PERIODICA POLYTECHNICA SER. MECH. ENG. VOL. 39, NOS. 3-4, PP. 225-246 (1995)

TRIBOLOGY OF HYDRAULIC SEALS FOR ALTERNATING MOTION

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Received: Febr. 3, 1995

Abstract

The present paper summarizes the friction properties of hydraulic piston and piston rod seals and the concerned calculation methods, based on test results published (in the literature) and on the own test results of the author. It includes the author's advanced friction force calculation method to estimate the expected friction force (loss) of different elastomeric and composite material seals.

The major aim of this paper is to recommend an advanced calculation method, which is based on the diagrams obtained from the 'Friction characteristic tests.' The method is suitable to calculate the expected friction force at any working parameter within the parameter range of the tests.

Keywords: tribology of seals, friction force calculation, alternating hydraulic seals.

1. Introduction

For operating and designing hydraulic equipments it is essential to know their operation characteristics, like: starting friction force, friction loss and force during operation, etc. These characteristics are mainly depending on the friction behaviours of the applied seals, working mostly in mixed friction condition. Researchers have carried out many test programmes to reveal the friction properties of alternating hydraulic seals and some of them recommended methods for estimating the expected friction losses as well.

The present paper summarizes the friction properties of hydraulic piston and piston rod seals and the concerned calculation methods, based on test results published (in the literature) and on the author's own test results. It includes the author's advanced friction force calculation method to estimate the expected friction force (loss) of different elastomeric and composite material seals.

Endurance test or test for lasting operation are running as long as the main operating characteristics (friction loss or leakage) of the seal are exceeding the permitted limit, while the working parameters (test pressure, alternating speed, and fluid temperature) are set and constant. For seals, even for properly selected and applied seals, the friction force curve shows a rapid change – mostly reduction – during the beginning of running and then the friction force change is very much reduced. This first part is the running-in period of each seal (*Figs. 1.1* and *1.2*).

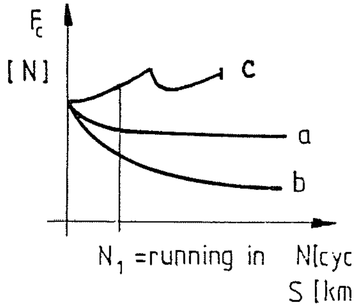


Fig. 1.1. Friction force curves of endurance test

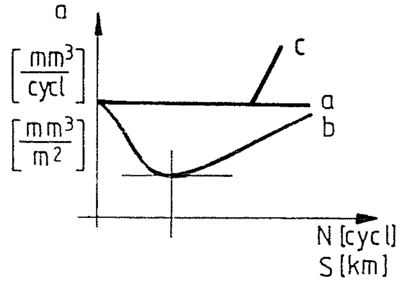


Fig. 1.2. Leacage curves of endurance test

Friction characteristic tests follow mostly the running-in period of the endurance test, in order to reveal the friction force behaviour in the whole parameter ranges of the test pressure and alternating speed on constant temperature levels. The obtained friction force – test pressure and the friction force – alternating speed diagrams (*Figs. 2.1* and *2.2*) give information on the friction force value changes in the whole parameter range of the test.

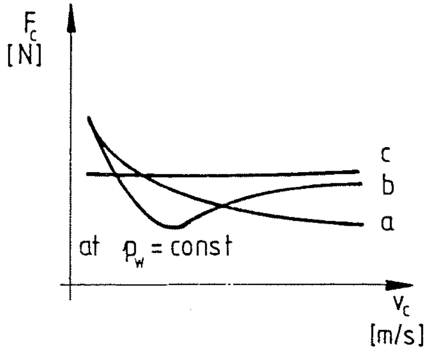


Fig. 2.1. Friction force alternating speed curves

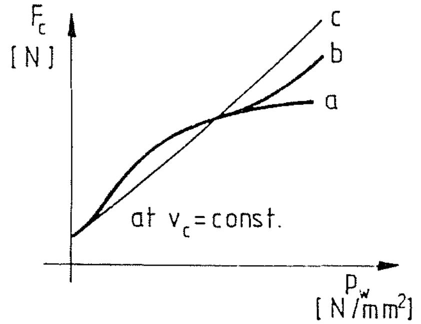


Fig. 2.2. Friction force-test pressure curves

By applying the concerned sealing or normal force the diagrams of friction force may be converted into generalized and more applicable forms. These forms are the friction coefficient – working pressure and the friction coefficient – alternating speed curves.

The recommended calculation method (*Table 1*) is based on the diagrams obtained from the friction characteristic tests and it is suitable to calculate the expected friction force at any working parameter within the parameter range of the tests (see Chapter 7).

2. Review on Friction Loss Determination

In this subject the first fundamental publication (in 1947) examined the friction behaviour of elastomeric piston and piston rod seals on theoretical basis [1]. Here, big number of tests were carried out and formulae were formed to calculate friction force and lubricating film thickness in pure fluid film lubrication condition, assuming rigid and conical seal-edge. However, the major effect of mixed friction condition during motion had not been considered.

Later, the effects of working pressure, standstill time, initial elastic deformation and surface roughness were examined on the friction losses of elastomeric O-ring seals [2]. Due to probable inaccuracies in measurements the friction force was found not dependent on the alternating speed.

Further on, many friction force – working pressure (curves) characteristics were published [3] for different piston and piston rod seals, at a given speed relation. In spite of some insufficiencies (e.g.: lacking data of test conditions, non-adequate driving mechanism for alternating speed...) these friction force – working pressure characteristics of the alternating seals, of different materials and designs, gave useful assistance for machine designers in selecting the proper types of seals.

