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Stability and Transient Analysis of Axial Groove, Multilobe and Tilting Pad Bearing By Using ANSYS

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ABSTRACT:

Flow of lubricant in bearing can generate unsteady forces on the surrounding parts of hydrodynamic journal bearing, which are unintentional but unavoidable. It is important to understand the impact of fluid forces on the surrounding equipment and its performance. In present work three bearing models are considered for analysis. Models are designed with AUTOCAD INVENTOR software. ICEM-CFD was used for structured grid formation. Simulation was carried out using ANSYS Fluent software package to predict the performance. Transient analysis was carried out to check stability of journal for 3 types of bearings.Three different cases of lubricantswereconsidered for each bearing. The performance of each bearing was observed with contours of Total Pressure.. Plots of Total Pressure were noted and results were concluded.

KEYWORDS:Hydrodynamic Journal Bearing, Inventor, ICEM, Ansys Fluent, Transient Analysis, Total Pressure, Wall Shear Stress

1. INTRODUCTION:

1.1Journal Bearing

Bearing is used to support journal, it also transfer loads from one machine element to another. Lubrication is used in the gap between the bearing inner surface and the journal wall. Lubrication provides damping and also reduces frictional losses. When the journal start its rotation, it moves upward in the bearing inner wall and as the speed is increased, wedge-shaped region is developed. The pressure is get developed inside the wedge shaped zone and hence the load is get supported [1]. Viscous shearing causes the bearing to heat up and thus causing loss of power.

1.2 Types of Bearing used for analysis [2] [3] [4]

1.2.1 Axial groove bearings:

Axial groove bearing can be simply explained as the casing with two grooves formed exactly opposite to each other in circumferential direction. May be used when the load angle changes as operational variables change. This design has slightly better stability than a plain bearing without grooves. Grooved bearings are available in axial or circumferential designs.

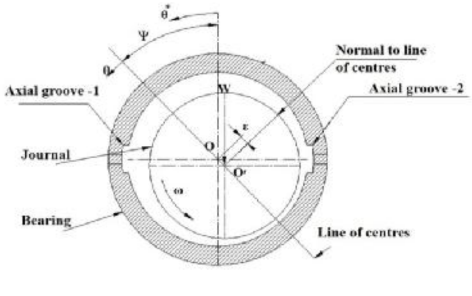

Fig. 1.1 Axial Grooved Bearing [5]

1.2.2 Three lobe bearings:

Many of the rotary applications are supported by the lobed bearing. Multilobe bearing have high stability. Bearings are characterized by the bearing bore profile. The bore of the multilobe bearing have special characteristic such as Non-cylindrical bore design, the radius of lobe is slightly larger the journal radius. This gap produces the efficient oil film in converging- diverging profile. [6]

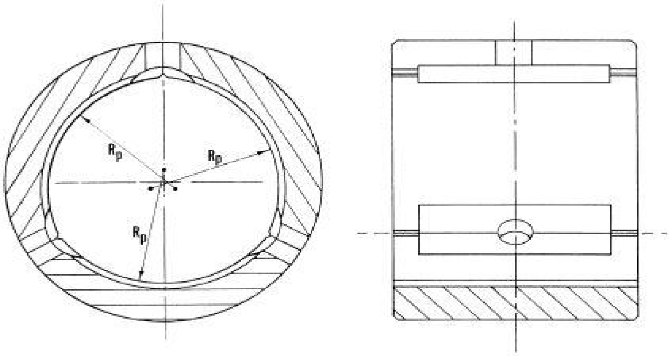

Fig. 1.2 Lobed Bearing [7]

1.2.3 Tilting pad journal bearing:

Tilting pad type of journal bearings provide stiffness efficiently.The minimum oil film clearance, pivoted pad are very useful for development of wedge shape oil profile.Pad are pivoted to the bearing by means of pin. Thus, there is no circumferential movement of the pads. Pads just make them tilting motion to support the variable loads and hence provide greater stability. The materials mainly used for shoes low-carbon steel. There is a coating of Babbitt on the bearing [8].

