

Friction Forces in O-ring Sealing

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Abstract: In the present study the focus was on developing a relationship as practical and convenient option for computing the friction force in O-ring sealing elements as used in the hydraulic and pneumatic equipments. For low-pressure applications, the developed relationship was applied for a different number of O-ring diameters, by investigating the obtained results, a good agreement has been observed for some of the tested diameters through the comparison between the obtained results from the developed relationship with those experimentally measured, which ensures the validity of the relationship. Regarding the low-pressure applications, theoretical and experimental studies were carried out in order to evaluate the effect of most important working parameters and conditions on the friction behavior. A remarkable advantage of using the relationship that reveals the accuracy of the relationship was detected through the comparison between the computed results with those obtained from the friction characteristic curves. Because the developed relationship having more important and quantified working parameters, it gives more accurate results, while in the characteristic tests curves the working parameters cannot be quantified, because they overlap and then they act cumulatively. The experimental study was performed and the obtained results are presented for two working cases, in which the friction force was determined in absence of pressurized fluid as in the first case, while in the second case the friction force was determined on basis of the presence of a pressurized fluid's.

Key words: Sealing Element, O-ring, Friction Force, Working Pressure

INTRODUCTION

Different types of non-metallic sealing elements are used in most of pneumatic and hydraulic equipments, but absolute tightness cannot possibly be reached. Therefore, the necessity of using the sealing elements is strongly recommended. Sealing elements are available in different types, shapes and designs such as (O-rings, U, X, V, L, I and others). They are used for sealing purposes, in order to avoid the tendency of leakage or mixing between fluids, as being under different working pressures. Sealing elements are used to retain fluids under the specific pressure. The sealing effect is obtained through the sealing element deformation between the movable surfaces with the relative motion where a considerable friction forces are resulted with unwanted consequences on the system efficiency or on the sealing element itself. The operating parameters and conditions concerned, the friction and leakage performances can be obtained from endurance and friction characteristic tests [1].

The lack of an accurate model for computing the friction force in O-ring sealing element, as same as the economical and technical advantages of the O-rings, are two important causes to encourage us to redirect this work toward developing a practical relationship for determining the friction forces acting on O-ring sealing element, practically a detailed analysis including the most important working conditions and factors must be firstly evaluated in order to predict the friction force

behavior. As compared with other sealing elements, the O-ring has a wide range of important and practical advantages as confirmed by Dragoni and Strozzi, [2] and Werencke [3].

- * O-rings offer an efficient and economical sealing element for a wide range of different standard and special static or dynamic applications, the inexpensive production methods and its ease of use have made the O-ring as the most applicable to a wide range of sealing problems and applications.
- * A wide choice of materials allows the O-ring to be used to seal practically all liquids and gaseous applications.
- * The single and double acting rotary seal for hydraulic and dynamic equipments have made the O-ring as the most applicable sealing element.
- * The initial squeeze acting in radial or axial directions, give the O-ring its initial sealing capability and to create the total sealing force that increase with the increasing system pressure.
- * Under a pressure O-rings behave in the similar way as the fluid, where the pressure is uniformly transmitted to all its sides.
- * O-rings can seal the shafts, which are simultaneously executing a reciprocating and rotating movement accurately.

On basis of the above-mentioned advantages, the work can be directed toward reaching its objective of

determining the friction force in the O-ring sealing element, so theoretical and experimental studies would proceed for studying the most important factors involved and their effect on the system efficiency.

In the dynamic applications the difference between the start-up friction and the running friction must be made, where the start-up friction must be overcome at the beginning of movement, while the running friction of seals practically depends on countless working factors, so a mathematical analysis practically is very hard to be reached and then it is difficult to establish an exact statement regarding the level of induced friction or to its magnitude [1]. The induced friction can be expected by considering the influence of the important working factors such as:

Related to the sealing element: the geometrical shape including the production tolerances and resulting deformation, hardness of the surface finish, friction values for dry and lubricated compound and temperature characteristics.

Related to the hydraulic fluid: tendency to build up a lubricating film and its distribution and the temperature- viscosity relationship.

Related to the working conditions: working pressure, velocity and direction of movement, working tolerances and working temperature.

The interaction of all the operating parameters have to be taken into consideration, but in the same time the contribution of these parameters to the magnitude of friction force will make the mathematical analysis as impossible or very hard, due to the simple reason that the above factors cannot be quantified and because they overlap and act cumulatively, so it is difficult to establish an exact or accurate statement regarding the expected friction level [4].

