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Optimum Design of Coaxial Hydraulic Sealing Systems Made from Polytetrafluoroethylene and Its Compounds

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Abstract: Fluidic actuation systems are optimizable as to energy consumption by reducing the friction in the hydraulic cylinders. Polymeric materials with special antifriction properties and good resistance to hydraulic fluids can be deployed to enhance the performance of hydraulic cylinders. Small friction forces can also be ensured by facilitating the hydrodynamic separation of the elements of the friction tribosystem, namely the seal and sealed-off surface, respectively. The study presented in this paper analyzed the hydrodynamic separation phenomenon in hydraulic cylinders with coaxial sealing systems of the pistons. The process underlying the forming of the fluid film between the seal and its contact surface was considered and the formula for calculating film thickness was deduced. This paper presents graphs that describe the variation of the fluid film thickness versus the dimensional and material characteristics of the sealing systems. The study yielded recommendations as to the most adequate polymeric material to be used and the optimum dimensional characteristics of the seal.

Keywords: coaxial sealing systems; hydraulic cylinders; hydrodynamic friction; fluid film; polytetrafluoroethylene

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1. Introduction

The role of sealing systems is crucial in various industrial applications that deploy hydraulic fluids (construction and transportation equipment: excavators, forklifts, telescopic handlers, basket elevators, booms; mining, robotics, and aviation, etc.) [1,2]. These systems are responsible for leakage-free and reliable operation, as well as for ensuring optimum performance and efficiency of the equipment they are part of. Thus, for example, the deployment of modern electronic control systems in hydraulic circuits allows the rapid and high-precision positioning of a cylinder. In order to make full use of the benefits afforded by electronics, the characteristics of new sealing systems have to include, in addition to a high degree of hermitization, also the following:

- low uniform friction, particularly at pressures $p < 150$ bar and speeds of $v < 0.5$ m/s;
- reduced (inexistant) stick-slip at small speeds of $v < 0.1$ m/s;
- long service life even at very short and rapid strokes, that is, for displacement lengths smaller than the length of the sealing.

Sealing systems influence decisively the design, operation, and service life of hydraulic motors. As their role in the assembly of a fluidic motor is of the highest importance, sealing systems have to meet a number of requirements of which the most relevant ones are as follows: hermitization capacity of the sealed-off enclosure, wear and extrusion strength, resistance to the working medium, resistance to high and/or low temperatures, low friction, compact form, simple installation, etc. [1–3].

Currently, sealing systems of hydraulic cylinders are complex assemblies consisting of seals placed on the piston and on the rod, and wipers that ensure guiding and cleaning. All these components have to be dimensioned in a correlated manner in order to prevent fluid leakage at increasing working pressures. Maximum performance levels for hydraulic cylinders require the usage of recent developments in seal materials and geometries. Also required is a significantly improved understanding of the operation of sealing systems, as well as modern simulation instruments. Modern sealing systems are operational at pressures up to 800 bar and at working fluid temperatures between −50 °C and +200 °C, while preventing stick-slip and maintaining minimum leakage [4,5].

A correctly conceived sealing system increases the service life of the entire assembly and is based on the optimized match of material and component design, requiring thorough knowledge and expertise by the designer. An optimum design for sealing systems is required because 44% of hydraulic cylinder breakdown is caused by sealings [6,7].

While in decades past the typical hydraulic cylinder sealing material was rubber, the 1970s and 1980s ushered in the use of polytetrafluoroethylene (PTFE) due to its qualities: a very low friction coefficient and excellent durability in hydraulic oil. The major disadvantage of PTFE is failure under load, thus rendering it undeployable at high pressures [8]. This impediment has been overcome by introducing additives such as bronze or glass fiber [9].

Selecting the optimum material for a sealing system is highly relevant for the durability of the entire hydraulic drive. The most frequently used materials and their main characteristics are as follows [10]:

- nitrile butadiene rubber (NBR), fluorocarbon (FKM), and hydrogenated nitrile butadiene rubber (HNBR) are popular materials that are used due to their resistance to fats, mineral oils, and other hydraulic fluids;
- polytetrafluoroethylene (PTFE) benefits from a remarkable durability and due to its low friction coefficient requires less lubrication, thus it may operate efficiently even under conditions of dry friction;
- rigid plastic materials are temperature and pressure resistant, being significantly more rigid than rubber or PTFE seals;
- thermoplastic elastomers (TPE) offer excellent strength and flexibility, are resistant to abrasion and breaking, and can maintain a constant pressure over long periods of time.

The current state of materials use in hydraulic sealing systems has been described in [11]. The main causes responsible for the failure of sealing systems (swelling, thermal degradation, deformation, and wear) have been identified. Recommendations have been made for polymer-based materials to be used for the manufacturing of seals.