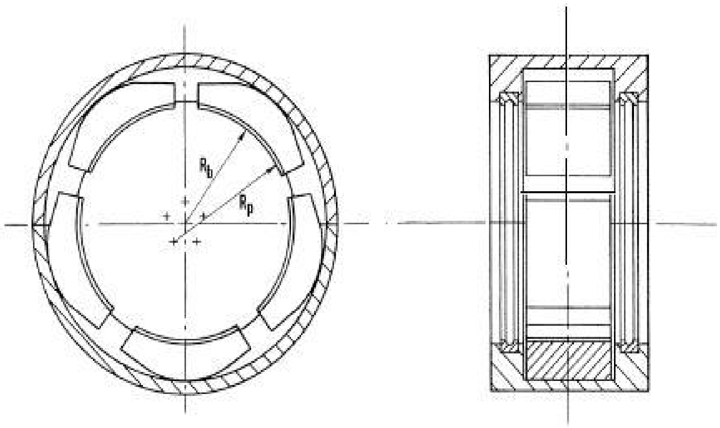

Fig. 1. 3 Tilting Pad Bearing [9]

2. LITERATURE SURVEY

S. Chaitanya Kumar, et al, [1]carried out CFD analysis of hydrodynamic journal bearing. Author stated that the load carrying capacity and the pressure developed in the layers of the lubricating oil used are interdependent. Author concluded that, for the short length bearing, displacement and stresses developed increases with the increasing eccentricity ratio. Thus, the deformation also increases with eccentricity ratio.

Swapnil M. Pawar, et al, [6] considered two lobe hydrodynamic journal bearing for the CFD analysis. In the present paper, author have validated theoretical results with Computational Fluid Dynamics (CFD) software. The circumferential variation of pressure over bearing surface is studied and compared with Simple Hydrodynamic Journal Bearing and it is found that Two Lobe Hydrodynamic Journal Bearing had much less maximum pressure than Simple Hydrodynamic Journal Bearing. From the results it was concluded that, at high speed, there is change in maximum pressure of cylindrical bearing in the fluid film and hence journal falls in the low pressure zone. This repeated cycle setups vibration in the bearing. On the other hand, maximum pressure variation is very less for two lobe journal bearing. So, the two lobed bearing, in comparison with simple hydrodynamic bearing, is more stable. Also the stability of the two lobe journal bearing increases with increased eccentricity ratio.

Nabarun Biswas, et al, [10], performed unsteady transient analysis of 3lobe journal bearing. Author performed the simulation for 1000 iterations, solution became steady after 110 sec. No slip conditions were given to walls. Material used for wall was Steel. Critical values of pressure is observed at fluid solid interface. Maximum pressure is studied at minimum region of oil film thickness.

K. M. Panday, et al, [11], investigated thin film lubricated bearing for numerical unsteady analysis. The thin film lubricated journal bearing was dynamically loaded and transient response was observed under unsteady conditions. Author found out that with increase in L/D ratio, there is reduction in shear stress on surface of bearing and journal but as the L/D ratio increases, turbulent viscosity of lubricant rises.

Rohit S. Kerlekar, et al, [12], worked on performance analysis of 3 lobed hydrodynamic journal bearing. Analysis included testing of bearing on an experimentation setup for various speeds at 600N and 300N loads.Author concluded that pressure generated is more for higher rpm and load. The CFD results were compared with experimental results and it is observed that, pressure and load on bearing increases with increase in rotation speed of journal.

MukeshSahu, et al, [13] studied journal bearing using CFD. Author found out circumferential & axial pressure distribution on journal surface, with and without considering temperature effect. Author concluded that temperature created from the frictional force results in decrease in the viscosity of the lubricant and lesser viscosity of lubricant causes decreases the maximum pressure of the lubricant inside the bearing.

This chapter takes an overview of the previous work and analysis done using ANSYS in the form of Literature Survey. Also the major problems and the approaches used to carry out analysis are surveyed.

3.OBJECTIVE

Based on the literature survey, need of the present work was interpreted and following objectives were fixed.

