EFFECT OF SURFACE ROUGHNESS ON STEADY PERFORMANCE OF HYDROSTATIC THRUST BEARINGS : RABINOWITSCH FLUIDS

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ABSTRACT

The present theoretical study is concerned with the analysis of surface roughness effects on the steady state performance of stepped circular hydrostatic thrust bearings lubricated with non-Newtonian Rabinowitsch fluids. To take the effects of surface roughness into account, Christensen theory of rough surface has been adopted. Expression for pressure gradient has been derived by means of energy integral approach. Results for film pressure, load carrying capacity and lubricant flow rate has been plotted and analysed on the basis of numerical results. Significant variations in the theoretical results of these properties have been observed.

KEYWORDS

Hydrostatic lubrication; pressurized bearings; thrust bearings; surface roughness, cubic stress fluids

INTRODUCTION

Amongst the externally pressurized bearings, circular hydrostatic thrust bearings are of great importance for industrial and engineering applications due to its wide uses in high speed rotating machines. Researchers have also paid significant attention towards improvement of performance of these bearings under various operating and lubricating conditions [1–5]. Several investigations have been carried out to analyse the dependence of different performance properties of these bearings on rotational inertia, fluid compressibility and temperature variation of the fluid [6–8]. Optimization of design of hydrostatic thrust bearings with special consideration on shape and radii of recess and supply hole, and different operating conditions have been done time-to-time [9–14] . Yadav and Kapur [15] considered the combined effect of temperature and inertia on the overall performance of hydrostatic step thrust bearings. Some investigators also focussed their attention to analyse the dynamic performance of hydrostatic thrust bearings lubricated with Newtonian and couple stress fluids [16,17].

On the same time, many researchers also emphasized that the surface roughness plays very important role on the performance of lubricated bearings [18]. Prakash and Tiwari [19] presented the analysis of porous bearing with surface corrugations. Singh, Gupta and Kapur [20] studied the characteristics of corrugated thrust bearings. Lin [21] analysed surface roughness effect on the dynamic stiffness and damping characteristics of compensated hydrostatic thrust bearings. Yacout, Ismaeel and Kassa [22] studied the combined effects of the centripetal inertia and the surface roughness on the hydrostatic thrust spherical bearings performance. Xuebing et al. [23] investigated the combined effect of the surface soughness and inertia on the performance of high speed hydrostatic thrust bearing. Walicka et al. [24] analysed various performance characteristics of thrust bearing with rough surfaces lubricated with non-Newtonian fluids by using Ellis fluid model.

Tribologists also worked to increase the stabilizing properties of non-Newtonian lubricants by addition of long chain polymer solutions (polyisobutylene and ethylene propylene etc.) as viscosity index improvers [25–27]. The use of additives reduces the sensitivity of the lubricant to the change in the shearing strain rate with temperature due which the lubricant's characteristics deviate from that of Newtonian fluids [27]. To study the lubrication regimes with such lubricants, many non-Newtonian models such as couple stress, power law, micropolar and Casson models are of common interest. Amongst these models, cubic stress (Rabinowitsch) fluid model [1–3] is an established model to analyze the non-Newtonian behavior of the fluid. Wada and Hayashi [26] showed that the lubricants blended with viscosity index improver behave like pseudoplastic fluids, which can be analyzed with cubic stress (Rabinowitsch) fluid model given by following relation for one-dimensional fluid flow:

$$\bar{\tau}_{rz} + \kappa\bar{\tau}_{rz}^{3} = \bar{\mu}\frac{\partial\bar{u}}{\partial\bar{z}} \tag{1}$$