Up to now, in many publications, the friction force behaviours were presented only by: the friction force – running time (or cycle) and the friction force characteristic curves at given test conditions [10, 12, 16, 18], but friction force calculation methods were not aimed to be presented by the authors.

In the way of test development it was found that the endurance tests and the friction characteristic tests provided proper description on the friction behaviour of hydraulic seals for axial alternating motion.

Table 1

Recommended friction force calculation formulae and coefficient

Denomination	Formulae	Remarks
U-rings,		c_3-direction change factor c_3 – 1.3 to 1.5 if
	$F_c = \mu_w F_t = \mu_w(A_t\overline{p_t} \cong \mu_w(bD\pi)p_w$	$v_c < 0.05$ m/s 1.1 to 1.2 if $0.5 < v_c < 0.3$ m/s
– stabilized friction force in the middle of the stroke	or $\mu_w(b \cdot d \cdot \pi)p_w$	c_4 – operation factor $c_4 = 1.5$ for motor operation, 0.5 for pump operation
– maximum friction force at the stroke ends		
O-rings and O-rings with back up rings		$A_t \cong \overline{b} \cdot D \cdot \pi$ for piston and $\overline{b} \cdot d \cdot \pi$ for piston rod seals
– stabilized friction force		$\overline{b} \geq \sqrt{\frac{d_2^2 \cdot \pi}{4}}$ for O-rings and $\overline{b} \cong \sqrt{\frac{d_2^2 \cdot \pi}{4}} + b_1 + b_2$
– maximum friction force		for O-rings with back up rings
Function or Stribec curve	$\mu_w = f(Z, p_w, t_w)$	$c_1(p_w)$ and $c_2(p_w)$ are parameter diagrams, see in *Fig. 4* and *Fig. 6*.
– hyperbola for U-rings and O-rings	$\mu = c_1 + \frac{c_2}{Z}$	– see in *Fig. 3* and *Fig. 5*
– horizontal line for O-rings with PTFE back-up rings	$\mu_w = c_1$	– see in *Fig. 7*
Z-dimensionless number for U-rings	$Z = \frac{\eta v_c}{p_t b} 10^7$	
O-rings	$Z = \frac{\eta v_c}{\overline{p_t} b} 10^7$ (or $\frac{\eta v_c}{\overline{p_t} d_2}$ as an approximation)	$\overline{p_t} \cong p_w$ (for $p_w > 4$ MPa)
O-rings with back up rings	$Z = \frac{\eta v_c}{\overline{p_t} b} 10^9$	

3. Endurance Tests

The endurance test – or test for lasting operation – is the basic test of all alternating seals to reveal their main operating characteristics, the friction loss and the leakage, during long lasting operation at fixed, or programmed, operating conditions.

For endurance tests the essential operating conditions are: the working pressure, speed, temperature and the medium, the friction surface quality, the seal and seal-space design.

The endurance tests may have two purposes:

- To qualify the seal for life, when the test is continued until the life of the seal is reached, where the life is determined by the permitted friction or leakage limitation, or until the minimum required service hours.
- To qualify the seal for friction characteristic tests the test is continued at least until the end of running-in period of the seal. During this first part of the endurance tests the measured operating characteristics – friction and leakage – show reasonable changes (*Figs. 1.1* and *1.2*). Before the friction characteristic tests are done the endurance test is recommended to be continued until the main operating characteristics of the seal show stabilized values. This portion of the endurance test is called running-in period of the seal.

  According to the endurance tests the running-in period was $S = 1.5$ to 3 km for alternating hydraulic U-rings and O-rings as well (when the working pressure value was $p_w \cong 16$ MPa and the working temperature values were $t_w = 20$ °C or 50 °C).

Endurance Test Curves

The most typical forms of endurance test curves are marked by a, b and c in the friction force – frictional way and in the leakage – frictional way curves (*Figs. 1.1* and *1.2*).

- Regarding the character of the friction force it shows some reduction at the first period of running for properly selected U-ring seals. Then, during the following long period of test, the friction force does not show reasonable change, curve 'a', indicating a balanced lubrication condition has been achieved between the seal and the alternating friction surface. From among the leakage curves 'a' and 'b' may correspond to answer the above when stabilized leakage values are measured following the running-in period.
- In certain cases, curve 'b', a continuous friction force reduction follows the running-in period. This indicates an improving but unbal-

anced lubrication condition between the seal and the frictioning surface. The above described is mostly typical for O-ring seals. (The unbalanced lubrication condition and the relatively higher leakage values, compared to U-rings and other developed seals, are the reasons why O-rings are not recommended to be applied for continuous long run operation in hydraulic cylinders.)

- If the operating pressure is too high for the applied elastomeric seal material, then the seal behaves like a high viscosity fluid and the seal's edge tends to be penetrated, by the operating pressure, into the clearance between the seal thrust and the friction surface. To eliminate this harmful effect either back up ring or reinforcement of the seal material, or both are applied, by which the critical applicable pressure of the seal is reasonably increased.

  When the operating pressures exceed the critical pressure then the seal material penetration tendency acts again. In such cases, curve '*c*', the seal was definitely not selected for endurance, it may work only for a certain life. Here, the friction force curve does not show a typical running-in period and the curve immediately stops at the seal failure which is indicated by a sudden or rapidly increasing leakage, following the exceeding of the permitted leakage limit.

