Effects of temperature on fluid lubricant consistency under adiabatic cylindrical rolling boundaries

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Abstract: **In this paper, the rigid cylindrical rollers, with rolling and normal squeezing motions, lubricated by a power law fluid are studied under the adiabatic boundary conditions. The consistency of the lubricant assumed to vary exponentially with pressure and mean temperature. By solving the modified Reynolds and Heat equations simultaneously to get pressures and temperatures. Different characteristics of roller bearings are analyzed and discussed. A non- uniform grid was employed to achieve more accurate predictions of pressure and temperature, especially in the high pressure region. The detailed analyze of the theoretical results obtained herein seems to suggest that the temperature dependence of the lubricant viscosity causes a reduction in both the load carrying capacity and surface traction of the system , whereas the normal squeezing motion leads to a substantial increase in pressure and also displaces the pressure peak towards the centre line of contact. Also results are compared with previous findings.**

Keywords: Cylindrical Roller, non-Newtonian, lubrication, squeezing velocity, pressure, mean temperature, lubricant consistency, load, traction.

I. INTRODUCTION

Machines have made our life a sophisticated one. Modernism has absolutely changed our life style. Along with machines, man too work twenty-four hours on shift basis. Like our physique, the machine too emit heat at work. Rest is the remedy to man but machines demand lubricant as remedy. Lubrication maximizes the life of the roller bearing. Dowson[1] was pioneer to propose a bond between relative film thickness and the capacity of the contacting surface to tolerate pitting. The efforts to propose Sigmoid curve by Lin et al.[2] was experimentally demonstrated by Skurka[3] and Danner[4].

In the beginning, grease was used as lubricant without oil circulation system and filtering. Wilson[5] proposed experiments for roller bearing via electrical capacitance measurements. He calculated film thickness to radially loaded double row spherical roller and single-row cylindrical roller. Later, “Lubcheck” device for surveying lubrication was delivered by Heemskerk et al[6]. This method was also relied upon electrical capacitance measurement. The probability of asperity contacts was calculated by using grease or oil in radially loaded deep- groove bell bearing. Later, the same device was used by Leenders and Houpert[7] and Wikstrom and

Jackobson for spherically roller bearings. Wardle et al.[8] and Jacobson used the same Lubcheck apparatus to experiment with refrigerant-lubricant mixture and Masen et al.[9] used it intwo-disc test rig to study surface micro-geometry on film formation.

Franke and Poll [10] exploited the capacitance technique to assess the speed, temperature and friction torque of the lubrication condition. This test was conducted using angular contact ball bearings with ten test grease lubrication. The lubrication in ball-on-flat and the lubrication in roller bearing is not similar. In roller bearing the lubricant flows fully inside the bearing. The level of starvation, the contact area geometry, and the dynamics of the rollers and cage, all has reasonable effect upon the film thickness. This is not possible in ball and disc model (Lugt [11]). Murer et al. [12] studied the load distribution in roller bearing using electrical capacitance. Schnabel and company [13] delivered that in mixed regime the character of contact capacitance is not clear. And it has to be augmented to know the additives play in impedance measurement. Recently, Jablonka et al.[14] used chromium-coated glass disc to assess the lubricant film thickness via optical interferometry and electrical capacitance. He experimented with 7 steel balls. Six ceramic balls(silicon nitrate) replaced the six steel balls. Because of the non-conductive character of silicon nitrate, the obtained figure responds to the film thickness between the steel balls and the rings.

A known fact is that lubricant failure is the primary cause of the failure of the bearing. Further, it is hard to calculate the life and properties of the fresh grease. The ageing can be classified both in terms of mechanical and chemical. It is quite natural to witness physical and chemical changes at high temperature operation due to mechanical and thermal stress. The way these changes affect the film is not clear. Still it is said that bearings fail at the cause of lubrication failure rather than surface fatigue [15]

Dowson[16] demonstrated that film thickness has a direct effect on steady-state wear rare for metal-on-metal joints tested in a hip joint simulator. When the film thickness increases the steady-state wear rate decreases in magnitude. Though the correlation is surprising, the strongest correlation would be with the lambda ratio provided the phenomenon is credited to asperity interactions. It is learnt that modern production techniques make sure that after bedding-in roughness and the surface form of several implants show same result, regardless of diameter, material design and clearance.