where $\bar{\mu}$ is the *initial viscosity* and κ is the non-linear factor responsible for the non-Newtonian effects of the fluid, which will be referred to as coefficient of pseudoplasticity in this paper. This model is applicable to Newtonian lubricants $(\kappa = 0)$, dilatant lubricants $(\kappa < 0)$ and pseudoplastic lubricants $(\kappa > 0)$. This model is one of the few models supported by experimental verification [26]. Many researchers used this model for theoretical study of bearing performance with non-Newtonian lubricants [28–31]. Singh, Gupta and Kapur [32] analysed the effects of inertia in the steady state pressurised flow of a non-Newtonian fluid between two curvilinear surfaces of revolution. Lin [33] studied the squeeze film characteristics between parallel annular disks. Singh, Gupta and Kapur [34–36] investigated the effects of non-Newtonian pseudoplastic lubricants on the various performances of squeeze film, hydrodynamic sliders bearings. Sharma and Yadav [37] presented the analysis of hydrostatic circular thrust pad bearing. In light of this discussion, it is observed that surface roughness and non-Newtonian lubricants are important considerations for analysis of hydrostatic thrust bearings, and none of the researchers has analysed the combined effects of surface roughness, inertia and non-Newtonian lubricants on hydrostatic thrust bearings using cubic stress fluid model: one of the experimentally verified fluid model.

In the present investigation, it is proposed to uncover the combined effects of surface roughness, non-Newtonian fluids and fluid inertia on the performance of a hydrostatic thrust bearing using cubic stress fluid model.

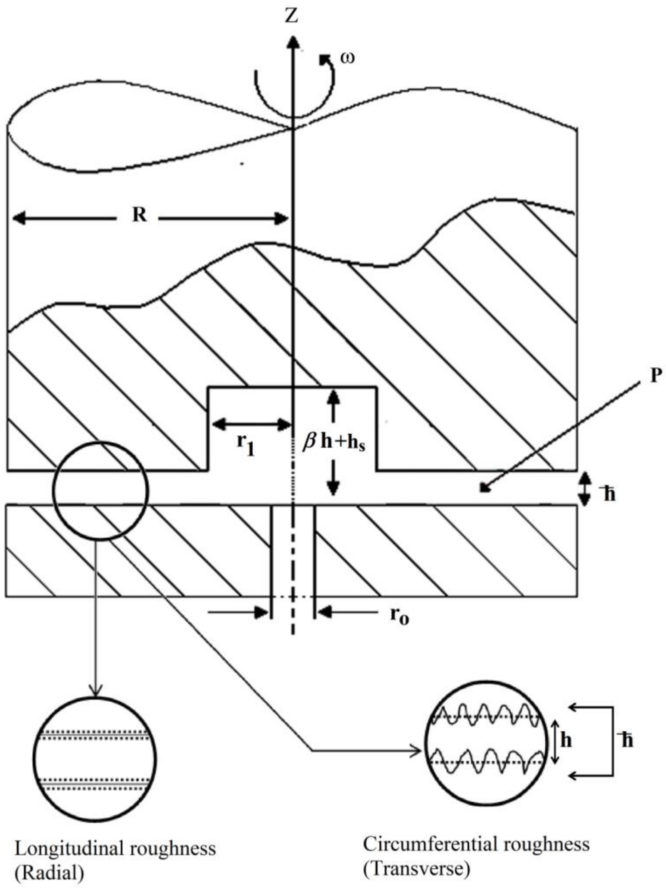

Fig. 1 : Schematic Diagram of Step Thrust Bearing with rough surfaces

ANALYSIS

A schematic configuration of a hydrostatic thrust bearing with rough surfaces is shown in Figure 1. It is assumed that the lubricant in the system is incompressible non-Newtonian fluid, the body forces and body couples are absent, and the assumptions of hydrodynamic lubrication are applicable to lubricant film. Following Singh, Gupta and Kapur [1], the expression for pressure gradient is :

$$\frac{\partial p}{\partial r} = -\frac{1}{6\mu Q}\left[r\left(\hbar^3 f^2 + \frac{3\alpha}{20}\hbar^5 f^4 \right) + \frac{3\Omega}{10} r^2 \left(\hbar^3 f + \alpha f^3 \frac{13\hbar^5}{84} \right) \right] \tag{2}$$