4. Friction Characteristic Tests

The friction characteristic tests usually, but not necessarily, follow the running-in period of the concerned endurance test of a seal. The purpose of the friction characteristic tests is to reveal the friction (loss) force behaviour of a seal during the whole range of the main operating (working) parameters, i.e: the working pressure, the alternating speed and the working temperature. During the friction characteristic tests the friction force is measured at constant operation conditions, when only one of the main working parameters is subject to change. The friction characteristic tests are usually presented in a form, when the friction force (or friction coefficient) are on the vertical axis, while the working pressure or the alternating speed is on the horizontal axis of the diagram (*Figs. 2.1* and *2.2*). These two regularly applied types of diagrams are referring to the varying main working parameter, such as: friction coefficient – working pressure and friction coefficient – alternating speed diagrams.

- In the first case the friction characteristic tests are expressed in the form of friction force (coefficient) – working pressure diagrams at fixed working temperature, and at different alternating speed levels. This type of friction characteristic diagrams were published first and provided effective help for the designers to select among the different alternating hydraulic seals, on comparative basis, in respect of the measured (and expected) friction force [3].

- The friction force (coefficient) – alternating speed diagrams, express the friction behaviour change in the function of alternating speed, on different working pressure levels, at constant working temperature. The friction coefficient – alternating speed diagrams reveal the seals friction behaviour and lubrication condition much better than the friction coefficient – working pressure diagrams.
  The experienced most typical forms of the friction force – working pressure and friction force – alternating speed diagrams are noted by '*a*', '*b*' and '*c*' (*Figs. 2.1* and *2.2*).

Friction Force – Working Pressure Curve

The friction force – working pressure curves express the frictional behaviour of the seal (*Fig. 2.2*), at constant alternating speed, when the working pressure is changed. For most of the laestomeric seals – U-rings and O-rings, the form of the curve is shown by curve '*a*', where the friction force shows at start a rapid increment which is gradually followed by a slower increment of the friction force.

The slow increment in the friction force indicates the improved lubrication condition between the seal and the friction surface. Above a certain working pressure limit – may be called critical pressure – the elastomeric seal cannot keep it's form and by the operating pressure it is penetrated into the gap (clearance) behind the seal.

Due to this penetration effect the alternating surfaces tear off particles from the seal's back edge. These moving particles penetrate between the friction surfaces and deteriorate the local lubrication conditions by which the friction force increases. The sudden rise of the friction force – working pressure curve may indicate the above phenomena and provides information on the critical pressure and the permitted pressure range of the concerned seal.

Some of the multiple V-ring seals show continuous friction force grows proportional to the increasing working pressure, curve '*c*'. The curve shows unfavourable friction – lubrication condition all along the working pressure range. There is no indication of any hydrodynamic effect which could improve the friction – lubrication condition.

Friction Force – Alternating Speed Curves

The friction force – alternating speed diagrams (*Fig. 2.1*) provide the most effective way to express the friction – lubrication condition of the seals. Both curves '*a*' and '*b*' prove the effect of hydrodynamic lubrication by the rapid friction force reduction resulted by the increasing alternating speed. The friction force – alternating speed diagrams are suitable to decide the optimum and recommended speed range of the tested seal.

In case of very low speed and high pressure, applied for O-ring seals having PTFE fibreglass reinforced back up rings, hydrodynamic lubrication effect could not be detected. Due to the PTFE material characteristics the friction force measured values were (about) constant along the range of the test speed (see curve '*c*').

Stribeck Diagrams

It, is quite simple to transform the friction coefficient – alternating speed diagram to Stribeck diagram which is a more generalized form for expressing the friction behaviour of alternating hydraulic seals. At the same time, it is suitable to determine the expected friction force at any working parameters and seal dimensions within the parameter ranges of the concerned friction characteristic tests.

5. Friction Loss (Friction Force) Calculation

Starting from the 60-s the methods for friction loss (force) calculation came into the spotlight and several calculation methods have been developed, including nomogram charts [8], to calculate the expected friction force of reciprocating elastomeric seals. Calculation methods were recommended for elastomeric (NBR and Vulkollan) O-ring from 0 to 10 MPa working pressure range, for U-rings [8], [13], etc.

The effecting factors and conditions on the friction force behaviour were also examined. These factors and conditions were as follows: working parameters, (temperature, viscosity, pressure and alternating speed), standstill time, hardness of elastomeric material, friction surface quality and the phenomena of stick-slip.

At first, LANG elaborated the results of friction characteristic tests of piston and piston rod seals (O- and U-rings) in the form of Stribeck-type diagram, which made possible the friction coefficients calculation at any parameters (p_w and v_c) within the parameter ranges of the tests [5]. However, these tests were carried out only at low values of the regular working pressure and speed range of alternating hydraulic seals. The initial sealing force was calculated by assuming parabolic sealing pressure distribution, according to the Hertz-contact pressure theory, and by calculating the average sealing force a constant correction factor was recommended (to modify the sealing pressure). Nevertheless, in line with theory and (later) experiments this factor cannot be constant (even at low working pressures range) and the Hertz equation cannot be applied as well for elastomeric materials.

The elaboration of static sealing pressure test method made a great step forward and made possible a more correct and accurate sealing force and friction force determination for the elastomeric reciprocating seals [6]. The axial static sealing pressure distribution diagrams of different reciprocating elastomeric seals, gave accurate sealing pressure values at any point along the contact surface of the seal and became the basis of both the estimation of accurate sealing force and leakage [6], [14], [20], [22], [24].