The initial phase of calculation of the friction force is based on considering the contribution of the most important working factors (Fig. 3), by considering these factors and their contribution we can initially estimate and express the friction force in O-ring by using the following relationship:

$$F_f = C \times \mu \times \pi \times D_3 \times b \times \Delta p \times i \quad (1)$$

Where, Δp is total pressure difference acting on the gasket, that is equal with the sum between the difference of the working pressures in two separated chambers by the sealing element and the pressure due to deformed assembling (0.05 – 0.06).

I: when more than one O-ring are used in the system, then the friction force for all rings must be combined in order to determine the total friction force. The influence of some of the above factors such as: (gasket shape, surface finish quality, speed of displacement and lubricating conditions) is globally appreciated by μ (the friction coefficient), that is a variable coefficient and due to its dependence on the different working

conditions it has following values as: (0.2 - 0.3) for lip-type of packing such as: U and L seals, (0.3 - 0.5) for X and O-ring types as in case of well finished and sufficient lubricated sealed surfaces and (0.7 – 0.8) indicates improper finished surfaces or insufficient lubrication conditions.

Regarding the seal's behavior, because it depends mostly on the magnitude of losses (friction force, leakage and wear) and on the operating parameters, then big losses must be avoided in order to meet the efficient requirements [3].

MATERIALS AND METHODS

Design Recommendations

O-rings: are vulcanized in moulds and are characterized by their inside diameter d_1 and the cross section diameter d_2 as shown in Fig. 1. The chosen cross section d_2 should be in an appropriate ratio to the O-ring inside diameter. Special catalogs and tables are available that usually are used to provide all the O-ring dimensions and lists the recommended range of squeeze value, also the working tolerances and the groove specifications are provided, these tables can be taken as a guideline for choosing the proper dimensions.

Generally, seal performance is considered to be primarily a result of sealing force that develops when a seal is compressed, so for optimum sealing performance a good balance of all physical properties is usually necessary; an important parameter for assessing the sealing behavior is the compression set of the O-ring compound [6].

Under load, elastomers exhibit not only an elastic component but also a permanent plastic deformation. The sealing behavior by means of the change of cross section height and for all the working states of the O-ring is shown in Fig. 2. The compression set can be determined in accordance with DIN 53517/ASTM 395B as:

$$CS = \frac{h_0 - h_2}{h_0 - h_1} \cdot 100(\%),$$

Where:

- 0% = indicates no relaxation has occurred
- 100% = indicates the total relaxation
- h_0 = Original height (d_2), Fig. 2-a
- h_1 = Height in the compressed state, 2-b
- h_2 = Height after releasing, 2-c.

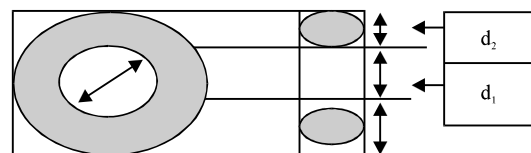


Fig.1: O-ring Dimensions

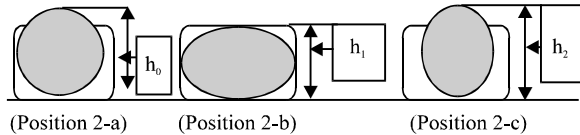


Fig. 2: Compression Set by Means of Cross Section Changes

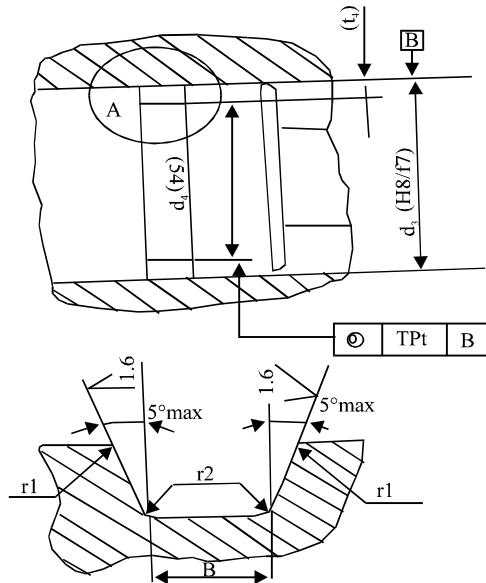


Fig. 3: Groove Design and the Main O-ring Dimensions

The importance of the initial squeeze in the O-ring groove (rectangular groove), is essential to ensure its function as a primary or secondary sealing element, initial squeeze will serve to achieve the initial sealing capability and to assure the defined frictional forces as same as to compensate for the compression set, depending on the nature of application, the following values can be applied for the initial squeeze as a proportion of the cross section d_2 :

