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Effects of velocity-slip and viscosity variation in hydrostatic step-seal

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HIGHLIGHTS

- *This study is done through various parameters using mathematical equations.*
- *The study will be very much helpful for the experimental investigators in various industries.*
- *The coefficient of friction in the hydrodynamic lubrication can be studied by taking different fluids in future in the experimental studies.*
- *To determine the slip effects in different squeeze film bearings which will be helpful in different fields like biomedicine.*

ABSTRACT

A generalized form of Reynolds equation for two symmetrical surfaces is taken by considering velocity-slip at the bearing surfaces. This equation is applied to study the effects of velocity-slip and viscosity variation in squeeze film lubrication of hydro static step-seal. Expressions for the load capacity are studied theoretically for various parameters. The load capacity decreases due to slip. The Load Capacity increases due to the presence of high viscous layer near the surface and decreases due to low viscous layer and it has been shown that the load capacities of squeeze film bearings decrease and those of step seals increase with an increase in the step height. With a hydrostatic step seal, the load capacity increases as the flow behaviour index of the fluid increases.

Keywords:
| Reynolds equation | Velocity-slip| Viscosity variation | Squeeze film lubrication | Load capacity | Squeezing time |

NOMENCLATURE

h Total film thickness
h_f Final film thickness
k Ratio of the viscosities
l Length of the bearing
P Hydrodynamic Pressure
R Radius of the surfaces in case of circular plates
T Squeezing time of for stiff surfaces
V Squeeze Velocity
W Load capacity for stiff surfaces
μ Viscosity of the purely hydrodynamic zone

1.0 INTRODUCTION

Most lubricated systems can generally be considered to consist of moving (or stationary) surfaces (planes/curves, loaded/unloaded), with a thin film of an external material (lubricant) between them. The presence of the thin film between these surfaces not only helps to support considerable loads, but also minimizes friction. System characteristics, such as pressure within the film, surface frictional force, lubricant flow rate etc., depend upon the nature of the surfaces, the lubricant's film boundary conditions, etc.

The equation governing the pressure generated within the lubricant film can be obtained by coupling the equations of motion and continuity. This was first derived by Reynolds (1866) and is known as "Reynolds Equation." In deriving this equation, thermal, compressibility, viscosity variation, surface slip, inertia and surface roughness effects were ignored. The Reynolds equation was later modified by Cope (1949) to include viscosity and density variations along the fluid film. Viscosity variation across the film's thickness was considered by Zienkiewicz (1957) and Cameron (1958), who also highlighted that temperature gradient and viscosity variation across the film should not be ignored. Later, Dowson (1962) unified all of these attempts in generalizing the Reynolds Equation by considering the variations of fluid properties across, as well as, along the fluid film's thickness, by neglecting the slip effects at the bearing surfaces. Since then, many researchers including myself (Shukla, 1964; Shukla et al., 1982; Sinha et al., 1983; Prakash and Sinha, 1977; Rao and Prasad, 2001; Rao and Prasad, 2002; Rao and Prasad, 2003; Rao et al., 2014) have studied the effects of viscosity variation in lubricated systems by considering the Reynolds Equation with an energy equation. Furthermore, a study by Patel et al. (2010) investigated the performance of a magnetic fluid based squeeze film between transversely rough triangular plates. Also, Shimpi and Deheri (2010) studied surface roughness and elastic deformation effects on the behaviour of a magnetic fluid based squeeze film between rotating porous circular plates with concentric circular pockets; which Shimpi and Deheri (2012) improved to include rotating curved porous circular plates. Furthermore, Rao et al. (2013) studied the effects of velocity-slip and viscosity variation in the squeeze film lubrication of two circular plates and spherical bearings. Sheikholeslami and Bhatti (2017) studied active methods for nanofluid heat transfer enhancement by means of EHD. Following that, Sheikholeslami and Shehzad (2017) studied the thermal radiation of ferrofluid in the presence of Lorentz forces by considering variable viscosity. Next, Sheikholeslami et al. (2017) studied numerical simulation of nanofluid forced convection heat transfer improvement in the presence of magnetic field using the lattice Boltzmann method. Also, Sheikholeslami (2017) studied both CuO-water nanofluid free convection in a porous cavity considering Darcy's Law and magneto hydrodynamic nanofluid forced convection in a porous lid driven cubic cavity using the lattice Boltzmann method. Additionally, Ram (2016) studied separately the performance of non-recessed hole-entry hybrid journal bearing operating under a turbulent regime and the numerical analysis of capillary compensated micropolar fluid lubricated hole-entry journal bearings. In this study, the effects of velocity-slip and viscosity variation in the squeeze film lubrication of hydrostatic step-seal have been discussed.