6. Recommended Calculation Method of Friction Force

The recommended calculation method regards to alternating hydraulic seals (piston and piston rod seals) over the running - in period (*Fig. 1.1*), after the friction force is stabilized. It gives formulae to calculate the stabilized friction force at rectilinear constant speed (F_c) and the maximum friction forces ($F_{\max}$) at the beginning of the stroke, where the lubrication condition is not stabilized yet. Recommended formulae are given to calculate the friction forces and friction coefficients (*Table 1*). This calculation method may be used for alternating hydraulic seals in general, provided the results of friction characteristic tests and axial sealing pressure distribution test are known. (From the friction characteristic test results the friction force – alternating speed $F_c - v_c$ curves, *Fig. 2.1*, are plotted and used for the presented calculation method).

Here, in this paper, the friction force calculation method is introduced through the examples of:

- Polyurethane U-ring,
- NBR O-ring in the regular parameter range of hydraulics and
- O-ring with back-up rings for extreme parameter ranges of hydraulics.

Formulae and Relationships

To obtain the dimensionless number

$$Z = \frac{\eta v_c}{\overline{p_t} b} \tag{1}$$

for the Stribeck diagram the following should be considered:

- The viscosity (η) can be decided for known working fluid and temperature (t_w),
- The alternating speed stabilized value (v_c) may be measured and applied, or substituted by the value of average alternating speed (v), calculated from the stroke length (s) and alternating speed (n).
- The seal width is given (b) or it can be determined easily.
- From the axial sealing pressure distribution curve the average sealing pressure is:

$$\overline{p_t} = \left[\int_0^b p_t(x) \cdot dx \right] / b \cong p_w \ . \tag{2}$$

For higher pressure the average sealing pressure value is about equal to the value of working pressure for U-rings and similar profiles while in case of O-rings the difference between the two pressure values is reasonable. (The effect of this pressure difference results a friction coefficient change of 5 to 8% at the most steep part of the friction coefficient curve.)

According to the test results and the above consideration $p_w \cong \overline{p_t}$ still may be accepted as a good approximation for $p_w \geq 4$ MPa. (In case of $p_w \leq 4$ MPa a correction factor should be applied [20]).

The Stribeck *diagrams of U-rings* (*Fig. 3*) having different profiles, and the *Stribeck diagrams of O-rings* (*Fig. 5*) having different initial cross-section deformation, *can be represented* well *by the enveloping* friction *hyperboles*

$$\mu = c_1 + \frac{c_2}{Z} \ . \tag{3}$$

These friction hyperboles are enveloping, by covering from above, the test result fields separately at each test pressure level. These fields of test results include the curves of U-rings for each geometry and the curves of O-rings for each cross-section deformation. Thus the enveloping hyperbolas are representing the concerned seals at the concerned test pressure levels and may be recommended to be used for deciding the value of friction coefficient when accurate seal profile geometry of U-rings and initial cross-section deformation of O-rings are not known.