6-20%- for dynamic applications

15-30%- for static application

Groove Design: As compared with the Triangular or with the Trapezoidal grooves that are characterized with the difficult manufacturing processes, the rectangular groove is preferred for all new designs; rectangular designs with beveled groove flanks up to 5° are permissible. The groove choice is based on the guide values for the initial squeeze and on the relationships between the Shore A hardness, loads and cross-sections. The recommended and specified groove width already takes into account the swelling value for O-rings as described in DIN 3771 part 5, September 1989, where 8% swelling is permissible for moving parts and 15% for stationary parts [2]. Figure 3 shows some of the required dimensions and characteristics of the rectangular groove.

The Shaft Design: is of a vital significance for the performance as well as for the useful life of the seal, so the shaft run out and the eccentricity between the shaft and housing bore centers should be avoided or kept within a minimum [7].

Principles of Sealing Mechanism: The demand for higher reliability and efficiency rates for both hydraulic and pneumatic systems necessitate an extensive research and development programs in sealing technologies. Generally the accepted basic principles for proper sealing effect and sealing mechanism include the following statements:

- * The starting sealing effect (p_{start}) as before the operating pressure is applied, the sealing effect is produced by the overlapping of the sealing edges on the cylindrical contact surfaces for any type of seals.
- * When the operating pressure value is elevated, the sealing pressure is automatically and proportionally increased, in which: ($P_{total}=P_{start}+P_{working}$), so the sealing element behavior on a different pressure ranges will acts during the operations as same as the involved working fluid, due to the effect of this phenomenon, the automatic and proper sealing is resulted along the operating pressure range [8].

Unfortunately, the lack of an accurate and useful model for computing the frictional forces acting on the sealing elements, the sealing technology and developments was based for several decades on the results of the friction characteristic tests as obtained from the endurance and friction curves, such as to use the Friction Coefficient curves or diagrams related to the O-rings, on basis of these tests and curves the possibility of comparing the friction behavior of different seals will give an information about the average measured friction force or leakage values during the testing periods, but these values are not accurate due to some deficiencies and because they indicate only the relationship of the represented factors, while the interaction of all the working parameters are not taken into consideration in the given tests and curves [8, 3].

However, In spite of all the latest developments, the friction characteristic tests curves are still remained to have the deciding role as the essential source of information in the research works and in the sealing technology. The friction characteristic can be revealed for different designs and working conditions, when a balanced running out and proper characteristic test curves are applied, where a careful consideration should be taken for the most important and operating parameters to be within the recommended ranges.

Friction Characteristic Curves can be applied to predict the expected friction loss or behavior of the O-ring seal working under different working conditions, for a long

time the friction characteristic tests are applied and used by the designers as the major source of information to compare the different types of seals, so on basis of the results obtained from these tests it is possible to compare the friction behavior of the O-ring seal with other sealing types working in a certain conditions within the operating parameters ranges.

Two diagrams or curves as published by Bisztray-Balku [1], showing the effect of velocity and working pressure on the friction behavior and on the magnitude of losses (friction force, leakage and wear), these curves can be used in our work in scope of verification and comparison only as a guideline to verify the friction behavior in relation to the working pressure as same as the effect of speed on the friction force (as one of the most important factors). On basis of these tests as a source of information, we can make an initial prediction regarding the practical limits of working pressure and the average of the alternating speed, since the average measured friction can be used as a reference point in our verification.

Below a brief description is focused on the maximum and minimum limits of a working pressure as same as to identify the optimum velocity:

The Friction Force-Working Pressure Curve: It provide us with the information about the applicable range of operating pressure, where at fixed alternating speeds and for an elevated operating pressure but within the established limits ($P_w < P_{max}$), a balanced increase of a friction force behavior can be obtained, also for longer operating time the maximum Pressure can be also permitted as a working pressure with out an early breakdown of the seal. However reaching a certain pressure above the maximum permitted pressure ($P_w > P_{max}$), so for any sudden increase in the operating pressure above the maximum operating value, the friction force will increase proportionally.