2.0 BASIC EQUATIONS

Consider the laminar flow of a fluid between two symmetric surfaces, whose physical configuration is as shown in the Figure 1.

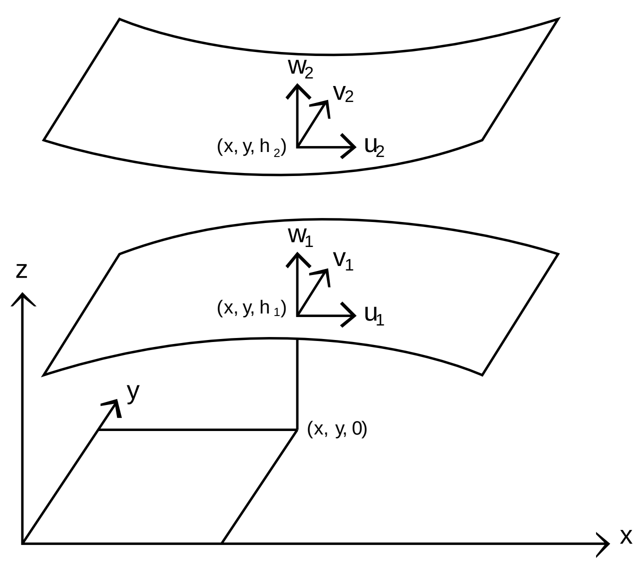

Figure 1: Coordinate system

Considering the variation of fluid properties across as well as along the film thickness, the basic equations of motion and equation of continuity in their general form for a Newtonian fluid can be written as follows:

$$\rho\,\frac{Du}{Dt} = \rho X - \frac{\partial P}{\partial x} + \frac{2}{3}\frac{\partial}{\partial x}\left\{\eta\left(\frac{\partial u}{\partial x} - \frac{\partial v}{\partial y}\right)\right\} + \frac{2}{3}\frac{\partial}{\partial x}\left\{\eta\left(\frac{\partial u}{\partial x} - \frac{\partial w}{\partial z}\right)\right\}$$

$$+ \frac{\partial}{\partial y}\left\{\eta\left(\frac{\partial v}{\partial x} + \frac{\partial u}{\partial y}\right)\right\} + \frac{\partial}{\partial z}\left\{\eta\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)\right\} \tag{1}$$

$$\rho\,\frac{\mathrm{Dv}}{\mathrm{Dt}} = \rho \mathrm{Y} - \frac{\partial \mathrm{P}}{\partial \mathrm{y}} + \frac{2}{3}\frac{\partial}{\partial \mathrm{y}}\left\{\eta\left(\frac{\partial \mathrm{v}}{\partial \mathrm{y}} - \frac{\partial \mathrm{u}}{\partial \mathrm{x}}\right)\right\} + \frac{2}{3}\frac{\partial}{\partial \mathrm{y}}\left\{\eta\left(\frac{\partial \mathrm{v}}{\partial \mathrm{y}} - \frac{\partial \mathrm{w}}{\partial \mathrm{z}}\right)\right\}$$

$$+ \frac{\partial}{\partial \mathrm{z}}\left\{\eta\left(\frac{\partial \mathrm{w}}{\partial \mathrm{y}} + \frac{\partial \mathrm{v}}{\partial \mathrm{z}}\right)\right\} + \frac{\partial}{\partial \mathrm{x}}\left\{\eta\left(\frac{\partial \mathrm{v}}{\partial \mathrm{x}} + \frac{\partial \mathrm{u}}{\partial \mathrm{y}}\right)\right\} \tag{2}$$

$$\rho\,\frac{Dw}{Dt} = \rho Z - \frac{\partial P}{\partial z} + \frac{2}{3}\frac{\partial}{\partial z}\left\{\eta\left(\frac{\partial w}{\partial z} - \frac{\partial u}{\partial x}\right)\right\} + \frac{2}{3}\frac{\partial}{\partial z}\left\{\eta\left(\frac{\partial w}{\partial z} - \frac{\partial v}{\partial y}\right)\right\}$$

$$+ \frac{\partial}{\partial x}\left\{\eta\left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x}\right)\right\} + \frac{\partial}{\partial y}\left\{\eta\left(\frac{\partial w}{\partial y} + \frac{\partial v}{\partial z}\right)\right\} \tag{3}$$

$$\frac{\partial \rho}{\partial \mathrm{t}} + \frac{\partial}{\partial \mathrm{x}}(\rho \mathrm{u}) + \frac{\partial}{\partial \mathrm{y}}(\rho \mathrm{v}) + \frac{\partial}{\partial \mathrm{z}}(\rho \mathrm{w}) = 0$$

with the following usual assumptions of lubrication theory:

i. Inertia and body force terms are negligible compared with the pressure and viscous terms.
ii. There is no variation of pressure across the fluid film, which means $\frac{\partial \rho}{\partial \mathrm{z}} = 0$.
iii. There is no slip in the fluid-solid boundaries.
iv. No external forces act on the film.
v. The flow is viscous and laminar.
vi. Due to the geometry of fluid film the derivatives of u and v with respect to zare much larger than other derivatives of velocity components.
vii. The height of the film h is very small compared to the bearing length, *l*. A typical value of h/*l* is about 10^{-3}.