1. To study the performance characteristics of bearing in detail.
2. To study the variation of pressure and stresses developed inside the fluid film.
3. To compare the results of various lubrication oil considered for present work

4. METHODOLOGY AND ANALYSIS

Bearings selected for transient and steady state analysis are axial grooved bearing, 3 Lobe Bearing and Tilting pad bearing. The lubricating oils selected are SAE-50, ISO-VG-60 and Servo Prime Oil. Bearings are modeled using Autodesk Inventor software. Meshing is performed in ICEM-CFD. Then, simulation is carried out in ANSYS FLUENT Solver. Transient analysis is carried out with calculated time step size and number of time steps. Contours and Plots Total Pressure are obtained and studied.

4.1 Dimensions

- Axial Grooved bearing [14]: length= 80 mm, shaft dia = 99.658 mm, outer dia of bearing = 150 mm, inner dia of bearing = 100 mm, depth of groove =5 mm, length of groove = 16 mm, Min. Oil film Thickness = 0.021 mm
- Multilobe Bearing [15]: length= 180 mm, shaft dia = 179.64 mm, outer dia of bearing = 205 mm, inner dia of bearing = 180.36 mm, Min. Oil film Thickness = 0.21 mm, preload = 0.4
- Tilting Pad Bearing [16]: length= 300 mm, shaft dia = 299.526 mm, outer dia of bearing = 354.776 mm, inner dia of bearing = 404.776 mm, Min. Oil film Thickness = 0.0875 mm, pad arc angle = 103 degrees, pad width = 270 mm

4.2 Properties of Lubricants:

Table 1: Properties of Lubricants [16] [10] [6]

Parameters	Density	Viscosity	Sp. Heat	Oil thermal conductivity
ISO VG 46	860 kg/m3	0.46 Poise	2001 J/Kg ^{0}C	0.135 W/m^{0}C
SAE 50	899 kg/m3	0.44 Poise	2270 J/Kg ^{0}C	0.62 W/m^{0}C
SERVO PRIME 46	867 kg/m3	0.326 Poise	2000 J/Kg ^{0}C	0.126 W/m^{0}C

4.3 GRID Model Description

Table 2: Types and number of Mesh Elements

Element Type	Hexa	Quad	Total
Axial Grooved Bearing	67084	31360	100571
Three Lobed Bearing	67776	47302	116929
Tilting Pad Bearing	185040	96408	285331

4.4 CFD Model Description

Flow was considered to be Viscous-Laminar. Pressure was applied Normal to Boundary. The operating pressure was set to 101325 Pa, No slip condition was used for inner and outer wall, while in case of Tilting Pad Bearing, and pad walls have been assigned with Specified Shear equal to zero.Gauge pressure was set to zero, the rotational axis origin was set to the value of eccentricity. For each case of bearing, inlet and outlet was set to pressure inlet and outlet respectively. Inner wall was considered as Moving Wall with specified RPM and outer wall was considered as stationary wall.Simulation was carried out with 3D Double Precision, Working fluid was Oil, Convergence criteria was used as $10^{-4}$for continuity, velocity parameters, Second Order Upwind was used for Momentum and velocity, PRESTO scheme was used for Pressure. Under relaxation factors were 0.3 for pressure, 0.7 for momentum equation, and 0.7 for energy.

5. RESULT AND DISCUSSION

5.1 Axial Grooved Bearing:

Rotational speed of journal for the case of axial grooved bearing considered is 4000 RPM [14]. The time step size used is 0.0015 sec. Total 1334 time steps were used. Mass flow rate and total pressure convergence plot shows straight line after 500 Iterations, i.e. the properties don't change with time after 500 Iterations and flow becomes steady. The results are further iterated for the value of 13340.

The contours of Total Pressure are as shown below in Fig. 5.1.

