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Investigation of friction in rectangular Nitrile-Butadiene Rubber (NBR) hydraulic rod seals for defence applications †

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Abstract

Contact based FE simulations have been carried out to estimate the contact pressure distribution at seal/rod interface at sealed oil pressures of 10, 20 and 30 MPa and constant rod velocity of 0.12 m/s. Oil film thickness at the interface was then computed analytically at various combinations of oil pressures and rod velocities. Seal contact pressure and oil film thickness data along with surface roughness, intermolecular interaction between seal/rod interfaces has been perused to estimate the friction in Nitrile-Butadiene Rubber (NBR) rectangular hydraulic rod seals using theoretical models such as Inverse hydrodynamic lubrication (IHL), Greenwood-Williamson (GW) and Wassink's models. The friction at seal/rod interface was also measured experimentally using a specially designed test rig. The comparison of theoretical and experimental data revealed that, friction computed from GW and Wassink's models had good agreement with the experimental results.

Keywords: Reciprocating hydraulic seal; Seal friction; Seal test rig; NBR seal; Stribeck curve

1. Introduction

Reciprocating seals are essential components used in variety of industrial, automobile, aerospace, and marine applications that involve linear actuators. Schematic of a typical linear actuator consisting of stationary cylindrical tube, reciprocating rod and piston is shown in Fig. 1(a). The fluid pressure on piston side rises due to external shock load acting on the reciprocating rod. Owing to its relative density, viscosity, bulk modulus and more importantly load carrying capacity of oil compared to air, the study of performance characteristics of hydraulic seals becomes increasingly important than pneumatic seals. The seals are generally made of polymeric or thermoplastic materials including elastomers, rubber-like materials that undergo large deformations operated under wide range of pressures ranging from 1 to 100 MPa and sliding speeds ranging from 0.1 to 2 m/s. Seals fit securely into the groove exerting contact pressure on the mating surfaces, which increases with increase in oil pressure and preventing the leakage of fluid. The estimation of frictional force at seal/rod interface is important especially in servo applications, which demand high position accuracy and smooth stick-slip-free motion. Classical seal mechanics demonstrated by stribeck curve as shown in Fig. 1(b) indicates that, friction at seal/metal interface is directly proportional to speed and inversely proportional to load. The friction is high due to surface roughness, which is termed as boundary lubrication, when the speed to pressure ratio is small. As the speed to pressure ratio increases, a thin fluid film is formed across the rubbing surfaces and frictional force arising from viscous shear stress is minimum that is termed as mixed lubrication. Further, increase in speed to pressure ratio increases fluid film thickness causing increase in frictional force that is termed as hydrodynamic lubrication. The sealing performance in terms of friction is affected by type of fluid, seal material, interacting surface and geometry of sealing interface.

Numerous researchers have attempted to determining friction, leakage and wear in different seal geometries. Nikas has studied performance of rectangular seal under elasto-hydrodynamic lubrication conditions assuming uniform contact pressure distribution along seal width [1]. Salant discussed GW model to evaluate various aspects of friction and leakage [2]. Kaneta has described friction and leakage using a glass plate and mono-chromatic optical interferometry technique [3]. Nitta has described experimental evaluation of seal contact pressure using polycarbonate film [4]. Yang et.al has developed a numerical model for a tandem reciprocating hydraulic rod seal [5]. Harp has considered viscous heat generation aspects for rotary seals [6]. Bullock has investigated friction of

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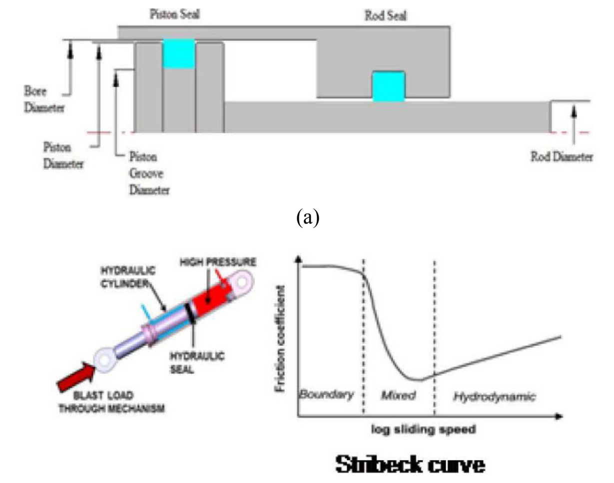