The Navier–Stoke's equation 1 can be simplified Dowson (1962) as follows:

$$\frac{\partial \mathrm{P}}{\partial \mathrm{x}} = \frac{\partial}{\partial \mathrm{z}}\left[\eta\frac{\partial \mathrm{u}}{\partial \mathrm{z}}\right] \tag{5}$$

$$\frac{\partial P}{\partial y} = \frac{\partial}{\partial z}\left[\eta \frac{\partial v}{\partial z}\right] \tag{6}$$

Where P = P (x,y) is the pressure in the film and η is the viscosity. The boundary conditions considering slip at the surfaces (Beavers and Joseph, 1967) are:

$$u = (u)_1 = (\lambda)_1\left[\frac{\partial u}{\partial z}\right]_1 + U_1 \tag{7}$$

At Z = H_1,

$$v = (v)_1 = (\delta)_1\left[\frac{\partial v}{\partial z}\right]_1 + V_1 \tag{8}$$

$$u = (u)_2 = -(\lambda)_2\left[\frac{\partial u}{\partial z}\right]_2 + U_2 \tag{9}$$

At Z = H_2

$$v = (v)_2 = -(\delta)_2\left[\frac{\partial v}{\partial z}\right]_2 + V_2 \tag{10}$$

Where ( )1 ( )2 denote the value at z = H1 and z = H2. Here, λ and δ s are molecular mean free path for gas lubrication and depend upon the lubricant temperature, pressure and viscosity. In liquid lubrication and depend on viscosity and the coefficient is sliding friction. However, with porous bearings and are functions of slip coefficient at the wall and the permeability parameter of the porous facing. Integrating equation 5 and equation 6 and using boundary conditions of 8, 9 and 10, the expressions for the fluid film velocities are obtained

$$u = U_1 + \left[\alpha_1 H_1 + \int_{H_1}^{z}\frac{z\,dz}{\eta}\right]\frac{\partial P}{\partial x} + \left[\frac{U_2 - U_1}{F_0} - \frac{F_1}{F_0}\frac{\partial P}{\partial x}\right]\left[\alpha_1 + \int_{H_1}^{z}\frac{dz}{\eta}\right] \tag{11}$$

$$v = V_1 + \left[\beta_1 H_1 + \int_{H_1}^{z}\frac{z\,dz}{\eta}\right]\frac{\partial P}{\partial y} + \left[\frac{V_2 - V_1}{F_0^1} - \frac{F_1^1}{F_0^1}\frac{\partial P}{\partial y}\right]\left[\beta_1 + \int_{H_1}^{z}\frac{dz}{\eta}\right] \tag{12}$$

Where

$$F_0 = \alpha_1 + \alpha_2 + \int_{H_1}^{z}\frac{dz}{\eta} \tag{13}$$

$$F_0^1 = \beta_1 + \beta_2 + \int_{H_1}^{z}\frac{z\,dz}{\eta} \tag{14}$$

$$F_1 = \alpha_1 H_1 + \alpha_2 H_2 + \int_{H_1}^{H_2}\frac{z\,dz}{\eta}$$

$$F_1^1 = \beta_1 H_1 + \beta_2 H_2 + \int_{H_1}^{z} \frac{z dz}{\eta} \tag{15}$$

$$\alpha_1 = \frac{(\lambda)_1}{(\eta)_1}, \ \alpha_2 = \frac{(\lambda)_2}{(\eta)_2}, \quad \beta_1 = \frac{(\delta)_1}{(\eta)_1}, \quad \beta_2 = \frac{(\delta)_2}{(\lambda)_2} \tag{16}$$

Integrating the equation of continuity (2) w.r.t.z. and taking limits from z = H_1 to z = H_2 gives

$$\int_{H_2}^{H_2} \frac{\partial \rho}{\partial t} dz + \int_{H_1}^{H_2} \frac{\partial}{\partial x}(\rho u) dz + \int_{H_1}^{H_2} \frac{\partial}{\partial y}(\rho v) dz + (\rho w)_{H_1}^{H_2} = 0 \tag{17}$$