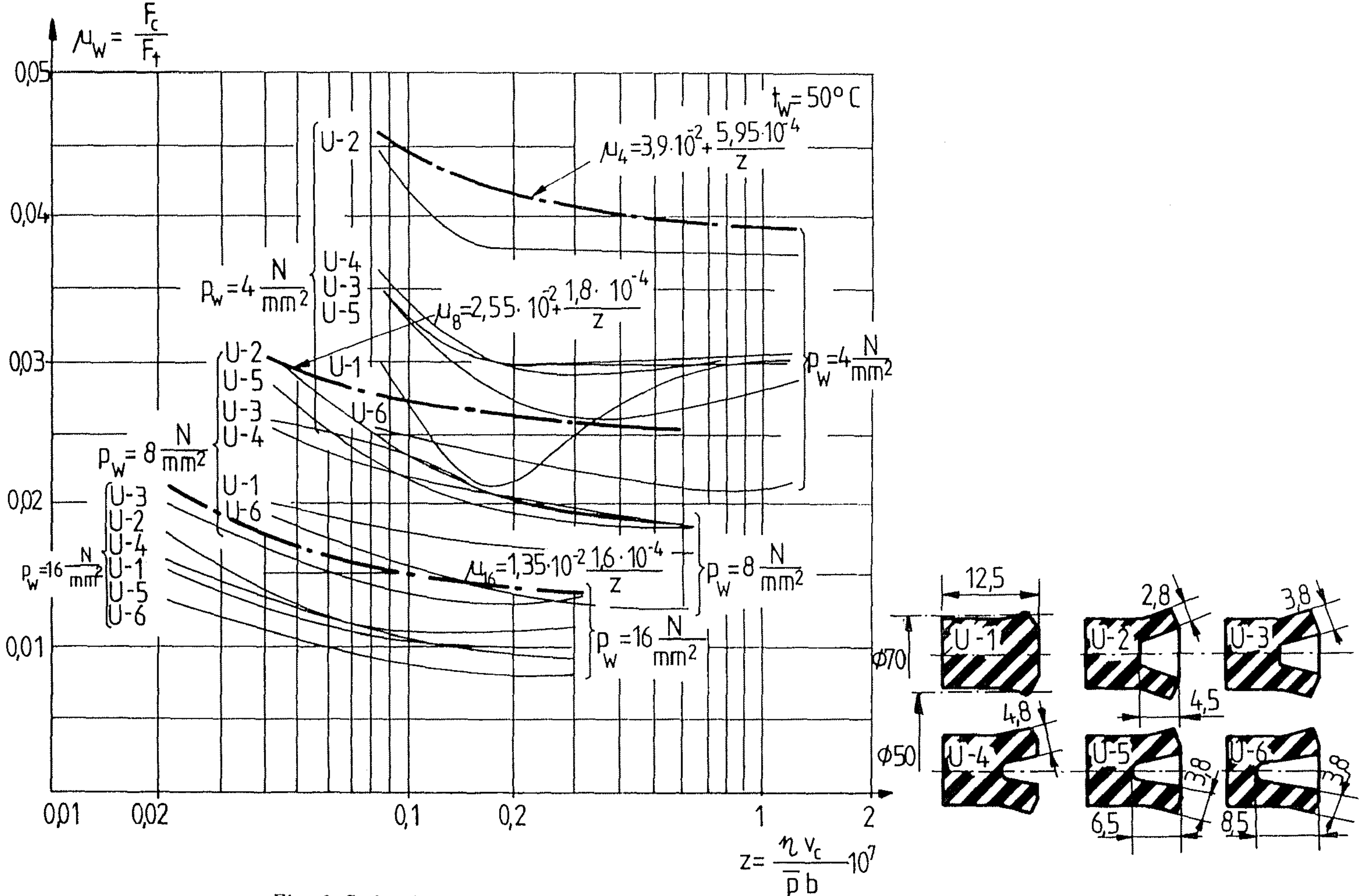


Fig. 3. Stribeck diagram of Polyurethan U-rings at $t_w = 10°C$ [20]

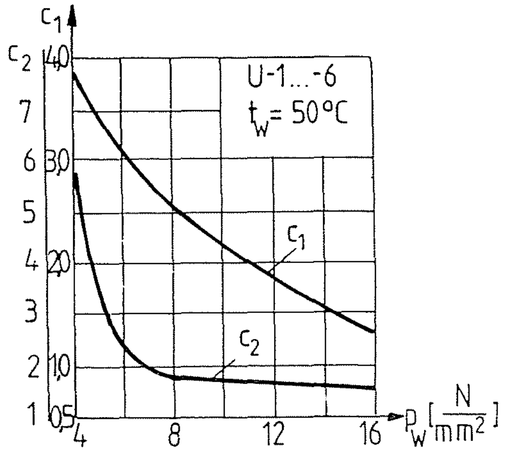


Fig. 4. Parameters of U-ring friction hyperbolas

Furthermore, if the expected working pressure does not fit to the test pressure, being an intermediate value, then the values of the constants c_1 and c_2 may be obtained from the concerned friction hyperbola parameter diagrams (*Figs. 4* and *6*) of $c_1(p_w)$ and $c_2(p_w)$.

In case of *special application of O-rings*, with PTFE back-up rings for very high pressure and low speed, *the enveloping curves are simplified to be constant lines* $\mu = c_1(p_w)$ (*Fig. 7*), where c_1, is in the function of the working pressure (*Fig. 8*). The friçtion coefficient depends mostly on the working pressure: if the pressure increases then the friction coefficient decreasas. This tendency may be explained by the effect of PTFE material, which penetrates – in the form of a thin layer – between the seal and the friction surface by the effect of pressure and displacement.

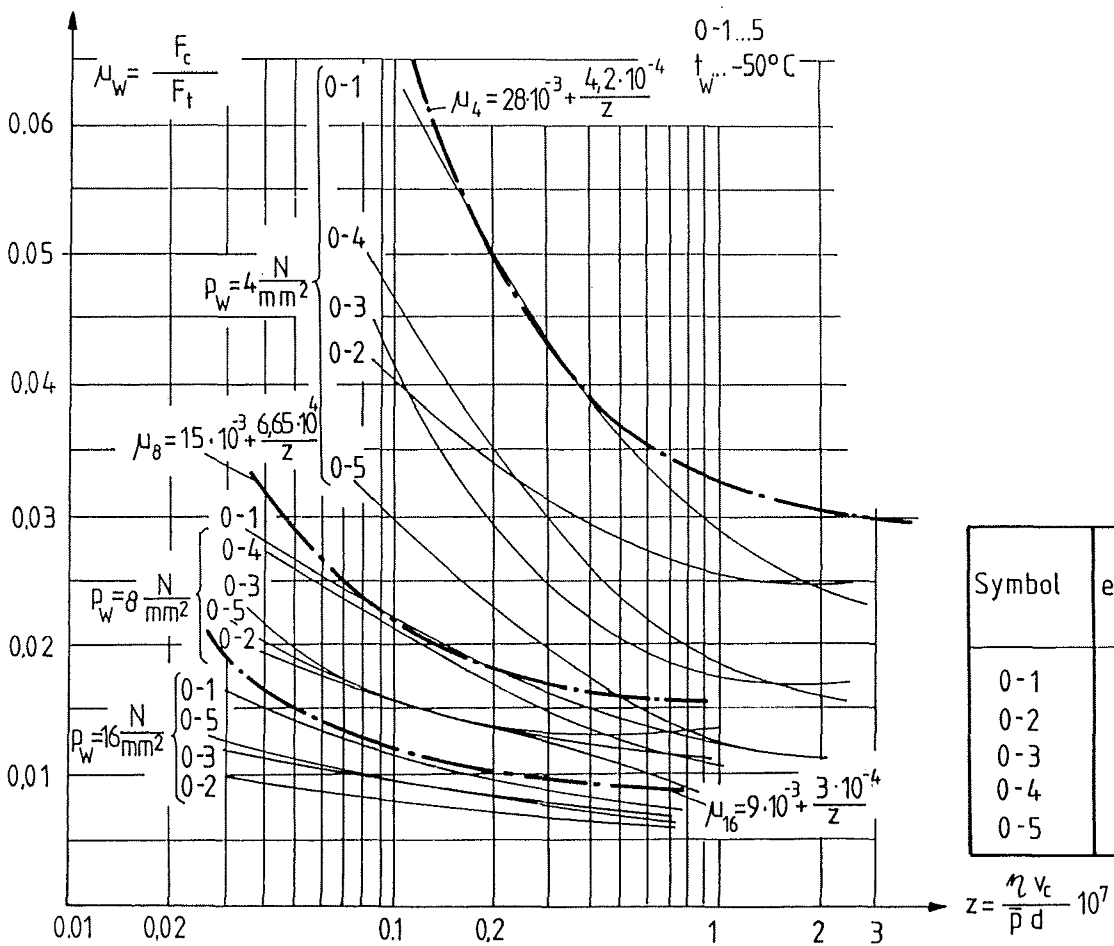
If the expected working pressure does not fit to any of the test pressure levels, being an intermediate value, then the value of c_1 may be obtained from the concerned diagram (*Fig. 8*).