Recommended materials for sealing systems have been enumerated and the need for these materials to adapt to the unevenness of metal surfaces in order to block fluid passage (the seal has to be able to expand or compress swiftly following the dimensional variations of the metallic components) has been emphasized [12]. The operational behavior at high temperatures of seals made from various types of polytetrafluoroethylene (pure or with additives) has been evaluated. Mixed materials contribute to increasing wear strength of the entire sealing assembly [13].

The properties of materials in sealing systems must be matched to specific applications so as to ensure optimum results. Urethane bonded to PTFE and other rigid plastics has improved performance related to high pressure, temperature, and speed [14]. PTFE adulterated with bronze reduces the occurrence of extrusion without affecting sliding capacity and wear resistance [15]. Further findings confirm that cylinders fitted with PTFE seals benefit from reduced friction forces [16,17]. Research on the tribological characteristics at initiation of sliding (break-away friction) of several PTFE-based materials confirm the adequate behavior of hydraulic cylinders [18].

The geometry of seals has been less studied, bar a small number of papers. Thus, two sealing variants have been examined (M and T sealing systems, respectively) and optimum working conditions determined [19].

The type of friction between the sealing system and contact surface has also been analyzed. The geometry of the gap formed between the seal and its adjacent surface has been documented [20]. During operation, fluid is drawn into this gap—called the elastohydrodynamic (EHD) inlet zone—and the pressure gradient causes the forming of a thin lubricating film. The formation of a fluid film at the interface between the seal and the sealed-off surface has been described. The calculated average thickness of the fluid film has been confirmed experimentally [21].

The tribology of the contact between the seals and sealed-off surfaces has been studied theoretically and experimentally regarding the lubrication phenomena and contact mechanics [22]. Contact pressure, distribution of fluid film thickness, and heating due to friction have been calculated. The effects of surface roughness on sealing performance have been determined.

The effect of seal geometry on leakage, friction, and wear in reciprocating hydraulic seals has been researched. Thus, increasing the roughness of the sealed-off surface from 0.2 to 0.4 μm causes greater fluid leakage [23].

A numerical model has been developed that investigates the effects of textured rods on the wear of reciprocating seals. This model is focused on seal wear under mixed lubrication conditions by combining the EHD lubrication model and the Archard wear model. A rod of smaller rather than greater roughness reduces leakage while increasing sealing wear [24].

Finite element simulation has been used for an optimum bidimensional design of a sealing structure in a hydraulic cylinder [25]. Discrepancies between the inverse theory of hydrodynamic lubrication (IHL) and empirically measured film thickness have been discussed. The influence of speed, viscosity, and pressure on the generation of the hydraulic oil film in the sealing process of the piston rods has been evaluated experimentally [26].

Over recent years, coaxial sealing systems made from antifriction materials have been preferred for the construction of hydraulic cylinders. Elastomers have been replaced by plastomers, thermoplastic elastomers, or duromers on grounds of their remarkable antifriction qualities. The literature, however, has not offered sufficient useful information for designers and users of sealing systems. The applicability of various combinations of polymeric materials has rarely been discussed, and there has been a similar scarcity regarding the influence of certain dimensional characteristics of PTFE seals on the sliding conditions in friction tribosystems.

In this paper, the results of theoretical research are presented concerning the influence of seal dimensional and material characteristics on the thickness of the fluid film formed at the interface between seal and sealed-off surface. The studied materials were virgin PTFE, PTFE CF10 (90% PTFE + 10% carbon fiber), and PTFE D46 (53% PTFE + 46% bronze + 1% pigments). The significance of this paper consists in providing designers of coaxial sealing systems with useful information for correctly dimensioning seals and for their optimum utilization depending on their material. Exact knowledge of material performance under concrete operational conditions is paramount for maximum efficiency of hydraulic motors. The paper presents the theoretical results obtained consequently to the analysis of the sealing process. The experimental validation of such results is discussed in [15].

The paper is structured into seven sections. The introduction is followed by a second section that details the components of the coaxial sealing systems and their respective roles. The sealing mechanism is described and the phenomena occurring at the interface between the seal and the sealed-off surface are explained. The necessity of ensuring small friction forces is emphasized in section three, attainable by generating hydrodynamic friction. The fourth section presents the geometry of the studied sealing system and the material properties of its components. The fifth section presents the obtained results and illustrates the variation of fluid film thickness versus the dimensional and material characteristics and the operational conditions of the sealing system. The sixth section is dedicated to the discussion of the results and puts forward recommendations for selecting the most adequate sealing system. The last section formulates the conclusions of the reported study.

2. The Coaxial Sealing System and the Sealing Process

Coaxial sealing systems are recommended for use in hydraulic cylinders due to their simple construction and good packing capacity of the pressurized enclosures. The coaxial sealings of the piston and rod of a hydraulic cylinder are shown in Figure 1.

Each such sealing system consists of a seal and an O-ring. The seal comes into contact with the metallic surface of the cylinder and of the rod, respectively, hence the importance of studying the tribological phenomena occurring at the interface.