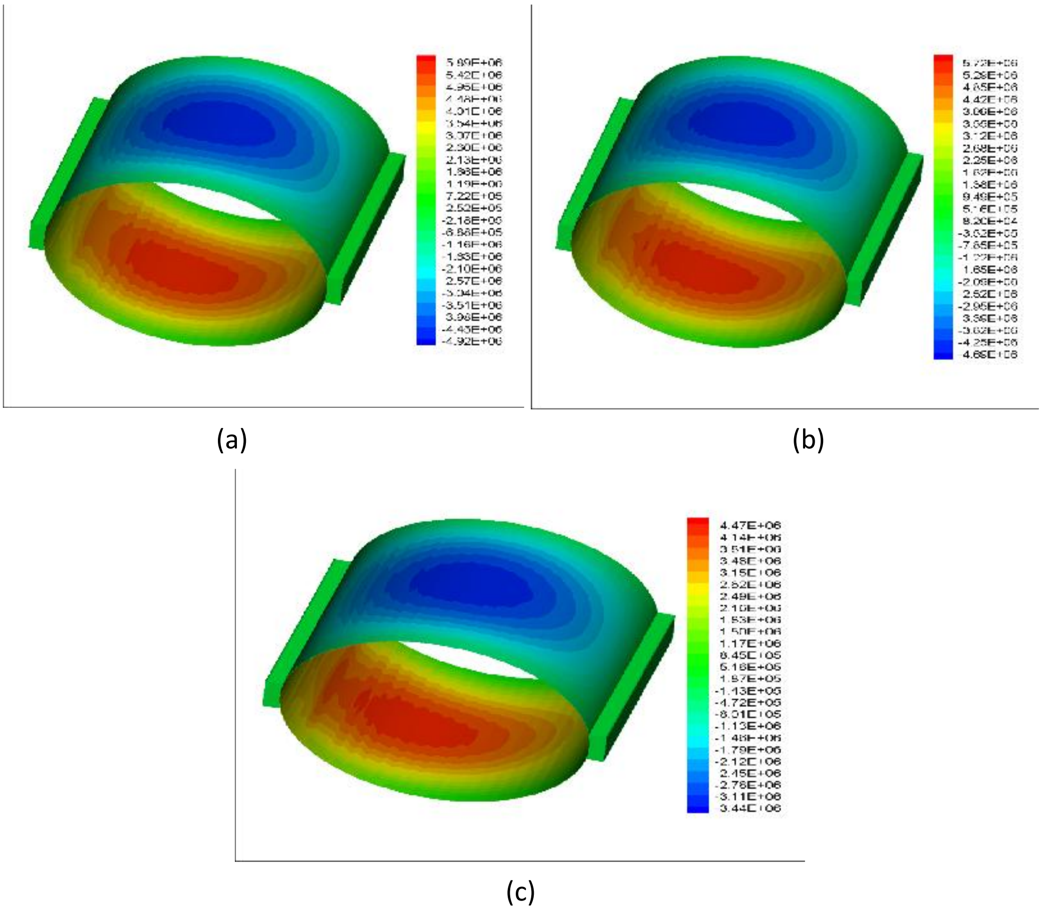

Fig. 5.1 Contours (a), (b), (c) are Total Pressure contours for ISO-VG-46, SAE-50 and SERVO PRIME-46 oil respectively.

Total Pressure variations on the walls of the bearing is plotted against the length of the bearing as shown in the figures 5.2