Cann [17] considered on the ball on disc test rig to bring relationship between starved film thickness and different temperatures. To study the above said point he used lithium grease. It was realized that film thickness decreased faster at high speed. At the same time reflow was stronger at lower speed and higher temperature. Further Cann[18] discovered that starvation was more when lithium grease of high thickener concentration and base oil viscosity was used. Hurley et al.[19] from his studies suggested that thermally aged and heavily aged lithium grease gave higher film thickness. Couronne et al.[20] tested 4 different grease using a ball on disc and showed 2 out of 4 lubricant showed increase and then decrease in film thickness.

In this present paper analysis is fully focused to examine the qualitative behavior of thermal effects on non-Newtonian power law lubrication of two heavily loaded rigid cylindrical roller bearings underneath adiabatic and isothermal boundaries. The lubricant consistency variation assumed to vary along with the pressure and the mean film temperature. The rolling ratio are used to study the rolling/ sliding effects of surfaces on the pressure, the temperature and the lubricant consistency along with load and traction. however, the effects of compressibility and surface roughness are neglected.

II. MATHEMATICAL ANALYSIS

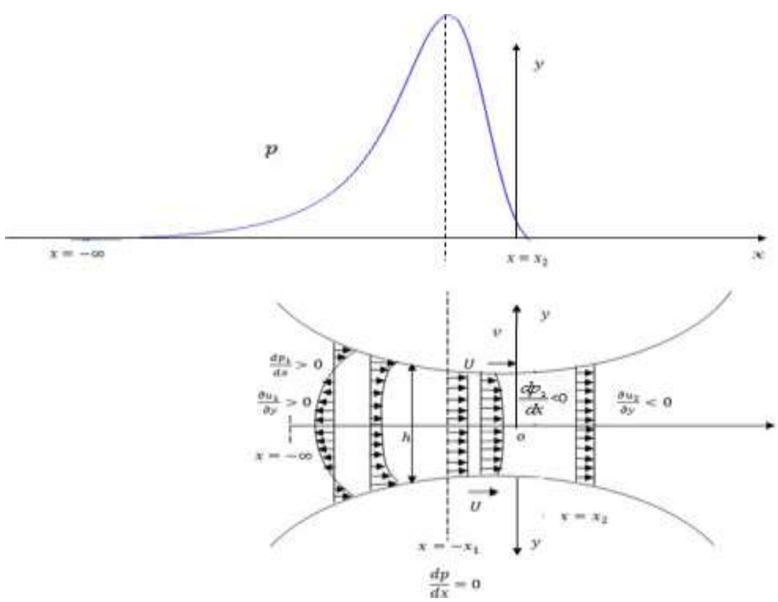


Fig 1: Lubrication of cylindrical rollers

The governing equations for the one dimensional fluid flow are [21]

dp □ □□ . (1)

dx □x □y

□

y

□u □

□v

$$= 0. \qquad (2)$$

Where the shear stress relation for this case is

$$\tau = m\left|\frac{\partial u}{\partial y}\right|^{n-1}\frac{\partial u}{\partial y}. \qquad (3)$$

The consistency m of the above power- law is taken as:

$$m = m_0 e^{\square p \square \square \frac{T_0}{T} \square m \square}. \qquad (4)$$

Where the mean temperature T_m is defined as

$$T_m = \frac{2}{h}\int_0^{h/2} T\,dy. \qquad (5)$$

u and v are the velocity components in x and y directions and p and T are the hydrodynamics pressure and the temperature respectively.

The boundary conditions for the equations (1) and (2) are

When $y = 0$, $\frac{\partial u}{\partial y} = 0$, at $y = \frac{h}{2}$, $u = U$, $v = \frac{U}{2}\frac{dh}{dx}$ and $v = 0$

Integrating equation (1) using the boundary conditions mentioned above, one may get f u □1□ s .

h (2n □1) n

x2

□

1 □ 1 1 □1 □

n □

s □ 2 □ □ □ □ □ .