The integrals of (ρu) and (ρv) are evaluated by partial integration. Introducing the expressions for (ρu) and (ρv) and their derivatives in equation 17 gives

$$\frac{\partial}{\partial x}\left\{\left(F_2 + G_1\right)\frac{\partial P}{\partial x}\right\} + \frac{\partial}{\partial y}\left\{\left(F_2^1 + G_1^1\right)\frac{\partial P}{\partial y}\right\}$$

$$= H_2\left\{\frac{\partial}{\partial x}(\rho u)_2 + \frac{\partial}{\partial y}(\rho v)_2\right\} - H_1\left\{\frac{\partial}{\partial x}(\rho u)_1 + \frac{\partial}{\partial y}(\rho v)_1\right\}$$

$$-\frac{\partial}{\partial x}\left\{\frac{(U_2 - U_1)(F_3 + G_2)}{F_0} + U_1 G_3\right\} - \frac{\partial}{\partial y}\left\{\frac{(V_2 - V_1)(F_3 + G_2^1)}{F_0^1} + V_1 G_3\right\}$$

$$+ \int_{H_1}^{H_2} \frac{\partial \rho}{\partial y} dz + (\rho w)_{H_1}^{H_2} \tag{18}$$

Where

$$F_2 = \int_{H_1}^{H_2} \frac{\rho z}{\eta}\left[z - \frac{F_1}{F_0}\right] dz \tag{19}$$

$$F_2^1 = \int_{H_1}^{H_2} \frac{\rho z}{\eta}\left[z - \frac{F_1^1}{F_0^1}\right] dz, \quad F_3 = \int_{H_1}^{H_2} \frac{\rho z}{\eta} dz \tag{20}$$

$$G_1 = \int_{H_1}^{H_2} \left[z \frac{\partial \rho}{\partial z} \left\{ \alpha_1 H_1 + \int_{H_1}^{z} \frac{zdz}{\eta} - \frac{F_1}{F_0} \left[\alpha_1 + \int_{H_1}^{z} \frac{dz}{\eta} \right] \right\} \right] dz \tag{21}$$

$$G_1^1 = \int_{H_1}^{H_2} \left[z \frac{\partial \rho}{\partial z} \left\{ \beta_1 H_1 + \int_{H_1}^{z} \frac{zdz}{\eta} - \frac{F_1^1}{F_0^1} \left[\beta_1 + \int_{H_1}^{z} \frac{dz}{\eta} \right] \right\} \right] dz \tag{22}$$

$$G_2 = \int_{H_1}^{H_2} \left\{ z \frac{\partial \rho}{\partial z} \left[\alpha_1 + \int_{H_1}^{z} \frac{dz}{\eta} \right] \right\} dz \tag{23}$$

$$G^2{}_1 = \int_{H_1}^{H_2} \left\{ z \frac{\partial \rho}{\partial z} \left[\beta_1 + \int_{H_1}^{z} \frac{dz}{\eta} \right] \right\} dz, \quad G_3 = \int_{H_1}^{H_3} z \frac{\partial \rho}{\partial z} dz \tag{24}$$

Equation 18 represents a generalized form of Reynolds equation for compressible fluid film lubrication considering slip velocities at the bearing surfaces. The two sets of functions F and G depend upon the variation of fluid properties along, as well as across, the film and on the slip conditions at the surfaces

i.e., $(\lambda)_1 = (\lambda)_2 = (\delta)_1 = (\delta)_2 = 0$

$$\alpha_1 = \alpha_2 = \beta_1 = \beta_2 = 0$$

The velocity of the lubricant can vary across the film and may be different near the bearing surfaces owing to the reaction of additives and surfactants with the surfaces (Shimpi and Dehari, 2012; Zienkiewiez, 1957). Considering a reasonable case where the density and viscosity of the lubricant near the bearing surfaces may be different from the central region, we can have

$\rho = \rho_1 (x, y), \eta = \eta_1 (x, y)$ $\quad H_1 \leq z \leq H_1 + h_1$

$\rho = \rho_2 (x, y), \eta = \eta_2 (x, y)$ $\quad H_1 + h_1 \leq z \leq H_1 + h_1 + h_2$

$\rho = \rho_3 (x, y), \eta = \eta_3 (x, y)$ $\quad H_1 + h_1 + h_2 \leq z \leq H_1 + h_1 + h_2 + h_3$

This introduces the concept of multiple-layer lubrication. By taking

$U_1 = U$ $\quad U_2 = V_1 = V_2 = 0$

$\alpha_1 = \beta_1$ $\quad \alpha_2 = \beta_2$

$\frac{\partial \rho_i}{\partial z} = 0$ $\quad i = 1,2,3, \ldots\ldots\ldots$

The generalized equation with slip reduces to the following form.