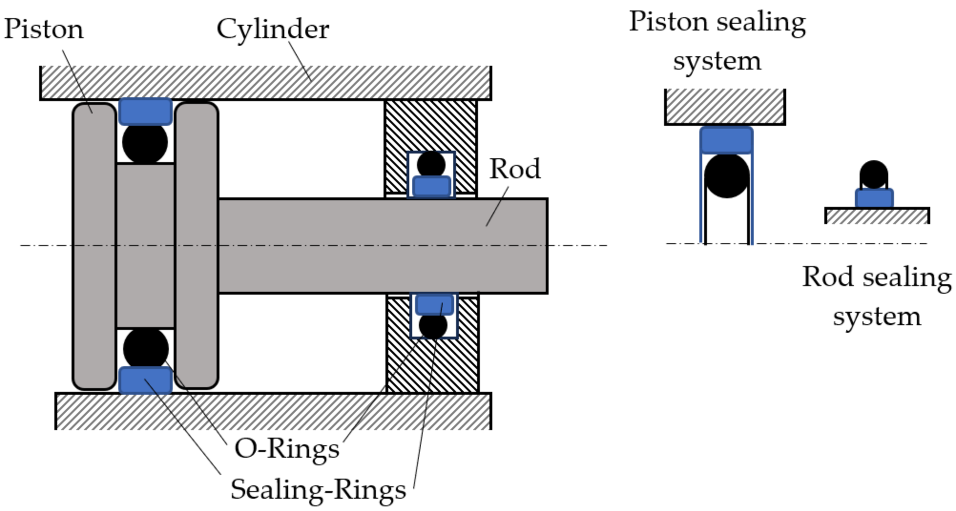
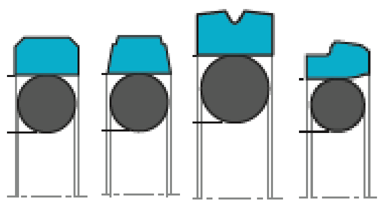


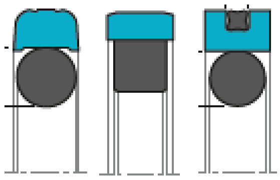
Figure 1. Coaxial sealing system.

At zero fluid pressure, the O-ring ensures the initial pressing of the seal onto the sealed-off surface in order to avoid leakage. For this purpose, the initial specific radial deformation of the O-ring is ε_{r0} = 10 to 25%. As the fluid pressure in the cylinder increases, the O-ring is further deformed and presses with a greater force onto the seal. O-rings are usually made from the following elastomers: nitrile butadiene rubber (NBR), ethylene propylene diene monomer (EPDM), fluorocarbon (FKM), hydrogenated nitrile butadiene rubber (HNBR), silicone rubber (Q), and fluorosilicone (FVMQ).

The cross-section geometry of seals varies according to the variant recommended by manufacturers so as to match specific working conditions. Variants of coaxial sealing systems of pistons are shown in Table 1 [3].

Table 1. Variants of seal cross-sections [3].

Schematic	Recommendations for Utilization
	Pressure: up to 60 MPa Speed: up to 15 m/s Frequency: up to 5 Hz. Temperature: −45 °C to +200 °C Media: Mineral oil-based hydraulic fluids Low friction, no stick-slip effect Recommended for injection molding machines, machine tools, presses, excavators, etc.

	Pressure: up to 50 MPa Speed: up to 2 m/s Frequency: up to 5 Hz. Temperature: −45 °C to +110 °C Excellent abrasion and extrusion resistance Media: Mineral oil-based hydraulic fluids

3. Methodology

One of the most important requirements in the operation of a hydraulic cylinder is keeping the friction forces as small as possible. Thus, seals are manufactured from low-friction materials like polytetrafluoroethylene (virgin or filled PTFE). In addition to good sliding properties, low friction is also ensured by the interruption of contact between the seal and the surface it slides along. This phenomenon follows the laws of hydrodynamic lubrication and occurs with the onset of relative motion between the seal and the piston or rod surface. The separation of the two elements of the tribosystem causes a gap filled with fluid under pressure. Such detachment of the two surfaces is explained by dynamic pressure building up in the fluid in the area of initial contact. Consequently, friction passes from dry to mixed or even fluid (hydrodynamic), and depends on the relative speed v and the dynamic viscosity η of the working fluid. From the viewpoint of the friction forces, fluid friction would be preferable, namely a fluid film that is as thick as possible. This, however, is opposed to the very phenomenon of sealing that means the absence of fluid, for which reason a compromise is required.

In a general situation, hydrodynamic friction occurs only if the thickness of the fluid film is at least equal to the sum of the roughness values (Ra) of the two surfaces. It has been demonstrated, however, that fluid friction is possible even for smaller thicknesses of the oil film [27]. This is due to the seal material being softer than that of the cylinder sleeve. Hence the seal adapts to the form of the asperities of the harder surface (it molds to the asperities). In areas where the asperities of the two surfaces are very close to one another, local pressure peaks occur that cause the seal to detach from the surface of the cylinder sleeve up to a distance at which the fluid pressure balances the tension created by the deformation of the material. This situation is shown in Figure 2. The two surfaces form a gap that is smaller than the sum of the values of roughness Ra.