□y□

Where 2□ n □ 2n □ □ h □ □

1 □ □ 1□ n □ 2 □ □

□ n □ 1 □

□

□ □ □ □

□ □

Integrating the above continuity equation (2) using the conditions (6) and $\frac{dp_1}{dx}$ □ 0 at x □□ and h □ h, one can get in dimensionless scheme

dx 1 1

d p1 / d x = m0 E / (2n□1) . f / h n □ □ □ x □ □x1 (7)

/ $\sqrt{2Rh_0}$ d p 2 m0 E n / 2n□1 = − d h / x g .

$\bar{\ } x_1 \ \bar{x} \ \bar{\ }$ (8)

x_2

$\bar{x}$ x ; $\bar{y}$ y / h ; $\bar{h}$ h h ; $\bar{p}$ p ; $\bar{g}$ $\bar{f}$; E $e^{p \ T_0 /}$ T_m ; $\bar{m}$ 2mc .

0 0 n^2 2 □ 2□2n□1□□^{n}

2R □ U □^{n}

—

f □ x – x □2q □x□ x_1 c_n □ □ □ □ T □ □T etc.

1 □; □ ; ;

□ n □ h_0 □ $\overline{h_0}$ □

The energy equation for the one dimensional flow case may be assumed to be

$$k \frac{\partial^2 T}{\partial y^2} \square \square \square \square \frac{\partial u}{\partial y} \square \square c u_m \frac{dT}{dx}. \quad (9)$$

Where k is the heat conduction of the fluid and is assumed to be constant.

Assuming, at y □ $\frac{h}{2}$, $\frac{\partial T}{\partial y}$ □ 0; and at □0 y, T □ T_{01}.

This above equation (10) is solved under boundary conditions mentioned above and is obtained with dimensionless scheme

dT □ $l f$ □ n□1 $m\ E\ f$ γ 2

T □ T □ m_1 □ hy □□ □ □ □ $_0$ t . □ □ □ x □ (10)

6□ □ □ □x

y

1 $2n\square1$ □ $2n^2$ □$3n$□1 1

0 dx

1 □

□ h n h n

dT □ □ n□1 γ

m $E g$

$l g$

2

□

T □ T □ m2 □ hy□□□ □ □ 0 t . $- x$ □ x □ x (11)

6□

y

2 01 $2n\square1$ □ $2n^2$ □$3n$□1 1 2

dx

□

□ h n □ h n

1 -1 1 □ 2 2 □ 1- □

1

□

Wher 2 □ □ $2n$ □1 □ □□ □ h □ n □1 □ h □ n □

e n^2 □ $n y$

3 □

l 2 n y n y

n 1 2 2 2n 1 2

3n 1 2n

1

1 -1

2 1 $\frac{\beta U h_0}{k\alpha}\sqrt{\frac{h_0}{2R}}$

1 n n h n y ;

3

t 2 n y ; ;

y y

2 h_0

3n 1

Using the dimensionless scheme, the mean temperature T_m, as given in (5), can be calculated as

1

$d\bar{T}$ b m E f^{n} 1 -2

m_1 □ □ $T m$ □ n 0 / □ □ $f h$ □ □□ .

□ T □ □ □ h

a

$d x$ □ 01 2n □ □ □ □ □ □ x □ (12)

1 n □x_1

h □ □

$d\bar{T}$ □ b $m E g^{n □1}$ □ □ -2

m_2 □ □ $T m$ □ n 0 / □ □□ gh □ □ □ .

□ T □ □ □a h

$d x$ □ 01 2n □ □ □ □ x_1 □ x □ (13)

2 n x_2

h □ □

= □ 18n^3 □ 31n^2 □ 10n^2 □ 7n □c Uh_0^2 ;

□15n □ 2 □ □1 □

Where □ 2□n □1□□3n □1□□4n □1□ □ b_n □ 8□3n □1□□4n

a_n □1□ □ □ □ . .