The Friction Force-Reciprocating Speed Curve: By running the test at a fixed operating pressure and at an optimum velocity, the friction force will remain within the balanced conditions, while in case of increasing the speed above the established optimum speed, the friction force will increase proportionally.

The use of these curves can be used for constructing functions or diagrams to determine the expected friction force for different operating conditions within the operating parameter limits, practically they are used as a guideline for the recommended or not recommended speed ranges but with the condition of establishing an optimum alternating speed.

Friction Force Determination

Theoretical Considerations: For low-pressure systems, seal friction can raise the required actuating pressure to many times, therefore, seal friction must be minimized and maintained within the reasonable

working limits in order to operate the system properly. The need for calculating the compression force developed in the O-ring as directly related to the sealing ability, will serve to evaluate the width and the contact pressure for the sealing element by one-hand and to show the amount of the peak contact by other hand [4].

The sealing performance can be easy predicted by using two methods, by knowing the peak contact value as (greater or less than the system pressure), or by determining the amount of squeeze. Both values can help us to predict the sealing performance in away that:

- * If the peak contact is greater than the system pressure, then the O-ring will seal the joint, while in case that the calculated value will be less than the system pressure then the ring will leak which means that the compressive force is lower than the required and also the contact stress is lower.

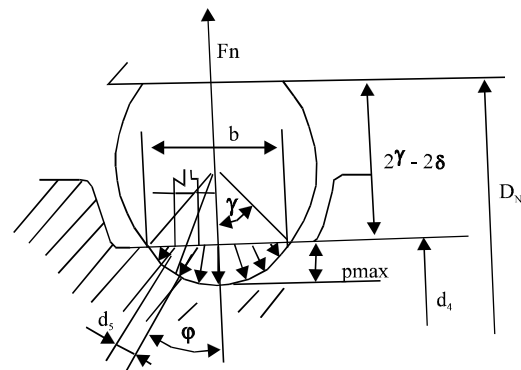


Fig. 4: Identical Pressure Distributions

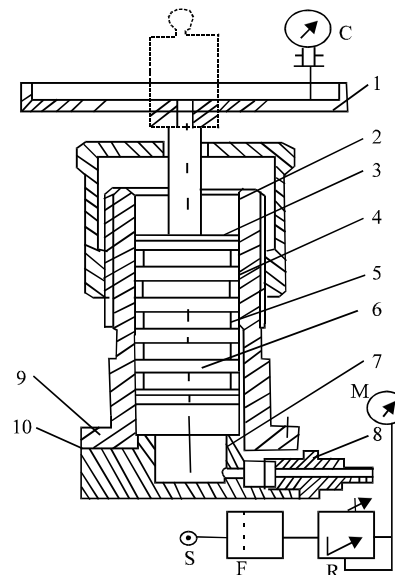


Fig. 5: Experimental Device Components, where; (1) Tray, (2) Auxiliary Masses (3) Stopper Mechanism, (4) Cylinder, (5) O-ring, (6) Ram, (7) Cover, (8) Sleeve, (9) Frame and (C) Comparator, (M) Manometer, (S) Source, (F) Filter, (R) Regulator

* Sealing performance can be also shown by the amount of squeeze, because the increased amount of squeeze above the established limits it can cause a high friction and excessively high actuating forces, also for low-squeeze below the limits means lowering the sealing pressure and increasing the leakage tendency.

Designing or selection of O-ring seals for low-pressures is not simply a matter of reducing the amount of squeeze, but it involves the difficulty of balancing between the material hardness, dimensional tolerances, stress relaxation and friction characteristics. The width and contact pressure evaluation in O-ring sealing as in Fig. 3, has to follow a rigorous procedure in order to show the important component of the friction from the sealing element. For this reason, a theoretical model was proposed and experimental measurements were performed to ensure and to compare the results. The appearance of an identical pressure distribution between the O- ring and the cylinder with diameter $D_3=D_N$, respectively between the O-ring and the ram with diameter d_4 (Fig. 4), the pressure distribution appears having the resulting force F_n (the normal force from the section. Admitting that the contact pressure has a cosinoidal distribution, where $p=p_{max}\cos\phi$, so we can approximate its maximum value by using Hooke's law:

$$p_{max} = E \frac{\delta}{r} = \left(1 - \frac{D_3 - d_4}{4r}\right) \quad (2)$$

Where, δ -is the ring radial deformation
 r -is the radius of O-ring section
 E -is the ring material elasticity modulus

The normal force (Fig. 4), is the resultant of vertical components of the elementary forces due to pressure, by integrating the normal force it gives:

$$F_n = \int_{-\gamma}^{\gamma} p \cos \Phi dp = \int_{-\gamma}^{\gamma} p_{max} r \cos \Phi d\Phi = 2p_{max} r \sin \gamma \quad (3)$$

Where, $db = rd\phi / \cos \phi$ and γ - is the half of the central angle of the deformed zone.