where, $f = -\frac{6\mu Q}{r\hbar^3}\left(1 - \frac{27}{5}\frac{\alpha\mu^2 Q^2}{r^2\hbar^4}\right)$ and $\hbar = \beta h + h_s$. Here, h is the nominal smooth part of the film thickness, β is step ratio, h_s is the part of film thickness due to asperities on the surface measured from smooth level. In case of radial roughness, $h_s = h_s(\theta, \xi)$, and in case of transverse or circumferential roughness, $h_s = h_s(r, \xi)$; where, ξ is a random variable to characterize some definite arrangement of the surface asperities. The boundary conditions for eq.(2) are : $p(r_o)=1$ and $p(1)=0$ together with the continuity of pressure at the step.

Taking the dimensionless probability density function of the stochastic film thickness $f(h_s)$, Christensen [18]:

$$f(h_s)=\begin{cases}\dfrac{35}{32c^7}\left(c^2-h_s{}^2\right)^3 & ; \quad -c<h_s<c\\ 0 & ; \quad \text{elsewhere}\end{cases} \tag{3}$$

and defining the expectancy function $E(\Theta)$ as :

$$E(\Theta)=\int_{-\infty}^{\infty}\Theta\, f(h_s)dh_s \ , \tag{4}$$

stochastic form of eq. (2) can be written as :

$$\frac{\partial E(p)}{\partial r}=-\frac{1}{6\mu Q}E\left(\left[r\left(\hbar^3 f^2+\frac{3\alpha}{20}\hbar^5 f^4\right)+\frac{3\Omega}{10}r^2\left(\hbar^3 f+\alpha f^3\frac{13\hbar^5}{84}\right)\right]\right) \tag{5}$$

where, c is the half range assumed by of the random film thickness variable ξ with the standard deviation σ. The function $f(h_s)$- defined in eq. (3) terminates at $c=3\sigma$. In the present analysis, c will be referred to as roughness parameter.

Denoting $E(p)$ by P and integrating eq. (5) with respect to r along the radius of the bearing, following expression is obtained :

$$P(r)=1-\frac{1}{6\mu Q}\int_{r_o}^{r}E\left(\left[r\left(\hbar^3 f^2+\frac{3\alpha}{20}\hbar^5 f^4\right)+\frac{3\Omega}{10}r^2\left(\hbar^3 f+\alpha f^3\frac{13\hbar^5}{84}\right)\right]\right)dr \tag{6}$$

where, the evaluation of integral on the right hand side of eq. (6) requires the exact expressions for expectation function occurring in the integrand.

The dimensionless load carrying capacity is calculated as :

$$W=\int_0^1 rP(r)dr=r_o{}^2+2\int_0^{r_1}rP(r)\mathrm{d}r+2\int_{r_1}^{1}rP(r)\mathrm{d}r \tag{7}$$

PRESSURE DISTRIBUTION

Case I: Radial Roughness:

For the radial roughness, the stochastic film thickness is $\hbar = \beta h + h_s(\theta,\xi)$ and stochastic form of eq. (6) is

$$P(r) = 1 - \frac{1}{6\mu Q}\int_{r_o}^{r}\left[r\left(E(\hbar^3)E(f)^2 + \frac{3\alpha}{20}E(\hbar^5)E(f)^4 \right) + \frac{3\Omega}{10}r^2\left(E(\hbar^3)E(f) + \alpha\frac{13E(\hbar^5)}{84}E(f)^3 \right)\right]dr \tag{8}$$

where,

$$E(f) = -\frac{6\mu Q}{r}\left(\frac{1}{E(\hbar^3)} - \frac{27}{5}\frac{\alpha\mu^2 Q^2}{r^2 E(\hbar^7)} \right) \tag{9}$$

$$E\left(\hbar^3\right) := \frac{1}{3}c^2 h\beta + h^3\beta^3 \tag{10}$$

$$E\left(\hbar^5\right) := \frac{5}{33}c^4 h\beta + \frac{10}{9}c^2 h^3\beta^3 + h^5\beta^5 \tag{11}$$

$$E\left(\hbar^7\right) := \frac{35}{429}c^6 h\beta + \frac{35}{33}c^4 h^3\beta^3 + \frac{7}{3}c^2 h^5\beta^5 + h^7\beta^7 \tag{12}$$

