet$ is the number of cycle where (burst out) leakage started [10]

sealing gap during reciprocating motion.

- For very high operating pressure – on very low reciprocating speed – and poor lubrication condition the abrasive wear of the seal is dominant and shows rather high values. As a result the seal life ($N_1^\bullet$) is limited by the amount of wear and consequently the remaining strength of the back up (elements) rings (*Fig. 7*).

6. Friction Characteristic Curves, Friction Force Determination

In order to reveal the friction characteristics of the seal – subject to the different design and working conditions – friction characteristic tests are applied. On the basis of these test results it is possible to compare the friction behaviour of different seals working in certain design and working conditions within the operating parameter ranges of the tests. Furthermore these results provide information – to a certain extent – to assume the expected approximate friction loss.

The friction characteristic tests are carried out after a certain cycle (N_1 is advised to be at least the 'running in' cycle) or length of path and provide the following two curve series:

- The *friction force – working pressure diagrams* (*Fig. 8*) were used first and applied for a long time by customers and designers as the major source of information to compare different types of seals and assess their friction behaviour [2].

  The friction force – working pressure curves at fixed alternating speeds show the friction force behaviour for elevated operating pressure grades. The curves refer to the applicable range of operating pressure ($p_w < p_{max}$) and

the possible maximum operating pressure $p_{\max}$ which can be permitted for longer operation time without early breakdown of the seal (*Fig. 9*).
While elevating the operating pressure for elastometric (and reinforced) seals the friction force curves show a balanced increasing in almost all cases. However, reaching a certain pressure the friction force shows a sudden increase indicating the worsening of friction and lubrication conditions and the spot of the possible maximum operating pressure, too.
(The resulted suddenly increasing friction force, due to the too high operating pressure $p_w > p_{\max}$, on the friction force can be seen in *Fig. 6*, too.)

- The *friction force – reciprocating speed curves* (*Fig. 10*) at fixed operating pressures show the change of the friction force for the change of the alternating speed. These curves give reference on the recommended and not recommended speed ranges and on the optimum alternating speed (v_{opt}) for application purposes. They may also be used for constructing functions or diagrams to determine the expected friction force for different operating conditions within the operating parameter limits of the tests [3], [6], [9].

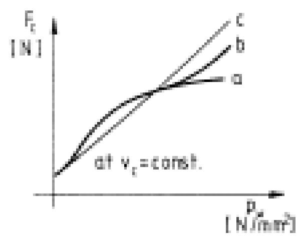

Fig. 8. Friction characteristic ($F_c - p_w$) curves at constant speed [9]

The idea to apply ‘Stribeck type’ diagram to describe and estimate friction forces was introduced already in the sixties [3] but due to some deficiencies of the measurements and other reasons it was not accepted for widespread application by researchers and professionals. However, the idea to express and estimate the expected friction force by applying this type of diagrams returned time and again and the ‘Stribeck diagrams’ (*Figs. 11–13*) were recently applied by a friction force calculation method [9,11].

7. Development in Modelling, Leakage Calculation

Regarding the tribological behaviour modelling of seals the first break-through was made by introducing the inverse hydrodynamic lubrication theory [4]. In the model, the flexible elastometric seal is moving on a lubricated rigid surface and the gap profile (gap size or lubricating film thickness change along the seal) points are decided from the static sealing pressure distribution curve (*Fig. 14*).