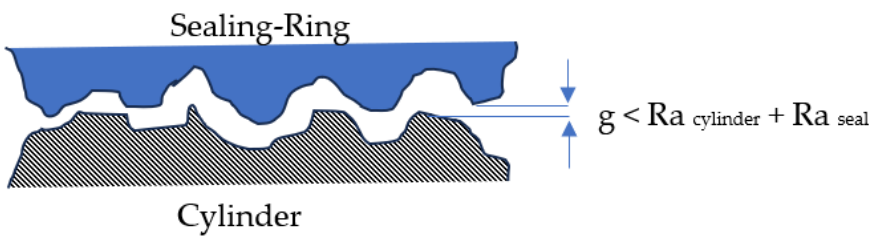


Figure 2. Fluid friction in coaxial sealing systems.

The thickness of the fluid layer depends on the variation of the pressure gradient in the gap dp/dx. A large gradient means a thin fluid film, while a small gradient causes a thicker fluid layer.

Perfect packing requires zero fluid flow in the gap [28] as shown in Equation (1):

$$Q = \pi \cdot D \cdot \int_0^g \dot{x} \cdot dy = 0, \tag{1}$$

where g is the average height of the fluid film and $\dot{x}$ is the fluid velocity in the gap calculated by Equation (2) [28]:

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

where v is the relative speed of the sealed surfaces. The equation above was obtained by integration of the Navier–Stokes motion equations that express the equilibrium of the pressure forces and tangential forces due to viscosity.

The two equations above lead to the following formula for the fluid flow in the gap:

$$Q = \pi \cdot D \cdot \left(\frac{v \cdot g}{2} - \frac{g^3}{12 \cdot \eta} \cdot \frac{dp}{dx} \right) \quad (3)$$

Equation (3) has two components: "transport" or the Couette component of flow, and "leakage" or its Poiseuille component. In Couette flow, the fluid is driven between the moving surfaces; the Poiseuille component refers to fluid set into motion by the pressure gradient. For fluid flow to be zero, the two components have to be equal. This is achieved when fluid velocity in the gap varies as shown in Figure 3.

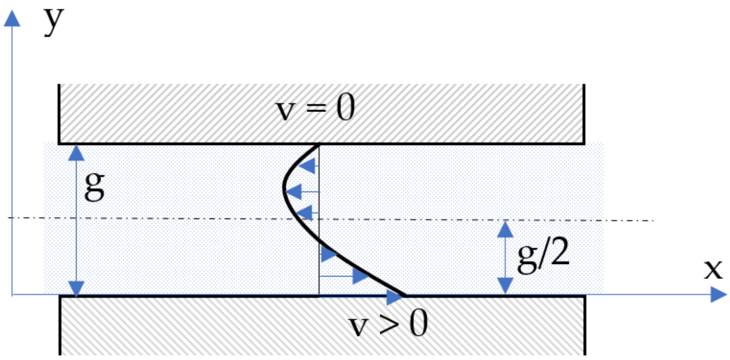


Figure 3. Required variation of fluid velocity in the gap for perfect sealing.

With the onset of motion ($v > 0$), the seal is radially deformed into the shape of a thin-walled tube subjected to the pressure of a fluid. The forming process of the gap in the case of a piston sealing is shown in Figure 4.

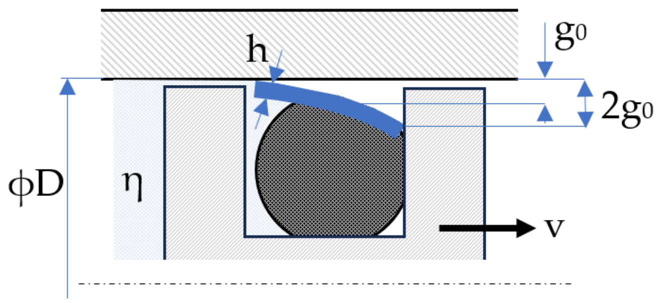


Figure 4. Forming of the gap between the elements of the tribosystem.

The computational formula for the average gap size is as follows [29]:

$$g_0 = \sqrt[3]{\frac{3}{16} \cdot \frac{(D-h)^2}{E_p \cdot h} \cdot \eta \cdot v \cdot L \cdot (1-\beta^2) \cdot \left[1 - \frac{2 \cdot \cosh(k \cdot L) \cdot \cos(k \cdot L)}{cosh^2(k \cdot L) + cos^2(k \cdot L)} \right]}, \quad (4)$$

where

$$k = \sqrt[4]{\frac{12 \cdot (1 - m_p^2)}{h^2 \cdot (D-h)^2}}, \quad (5)$$

m_p and E_p are Poisson's ratio and Young's modulus of the seal material, respectively; h = seal thickness; D = piston diameter; L = seal width.