Equations (8–12) represent the expression for pressure in the case of radial roughness, which give the pressure in the recess region $(r_o \le r \le r_1)$ for $\beta \ne 1$, and pressure in the land region $(r_1 \le r \le 1)$ for $\beta = 1$.

Case II: Circumferential or Azimuthal Roughness:

For the circumferential roughness, the stochastic film thickness is $\hbar = \beta h + h_s(r,\xi)$ and stochastic form of eq. (6) is

$$P(r) = 1 - \frac{1}{6\mu Q}\int_{r_o}^{r}\left[r\left(\frac{E(f)^2}{E(\hbar^{-3})} + \frac{3\alpha}{20}\frac{E(f)^4}{E(\hbar^{-5})} \right) + \frac{3\Omega}{10}r^2\left(\frac{E(f)}{E(\hbar^{-3})} + \alpha\frac{13}{84}\frac{E(f)^3}{E(\hbar^{-5})} \right)\right]dr \tag{13}$$

where,

$$E(f) = -\frac{6\mu Q}{r}\left(E(\hbar^{-3}) - \frac{27}{5}\frac{\alpha\mu^2 Q^2}{r^2}E(\hbar^{-7}) \right) \tag{14}$$

$$E\left(\hbar^{-3}\right) = \frac{35}{32c^7}\left(3\left(6c^2h^2\beta^2 - c^4 - 5h^4\beta^4\right)\log\frac{(h\beta + c)}{(h\beta - c)} + 30ch^3\beta^3 - 26c^3h\beta \right) \tag{15}$$

$$E\left(\hbar^{-5}\right)=\frac{35}{32c^{7}\left(c^{2}-h^{2}\beta^{2}\right)}\left(3\left(c^{4}-6c^{2}h^{2}\beta^{2}+5h^{4}\beta^{4}\right)\log\frac{\left(h\beta+c\right)}{\left(h\beta-c\right)}+26c^{3}h\beta-30ch^{3}\beta^{3}\right) \tag{16}$$

$$E\left(\hbar^{-7}\right)=\frac{-7}{96c^{7}\left(c^{2}-h^{2}\beta^{2}\right)^{3}}\left(15\left(c^{2}-h^{2}\beta^{2}\right)^{3}\log\left(\frac{h\beta+c}{h\beta-c}\right)-80c^{3}h^{3}\beta^{3}+66c^{5}h\beta+30ch^{5}\beta^{5}\right) \tag{17}$$

Equations, eq. (13 –17) give the pressure in the recess region $(r_o \le r \le r_1)$ for $\beta \ne 1$, and land region $(r_1 \le r \le 1)$ for $\beta = 1$.