Also from the same figure, we can obtain:

$$\sin \gamma = \frac{b}{2r} = \sqrt{1 - \frac{(D_3 - d_4)^2}{16r^2}} \quad (4)$$

The friction force due to ring deformation at assembly is:

$$F_f = \mu \pi D_3 F_n \quad (5)$$

Replacing equations (2,3,and 4) in equation number (5), we get the developed expression of the friction force:

$$F_f = 2\pi D_3 r E \left(1 - \frac{D_3 - d_4}{4r}\right) \sqrt{1 - \frac{(D_3 - d_4)^2}{16r^2}} \quad (6)$$

RESULTS AND DISCUSSION

The validity of expression 6

$$(F_f = 2\pi D_3 r E \left(1 - \frac{D_3 - d_4}{4r}\right) \sqrt{1 - \frac{(D_3 - d_4)^2}{16r^2}})$$

was tested with an experimental device (Fig. 5), which was used for obtaining the experimental results. The Ram has five different sizes of sockets or grooves to increase the possibility of testing different number of O-ring sizes (Fig. 5). Each groove is designed for a specific diameter, The ram and cylinder geometry is characterized by the values of the diameters, both diameters are clearly shown in Fig. 3 and 4, in our case the cylinder diameter $D_3 = 28.03$ mm, the inside ram diameter d_4 having five different sizes, in our case ($d_4 = 21.41, 21.45, 21.59, 21.74$ and 22.10 mm), all of these sizes were tested and the obtained results are indicated in Fig. 7 (in absence of the pressure), in which each size has its proper value of friction, Fig. 8 indicates the measured values for different O-ring sizes where the pressurized agent was involved. Experimentally we follow to ensure the validity of using the developed relationship for calculating the friction force, the results obtained through the use of this formula are used to show the degree of agreement between the computed and measured values for each inside diameter. On basis of a comparison between the obtained results for all the tested O-rings (Fig. 7), in which a good agreement between the measured and computed values is represented by the diameter $d_4 = (21.5$ and 21.6 mm) as the optimum diameter with less friction.

In each experiment the contact pressure between the ram and the cylinder will be modified to suit the given O-ring dimensions, so the friction force was determined on basis of the minimum and a necessary force needed to move the ram for all the five positions of the O-ring. In order to compare and evaluate the effect of the working pressure, the experimental procedure was focused on two cases, in the first case; the obtained results show the effect of the pressurized agent, while in the second case; the experiment was carried out in the absence of the pressurized fluid.

In the first experiment case an auxiliary mass of 1.45 kg was added to the ram mass in order to move it and to register the amount of the friction force in the same time to observe the amount of resistance for whole the mass.

In the second case it determine the necessary pressure to move the ram with out the auxiliary masses (device is free of the added masses).

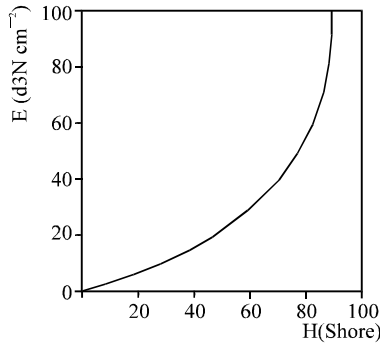


Fig. 6: Rubber Modulus of Elasticity vs. the Shore Durability

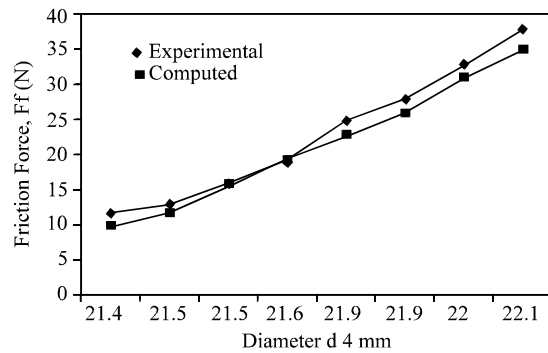


Fig. 7: Comparison between the Experimental and the Computer Values (Without Pressurized Fluid)