$$\frac{\partial}{\partial x}\left[F_2\frac{\partial P}{\partial x}\right]+\frac{\partial}{\partial y}\left[F_2\frac{\partial P}{\partial y}\right]=H_2\left\{\frac{\partial}{\partial x}(\rho u)_2+\frac{\partial}{\partial y}(\rho v)_2\right\}-H_1\left\{\frac{\partial}{\partial x}(\rho u)_1+\frac{\partial}{\partial y}(\rho v)_1\right\}$$

$$+U\frac{\partial}{\partial x}\left[\frac{F_3}{F_0}\right]+[\rho w]_{H_1}^{H_2} \tag{25}$$

Where

$$F_0=\alpha_1+\alpha_2+\frac{h_1}{\eta_1}+\frac{h_2}{\eta_2}+\frac{h_3}{\eta_3}$$

(26)

$$F_1=\alpha_1 H_1+\alpha_2 H_2+\frac{h_1(2H_1+h_1)}{2\eta_1}+\frac{h_2(2H_1+2h_1+h_2)}{2\eta_2}$$

$$+\frac{h_3(2H_1+2h_1+2h_2+h_3)}{2\eta_2} \tag{27}$$

$$F_2=\frac{\rho_1}{3\eta_1}\left\{(H_1+h_1)^3-H_1^3\right\}+\frac{\rho_2}{3\eta_2}\left\{(H_1+h_1+h_2)^3-(H_1+h_1)^3\right\}$$

$$+\frac{\rho_3}{3\eta_3}\left\{H_2^3-(H_1+h_1+h_2)^3\right\}-\frac{F_1F_3}{F_0} \tag{28}$$

$$F_3=\frac{\rho_1 h_1}{2\eta_1}(2H_1+h_1)+\frac{\rho_2 h_2}{2\eta_2}(2H_1+2h_1+h_2)+\frac{\rho_3 h_3}{2\eta_3}(2H_1+2h_1+2h_2+h_3) \tag{29}$$

$$(\rho u)_1=\rho_1\alpha_1\left[H_1-\frac{F_1}{F_0}\right]\frac{\partial P}{\partial x}+\rho_1 U\left[1-\frac{\alpha_1}{F_0}\right] \tag{30}$$

$$(\rho u)_2=-\rho_3\alpha_2\left[H_2-\frac{F_1}{F_0}\right]\frac{\partial P}{\partial x}+\rho_3 U\frac{\alpha_2}{F_0} \tag{31}$$

$$(\rho v)_1=\rho_1\alpha_1\left[H_1-\frac{F_1}{F_0}\right]\frac{\partial P}{\partial y} \tag{34}$$

$$(\rho v)_2=-\rho_3\alpha_2\left[H_2-\frac{F_1}{F_0}\right]\frac{\partial P}{\partial y} \tag{35}$$

$$[\rho w]_{H_1}^{H_2}=(\rho u)_2\frac{\partial H_2}{\partial x}+(\rho v)_2\frac{\partial H_2}{\partial y}-(\rho u)_1\frac{\partial H_1}{\partial x}-(\rho v)_1\frac{\partial H_1}{\partial y}-V_s \tag{36}$$

Here, V_s is the resultant velocity towards the film. To see the effect of slip, consider three symmetrical incompressible layers between two solid boundaries.

$\eta_1 = \eta_2 \quad \rho_1 = \rho_2 = \rho_3$

$H_1 = 0$ $\qquad$ $H_2 = (h+a)=h,\ h_1 = h_3 = a/2,\ h_2 = (h-a)$

$\alpha_1 = \alpha_2 = \beta_1 = \beta_2 = 1/\beta$

The Reynolds equation can be written from equation 25 as follows:

$$\frac{\partial}{\partial x}\left[F_4 \frac{\partial P}{\partial x}\right] + \frac{\partial}{\partial y}\left[F_4 \frac{\partial P}{\partial y}\right] = U\frac{\partial}{\partial x}(h) - V \tag{37}$$

Wwhere, $F_4 = \dfrac{(h-a)^3}{12\eta_2} + \dfrac{a^3 + 3a^2(h-a) + 3a(h-a)^2}{12\eta_1} + \dfrac{h^2}{2\beta}$ and taking $\beta = \dfrac{\eta_1}{\lambda}$ as the slip parameter.

3.0 SQUEEZE FILM LUBRICATION OF HYDROSTATIC STEP-SEAL

Consider a Non-conducting Hydrostatic step seal as shown in the Figure 2, fluid is flowing through the stepped clearance due to high pressure P_i at k = 0. Since the seal width L is small in comparison to the outside radius of the seal, the new pressure distribution (Shukla and Isa, 1974) is like in equation 38.