The quantity β in Equation (4) is the so-called real nondimensional area, representing the share of the nominal area corresponding to the surfaces in direct contact. The quantity β is defined as the ratio of the real contact area of the two surfaces and the nominal area $A_n = \pi \cdot D \cdot L$.

$$\beta = \frac{A_r}{A_n} \tag{6}$$

The real contact area is the sum of the direct contact areas of the asperities on the surfaces of both elements of the tribosystem (Figure 5). The real nondimensional contact area β is a quantity with values less than unity that takes into account seal characteristics and cylinder material and the initial specific radial deformation ε_{r0} of the O-ring.

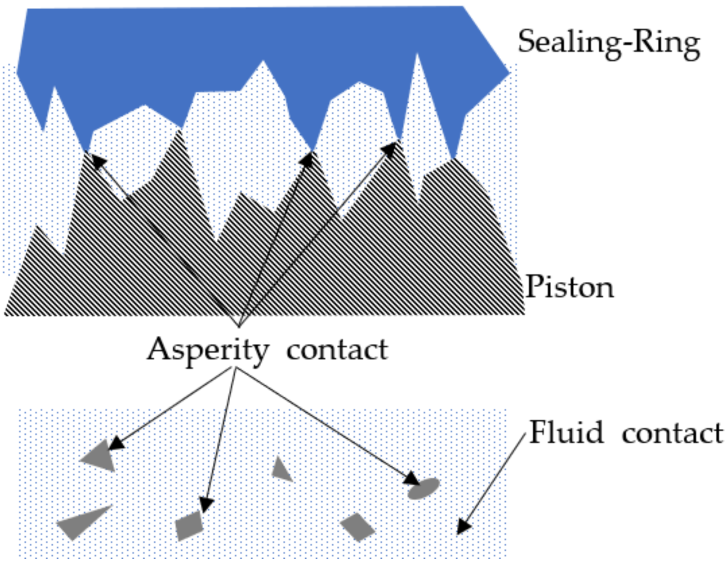


Figure 5. Contact surfaces between the elements of the tribosystem.

4. Materials

The research considered the sealing of a hydraulic cylinder piston by means of coaxial sealing systems whose seals are made from polytetrafluoroethylene polymers (virgin PTFE, PTFE CF10, and PTFE D46—produced by DMH Dichtungs- und Maschinenhandels GmbH, Traboch, Austria). The O-rings were made from nitrile butadiene rubber (NBR) with a hardness of 70 Shore A. The main characteristics of the tested materials are shown in Table 2 [15,30].

Table 2. Characteristics of the seal materials.

Material	Composition	Hardness Sh D	Young's Modulus [MPa]
PTFE CF10	90% PTFE + 10% carbon fiber	58 ± 3	300
Virgin PTFE	100% PTFE	55 ± 3	540
PTFE D46	53% PTFE + 46% bronze + 1% pigments	63 ± 3	1420

The first material, PTFE CF10, contains added carbon fibers that contribute to increasing wear strength, decreasing the friction coefficient and improving thermal expansion properties. On the other hand, PTFE D46 includes 46% bronze, which improves compression strength, thermal conductivity, and electrical conductivity.

Variation graphs of the nondimensional contact surface area β were plotted for these materials. The dependency $\beta = f(p)$ for the studied materials is presented in Figure 6 [15].

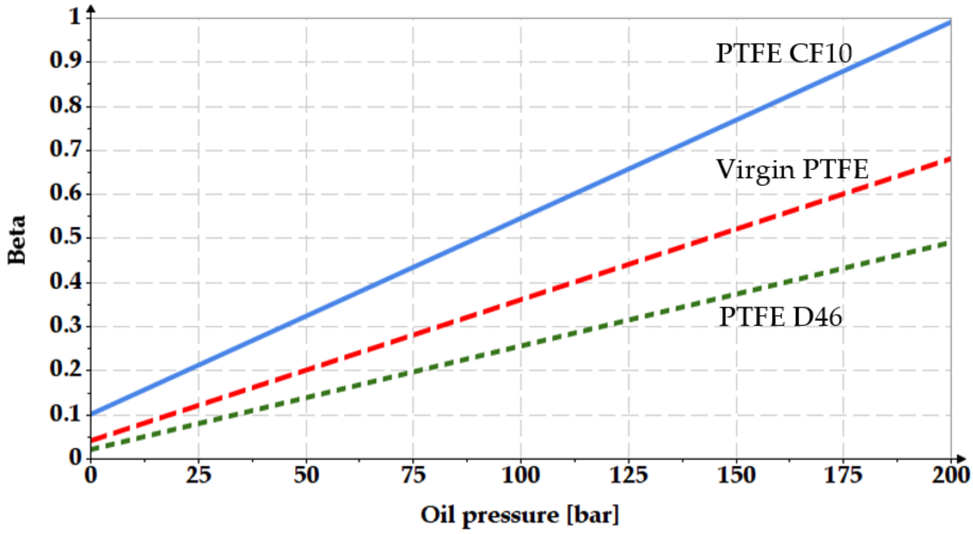


Figure 6. Dependency of the nondimensional contact surface area on seal material and sealed-off pressure [15].