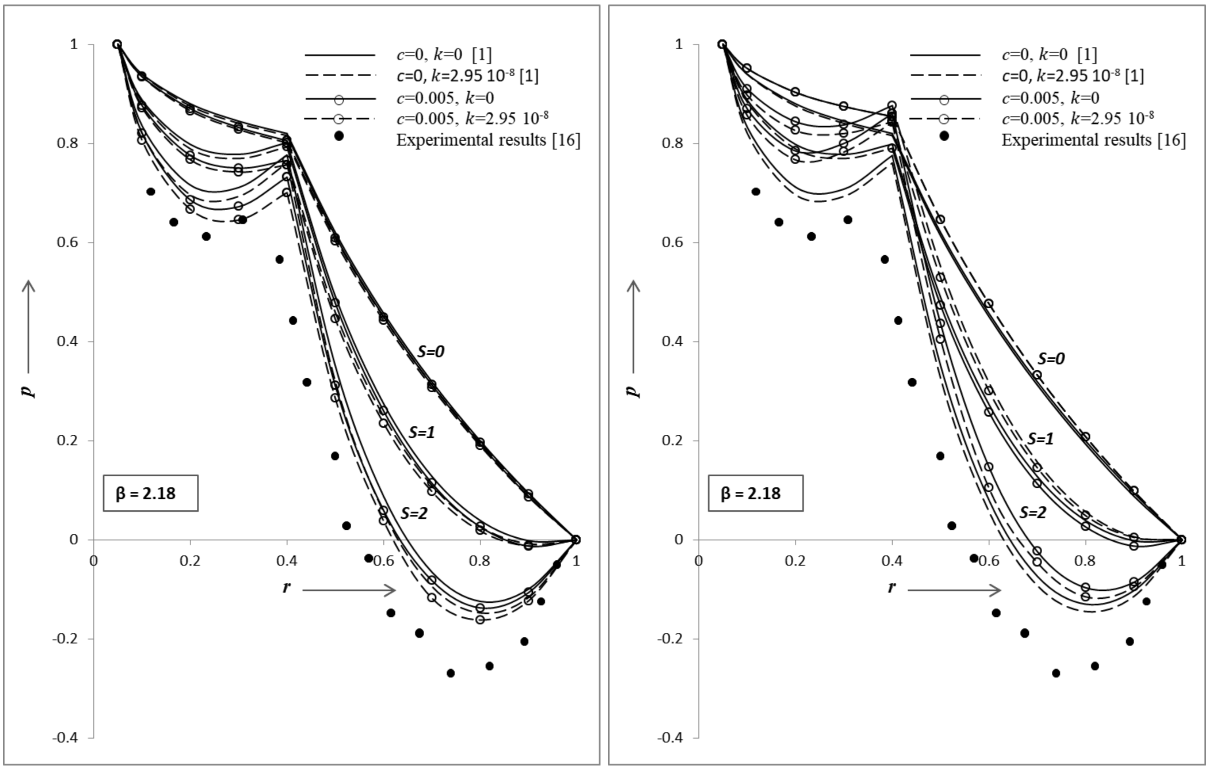

Fig. 2 **(a) :** Variation of film pressure with radius for different values of inertia parameter (S) and roughness parameter (c) [Longitudinal Roughness]

(b) : Variation of film pressure with radius for different values of inertia parameter (S) and roughness parameter (c) [Circular Roughness]

RESULTS AND DISCUSSION

In order to analyse the effect of roughness on the steady state performance characteristics of externally pressurized thrust bearings lubricated with non-Newtonian (pseudoplastic) lubricants, numerical results for pressure and load capacity have been obtained for the different values of roughness parameter c $\left(0 \le c \le 0.005\right)$ and parameter of pseudoplasticity, α $(\alpha = \kappa P_o^{\,2})$. For the practical justification, results have been discussed for the experimental values for coefficient of pseudoplasticity, $\kappa = 5.65\times10^{-6}$ m^4/N^2 (0.3 %

additive) and 3.05×10^{-6} m^4/N^2 (1 % additive) have been taken [26]. The values of operating parameters, namely, film thickness ratio β = 2, 5; parameter of centrifugal inertia S = 0, 1.0, 2.0; supply radius $r_o = 0.05$, and step radius $r_1 = 0.4$ have been taken from experimental results of Dowson [6]. Further, the theoretical results of pressure for Newtonian and non-Newtonian (pseudoplastic) lubricants have been compared with the experimental results of Dowson [6], and the results for both load capacity and pressure have been compared with the established theoretical results of Singh, Gupta and Kapur [1]. It was observed that pressure and load capacity for smooth surfaces ($c = 0$) in the present analysis are same as those obtained by Singh, Gupta and Kapur [1].