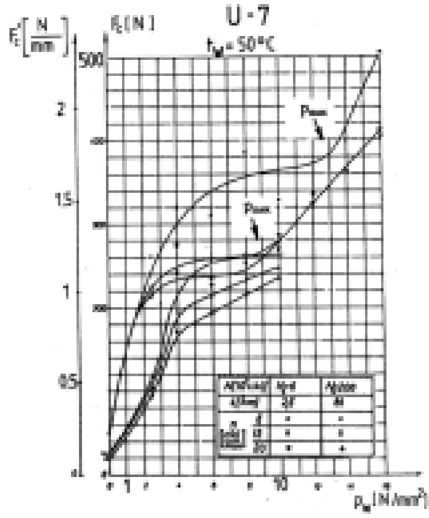

Fig. 9. Silicon rubber U-ring friction force operating pressure $(F_c - p_w)$ curves [9]

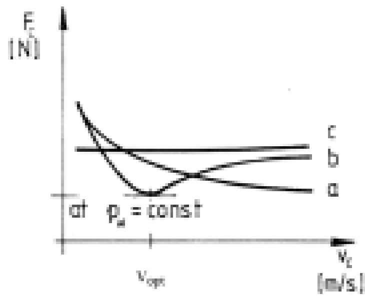

Fig. 10. Fricxtion characteristic $(F_c - v)$ curves at constant operating pressure [9]

From the differential equation – describing the inverse hydrodynamic lubrication theory – was derived the formula to calculate the characteristic lubrication film thicknesses (*Fig. 14*) both for out and instroke:

$$h^* = \text{constant} \cdot \sqrt{\frac{\eta v}{(\mathrm{d}p_t/\mathrm{d}x)_{\max}}}\,.$$

According to the relationships the film thickness value is influenced by the working medium viscosity, the speed of the motion and moreover by the maximum gradient

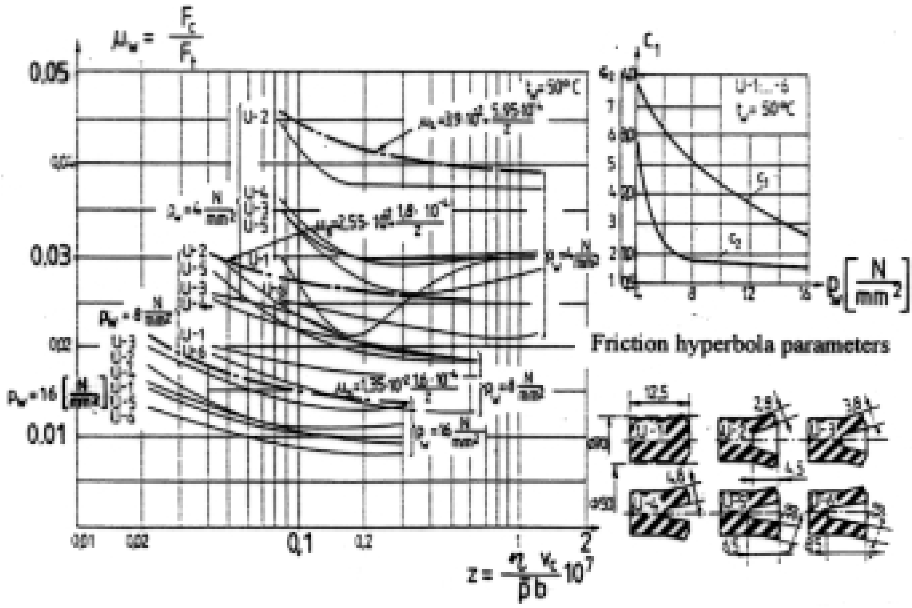

Fig. 11. Friction coefficient (Stribeck type) curves of different shaped Polyurethan U-rings at constant operating pressure [9], [11]

of the sealing pressure distribution curve taken from the direction of the motion. By the help of the characteristic film thickness values, related to the in and outstroke motion, an applicable leakage calculation method was obtained:

$$Q = \frac{\pi Ds}{2} \left(h^*_{\text{out}} - h^*_{\text{in}} \right) .$$

The relations revealed by the leakage formula provided useful guidelines for the methodical development of the elastomeric hydraulic piston and piston rod seals, with special regard to the sealing profiles and edges.

By applying proper seal design – profile and edge shape – and material characteristics, the leakage value rates can be well controlled and set for the requirements, which may even be zero leakage value as an output. In this method the gradients of the sealing pressure distribution curves were found to be the major influencing factors of the expected leakage value determination.