□ □ □ □

$= | \square | ;$ $k\sqrt{2Rh_0}$ 12

The normal load is given by

$$w_y = \int_{x_1}^{x_2} p \, dx \qquad (14)$$

The dimensionless load $\bar{w}_y = \frac{W}{\sqrt{2Rh_0}}$ is given by

$$\bar{w}_y = -\int^{\bar{x}_2} \bar{x} \frac{d\bar{p}}{d\bar{x}} \, d\bar{x} \qquad (15)$$

The tangential load is given [22] by

$$w_x = -2\int_{h_0}^{h_2} p\,dh = -2\int^{x_2} x^2 \frac{dp}{dx}\,dx \qquad (16)$$

The dimensionless load $\bar{w}_x = \frac{W_x}{2h_0}$ is given by

$$\bar{w}_x = -\int^{\bar{x}_2} \bar{x}^2 \frac{d\bar{p}}{d\bar{x}}\,d\bar{x} \qquad (17)$$

The load W is calculated by $\bar{W} = \sqrt{\bar{w}_x^2 + \bar{w}_y^2}$

(18)

the surface traction force T_F, obtained from the integration of shear stress τ over the entire length , may be written as

$$T_{Fh} = \int^{x_2} \frac{h}{2} \frac{dp}{dx} \, dx \tag{19}$$

Then, the dimensionless traction may be written as

$$\bar{T}_{Fh} = \int^{\bar{x}_2} \frac{\bar{h}}{} \frac{d\bar{p}}{d\bar{x}} \, d\bar{x} \; ; \tag{20}$$

Finally, one can get the consistency expression in the form

$$\bar{m} = \bar{m}_0 \, \bar{E} \, . \tag{21}$$

III. RESULT AND DISCUSSION

Calculation for a semi analytical solution of the Reynolds equations (7,8) and the heat energy equations(12,13) is done using the values below:

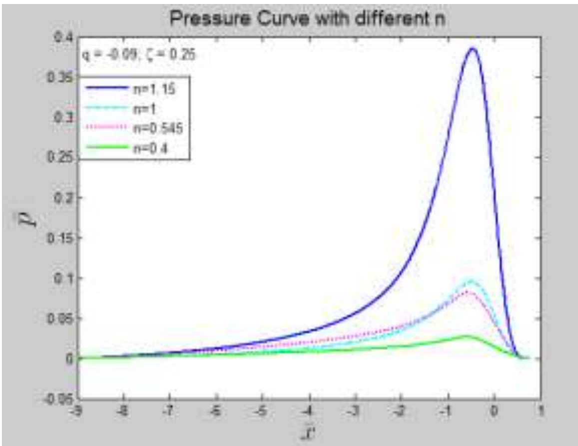
U = 4 m/s; R = 0.03 m; -0.09 < q < 0.09; 0.4 □ n □ 1.15; $\bar{T}_{01}$ = 8; $\bar{T}_{0}$ = 3; □ = 4; h_0 = 6 x 10 $^{□5}$ m;

□ = 6 x10 $^{□8}$ pa $^{□1}$ m ^{2}

3.1 Pressure Profile –

Figures 2 and 3 show the distribution of pressure $\bar{p}$ is a function of U and 'n'. In figure 2 it could be seen the pressure profile $\bar{p}$ increases mostly with rolling ratio U for different n. In figure 3 it could be seen that $\bar{p}$

increases with different with fixed n. This module was displayed by Hajishafiee et al. [23] and Tobais Hultqvist et al [24].



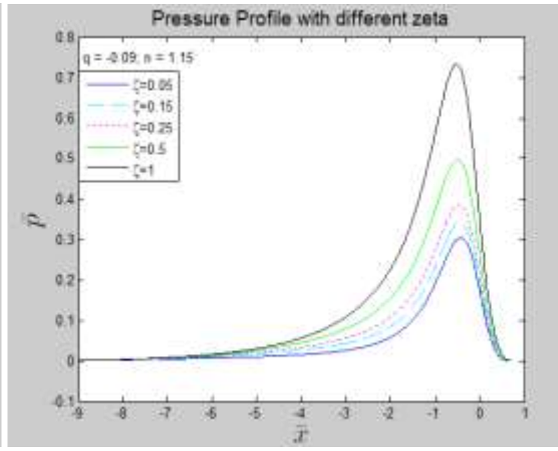


Fig 2: $\bar{p}$ against $\bar{x}$ Fig 3: $\bar{p}$ against $\bar{x}$

3.2 *Temperature Profile –*

Figures 4 and 5 show the mean temperature $\bar{T}_m$ for various values of n with fixed □ . The increase in mean temperature $\bar{T}_m$ with n shows that the temperature for dilatants fluid is higher than Newtonian and pseudo plastic

fluid. Qualitatively, the mean temperature $\bar{T}_m$ against $\bar{x}$ is similar to that of the temperature profile received by

Prasad et al. [21], the mean temperature $\bar{T}_m$ increase with q as per figure 5. From this it could be learnt that sliding

temperature is higher than pure rolling. Further, it could be marked that the mean temperature

T_m when 0 refer to the case without convection in Fig 4.