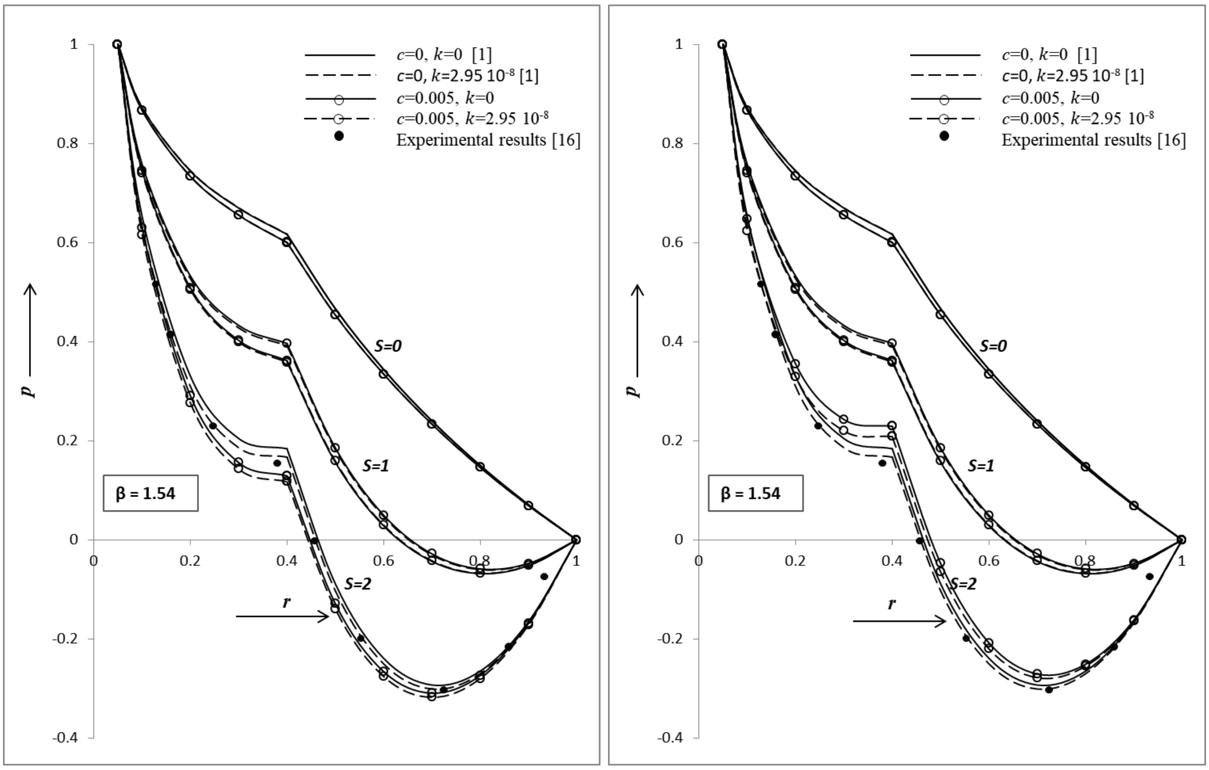

Fig. 3 **(a) :** Variation of film pressure with radius for different values of inertia parameter (*S*) and roughness parameter (*c*) [Longitudinal Roughness]

(b) : Variation of film pressure with radius for different values of inertia parameter (*S*) and roughness parameter (*c*) [Circular Roughness]