(b)

Fig. 1. Schematic of a linear actuator with reciprocating hydraulic seals.

reciprocating hydraulic seals through simulation validated by experimental results [7]. Wassink has accounted for intermolecular forces contributing towards friction in U-cup seal geometry [8]. Mao has investigated friction and leakage for a combined reciprocating hydraulic seals [9]. Fatu and Hajjam have performed numerical modeling of hydraulic seals by IHL theory [10].

However, none of the investigators have considered frictional aspects in high pressure rectangular rod seals made of NBR material. In order to fill this gap albeit partially, the current investigation is taken up to study frictional aspects in NBR rod seals at different oil pressures and rod velocities through IHL, GW and Wassink's models. Oil film thickness distribution along seal width, contact pressure at seal/rod interface, surface roughness, and intermolecular interaction between rubber seal / steel rod was also considered during estimation of friction. Frictional force was also measured experimentally by a specially designed test rig described later under Sec. 4. A comparison between theoretical and experimental results revealed that, GW and Wassink's models had good correlation with the experimental output.

2. Numerical modeling

Elastic materials undergoing finite (large) deformations may be formulated using hypoelastic or hyperelastic approach of which hyperelastic formulation is most commonly used. Laws of solid and fluid mechanics are applicable to rubber and laws of thermodynamics are considered for obtaining constitutive relationships. Stress in a hyperelastic material is a function of strain energy and deformation gradient. The strain energy function may be obtained by fitting experimental stress-strain data generated from uniaxial, biaxial, pure shear and volumetric tests using Ogden, Mooney-Rivlin, Yeoh models etc. The deformation gradient F is given by $\partial x / \partial X$ considering that, there is a relationship between deformed (x) coordinate and undeformed (X) coordinate of a body.

Table 1. Mooney-Rivlin model for NBR.

Temperature(^{0}C)	C_1 (MPa)	C_2 (MPa)	Poisson's ratio
20	40	10	0.4995
40	120	30	0.4995

Mixed or Hybrid approach is used for incompressible materials in which, the variables are displacement, volume and Lagrangian pressure. This approach is preferred for rubber FE analysis, since it reduces mesh locking when it is used in conjunction with reduced integration i.e. an aspect of zero deformation (nodal displacements) when Poisson's ratio approaches 0.5. The first order strain energy function for a hyperelastic Mooney-Rivlin model is described as,

$$\Psi = C_1\left(\bar{I}_1 - 3\right) + C_2\left(\bar{I}_2 - 3\right) + \frac{1}{D_1}(J-1)^2$$

where $\bar{I}_1 = J^{-\frac{2}{3}} I_1; \bar{I}_2 = J^{-\frac{4}{3}} I_2; J = |F|$; D_1 = 2/(bulk modulus) for an incompressible material J = 1.

FE analysis is a powerful tool for solving nonlinear hyperelastic deformation behavior of rubber under different loading/boundary conditions. The output of the analysis may be measured in terms of seal's pressure on the rod due to their contact interaction. The effort and cost of conducting experiments may be minimized by FE Analysis and seal's behavior may be studied in greater detail. Number of simulations has been carried out using ABAQUS software to understand the contact pressure distribution across the seal width for different test conditions and experiments were conducted to check the validity of analysis. FE Analysis was done using three different methods viz. axisymmetric Lagrangian method, coupled Eulerian-Lagrangian method and smoothed particle hydrodynamic method. Apart from axisymmetric Lagrangian method other methods are computationally expensive. Two parameter Mooney-Rivlin (Table 1) was used in the FE model with minimum element size of 0.05 mm [9].