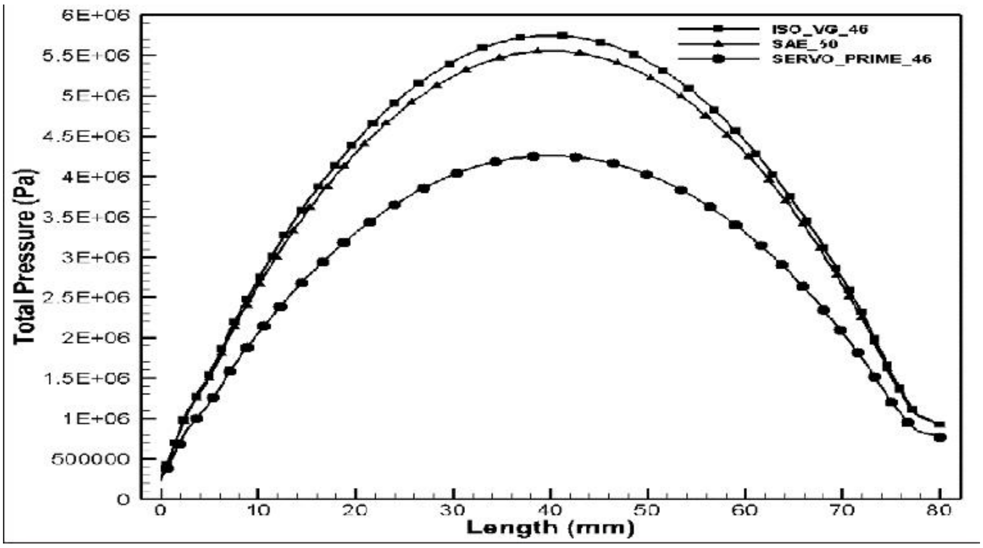

Fig. 5.2 Total Pressure over Length

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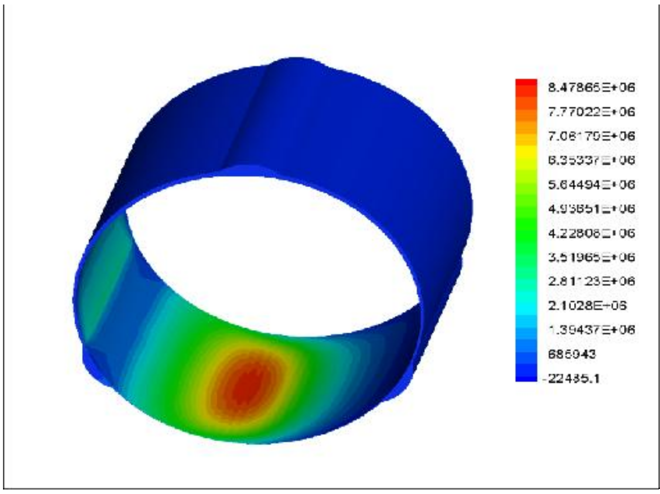
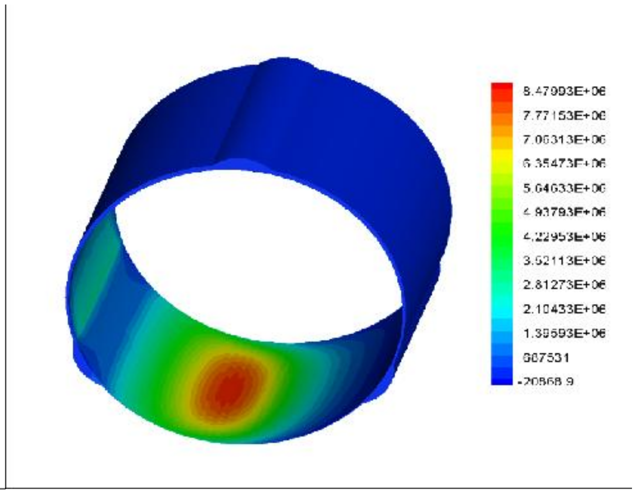
5.2 Multilobe Bearing:

Rotational speed of journal for the case of Multilobe bearing considered is 7448 RPM [15]. The time step size used is 0.0008 sec. Total 2500 time steps were used. Mass flow rate and total pressure convergence plot shows straight line after 1250 Iterations, i.e. the properties don't change with time after 1250 Iterations and flow becomes steady. The results are further iterated for the value of 12500.

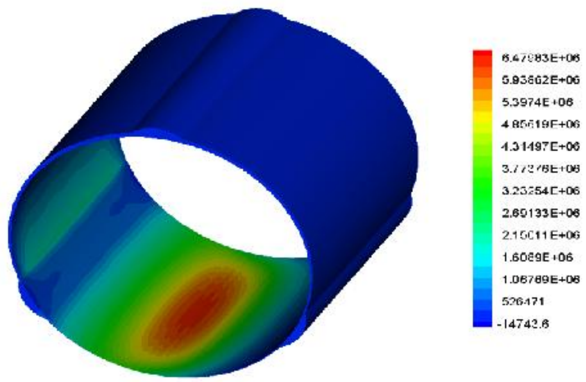
The contours of Total Pressure are as shown below in Fig. 5.3



(a)



(b)



(c)

Fig. 5.3 Contours (a), (b), (c) are Total Pressure contours for ISO-VG-46, SAE-50 and SERVO PRIME-46 oil respectively.