7. Validity Range and Accuracy of the Friction Force Calculation

The validity range of the calculation is certainly set by the conditions and the parameter range of the friction characteristic tests concerned.

For polyurethane U-rings (90 IRHD) and NBR O-rings (70 IRHD)

- the working pressure range was 4 MPa $\leq p_w \leq$ 16 MPa
- the alternating speed range was 0.01 m/s $\leq v_c \leq$ 0.3 m/s



Symbol	Initial elastic def. δ %
0-1	5
0-2	10
0-3	14
0-4	18
0-5	22

Fig. 5. Stribeck diagram of NBR O-rings [20]

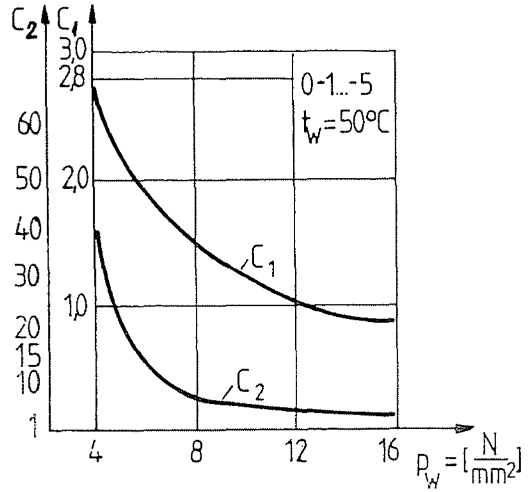


Fig. 6. Parameters, c_1 and c_2

- the working temperature values were t_w = 30 and 50 °C and the applied fluid was Hydro 20 oil.

For O-rings (83 FKM and 90 VI) with 2 mm thick, fibre glass reinforced PTFE, back-up rings the extreme working conditions were as follows:

- the working pressure range was 20 MPa $\leq p_w \leq$ 100 MPa
- the alternating speed range was 0.8 mm/s $\leq v_c <$ 12 mm/s.
- the working temperature values were t_w = 180 (190) and 30 °C and the applied fluid was water and water-oil emulsion.

Accuracy

Strain gauge membrane dynamometer was applied for the measurements with an estimated error of 3%. For seals position and alternating speed determination a standard inductive transducer was used.

The friction force measurement was made by calibrated strain gauge dynamometers and a measuring amplifier, having an accuracy of 3%.

Concerning the measured friction force values, they showed a maximum of ±40 (50)% deviation – meaning ±20 (25)% quadratic mean deviation value – from their own friction characteristic curves at low pressure (p_w = 4 MPa) and low speed (v_c = 0.02 m/s) values. The measured friction force values provided much smaller, about the half, of the above given deviation at