An axisymmetric FE model for rectangular profile with hybrid formulation using ABAQUS is shown in Fig. 2. The CAX4RH element was used for seal and CAX4R element for rod and gland. In any numerical approach, the number of nodes or in other words number of elements considered in the model generally has a direct effect on the results. Therefore, a mesh convergence study has been carried out by increasing number of mesh elements from 10000 to 15000 in steps of 200. No significant change in results was observed beyond 12600 elements; therefore, simulations were carried out considering 12600 elements to reduce the computational effort. Axisymmetric model has U_r and U_z degree of freedom at every node.

Load is applied as pressure on the line, which represents seal surface in 3D.

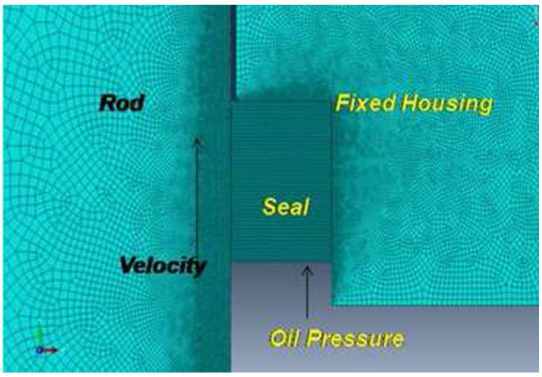

Fig. 2. Axisymmetric FE model.

3. Theoretical models

The friction arising from viscous shear force due to oil film formed, rubbing of asperities and intermolecular adhesion at seal/rod interfaces can be estimated by theoretical models such as IHL, GW and Wassink's models.

3.1 IHL model

According to IHL model, the only source of friction is assumed to be given by shear stress of the oil film formed at seal/rod interface. This model was perused to calculate the oil film thickness through the contact pressure distribution obtained from FE analysis. Fluid pressure and film thickness across the seal were assumed to be governed by 1D Reynold's equation instead of exponential relationship between pressure and viscosity [7]. If 'h' is the film thickness, η is the dynamic viscosity, p is contact pressure, x is seal co-ordinate across width and U is the rod velocity then, Reynold's equation is described as,

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\eta}\frac{\partial p}{\partial x}\right) = 6\mathrm{U}\frac{\partial h}{\partial x}. \tag{1}$$

Integrating Eq. (1),

$$\left(\frac{\partial p}{\partial x}\right) = 6\mathrm{U}\eta\left(\frac{h-h_i}{h^3}\right). \tag{2}$$

Rearranging Eq. (1),

$$h^3\frac{d}{dx}\left(\frac{1}{\eta}\frac{\partial p}{\partial x}\right) - \frac{dh}{dx}\left(6U - \frac{3h^2}{\eta}\frac{dp}{dx}\right) = 0. \tag{3}$$

Inflection point is obtained where,

$$\frac{d}{dx}\left(\frac{1}{\eta}\frac{\partial p}{\partial x}\right) = 0. \tag{4}$$

From Eq. (3), at the inflection point,

$$h_a = \sqrt{\frac{2\mathrm{U}\eta}{\left[\frac{\mathrm{dp}}{\mathrm{dx}}\right]_a}}. \tag{5}$$

h_a is the film thickness at inflection point. Substituting Eq. (4) in Eq. (2),

Integration constant: $h_i = \frac{2}{3}h_a$. Film thickness at all locations along the seal width is calculated using Eq. (2) for all $\frac{dp}{dx}$, Frictional force can be calculated as [10],

$$F_{friction} = \pi D\int_0^L \left(\frac{h}{2}\frac{dp}{dx} + \eta\frac{U}{h}\right)dx.$$