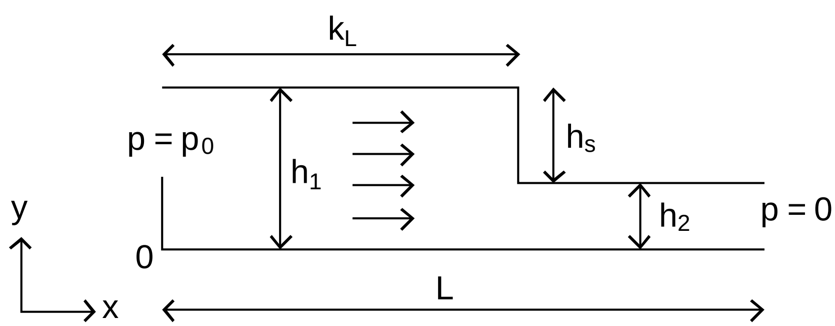

Figure 2: Squeeze film in Hydrostatic Step-Seal

$$\frac{dP_j}{dx} = \frac{-6\mu q}{bg_{4j}} \tag{38}$$

Where j = 1 is the region $0 < x < \kappa L$, and j = 2 is the region $\kappa L < x < L$ and Q is the flow flux which is a constant. On integrating the equation 38 using the following boundary conditions:

$P = P_i$	at $x = 0$
$P_1 = P_2$	at $x = \kappa L$
$P_2 = 0$	at $x = L$

We get the following expressions for the pressure in the regions $0 < x < \kappa L$ and $\kappa L < x < L$, $\kappa < 1$.

$$P_1 = \frac{P_i}{f}\left[f - \frac{1}{\overline{\overline{g}}_{41}} \frac{1}{h_1^3} \frac{x}{L} \right] \quad 0 < x < \kappa L \tag{39}$$

$$P_2 = \frac{P_i}{f} - \frac{1}{\overline{\overline{g}}_{42}} \frac{1}{h_2^3}\left[1 - \frac{x}{L} \right] \quad \kappa L < x < L \tag{40}$$

where:

$$f = \left[\frac{\kappa}{g_{41}} + \frac{(1-\kappa)}{g_{42}} \right] \tag{41}$$

$$g_{42} = h_2^3\left[(1-\bar{a})^3\left[\frac{k-1}{k} \right] + \frac{1}{k} + \frac{6}{\overline{\overline{\beta}}} \right] = h_2^3\left[\overline{g_{42}}\right] \tag{42}$$

$$\bar{a} = a / h_2 \; ; \overline{\beta} = \left[\frac{\beta h_2}{\mu} \right]$$

$H = (h_1/h_2)$ and $h_1 = h_2 + h_s$

The load capacity is given by

$$W_s = b\int_0^{\kappa L} P_1 \, dx + b\int_0^{L} P_2 \, dx \tag{43}$$

Which on using equations 39 and 40 gives

$$\overline{W_s} = \frac{W_s}{bP_oL} = \frac{\kappa^2 \overline{g_{42}} + (1-\kappa^2)H^3 \overline{g_{41}}}{2\left[\overline{\overline{\kappa g_{42} + (1-\kappa)H^3 g_{41}}}\right]} > 0 \tag{44}$$

From equation 44,

$$\frac{\partial \overline{W_s}}{\partial H^3} = \frac{\kappa(1-\kappa)\overline{g_{41}}\,\overline{g_{42}}}{2\left[\kappa\overline{g_{42}} + (1-\kappa)H^3\overline{g_{41}}\right]} > 0 \tag{45}$$

Again from the equation 43, we obtain

$$\frac{\partial \overline{W_s}}{\partial \kappa^3} = \frac{\left[\overline{g_{42}} - H^3\overline{g_{41}}\right]\left[\kappa^2\overline{g_{42}} - (1-\kappa)^2 H^3\overline{g_{41}}\right]}{2\left[\kappa\overline{g_{42}} + (1-\kappa)H^3\overline{g_{41}}\right]^2} \tag{46}$$

4.0 RESULTS AND DISCUSSION

From the equation 46, since H > 1, H^3 increases as H increases. Thus, we observe that $\overline{W}$ increases for fixed κ, which is seen from Figure 3 to Figure 6.