The increase in n hints an increase in effective viscosity [25]. Figures 6 and 7 show the two dimensional temperature distribution in x and y plane. □ is almost zero and at this cause the figure 6 is drawn without convection. Whereas, figure 7 has the absolute distribution of temperature with convection and conduction.

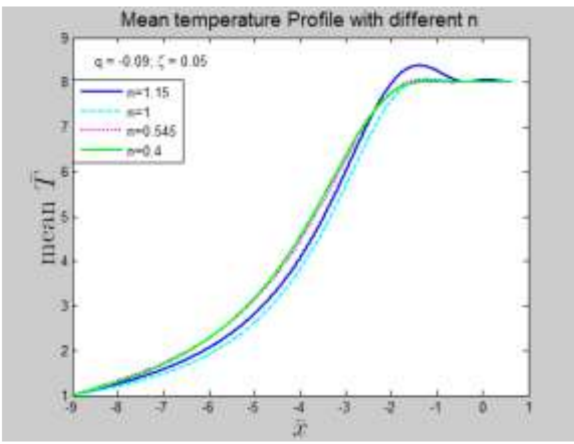


Fig 4: Mean $\bar{T}$ against $\bar{x}$

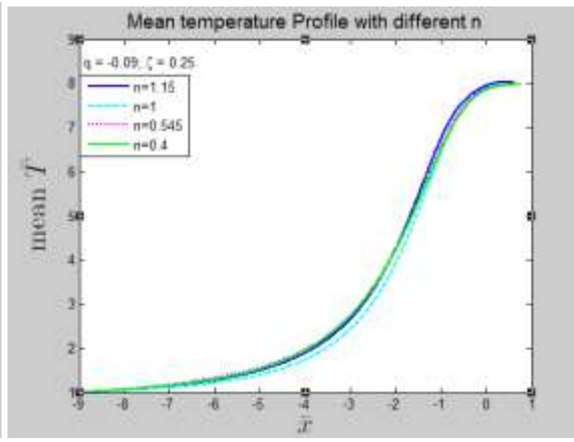


Fig 5: Mean $\bar{T}$ against $\bar{x}$

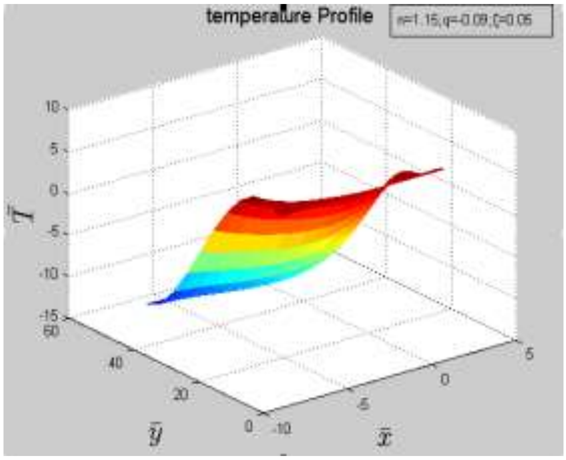


Fig 6: $\bar{T}$ against ($\bar{x}$ & $\bar{y}$)

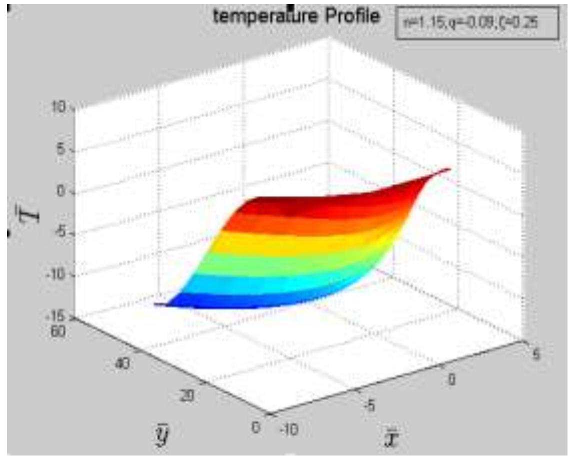


Fig 7: $\bar{T}$ against ($\bar{x}$ & $\bar{y}$)