According to Equation (4), the average value of the gap between seal and cylinder surface (that coincides with the thickness of the fluid film between the two elements) depends on the constructive parameters of the seal, on the material properties, and the operational characteristics of the hydraulic cylinder. With regard to seal dimensions, the dependency of fluid film thickness on seal width L, piston diameter D, and thickness h is discussed further on. The imposed limit condition is the maximum admissible value of fluid film thickness g_0 ($g_0 < 10$ μm). At this value, fluid transport is still within acceptable limits in the case of pistons, and fluid flow is still laminar in the gap; beyond this value, the flow becomes turbulent causing undesirable losses due to friction [31].

The geometry of the studied sealing systems is shown in Figure 7. All dimensions are in mm.

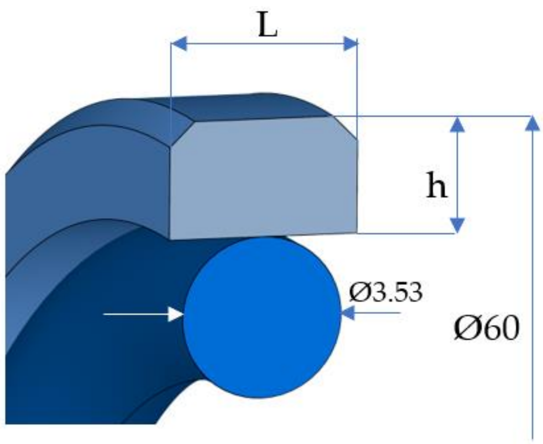


Figure 7. Dimensions of the sealing system components.

5. Results

The influence of dimensional parameter *h* (seal thickness) on fluid film thickness is shown in Figure 8.

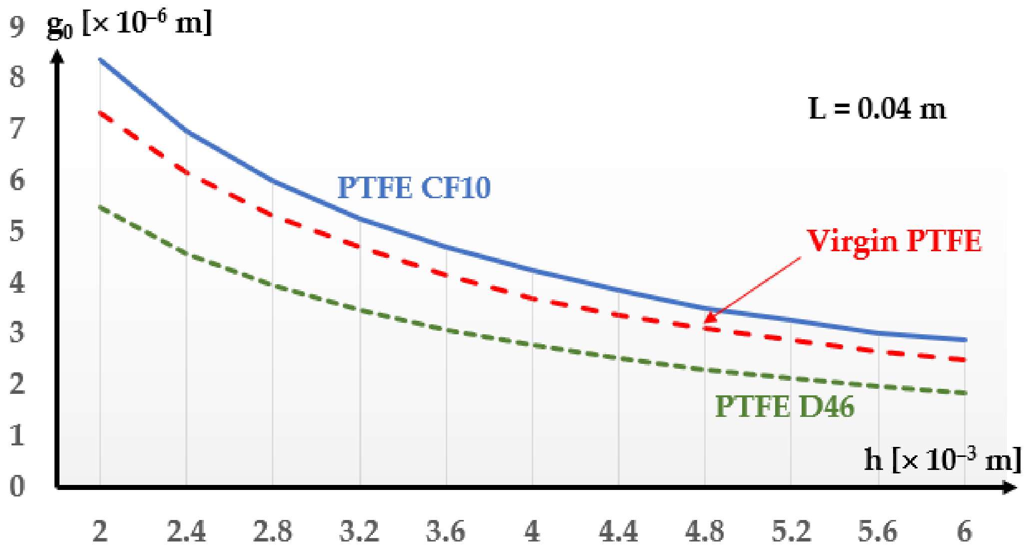


Figure 8. Variation of fluid film thickness g_0 versus parameter *h*.

For seal thickness values between 0.002 and 0.006 m, the lubricant film thickness follows a decreasing trend for all three materials. For all three materials the imposed limit condition of 10 μm is not exceeded.

The variation of fluid film thickness versus parameter *L* (seal width) is shown in Figure 9.

