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Investigation on Thermal Effects in Journal Bearing Using Bingham Lubrication

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Abstract : *Hydrodynamic journal bearing are broadly utilized as a part of industry due to their effortlessness, efficiency and minimal effort. They support rotating shafts over various years and are regularly subjected to many stops and begin. Thermo hydrodynamic (THD) examination ought to hence be done to acquire the reasonable execution qualities of the bearing. Such a large number of analysts have been done in this field. The vast majority of these investigations utilized two dimensional vitality conditions to discover the temperature conveyance in the liquid film. This paper tries to display the thermo hydrodynamic execution of a hydrodynamic diary bearing. The investigation manages an alternate l/d proportion. Our fundamental concentrate was on three distinctive l/d proportion of hydrodynamic diary bearing weight, temperature dispersions at the film/shrub interface. The vast majority of these examinations utilized two dimensional vitality conditions to discover the temperature circulation in the liquid movie by disregarding the temperature variety in the hub heading and two dimensional Reynolds conditions was utilized to get weight dissemination in the ointment stream by dismissing the weight variety over the film thickness. In this paper a numerical approach which can precisely foresee the execution qualities of a surface finished diary bearing have been displayed. After examination the outcomes are approved through test investigation.*

Keywords – *Computational Fluid Dynamics, Hydrodynamics Journal Bearing, l/d Ratio.*

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I. Introduction

Journal bearings are machine elements in which the applied force is entirely supported by an oil film pressure. As with motion, load is applied to a bearing in either of two directions, or in both directions simultaneously. The centrifugal pump rotor is one of the most complex and vital assembly units that determine reliability of the whole pump to a large extent. Major components of a rotor are a shaft, impellers, shaft couplings,shaft sleeves, parts of mechanical seals, balancing devices, etc. Depending on a pump design the rotor is manufactured with different mutual arrangement of parts. A cantilever pump impeller is placed on the shaft end. As a rule, rotor in a single-stage double entry pump and ring-section type pumps is kept by symmetrically located outer bearings. Bearings are placed in bearing housings. Bearing housings could have different shapes and arrangements. Rotor dynamics of the barrel casing centrifugal pump rotor depends on various entities, the most important of which are hydrodynamic processes within plain journal bearings and annular seals, and the static stiffness of bearing housings. The inclusion of the stiffness of bearing housing designs will allow determining rotor dynamics of the barrel casing centrifugal boiler feed pump rotor more precisely. It has been analysis on thermos-hydro-dynamics analysis of a worm plain journal bearing. Their deal with 100 mm diameter bearing they found that wear defect could lead to an increase in thermos-hydro-dynamics performance with highly loaded bearing operating at low speed and critical parameter are film thickness and maximum pressure[1]. The influence of surface texture hydrodynamics journal bearing through Computational Fluid Dynamics. has been carried out on the review on design and thermal performance of non-circular journal bearing. They worked on the precision of the journal bearing for worked at high speed operation and increases the high pressure and temperature poses; and increase the lubricant temperature and reduces the lubricant viscosity that reason chances of failure are increased.[3] has investigation on the failure analysis of journal bearing. has been carried out on the thermal effect in journal bearing. They worked on the rise in oil temperature, thermal pressure and load carrying capacity at three different commercially oils[5]. has investigation on journal bearing through Raimondi and boyd chart. They worked on the pressure profile and oil film thickness in journal bearing through FEM approach. They also take help through Raimondi and Boyd chart. They found that temperature increase was shown to give a decrease of attitude angle β and an increase in pressure peak. Increase of viscosity-to-pressure sensitivity gives a general increase of peak pressure[7]. has been analysis on the thermos-hydrodynamic journal bearing. has numerically investigation on the thermos-hydrodynamics of journal bearing. Fundamentally they dealt with the execution attributes and weight and temperature dispersion of a plain journal bearing. Priyanka Tiwari et. al. [10] has investigation on hydrodynamic journal bearing. They found that indistinguishable answer for interminably L/D = 1.5 was greatly improved[9].have utilized computational liquid dynamic (CFD) for dissect the hydrodynamic execution of roundabout journal bearing. From their outcome discovered that when the thickness is put steady , temperature and in addition weight increments is more in grease however in reasonable idea expanding temperature decreases the ointment consistency and it influence the bearing stacking limit consequently at consistent consistency may give wrong prevision, so the investigation is useful for considering the working state of bearing[14]. broke down the execution of limitlessly long journal bearing utilizing CFD and FSI approach. Their targets of her paper need to establish out weight and temperature variety of journal bearing under unfaltering state condition[15].