Consequently, the leakage could be determined with reasonable accuracy only by following the development of an adequate static sealing pressure distribution tester and method [5]. Typical static sealing pressure distribution curves are shown in *Figs. 14* to *17* [9].

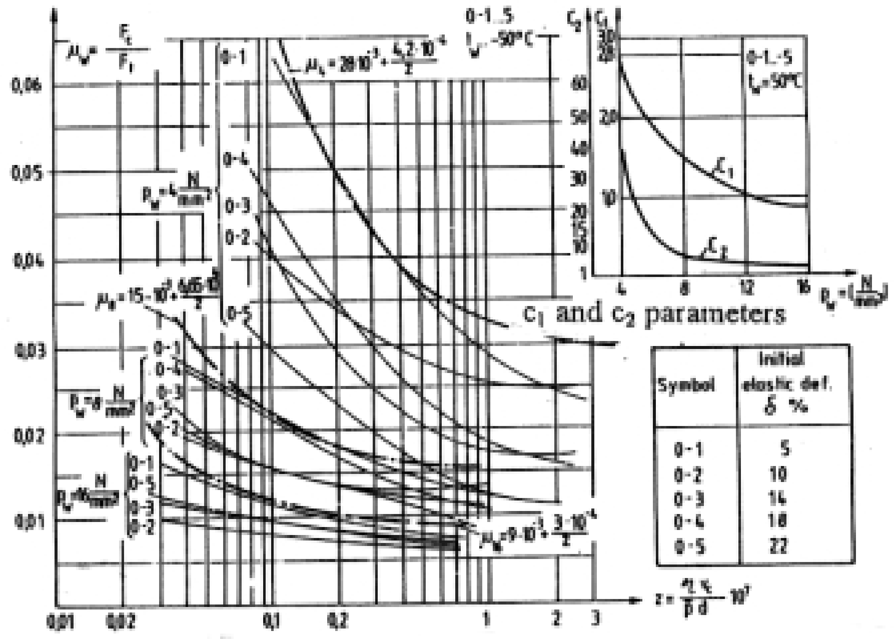

Fig. 12. Friction coefficient curves of *O*-rings, having different deformation [9], [11]

By applying the flexible model and the static sealing pressure distribution diagram, similar – or equal – leakage values were obtained both for measurements and calculation in case of the elastomeric seals (0-, *U*-rings, etc.) [9].

Anyhow, due to some deficiencies, this flexible model was not adequate to be the basis to calculate friction force values. As a tendency, the calculated friction forces were far below the measured values.

Returning to the sealing pressure measurements, the time dependent material behaviour of elastomers and plastomers and the speed of motion have reasonable effect on the sealing pressure distribution. In principle the dynamic pressure distribution provides theoretically correct information during operation on the pressure distribution in the fluid film between the seal and the frictioning surface but the accurate dynamic measurement has unfortunately serious difficulties.

While due to the sealing behaviour and measurement characteristics the static sealing pressure distribution has some remarkable features and advantages:

- The sealing edge is quite rigid, i.e. form keeping, for bmodern hydraulic piston and piston rod seals due to the design and material hardness of the sealing edges. Therefore the directions of the most critical tangents to the static and dynamic pressure distribution curves do not differ a lot. Also, in