Figure 2 shows the variation of film pressure along the radial direction for step ratio $\beta = 2.18$ and different values of roughness parameter (*c*), coefficient of pseudoplasticity (κ) and inertia parameter (s). To maintain the graphical clarity, the results for longitudinal and circular roughness have been presented as separate figures. In both the case of longitudinal and circular roughness patterns, the dimensionless pressure for $c = 0$ is same as obtained by Singh, Gupta and Kapur [1] for each value of s, which validates the present results for smooth surfaces. It is, further, observed that the longitudinal roughness decreases the dimensionless values of film pressure, while the circular roughness pattern increases the dimensionless pressure. In case of the longitudinal roughness patterns $(c = 0.005)$, the pressure is less than that for $c = 0$ for Newtonian as well as non-Newtonian lubricants, and the trend holds for each value of $s = 0,1$ and 2, while the case is reversed in case of circular roughness patterns. Further, in comparison to smooth surface, the theoretical values of pressure obtained for rough surfaces with longitudinal roughness are closer to the experimental values. The pressure for longitudinal roughness ($c = 0.005$) and non-Newtonian lubricants $\kappa = 5.65\times10^{-6}$ are more close to the experimental results [6] than the theoretical results [1]. In figure 3, similar variation of profile of dimensionless pressure has been obtained and compared with established theoretical and experimental results for step ratio $\beta = 1.54$. This establishes the sustainability of present results of film pressure for surface roughness. Furthermore, all the figures for pressure also shows that the effects of surface roughness and non-Newtonian lubricants are more significant at higher values of inertia parameter s.

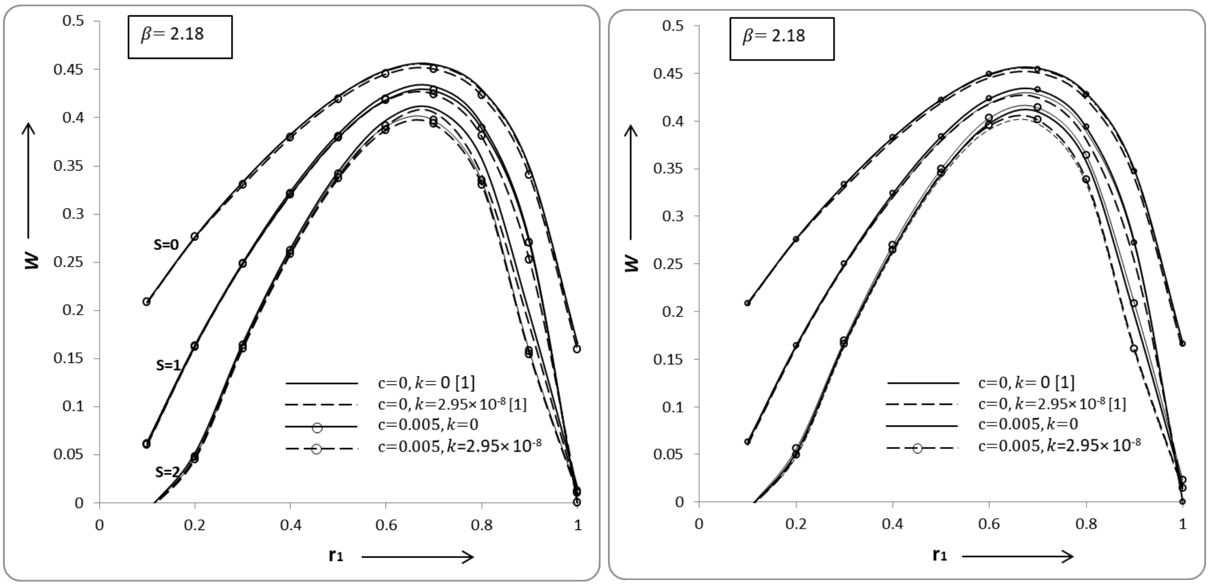

Fig. 4 **(a) :** Variation of load capacity with step parameter (r_1) for different values of roughness parameter (c), inertia parameter s and coefficient of pseudopasticity κ [Longitudinal Roughness]

(b) : Variation of load capacity with step parameter (r_1) for different values of roughness parameter (c), inertia parameter s and coefficient of pseudopasticity κ [Circular Roughness]