Ravindra M. Mane et al have exhibited weight conveyance in 3D model of plain journal bearing utilizing COMSOL Multi-material science and investigative model created utilizing reynold condition. Weight dissemination was discovered on unendingly short and interminably long hydrodynamic bearing under enduring state condition. Outline Improvement of journal bearing plan is unpredictable. It includes advancing clearances, bearing length, least film ointment, consistency, stream rate, and bay spaces. Plan conditions are accessible, yet their answer is tedious unless done on a PC. Luckily, these conditions have been decreased to diagram shape, and a wide assortment of plan issues can be settled with different outlines in the writing. Powerful thickness for the bearing ought to be gotten from the mean oil working temperature. Utilizing mineral oil-based greases, this temperature normally runs from 48 °C to 82°C, yet ought to be under 121°C. In any case, Oil stream rates are resolved from the oil temperature rise and power misfortune. At the point when the required oil stream is resolved, a gauge ought to be made with respect to whether the required measure of oil is drawn through the freedom space in the bearing. Least film thickness is frequently appeared on configuration graphs and is found from (1 - ec)

where e = capriciousness proportion and c = bearing freedom. Another critical thought in hydrodynamic oil is warm part of outline.

The warmth produced in the bearing ought to be viably scattered so the balance conditions are come to in a brief span. Further, the normal or balance temperature of the oil ought not surpass 93 to 123°C to anticipate speedy disintegration of the oil. The frictional warmth created can be found from the heap (F) coefficient of contact (f), and the journal speed (n).

Frictional power misfortune: Hg = f v

Where Hg is communicated in Nm/s or W

The oil temperature rise can be assessed , formulated by Raimondi and Boyd or from the warmth adjust condition on account of independent orientation as on account of ring, neckline or oil shower grease. Modern uses of independent course can be found in fans, blowers, pumps, engines et cetera.

$$T_{var} = \gamma C_H \left(\frac{\Delta T}{p}\right)$$

γ is the density of the oil 861 kg /m^3

C_H is the specific heat of the oil, an average

value of 1760 J/ kg.^{o}C may be taken. ΔT is the temperature rise ^{o}C and P is the film pressure in Pa.

Heat dissipated: H_d = C A (T_H –T_A)

Where, H_d = in W or Nm/s

C = combined the heat transfer coefficient (radiation and convection), W/m^2 .^{o}C

A = exposed surface area of the housing, m^2= 20 d l

T_H = surface temperature of the housing, ^{o}C

T_A= temperature of surrounding air, ^{o}C.

The value of C depends on the material, colour, geometry and roughness of the housing, temperature difference between the housing and surrounding objects and temperature and velocity of the air.

C = 11.4 W/m^2.^{o}C for still air

C = 15.3 W/m^2.^{o}C for average design practice

C = 33.5 W/m^2.^{o}C for air moving at 2.5 m/s

The temperature difference T_o –T_H between the lubricant oil film and the housing.

The relationship depends on the lubrication system and the quality of lubricant circulation. Oil bath lubrication system in which a part of the journal is immersed in the lubricant provides good circulation. A ring oiled bearing in which oil rings ride on top of the journal or an integral collar on journal dip into the oil sump and provides fair circulation for many purposes. Wick feeding will result in inadequate circulation and should be limited to very light load application and is not considered here.

$T_O - T_H = b\,(T_H - T_A)$

Where T_O be the average oil film temperature and b is a constant depending on lubrication system. Since T_O and T_A are known.

$$H_D = CAx\left(\frac{1}{b+1}\right)x(T_o - T_A)$$

$H_d = CAB(T_O - T_A)$

Where B = 1/ (b+1) and a rough estimate of this is given in Table 3.1 In heat balance computation, the oil film temperature and hence the viscosity of the lubricant in a self- contained bearing are unknown. The determination is based on iterative process where the heat generated and heat dissipated match giving the equilibrium temperature. This is a time involving procedure.