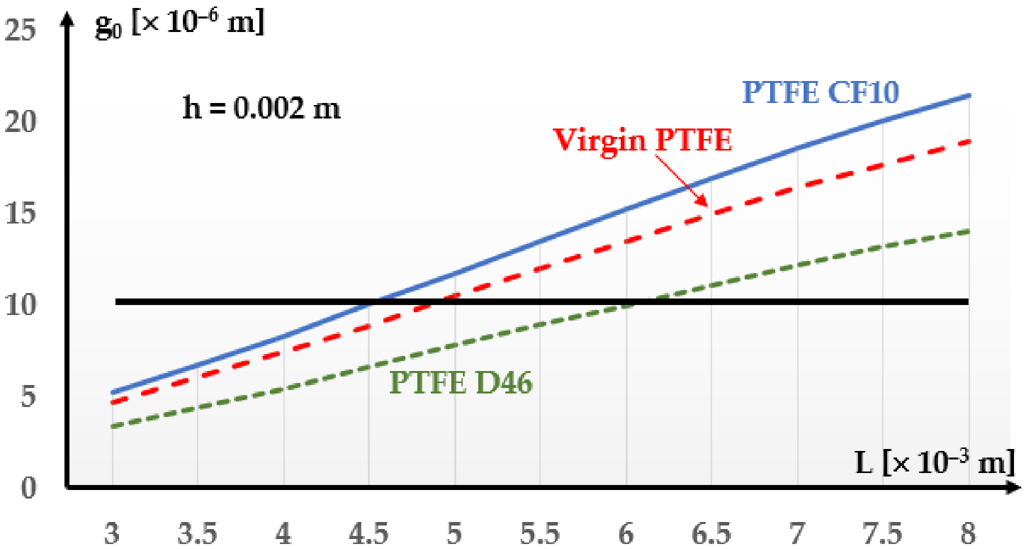


Figure 9. Variation of fluid film thickness g_0 versus parameter *L*.

In the figure above, a thick black line marks the upper limit that must not be exceeded by the thickness g_0 of the fluid film. The vertical position of the line was set to g_0 = 10 μm, according to the recommendations in [31]. The figure shows that a greater seal width favors gaps exceeding 10 μm. The reason for this behavior is, that a wider seal is subjected to a larger bending couple generated by the pressure of the working fluid. Under such conditions, the radial deformation of the seal is more pronounced. For seals made from

PTFE D46, maximum admissible seal width is 0.006 m, while for the other two materials L = 0.0045 to 0.005 m.

Thus, a fluid film of acceptable thickness is obtained between the seal and cylinder for seal thicknesses greater than 0.002 m and widths smaller than 0.0045 m. The values considered in the study of the further dependencies are h = 0.002 m and L = 0.004 m.

The influence of piston diameter on fluid film thickness g_0 is shown in Figure 10.

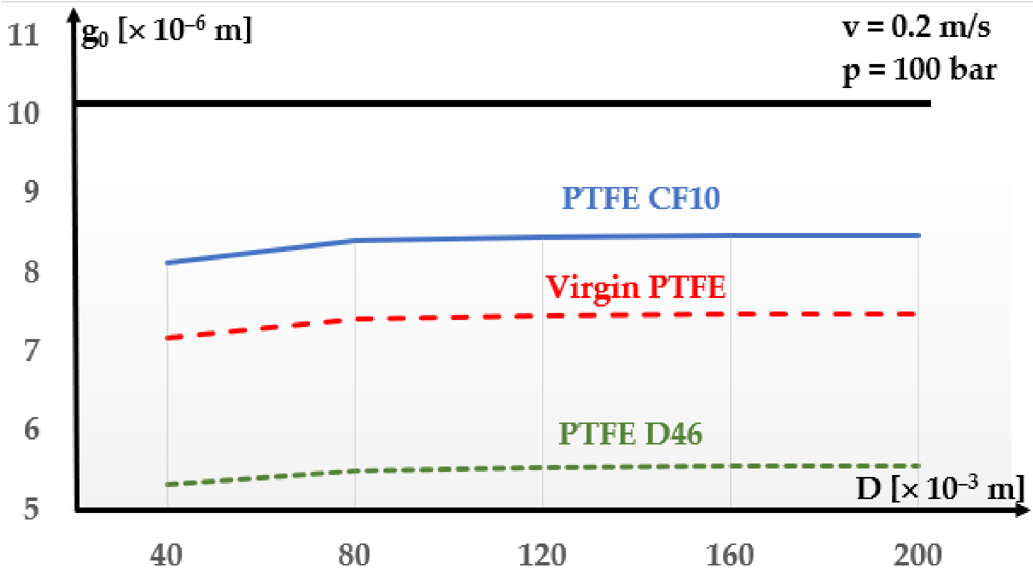


Figure 10. Variation of fluid film thickness g_0 versus parameter D.

Thus, piston diameter does not influence lubrication conditions between the seal and sealed-off surface significantly. For all materials, as the piston diameter increases from 0.04 m to 0.2 m, fluid film thickness increases only slightly up to 0.4 µm. In all analyzed cases, the imposed limit of 10 µm is not exceeded.

The values yielded by this theoretical study are close to the experimental results described in [10], with fluid thickness varying between 1 and 20 µm.