3.2 GW model

GW model assumes that, rubbing of crests and troughs of asperities and fluid viscous shear contributes towards friction [2]. Additional pressure arising due to asperities can be calculated assuming parabolic shaped asperities with Gaussian distribution; the asperity contact pressure is given by,

$$\mathrm{P_c} = \frac{4}{3}\frac{1}{\left(1-\vartheta^2\right)}\mathrm{E}\hat{\sigma}^{3/2}\frac{1}{\sqrt{2\pi}}\int_{\mathrm{H}}^{\infty}(\mathrm{z}-\mathrm{H})^{3/2}\,\mathrm{e}^{-\frac{1}{2}\mathrm{z}^2}\,\mathrm{dz}. \tag{6}$$

The average fluid shear stress is subsequently determined from Reynolds equation taking into account cavitations by Eq. (7) given below.

$$\tau_{\mathrm{av}} = \frac{\hat{\sigma}}{\mathrm{H}}\mathrm{e}^{-\alpha\hat{F}\varphi}\left(\varphi_{\mathrm{f}} - \varphi_{\mathrm{fss}}\right) - \varphi_{\mathrm{fpp}}\frac{\hat{\sigma}\,\mathrm{H}}{2}\frac{\mathrm{dF}\varphi}{\mathrm{d}\hat{\mathrm{x}}} \tag{7}$$

$$\hat{\tau}_c = -f\,P_c\left(\frac{\tau}{|\tau|}\right). \tag{8}$$

Computations of the above parameters are described by Salant [2] and Maser [11].

$$F_r = \pi D\int_0^L(\tau_{av} + \tau_c)dx. \tag{9}$$

3.3 Wassink's model

Wassink's model takes into account viscoelastic properties of rubber i.e. the frictional force is a combination of viscous shear, roughness induced and intermolecular forces at the sliding interface [8]. This is due to relaxation of rubber into the asperities of rod and intermolecular bond formation/rupture at the sliding interface.

Minimum oil film thickness can be obtained from

$$h = \sqrt[3]{\frac{\rho L U \eta_0}{P_{ct}}} . \tag{10}$$

Viscous shear force in the oil film can be obtained from

$$F_{viscous} = \rho\eta \frac{0.11DLU}{h} . \tag{11}$$

Viscoelastic energy loss contributing towards friction due to relaxation of rubber into the asperities on rod can be obtained from Eq. (12).

$$F_{roughness} = 1.1\times10^{5} D\left(\frac{P_{ct}}{P_{ct} + \left(2\pi\sigma\omega\frac{G'}{1-\vartheta}\right)}\sigma\right)^{2}\frac{G''}{\Delta x} \tag{12}$$

where P_{ct} is the average contact pressure distribution across the seal width obtained from FE analysis. Intermolecular force due to bond formation and rupture can also be accounted for stick-slip friction as described by Eq. (13) given below,

$$F_{intermolecular} = 0.23\pi DLP_{ct}\left(\frac{2}{\pi}\sin^{-1}\left(\sqrt{\frac{P_{ct}}{\left(2\pi\sigma\omega\frac{G'}{1-\vartheta}\right)}}\right)\right)\left(\frac{1}{\left(1+\frac{h}{h_m}\right)^{4}}\right)f . \tag{13}$$

The total frictional force is the addition of above three components.

4. Experimental

The schematic of a test rig specially developed as per ISO 7986 for measurement of friction in rod seals is shown in Fig. 3. A set of two seals will isolate the inside chamber that was initially charged to the sealed oil pressure. Hydraulic fluid was circulated into the chamber by a hydraulic pump and the desired test pressure was maintained by a pressure relief valve. The rod connected to a position/speed sensor was supported on bearings and due care was taken to ensure circularity and concentricity during reciprocating motion. During the test, the reciprocating motion of rod was obtained by an articulating cylinder and velocity of rod was controlled by an electronic controller. The minimum speed of the actuator was limited to 0.12 m/s based on minimum pump flow rate possible. The sealed oil pressure was considered in the range of 10-30 MPa that was within the normal working pressure existing in any hydraulic system. The friction at the interface was measured with and without test seals using load cell. The difference between the two readings was considered as the frictional force due to seals.