II. Detail Description of Hydrodynamics Journal Bearing

Table: - 1. Detail of Journal Bearing Design Parameters

Application of Bearing	Centrifugal Pump
Diameter of Journal Bearing	120 mm
Length of Journal Bearing	According to L/D ratio
Radial Clearance	145 µm
Load on Journal Bearing	10 KN
Rotation of Shaft	1800 rpm
L/D	0.75, 1.0, 1.25
Unit Load in Centrifugal pump	to 1.4 MPa

Table: - 2. Detail of Journal Bearing Material Specification

Types of Materials	Steel HSS
Density	8030 kg/m³
Young's Modulus	
Poisons Ratio	0.3
Thermal Conductivity	16.27 w/m K
Specific Heat	502.48 /Kg K

Table: - 3. Detail of Bingham Fluid Specification

Types of Fluid	Bingham Model (ISO VG 220)
Density	867 kg/m³
Poisons Ratio	0.34
Thermal Conductivity	0.95 w/m K
Specific Heat	1845.44 J/Kg K
Viscosity	200 cP@40 °C 14 cP@ 100°C
Temperature Range	-30° C to +220 °C

III. Analytical Formulation Used for Design of Hydrodynamic Journal Bearing

Another important consideration in hydrodynamic lubrication is thermal aspect of design. The heat generated in the bearing should be effectively dissipated so that the equilibrium conditions are reached in a short time. Further, the average or equilibrium temperature of the oil should not exceed 93 to 123°C to prevent quick deterioration of the oil. The frictional heat generated can be found from the load (F) coefficient of friction (f), and the journal speed (n).

Length of the Bearing:-

L/D = 0.75, (1)

Load on Centrifugal pump (P)

P = F/ (L x D) (2)

Velocity (V)

$v = 2\pi nd / 60000$ rad / s where n is in rpm & d in mm. (3)

Frictional power loss: $H_g = F \times f \times v$ (4)

Where H_g is expressed in Nm/s or W

The oil temperature rise can be estimated from chart in Fig.1 devised by Raimondi and Boyd or from the heat balance equation in the case of self-contained bearings as in the case of ring, collar or oil bath lubrication. Industrial applications of self-contained bearings can be seen in fans, blowers, pumps, motors and soon.

$$T_{var} = \gamma C_H \left(\frac{\Delta T}{P}\right) \tag{5}$$

γ is the density of the oil 861 kg /m^3

C_H is the specific heat of the oil, an average value of1760 J/kg.^oC may be taken.ΔT is the temperature rise^oC and P is the film pressure in Pa.

Heat dissipated: $H_d = C\ A\ (T_H - T_A)$ (6)

Where, H_d= in W or Nm/s

C = combined the heat transfer coefficient (radiation and convection), W/m^2.^OC

A = exposed surface area of the housing, m^2=20 dl

T_H=surface temperature of the housing,^oC

T_A= temperature of surrounding air,^oC.

The value of C depends on the material, colour, geometry and roughness of the housing, temperature difference between the housing and surrounding objects and temperature and velocity of the air.

C = 11.4 W/m^2.^oC for still air

C = 15.3 W/m^2.^oC for average design practice

C = 33.5 W/m^2.^oC for air moving at 2.5 m/s

An expression similar to eqn. (6) can be written between the temperature difference $T_O - T_H$ between the lubricant oil film and the housing. The relationship depends on the lubrication system and the quality of lubricant circulation. Oil bath lubrication system in which a part of the journal is immersed in the lubricant provides good circulation. A ring oiled bearing in which oil rings ride on top of the journal or an integral collar on journal dip into the oil sump and provides fair circulation for many purposes. Wick feeding will result in inadequate circulation and should be limited to very light load application and is not considered here.

$T_O - T_H = b\ (T_H - T_A)$ (7)

Where T_O be the average oil film temperature and b is a constant depending on lubrication system. Since T_O and T_A are known, combining eqn. (6) & (7),

$$H_D = CA\ x\ \left(\frac{1}{b+1}\right)\ x\ (T_o - T_A) \tag{8}$$

$$H_d = CAB(T_O - T_A) \tag{9}$$

Where B = 1/ (b+1) and a rough estimate of this is given in data sheet. In heat balance computation, the oil film temperature and hence the viscosity of the lubricant in a self- contained bearing are unknown. The determination is based on iterative process where the heat generated and heat dissipated match giving the equilibrium temperature. This is a time involving procedure.