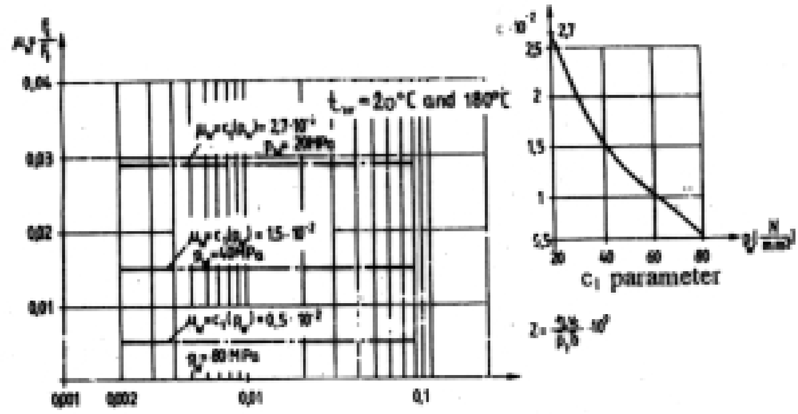

Fig. 13. Friction coefficient curves of *O*-ring seals having (fibre glass reinforced) back up rings [10]

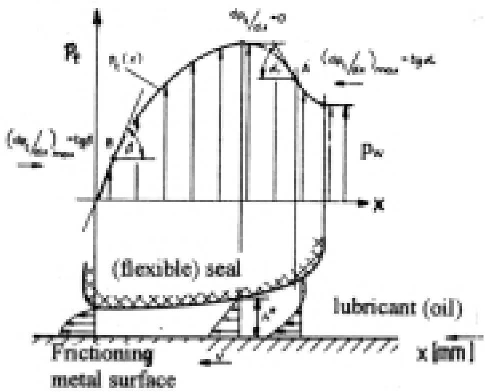

Fig. 14. Flexible tribological model, of reciprocating elastomeric seals

static sealing pressure distribution curves the pressure gradients – denoted by α and β – do not show reasonable differences for outstroke and instroke motion (*Fig. 16*). In this regard – for leakage calculation – the static curve may substitute the dynamic pressure distribution curve.

- Furthermore it does not require a big effort to develop a static sealing pressure tester with the required high accuracy.

As a result of the listed advantages the static sealing pressure distribution test

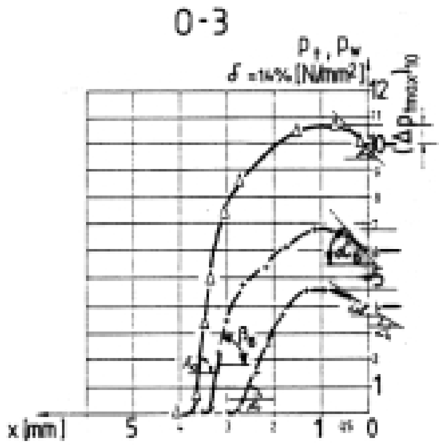

Fig. 15. Static sealing pressure distribution curve of NBR '*O*'-rings [4]

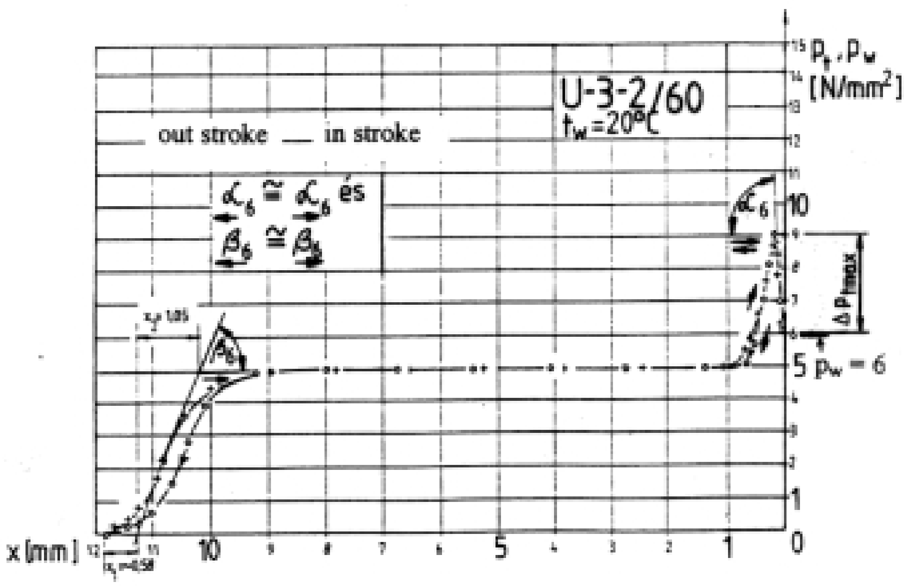

Fig. 16. Static sealing pressure distribution curve of Polyurethan *U*-ring for outstroke and instroke [9]