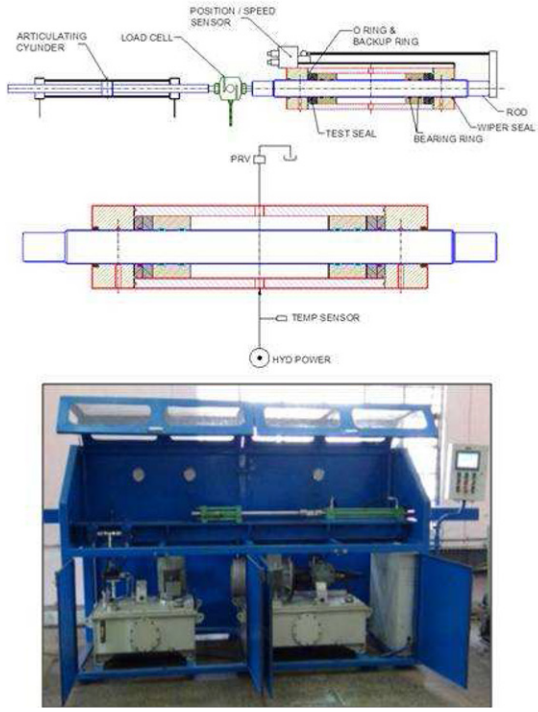

Fig. 3. Schematic of a seal test rig.

Table 2. Test parameters considered in the experiment.

Test parameter	Value
Seal geometry	Inner diameter : Ø36 mm
	Outer diameter : Ø46 mm
	Seal width : 6 mm
	Gap between seal housing and rod: 0.27 μm
Seal material properties	Nitrile-Butadiene Rubber (NBR) Shore hardness A90, Modulus of elasticity: 43 MPa Poisson's ratio : 0.49
Seal's pre-compression after assembly	10 %
Oil properties	Test temperature : 30^0C
	Oil viscosity : 0.043 Pa.s
	Oil density :890 kg/m^3
Sealed oil pressure	10, 20 and 30 MPa
Rod speed	0.12, 0.3, and 0.5 m/s
Rod average surface roughness	0.4 μm
Steel rod outer diameter	Ø36 mm

A macroscopic experimental approach was perused [12] to determine the sliding friction forces in a pneumatic actuator. Several empirical models and repeatability studies revealed random behavior of friction forces. An efficient alternative method [13] for testing and qualifying large diameter piston

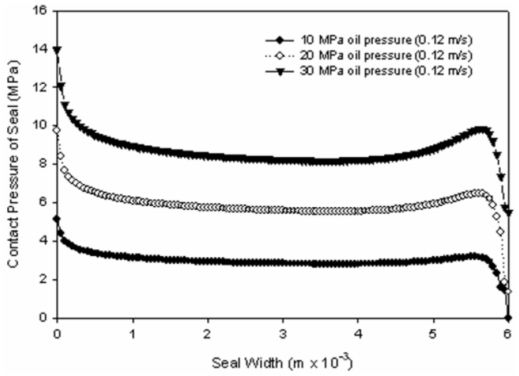

Fig. 4. Contact pressure distribution across seal width.

rod seals has been considered rather than conventional tests, in which seals are tested as part of complete hydraulic cylinder assembly. The method employs an articulating cylinder, which moves a test piston rod through a pressurized chamber containing the test seals, thus reducing test time and energy. The ISO standard [14] brings out methods of performance test of reciprocating hydraulic seals. The test parameters considered in the experiment are given in Table 2.

5. Results and discussion

5.1 Estimation of contact pressure distribution across the seal width

The FE model as described under Sec. 2 was solved for sealed oil pressures of 10, 20 and 30 MPa and rod velocities of 0.12, 0.3 and 0.5 m/s. The contact pressure distribution across seal width at rod velocity 0.12 m/s is shown in Fig. 4. It may be observed that, the contact pressure is higher at 30 MPa and lower at 10 MPa and is more or less uniform across the seal width except near the edges. This is due to the fact that, since the seal housing is constrained, the step of the groove exerts reaction on the seal. Further, as the seal gets compressed due to applied oil pressure, the seal contact pressure increases near the reaction zone. In addition, seal's nonlinearity and the boundary conditions i.e. degree of freedom in the vertical direction allow the seal to expand therefore, the seal experiences less contact pressure on the oil side. It may also be noted from Fig. 4 that, the contact pressure followed a similar pattern at different oil pressures.