3.3 Velocity Profile –

For constant values of q and n, the velocity distribution at various values of $\bar{x}$ between $\bar{x}$ =-9 and $\bar{x}$ =-0.6 is bestowed. Here, the velocity upsurges with $\bar{y}$, as the figure 8, shows, and this is similar to the work by Lorenzo

Fusi[26]. As shown in figure 3 the velocity remains fixed when pressure peak is $\bar{x}$ =-0.6. The complete distribution of velocity with convection and conduction is shown figure 9.

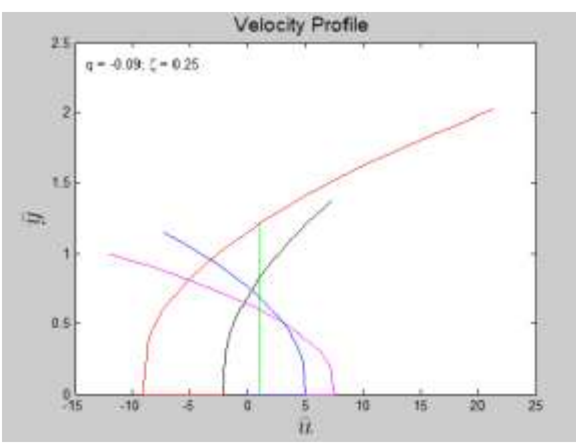


Fig 8: $\bar{y}$ against $\bar{u}$

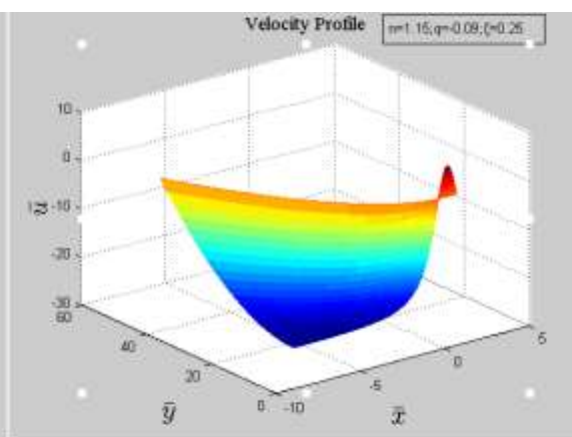


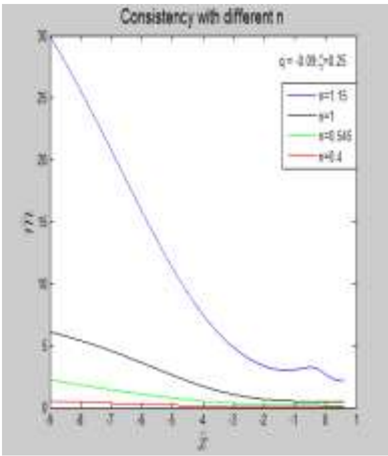
Fig 9: $\bar{u}$ against ($\bar{x}$ & $\bar{y}$)

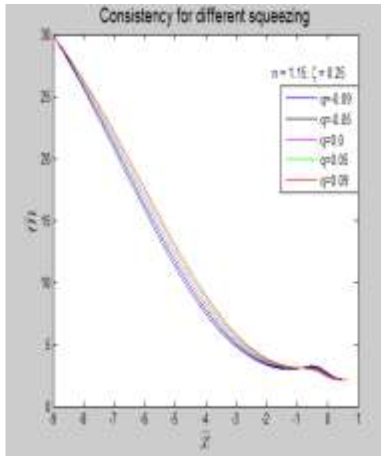
3.4 Consistency Profile –

The lubricant consistency and its change in m with p and mean temperature T_m

is the main subject of

this article and it is clearly shown in the figures below. From figure 10 to 12, it is clear that the overall consistency changes with x for different n and different q and. It specifies the supremacy of pressure over the temperature for value below 0.1 and vice versa for value 0.1 and above. So, the consistency variation with pressure and temperature is well justified [27,28].