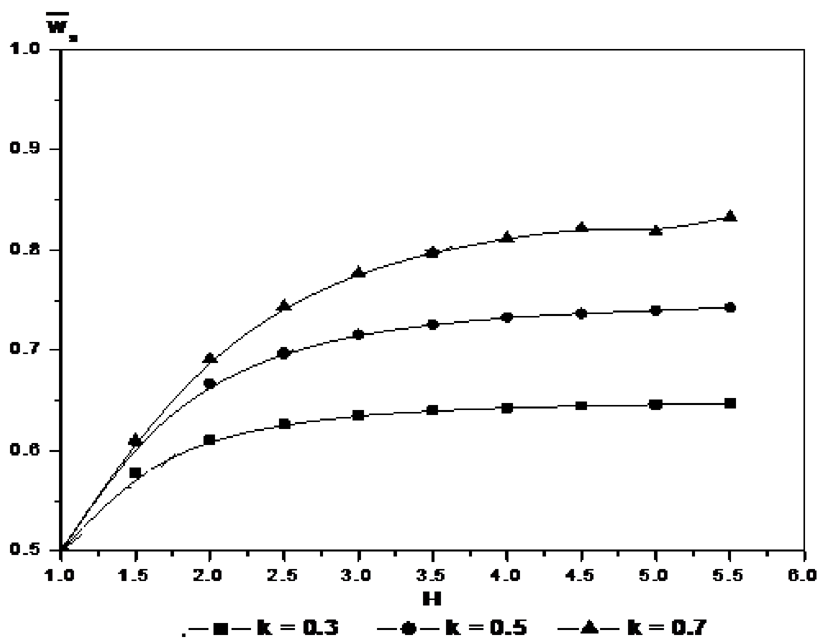

Figure 3: Variation of $\overline{W}s$ with H for various k with $\bar{a}$= 0.1

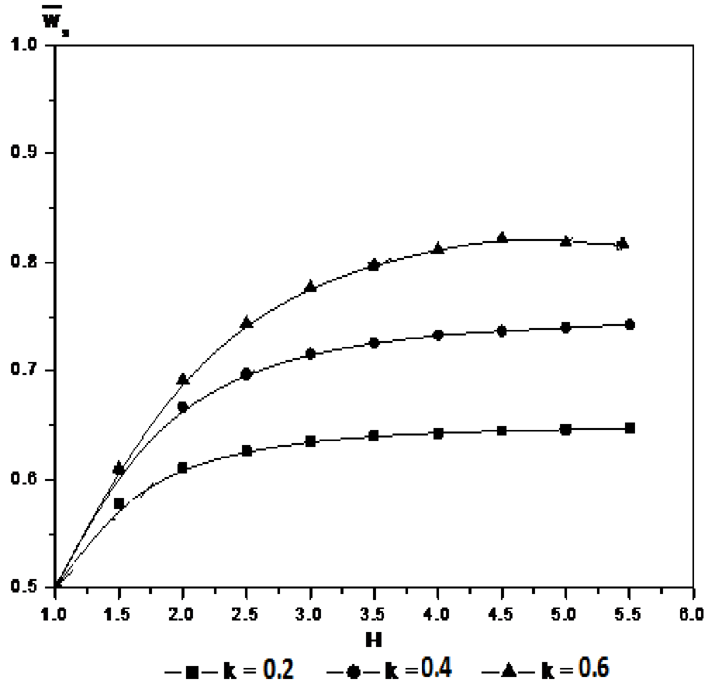

Figure 4: Variation of $\overline{W}s$ with H for various k with $\bar{a}$= 0.1

From equation 46, it is seen that $\frac{d\overline{Ws}}{d\kappa}$ depends on various parameters. Hence $\overline{W}s$ can increases (or) decreases w.r.t κ depending on the various other parameters. The result is also seen from Figure 3 and Figure 5.