Figure 4 shows variation of dimensionless load capacity with respect to the step position r_1 for different values of roughness parameter (c), coefficient of pseudoplasticity (κ) and inertia parameter (s). In case of smooth surfaces $(c = 0)$, the load capacity obtained in the present analysis is same as that obtained by Singh, Gupta and Kapur [1] for each value of $s = 0,1$ and 2. The load capacity for surface with longitudinal roughness pattern $(c = 0.005)$ is less than that for smooth surfaces in both the cases of Newtonian and non-Newtonian lubricants, and the trend of variations sustain for each value of s. In case of circular roughness patterns, the results are reversed. Furthermore, it can be clearly observed from the figures that higher is the value of inertia parameter s, more significant are the effects of surface roughness and non-Newtonian lubricants.

Figure 5 show the variation of load carrying capacity with roughness parameters (c) for inertia parameter $(s = 0,1,2)$, Newtonian lubricants $((\kappa = 0)$, pseudoplastic lubricants $(\kappa = 2.95\times10^{-8})$ and a typical value of step parameter $r_1 = 0.4$. Results for both the type of longitudinal and circular roughness patterns are plotted simultaneously to observe the influence of surface roughness and its pattern on the load capacity more precisely. It is easy to observe that load carrying capacity decreases with the increase of longitudinal roughness, and increases with the increase of circular roughness for Newtonian as well as pseudoplastic lubricants at each value of inertia parameter *s*. It is also clearly observed that surface roughness has more significant influence on load capacity for higher values of inertia parameter *s*.

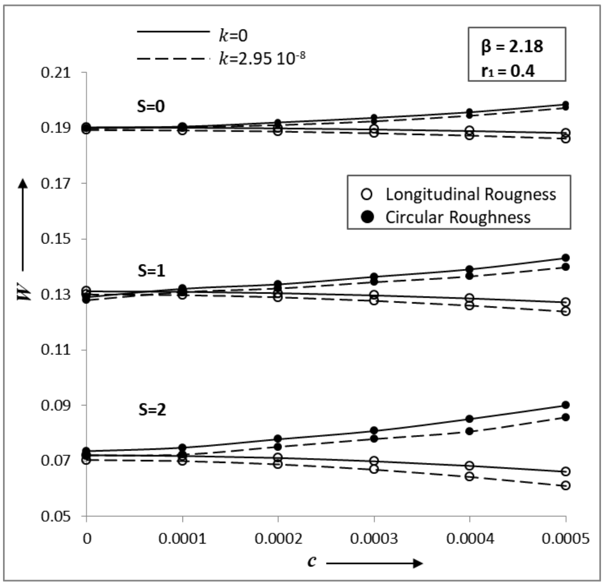

Fig. 5 : Variation of load capacity with roughness parameter (c) for different values of inertia parameter s and coefficient of pseudopasticity κ [○ – Longitudinal Roughness, ● – Circular Roughness]

CONCLUSIONS

Combined effects of surface roughness, non-Newtonian pseudoplastic lubricants and lubricant inertia on the steady performance of hydrostatic thrust bearings, neglecting the radial inertia of lubricant and cavitation effects, have been presented. Cubic stress-strain model for non-Newtonian nature of the fluid, Christensen theory for surface roughness and *energy integral method* to derive pressure gradient were used in the analysis, based on which, following conclusions are drawn –

1. In comparison with smooth surfaces, dimensionless film pressure and load capacity is lower for longitudinal roughness and higher for circular roughness patterns with Newtonian as well as non-Newtonian lubricants.
2. In case of longitudinal roughness patterns, load carrying capacity decreases with the increase roughness; and in the case of circular roughness, load capacity increases with the increase roughness.

3. In comparison with Newtonian lubricants, dimensionless pressure and load is lower for pseudoplastic lubricants, which is also the established result [1].
4. Effects of surface roughness and non-Newtonian lubricants become more significant for larger values of inertia parameter, that is, for larger bearing radius and higher operating speed.

Hence, the present analysis is expected to be helpful for better bearing designs.

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