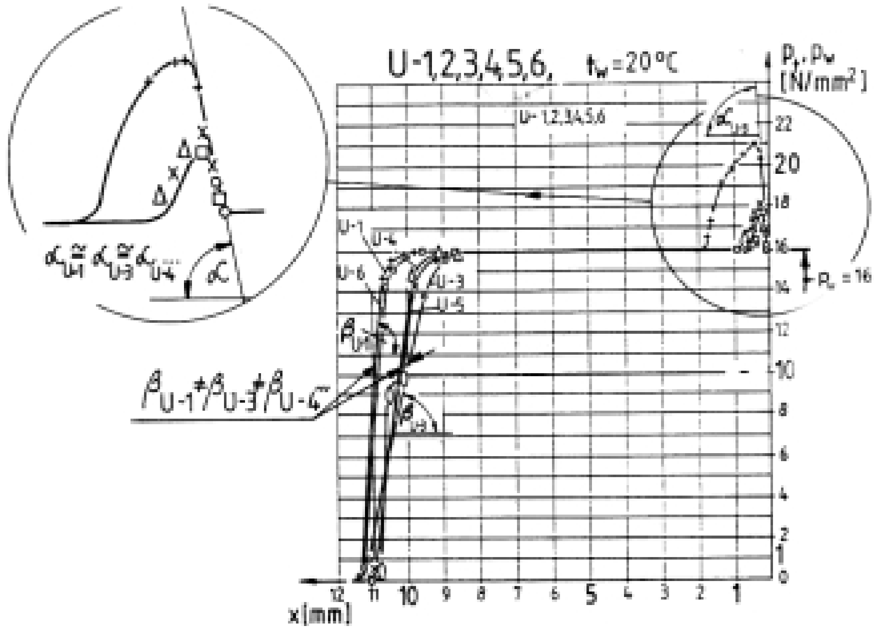

Fig. 17. Static sealing pressure distribution curve of Polyurethan *U*-rings (having different cross-sections) [9]

method and curves have been used for research and development up to present day.

8. Analysing Tribological Features

For deeper analysis of tribological features of seals the concerned relationships should be examined and projected (*Table 1*).

By time, reasonable advancements were achieved in testing the tribological features of the reciprocating seals:

- Seal lip and lubricating film temperature could accurately be measured and the temperature profiles (curves) were plotted along the frictioning surface of the seals. The plotted temperature distribution showed differences during outstroke and instroke for – the tested – single lip elastomeric seal. In case of the better lubricated outstroke the btemperature profile showed lower values. The maximum temperature difference between the out and instroke temperature profile values reached 30% and the highest total temperature of the lubricating film did not exceed 200 °C for regular piston rod seals in the speed range of the measurements [7].

Table 1. Relationships of tribological features

Design and working conditions	→	Tribological features	→	Main operating characteristics
Design characteristics, shape, dimensions...		Sealing force and pressure distribution		Friction force
Materials and surface properties		Temperature magnitude and distribution		Leakage
Operating conditions: operating pressure, alternating speed, temperature		Film thickness (height) and profile +influencing factors of lubricant and contact surfaces		Wear, life, endurance
	→		→	