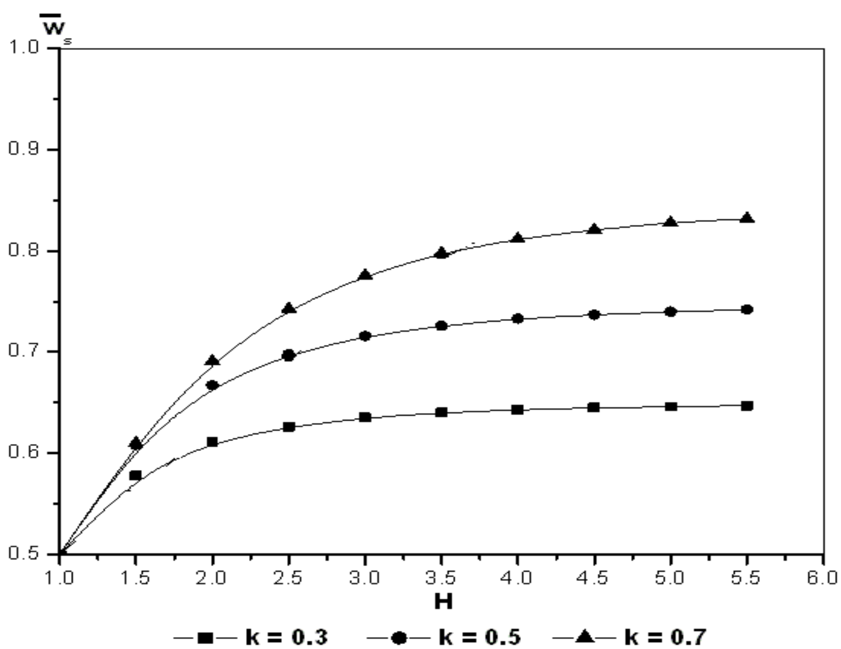

Figure 5: Variation of $\overline{W}s$ with H for various k with $\bar{a}$= 0.01

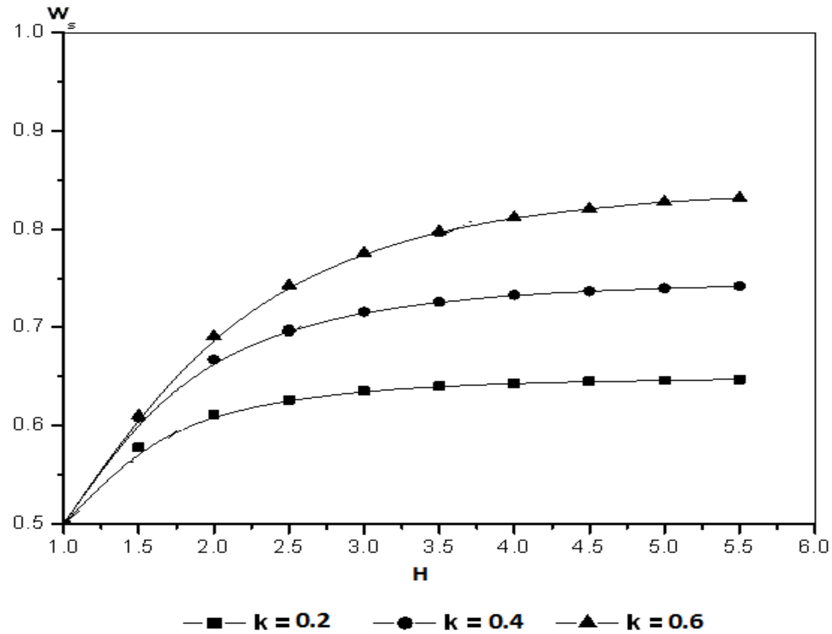

Figure 6: Variation of $\overline{W}s$ with H for various k with $\bar{a}$= 0.01

It is seen from the Figure 7, that the $\overline{W}s$ increases (or) decreases $\bar{a}$ according as $k>1$ (or) $k<1$.There is no variation of $\overline{W}s$ w.r.t $\bar{a}$ when $k = 1$ as seen from the Figure 7. It is similar to the earlier results obtained.

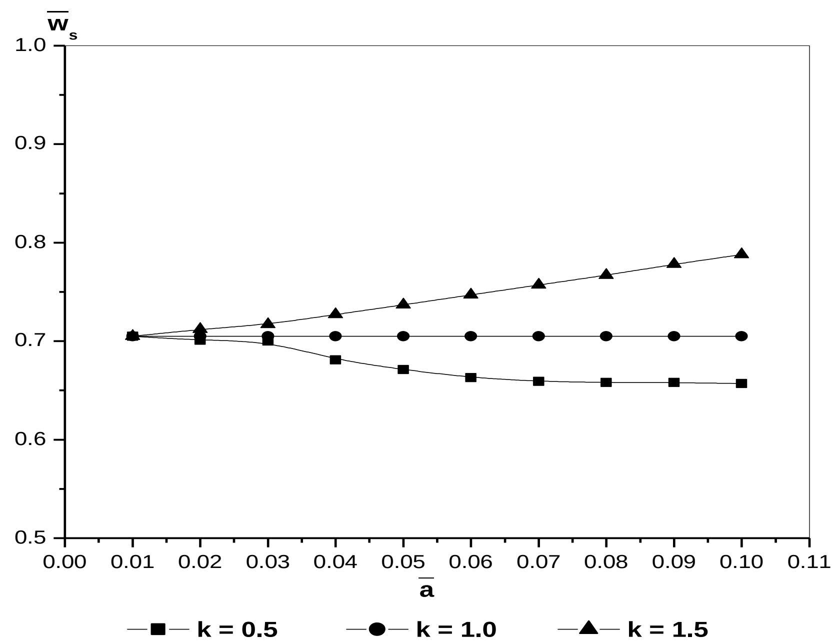

Figure 6: Variation of $\overline{W}s$ with H for various k with H=4.5

CONCLUSION

A generalized form of Reynolds equation applicable to fluid film lubrication was derived considering the variation of fluid properties, both across and along the film thickness, with velocity-slip at the bearing surfaces. The effects of velocity-slip and viscosity variation in squeeze film lubrication of hydrostatic step-seal have been studied. The beneficial result for hydrodynamic lubrication due to the presence of increased viscosity near the bearing surface was indicated.

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