5.2 Estimation of oil film thickness and friction at seal/rod interface

Experimental evaluation of film thickness is rather a complex phenomenon. Oil film thickness is bound to vary as a function of displacement and time due to seal's viscoelasticity, rod's surface roughness and intermolecular adhesion, Therefore, oil film thickness across the seal width at different oil pressures and rod velocities has been calculated through IHL theory assuming perfectly smooth rod and plotted in Fig. 5. The oil film thickness and the frictional force are

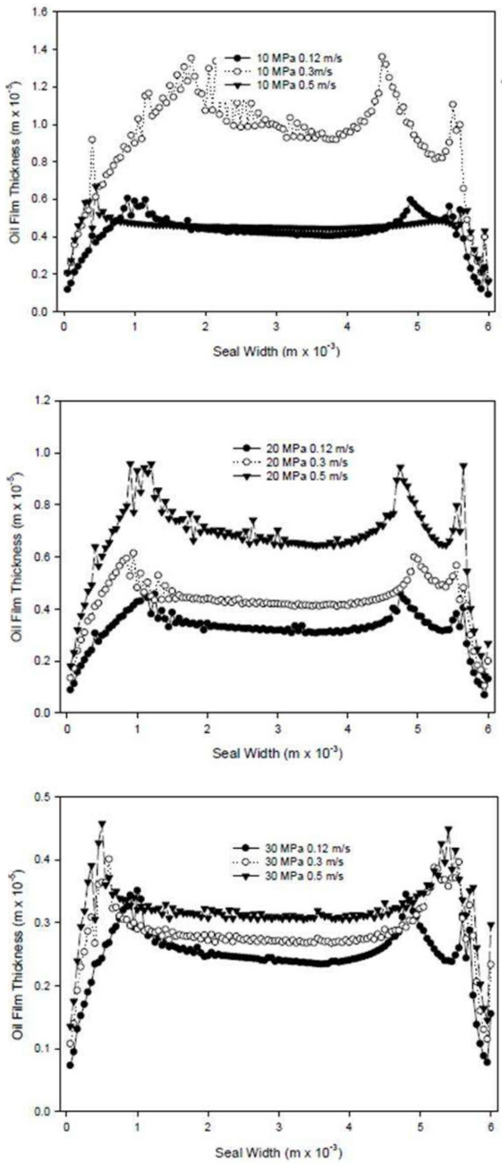

Fig. 5. Oil film thickness across seal width.

calculated from a nonlinear equation, which depends on the contact pressure gradient and rod velocity. Experiments were also carried out as described under Sec. 4 to measure the frictional force which is assumed to be a function of oil film thickness. It has been observed that, the magnitude of the frictional force given by IHL model is very small compared to experimental results as shown in Fig. 6, because it considers only the contribution of viscous shear stress. Further, theoretical and experimental values show a trend corresponding to Stribeck curve that indicates the variation of coefficient of friction as a function of $\eta U/P$. Bullock et al. [7], described that, if h_i indicates average film thickness, the friction is proportional to $U^{0.5}$. However, there is no such correlation found in the current investigation. This is due to the fact that, the pressure-viscosity was assumed to be absent in the present study.

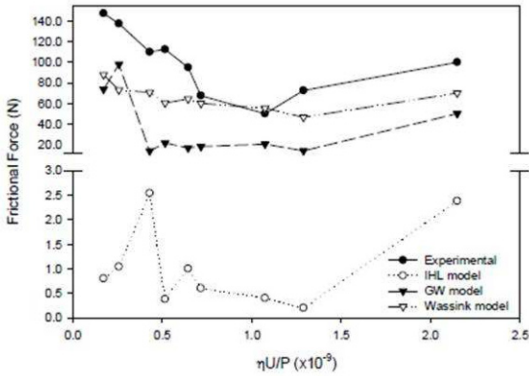

Fig. 6. Friction (one seal) as a function of ηU/P.