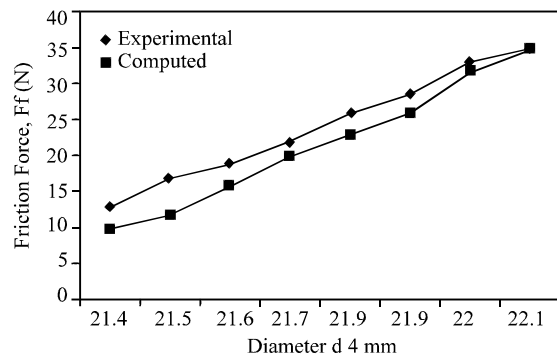


Fig. 8: Comparison between the Experimental and the Computer Values (in the Presence of Pressurized Fluid)

The tested sealing element in our work is the O-ring with the symbol 35502120, it is characterized by the section diameter $d_2 = 2r = 3.55\text{mm}$ and exterior diameter $d_3 = 28\text{ mm}$. In order to compute the friction force in absence of the pressurized fluid, the dependence of the rubber modulus of elasticity toward the Shore durability was used (Fig. 6), for most applications a Shore A hardness of 70° is sufficient, where 70° Shore correspond to $E = 4\text{ Mpa}$. Fig. 7 and 8 show the comparison between the experimental and computed results as carried out for both cases.

Figure 7 shows the comparison between the experimental and the computed values but in the absence of the pressurized fluid and in the area corresponding to the diameter d_4 , a good agreement is observed between the computed and the experimental results in the included area between 21.5 and 21.6mm. The maximum deviation of about 30% between the computed and experimental values appears in the lower value of the diameter ($d_4 = 21.41\text{mm}$), but in the technical standards it is considered as acceptable deviation, because in the given working conditions there are many factors whose their contribution to the magnitude of the friction force can not be quantified.

An advanced resistance of the ram is obtained under the effect of the pressurized fluid while trying to move it (Fig. 8). When computing the friction force we have to make summation between the obtained values from equation Nr. 6, with the fluid pressure contribution from equation Nr. 1, because the resulted value by using equation (Nr.1), represent in most cases a third of that corresponding to relationship (Nr.6).

CONCLUSION

In this study, a mathematical relationship was developed for determining the friction force in O-ring sealing element as due to its deformation between the working surfaces. In absence of the pressurized fluid, the comparison between the obtained results shows that the computed results through using the developed relationship were in a good agreement with those experimentally obtained, that means a good indication for ensuring the validity of using this formula to be as a practical formula for determining the friction force in O-ring sealing elements. Experimentally and technically the relationship was confirmed within the acceptable working limits. Theoretical study has been performed to investigate the sealing behavior concerning to the effect of the most important working factors and to predict their effect on the magnitude of the friction. Through the experimental study the following key conclusions and observation are obtained:

- * An important contribution of the working pressure and then the peak contact effect on the magnitude of the friction force and sealing performance was strongly imposed, in which a high friction increase is proportionally to the working pressure increase, while for low pressure below the established limits, the leakage tendency will increase proportionally.
- * The friction is directly influenced by the seal diameter because the wear-area is greater.
- * An important consideration must be given to the amount of squeeze, because a high friction is obtained when the squeeze exceeds the established limits and for low-squeeze below the limits will

results in increasing the leakage tendency, so a careful balancing of O-ring material is necessary to ensure leak-free operation at low-pressure.

- * The need for keeping the seal friction as low as possible is essential, especially in case of the pneumatic equipments working at low pressures.

Nomenclature

b	[mm ²]	Seal contact area (width)
C	-	Correction factor
D ₃ =D _N	[mm]	Sealed surface diameter
d ₁	[mm]	O-ring inside diameter
d ₂	[mm]	Cross section diameter
d ₄	[mm]	Ram inside diameter
E	[daN/cm ²]	Modulus of elasticity
F _f	[N]	Friction force.
F _n	[N]	Resulting normal force.
H	[mm]	Height
I	-	Identical Nr. of gaskets
P _{max.}	[N]	Maximum working pressure.
P _w	[N]	Working pressure.
R	-	Radius of O-ring
Δp	[Mpa]	Total pressure difference
δ	-	Ring radial deformation.
μ	-	Friction coefficient

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