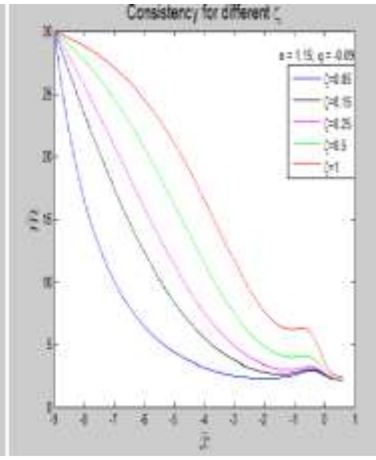


Fig 10: $\bar{m}$ against $\bar{x}$ Fig 11: $\bar{m}$ against $\bar{x}$ Fig 12: $\bar{m}$ against $\bar{x}$

3.5 Load and Traction –

Here, the table 1 represents the eccentric features of bearings namely the load $\bar{w}$ and the traction force

increase with n and this is

$\bar{T_F}$ with different n and q values. Reading the table shows that

both $\bar{w}$ and $\bar{T_F}$

correlation with the earlier findings stated in [29]. Figures 13 to 15 denote load, traction, coefficient of traction against $\bar{x}$ and figure 16 shows the traction fore against normal load W_y .

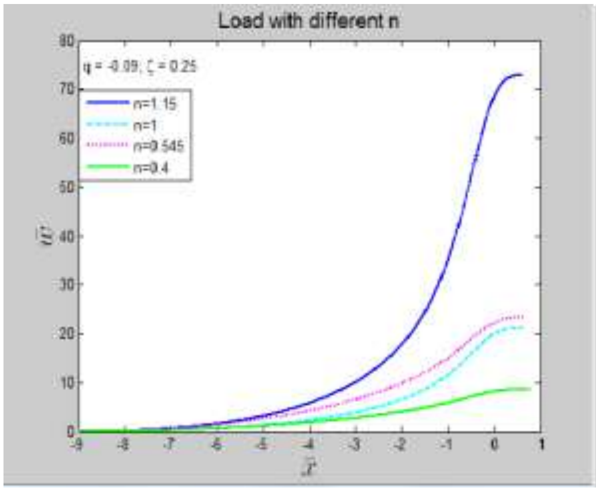


Fig 13: $\bar{w}$ against $\bar{x}$

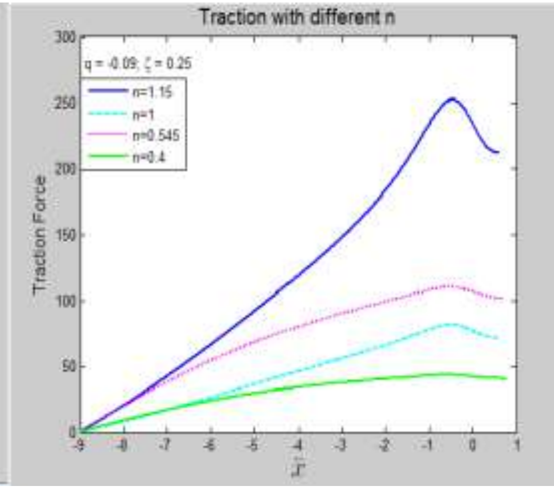


Fig 14: $\overline{T_F}$ against $\bar{x}$

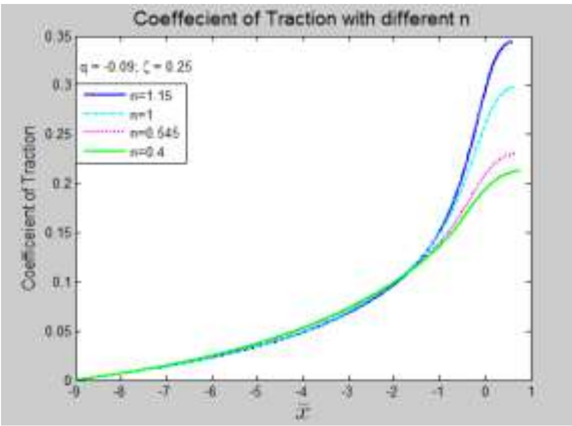


Fig 15: Co-efficient of T_F against $\bar{x}$

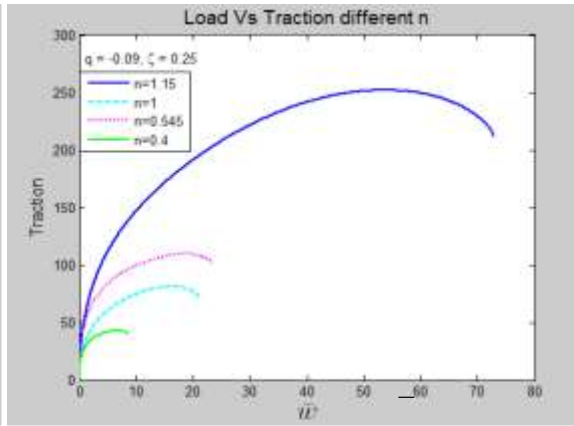


Fig 16: T_F against $\bar{w}$

Table -1 Load and Traction

n/ m_0	**q=-0.09**	**q=-0.05**	**q=0.00**	**q=0.05**	**q=0.09**
			x_1 **values**		
1.15/0.56	0.457460	0.484745	0.520866	0.558701	0.590000
1.00/0.75	0.477936	0.506334	0.543404	0.582658	0.615506
0.545/86.0	0.529518	0.562186	0.610295	0.667080	0.702344
0.40/128.0	0.596048	0.625888	0.665662	0.714949	0.749860
			x_2 **values**		
1.15/0.56	0.637460	0.584745	0.520866	0.458701	0.410000
1.00/0.75	0.657936	0.606334	0.543404	0.482658	0.435506

0.545/ 86.0	0.709 518	0.662 186	0.610295	0.567 080	0.522 344
0.40/1 28.0	0.776 048	0.725 888	0.665662	0.614 949	0.569 860
Normal load					
1.15/0. 56	0.728 287	0.697 383	0.661080	0.626 262	0.599 713
1.00/0. 75	0.211 555	0.206 124	0.199348	0.192 612	0.187 270
0.545/ 86.0	0.233 583	0.234 338	0.235542	0.237 062	0.236 323
0.40/1 28.0	0.085 741	0.086 481	0.087233	0.087 964	0.088 366