6. Discussion

- A greater seal thickness h eliminates the risk of leakage, but worsens friction (thinning of the fluid film). For the tested materials, a thickness of h = 0.002 m can be considered optimum. The decrease in fluid film thickness with the increase of parameter h is explained by the fact that the thicker seal walls entail its greater resistance to radial deformation.
- Seal width L is essential for dimensioning the sealing system. Fluid film thickness increases rapidly with seal width and exceeds the imposed limit of 10 µm. This phenomenon is explained by the fact that the seal exhibits a behavior similar to that of a thin-walled tube subject to the pressure of the working fluid. The longer the tube is, and implicitly the wider the seal is, the greater the bending couple generated by the fluid pressure will be, and the radial deformation will be more pronounced. Thus, for the studied materials and considered piston diameter it appears adequate to limit seal width to 0.004 m.
- Piston diameter influences fluid film thickness insignificantly. For typical piston diameters used in industry (D = 0.04 to 0.2 m), fluid film thickness varies between 5 µm and 8.5 µm and remains under the imposed threshold of 10 µm.
- The recommended maximum roughness (maximum peak-to-valley height) of cylinder inner surfaces is $Rmax$ = 0.63 and 2.5 µm [31]. According to the presented graphs, film thickness ranges from 1 to 20 µm; hence the friction is of fluid (hydrodynamic)

type. For a fluid film thickness g_0 of maximum 10 μm, the recommendations for selecting seal material are shown in Figure 11.

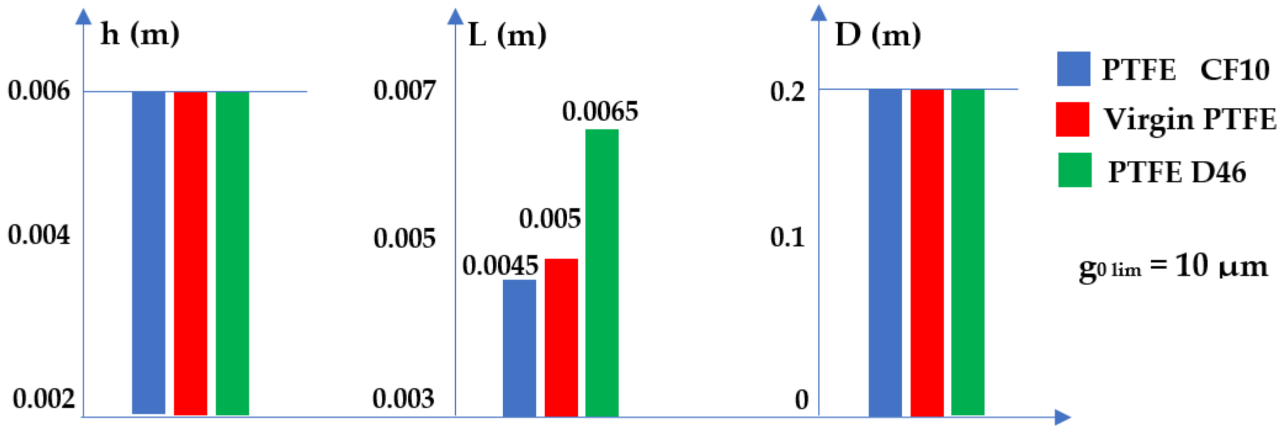


Figure 11. Recommended utilization intervals of the various materials.

Thus, PTFE D46 adulterated with bronze is the most adequate under the imposed conditions.

7. Conclusions

Coaxial sealing systems are widely used in linear hydraulic motors due to their simple construction and solid performance. This includes good packing capacity of pressurized enclosures and small friction forces. This latter benefit is due to polytetrafluoroethylene (PTFE), an excellent low-friction material. This paper has presented the theoretical results of a detailed study on the influence of certain dimensional and material characteristics on the performance of sealing systems. The study concerned three coaxial sealing systems made from PTFE-based materials, with 100% PTFE concentration, with 10% carbon fibers, and with 46% bronze, respectively. With the onset of motion, the seal is detached from the surface of the cylinder and the gap is filled by a fluid film of a certain thickness. The dimensions of this fluid film are calculated and the presence of hydrodynamic friction is demonstrated.

Significant conclusions are as follows:

- A greater seal thickness h can prevent hydrodynamic friction and generate the risk of mixed or even dry friction. From the analyzed dimensional range (h = 0.002 to 0.006 m), the smallest value is recommended (h = 0.002 m). All studied materials are adequate for seals.
- At large values of seal width L, leakage may occur due to the increased thickness of the fluid film. Thus, limiting seal width to 0.004 m is adequate for the studied materials.
- Piston diameter does not influence fluid film thickness decisively, so that its maximum admissible thickness is not exceeded.

Of the three studied polymer materials, the most adequate for deployment in sealing systems is PTFE D46 with added bronze. The other two materials can also be used for seals, provided width L is limited to smaller values.

The obtained results provide designers and users of coaxial sealing systems with useful information for ensuring superior performance of hydraulic cylinders. A future avenue of research envisages research on the influence of nonrectangular section seal characteristics on the type of friction.

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