Total Pressure variations on the walls of the bearing is plotted against the length of the bearing as shown in the figures 5.4 Readings of ISO VG 46 and SAE 50 found to be matched to each other.

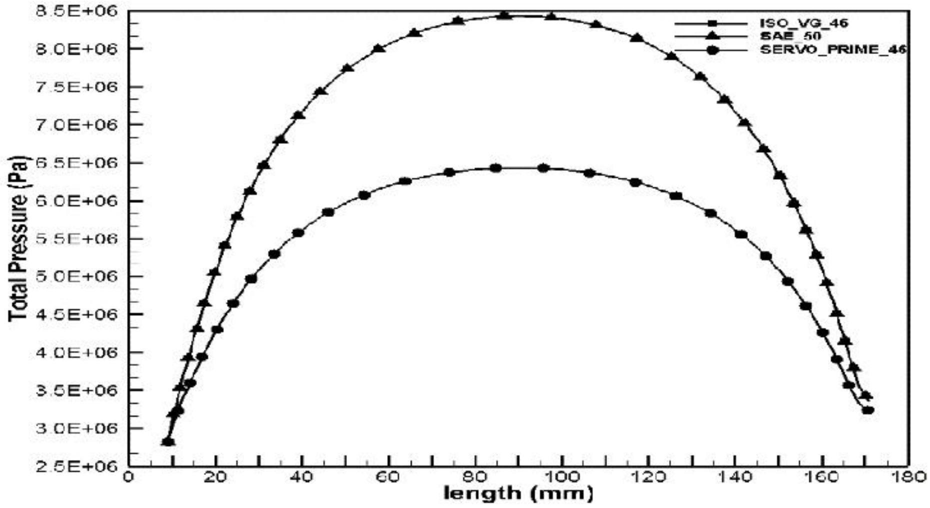
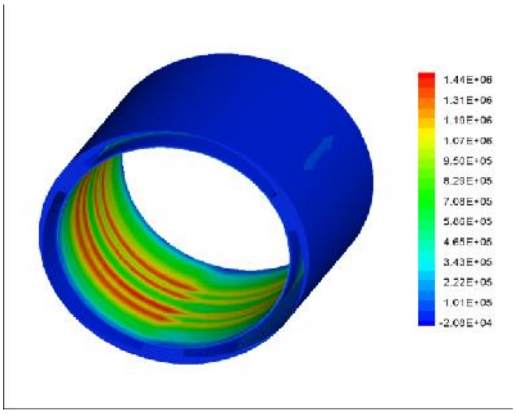
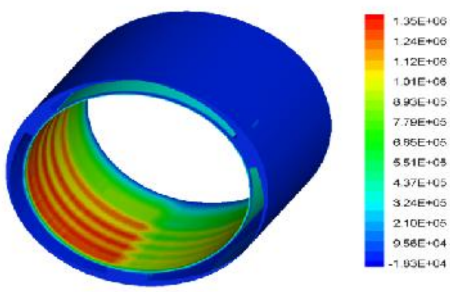


Fig. 5.4 Total Pressure over Length

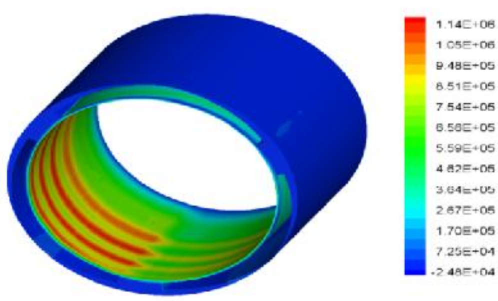
5.3. Tilting Pad Bearing: Rotational speed of journal for the case of Multilobe bearing considered is 5000 RPM [16]. The time step size used is 0.0012 sec. Total 1667 time steps were used. Mass flow rate and total pressure convergence plot shows straight line after 2000 Iterations, i.e. the properties don't change with time after 2000 Iterations and flow becomes steady. The results are further iterated for the value of 16670. The contours of Total Pressure are as shown below in Fig. 5.5



(a)



(b)



(c)

Fig. 5.5 Contours (a), (b), (c) are Total Pressure contours for ISO-VG-46, SAE-50 and SERVO PRIME-46 oil respectively.