Further, there is a significant improvement in the value of frictional force obtained by GW model, since it considers effect of surface roughness too in addition to viscous shear stress. Therefore, total frictional force can be assumed as a function of viscous force, roughness and intermolecular forces between the sliding interfaces. Roughness induced force takes into account a linear viscoelastic model based on storage and loss moduli, deformation and recovery of seal result in loss of energy contributing to total friction. In addition, intermolecular adhesion takes place at the interface and the resulting bond formation/rupture cycles occurring during sliding produce hysteresis losses in rubber leading to increased friction. If the average film thickness computed from IHL theory as shown in Fig. 5 is used then the theory proposed by Wassink et al. also predicts the frictional behavior reasonably well. Though various theories predict the friction, Wassink's theory has been found to have a better approximation with the actual values. It may be noted from Fig. 6 that, there is no exact correlation between analytical and experimental results, which may be due to contribution of viscous heating of rubber adding to contact pressure during the experimentation. The results may be further refined by performing the coupled temperature displacement analysis along with fluid to arrive at the exact contact pressure distribution. Prony series viscoelasticity [15], Mullin's effect, loading and unloading history of the seal should be taken into account for accurate estimation of friction.

6. Conclusions

(1) The seal contact pressure increased near the reaction zone and deceased on the oil side due to seal material nonlinearity and its degree of freedom in the vertical direction.

(2) The magnitude of friction at seal/rod interface given by IHL model is very small compared to experimental results, because it considers the contribution of viscous shear stress only.

(3) The frictional values obtained from GW model were more realistic, because this model will account for surface roughness too unlike IHL model.

(4) The values of friction obtained from theoretical models and experiments indicated a pattern resembling Stribeck curve.

(5) A comparison between theoretical and experimental results revealed that, GW and Wassink's models had good correlation with the experimental output.

(6) The data generated in the current investigation is useful in the design of a better sealing system for linear hydraulic actuators used in various industrial and defence applications.

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Nomenclature

D : Rod diameter, m
E : Modulus of elasticity of rubber, MPa
f : Coefficient of friction
G' : Storage modulus of rubber, Pa
G'' : Loss modulus of rubber, Pa
h : Film thickness, 10^{-5}m
H : Dimensionless film thickness, h/σ_h
I_1, I_2, I_3 : Three invariants of the green deformation tensor
$\bar{I}_1, \bar{I}_2, \bar{I}_3$: First, second and third invariants of deviatoric strain
L : Length of solution domain in x-direction, mm
N : Asperity density, 10^{13} m^{-2}
P : Contact pressure, MPa
P_{ct} : Average Contact Pressure, Pa
R : Asperity radius, 10^{-6} m
U : Rod Velocity, m/s
η_0 : Dynamic viscosity, Pa·s
σ_h : Average surface roughness height, µm
$\hat{\tau}_c$: Dimensionless roughness, $\sigma_h N^{2/3} R^{1/3}$
ν : Poisson's ratio
ζ : Dimensionless rod speed, η_0UL/pa σ_h^2
$\varphi_f, \varphi_{fss}, \varphi_{fpp}$: Flow factors
ρ : Density of oil, kg/m^3
ω : Spatial frequency of roughness, m^{-1}

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- Listed in Who's Who in the World (29th edition), 2012.
- Fellow, Institution of Engineers (India).
- Visited Lehigh University, Bethlehem, Pennsylvania, USA, Dec, 2013.
- Reviewer, International journal of Materials Science & Engg-B (Elsevier)
- Reviewer, Journal of Mechanical Science and Technology (Springer)
- Reviewer, International Journal of the Physical Sciences (Elsevier).
- Reviewer, International journal of Materials & Design (Elsevier).
- Recipient of National Technology Day Oration award (NTD-2015).

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