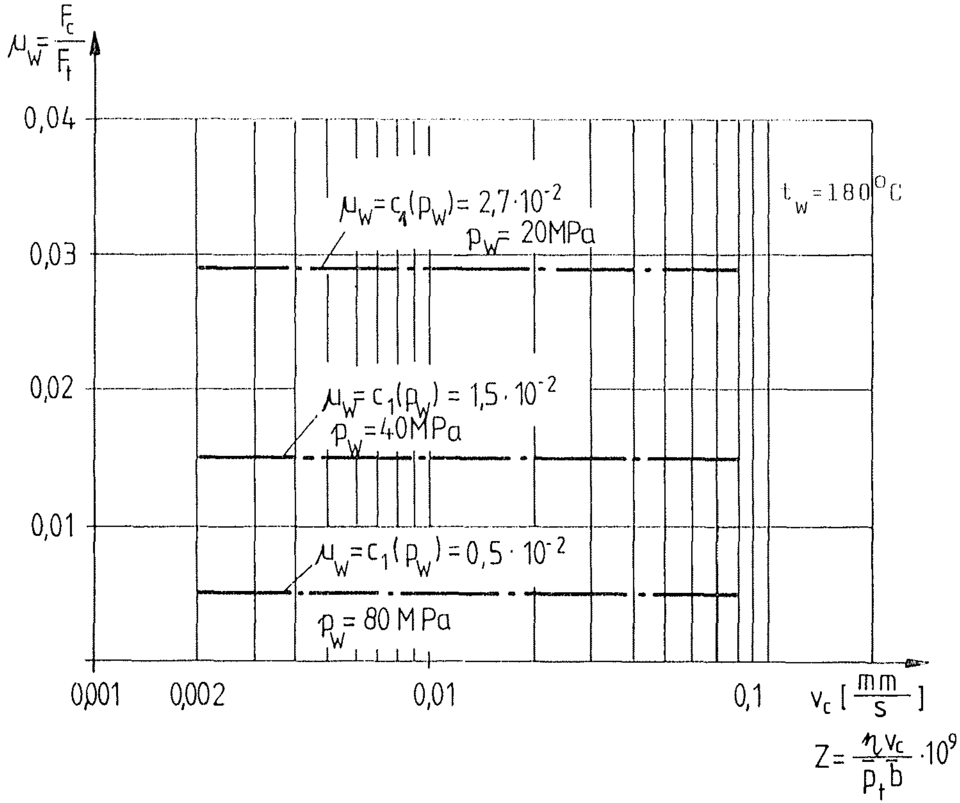


Fig. 7. Stribeck diagramm of O-rings seals having fibre glass black up rings (at $t_w = 180°C$ and $20°C$ the diagrams are similar) (23)

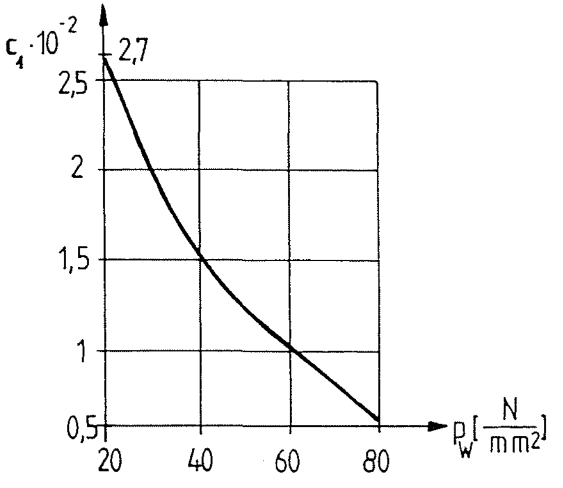


Fig. 8. Parameters $c_1(p_w)$ of O-ring having fibre glass back up rings

high pressure ($p_w \geq 16$ MPa). Considering the accuracy of the measured friction force values the measurement accuracy could be accepted, being much smaller value than the deviation values [20].

Regarding the accuracy of the recommended friction force calculation method, comparison was made among the values of some recommendations, provided by the literature, and the values provided by own measurements and recommendations.

Stribeck-curves were calculated and plotted from the information of references [8], [14] and they corresponded well to the recommended friction hyperbolas and to the measurement curves (*Fig. 9*).

For higher operating pressure the deviation between the reference value [14] and the value of the friction hyperbola was 0 (zero) at low speed and about + 40% at higher speed, while the deviation was only + 18% at low speed and −25% at high speed if the reference values were compared to the values of a curve obtained from own measurement for medium (predeformed) pretightened, $\delta = 14\%$, O-rings (see 0-3 in *Fig. 9*).

However, the deviations of the friction coefficient values are less favourable at low operating pressure.

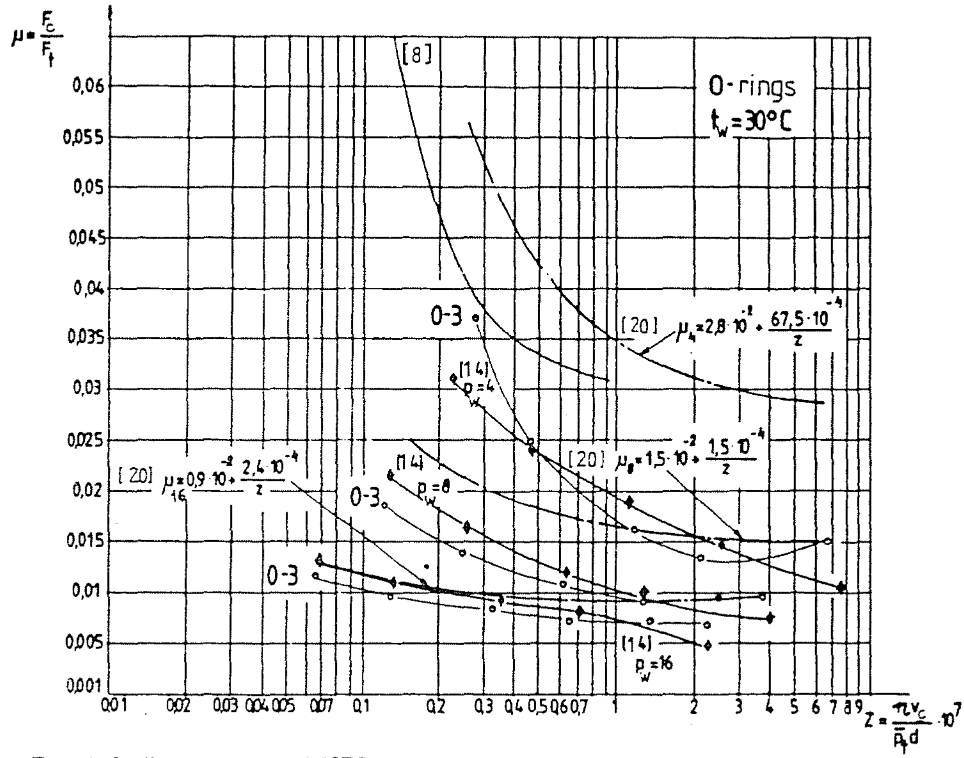


Fig. 9. Stribeck curves of NBR O-rings, comparing the published and own research results

8. Constant (or Stabilized) Friction Force

When the dimensionless number Z and the friction coefficient μ are decided, or obtained the stabilized expected friction force (at the middle of the stroke) may be determined (*Table 1*) as follows:

$$F_c = \mu F_t = \mu(A_t \overline{p_t}) \cong \overline{p_t}\bar{b}D\pi\mu_w \ , \tag{4}$$

where $\overline{p_t} \cong p_{u'}$ in the validity range.