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Total Pressure variations on the walls of the bearing is plotted against the length of the bearing as shown in the figures 5.6

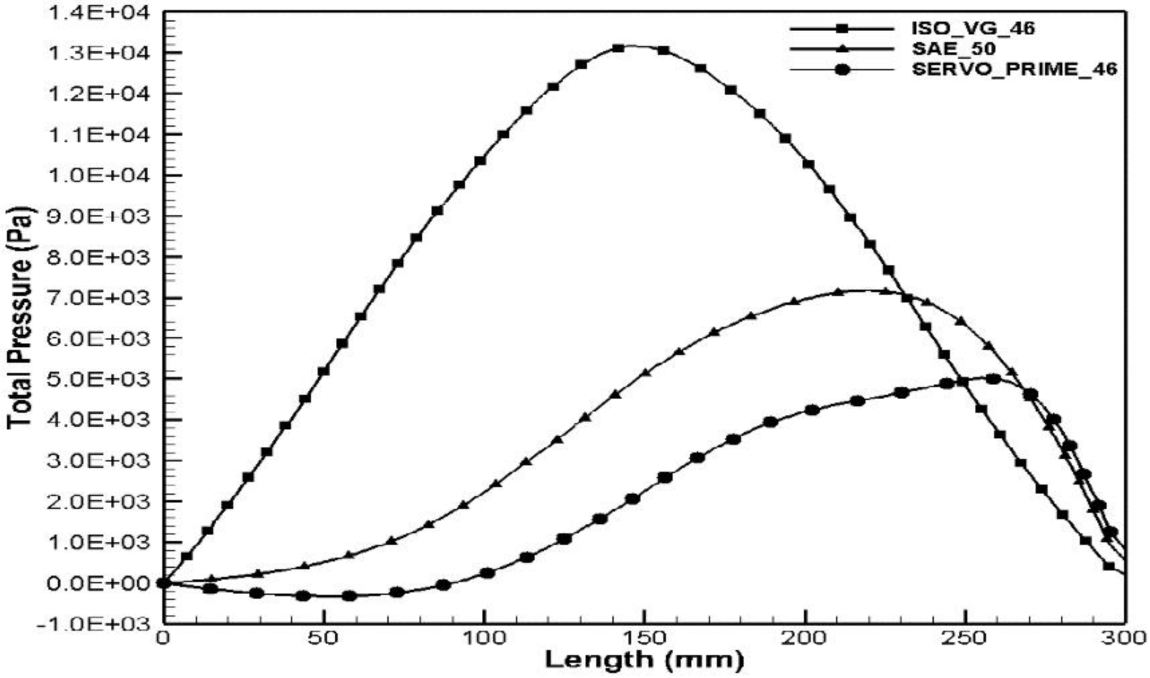

Fig. 5.6 Total Pressure over Length

For Axial Grooved bearing, the maximum total pressure, 5.86E6 Pa, was observed in case of ISO VG-46 oil while the minimum total pressure, 4.47E6 Pa, was observed in case of SERVO Prime-46 oil. For three lobe bearing, the maximum total pressure, 8.47993E6 Pa, was observed in case of SAE-50 oil while the minimum total pressure, 6.47983E6 Pa, was observed in case of SERVO Prime-46 oil. For Tilting Pad bearing, the maximum total pressure, 1.44E6 Pa, was observed in case of ISO VG-46 oil while the minimum total pressure, 1.14E6 Pa, was observed in case of SERVO Prime-46 oil.

6. CONCLUSION

Transient behavior of Axial Grooved, Multilobe and Tilting Pad Bearings had been conducted. From the contours, it was observed that the maximum total pressure is observed in case of ISO VG 46 for Axial Grooved and Tilting Pad bearing while in case of SAE-50 for Three Lob bearing. The minimum total pressure was observed in the case of SERVO PRIME 46 oil for each of the three bearings. Total Pressure Vs Length showed that ISO VG 46 and SAE 50 gave result in good agreement with each other but SERVO PRIME 46 oil developed least pressure at the minimum oil film for each bearing model. Therefore, in the present work, it was revealed that, SERVO PRIME 46 oil is best advisable lubricant amongst the cases considered.

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