Traction					
1.15/0. 56	2.116 632	2.110 030	2.096998	2.080 945	2.066 434
1.00/0. 75	0.711 910	0.714 393	0.715818	0.715 251	0.713 533
0.545/ 86.0	1.013 203	1.032 708	1.052576	1.067 242	1.080 937
0.40/1 28.0	0.402 739	0.413 053	0.424604	0.434 258	0.441 572
Coefficient of Traction					
1.15/0. 56	2.906 316	3.025 641	3.172077	3.322 802	3.445 702
1.00/0. 75	3.365 124	3.465 844	3.590789	3.713 430	3.810 186
0.545/ 86.0	4.337 666	4.406 923	4.468749	4.501 953	4.573 987
0.40/1	4.697	4.776	4.867469	4.936	4.997

28.0	139	235		783	085

IV. CONCLUSION

The problem attempts to study the thermal effects in hydrodynamic lubrication of roller bearings using incompressible power law fluid under adiabatic boundaries. The Reynolds and the thermal energy equations which are functions of consistency $\bar{m}$ and $\bar{u}$ and the consistency index n are obtained and solved semi analytically for pressure $\bar{p}$ and the mean temperature $\overline{T_m}$. The following are the inferences:

(i) The pressure increases significantly with n and □ .

(ii) The sliding temperature is higher when compared to pure rolling.

(iii) The temperature is subjected to bring down the load carrying capacity of the system..

(iv) The load and traction rises with 'n' and q.

(v) The traction at lower surface is more because of high speed at lower surface than the upper surface.

(vi) The velocity of the lower surface is high resulting in the move of the velocity profile above the x- axis.

(vii) The mean film temperature upsurges considerably with n and □ . So it is mandatory to treat the consistency $\bar{m}$

of the power law fluid to differ with temperature and pressure.

(viii) In the inlet domain, the effect of the peclet (□)

number is more. REFERENCES

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