9. Maximum Friction Force

The friction force is 'stabilized' in the middle of the hydraulic cylinder stroke but the change in the alternating speed (at the stroke ends) and certain operational conditions are influencing the friction force value, too. The friction force at the stroke ends is the maximum friction force and it is determined as follows:

$$F_{\max} = c_3 c_4 F_c \ . \tag{5}$$

At the end of the strokes, when the direction of the alternating motion is changed, the lubrication condition is usually deteriorating and some displacement is needed in the new direction to rebuild the lubrication layer and to obtain balance in the lubrication condition again [17]. The friction force change at the direction change of alternating motion is expressed by the direction change factor (c_3), may be taken $c_3 = 1.3-1.5$ at $v_c < 0.05$m/s and $c_3 = 1.1 - 1.2$ at 0.05 m/s $< v_c < 0.3$ m/s for elastomeric U-rings and O-rings.

From operational point of view the piston rod seals (and also the piston seals) operate in a different lubrication condition when the piston rod moves out and in. When the rod moves out from the hydraulic piston, it supplies lubricant continuously between the sealing edge and the rod surface, while in case of instroke the lubrication condition is poorer having no continuous lubricant supply. The outstroke mode is called pump operation and the instroke mode is the motor operation in references. The recommended operational factor, expressing the above, is: $c_4 = 0.5$ for pump and $c_4 = 1.5$ for motor operation [11], [17].

(It is important to be mentioned that the difference of p_t and p_w and also the phenomena of relaxation have reasonable effect on the axial sealing pressure distribution curve if $p_w < 4$ MPa. To compensate this effect a sealing pressure compensation factor may be applied in the determination of friction force [20].)

10. Summary

The friction force calculation methods of hydraulic alternating seals had been reviewed and based on references and own test results, an advanced friction force calculation method was recommended.

The recommended calculation method regards to hydraulic piston and piston rod seals in general but the prerequisites of the application are the availability of friction characteristic test results (in the form of Stribeck diagram) and the axial sealing pressure distribution curve (or it's approximation) of the concerned seal.

Three groups of seals (Polyurethane U-rings, NBR O-rings, and O-rings with reinforced PTFE back-up rings) had been tested, out of the great number of hydraulic reciprocating seals, and for the recommended friction force calculation method Stribeck diagrams were plotted from the friction characteristic test results.

The axial sealing pressure distribution curves of the tested seals were not published in this paper because $\overline{p_t} \cong p_w$ was considered for the tested seals (which assumption may be taken in the applied working pressure range, see Chapter 7).

Symbols, Units and Denominations

$F_c[N]$	– friction force stabilized value (at the middle of the stroke)
$F_i[N]$	– friction force at restart
$F_{\max}[N]$	– maximum friction force
$F_n = F_t \cong p_w \cdot A_t[N]$	– sealing or normal force (accurate value of F_t is determined from axial seal pressure distribution)
$\mu_w = F_c/F_t$	– friction coefficient
$A_t[\mathrm{mm}^2]$	– friction surface of seals $A_t \leq D \cdot \pi \cdot b$ for U-ring; $A_t \leq D \cdot \pi \cdot d_2$ for O-ring and $A_t \leq D \cdot \pi(d_2 + b_1 + b_2)$ for O-ring with back-up rings
$D[\mathrm{mm}]$	– outside diameter of the seal
$d[\mathrm{mm}]$	– piston rod, or inside diameter of the seal
$b[\mathrm{mm}]$	– width of seals (U-ring)
$\bar{b}[\mathrm{mm}]$	– compressed seal width $\bar{b} \geq \sqrt{d_2^2\pi/4}$ for O-ring
b_1 and $b_2[\mathrm{mm}]$	– back-up ring width
$p_w[\mathrm{MPa}][\mathrm{bar}]$	– working, operating (test) pressure
$\overline{p_t}[\mathrm{MPa}][\mathrm{bar}]$	– average sealing pressure
$n[\mathrm{cycle/min}]$	– alternating speed of the piston
$N[\mathrm{cycle}]$	– number of double stroke done
$s[\mathrm{mm}]$	– stroke length
$v_c[\mathrm{m/s}][\mathrm{mm/s}]$	– alternating speed
$v_c[\mathrm{m/s}][\mathrm{mm/s}]$	– stabilized value of alternating speed (at the middle of the stroke)
$\delta = \frac{\Delta d}{d}100\%$	– initial elastic deformation, predeformation, pretightening, or overlap of the O-ring
$\eta[\mathrm{Ns/m}^2]$	– dynamic viscosity of the fluid
$t_w[^\circ\mathrm{C}]$	– working (test) temperature
$Z = \frac{\eta v_c}{\overline{p_t} b}$	– dimensionless number
$q\left[\frac{\mathrm{mm}^3}{\mathrm{cycle}}\right]$	– leakage during one cycle and
$\left[\frac{\mathrm{mm}^3}{\mathrm{m}^2}\right]$	– leakage for one meter square

	surface under friction
c_1 and c_2	– parameters of friction hyperbolas
c_3	– direction change factor
c_4	– operation factor
S[km]	– frictional way, total distance done by the alternating seal

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