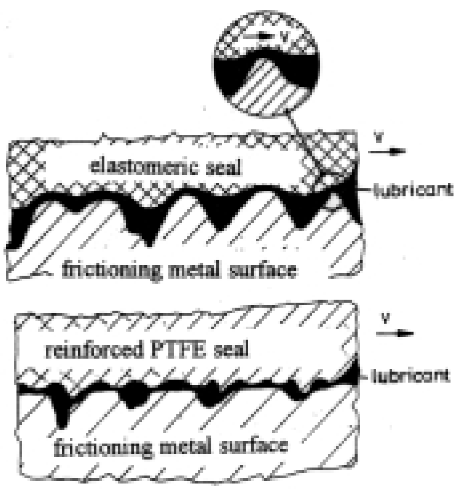

Fig. 18. True btribological model of elastomer and plastomer seals for alternating motion

- The film thickness – between the seal and the frictioning surface – showed very much varying values due to the surface roughness topography of the frictioning surface. Along the contact length (frictioning surface) of the seal the film thickness measured values differ by several order while other measurements proved the existence of continuous lubricating film even around $h = 0.01\ \mu$m thickness [7].

9. True Tribological Model and Friction Force

This result suggests the presence of some boundary layer lubrication, in some micro areas, beside the mostly hydrodynamically lubricated parts of the seal contact area. These micro areas of boundary layer lubrication – having 0.01 μm thickness range – greatly increase the friction force.

For reciprocating hydraulic seals, the operational lubrication condition is commonly called or characterised by ‘mixed friction state’, though there is a sophisticated, compound lubrication state which includes the operation of different lubrication conditions in the different micro areas, at the same time, under the seal.

The result of the film thickness measurements seems to be proved the existence and working of the true model even for up to date composite seals, both for elastomeric and plastomeric sealing edges (*Fig. 18*). In the ‘true model’ the lubricating film thickness is changing along the contact surface of the seal. The thickness change is depending on the topography of the frictioning metal surface and the material characteristics and surface quality of the elastomer or plastomer seal.

Applying the suggestion of the film thickness measurements and the true model there is not existing contradiction any more between the calculated and measured friction force values of reciprocating elastomeric seals. The calculated friction force values reach the measured values if only a minor portion of the frictioning surface of the seal is assumed to work under boundary layer lubrication state while the bigger part of the surface is under hydrodynamic lubrication (which may characterise the leakage).

Practically the greater portion of the measured friction force acts on some protruding peaks of the frictioning metal surface [8].

Consequently the true tribological model is raising very complex and sophisticated further problems, where the existing lubrication, friction, heat, etc. processes may probably be approached, described and solved by applying appropriate computer simulation methods.

Symbols, units and denominations

F_c [N]	– stabilized friction force value (at the middle of the stroke)
F'_c [N/mm] $= \frac{F_c}{\mathrm{d}\pi}$	– specific stabilized friction force value
F_c [N] $F_t \cong p_w \cdot A_t$	– sealing force
$\mu_c = \mu_n = F_c/F_n$	– (stabilized) friction coefficient
Q [mm^3]	– leakage
q [mm^3/cycle]	– leakage for one cycle
q' [mm^3/m^2]	– leakage for one sqm enveloped surface
A_t [mm^2]	– friction surface of the seal
D [mm]	– bore hole or groove dia. of the housing
d [mm]	– piston rod, or inside dia. of the seal
b [mm]	– seal width
$p_w = p_u$ [MPa] or [bar]	– working/operating/test pressure
pt_o [MPa]	– starting sealing pressure (before operating pressure is applied) at the sealing edge
p_t [MPa] $= p_{to} + p_w$	– sealing pressure
$\overline{p}$ [MPa]	– average sealing pressure
v [m/s]	– reciprocating speed
v_c [m/s]	– stabilized (constant) reciprocating speed
$Z = \eta v_c/\overline{p}b$	– dimensionless number
η [Ns/m^2]	– dynamic viscosity of the fluid
c_1 and c_2	– parameters of friction hyperbolas
s [mm]	– stroke length
S [km]	– frictional way, distance done by the seal
N [cycles]	– double strokes done
N_1 [cycles]	– double strokes done during running-in
$N_1^\bullet$ [cycles]	– seal life

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