Table:- 4 Values of the constant B

Lubrication System	Condition	Range of B
Oil Ring	Moving air	0.333-0.500
Oil Ring	Still air	0.667-0.500
Oil Bath	Moving air	0.667-0.500
Oil Bath	Still air	0.714-0.833

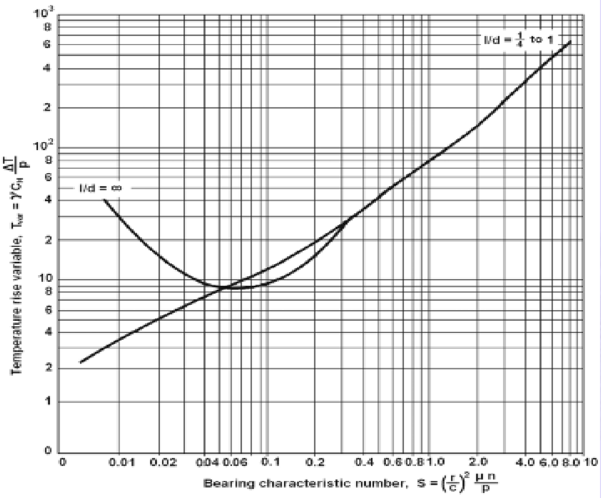

Fig.1. Chart for temperature variable, $T_{var} = \gamma\, C_H \left(\frac{\Delta T}{P}\right)$

The use of this chart will be illustrated with worked out problems in arriving at equilibrium temperature.

IV. Analytical Formulation Used for Hydrodynamics Journal Bearing

Length of the Bearing:-
L/D = 0.75,
L = 0.75 x 120 = 90 mm,

i. Load on Centrifugal pump (P)

P = F/ (L x D) = 10000/ (90 x 120) = 0.926 MPa

ii. Velocity (V)

V = π D N = π X 0.120 X (1800/60) = 11.31 m/s
For viscosity of Bingham Model for the centrifugal pump application is 0.2 Pa s [10].
Assume the bearing to operate between 40 to 60ºC and average temperature of 50ºC, and viscosity is 216 cP @ 40 ºC.

iii. Clearance Ratio

Ψ = (r/c) = (60 / 0.145) = 413.79 so assume that for the clearance value of centrifugal pump is 500.

iv. Summerfield Number or Bearing Characteristic Number (S) $S = \left(\frac{r}{c}\right)^2 X \frac{\mu\, X\, n}{P}$S = 1.62

Table:- 5 Clearance ratio Ψ = (r/c) in 10^{-3}

Working Pressure P MPa	Peripheral Speed m/s		
	Low < 2	Medium – 2 to 3	High > 3
Low to medium P < 8MPa	0.7 – 1.2	1.24 -2.0	2 – 3
High P > 8Mpa	0.3 – 0.6	0.8 – 1.4	1.5 – 2.5

v. At S = 1.62 and L/D = 0.75 then

$T_{var} = \gamma\, C_H \left(\frac{\Delta T}{P}\right) = 115$

$\Delta T = \frac{T_{var}\, x\, P}{\gamma\, x\, C_H}$ ΔT = 66.57ºC

$T_{av} = T_i + 0.5\, x\, \Delta T$

$T_{av} = 40 + 0.5 * 66.57$

$T_{av} = 73.28°C$

vi. For $T_{av} = 73.28°C$hen μ = 46 cP from graph

Recalculated $S = \left(\frac{r}{c}\right)^2 X \frac{\mu\, X\, n}{P}$

$S = (500)^2\, X\, \frac{0.045\, X\, 30}{0.926 X 10^6}$

S = 0.3725

Again for S = 0.3725 and L/D = 0.75, $T_{av} = 34.28°C$ from figure
Than
$$\Delta T = \frac{34.28\ X\ .926X10^6}{867\ X\ 1845.44}$$
$\Delta T = 19.88°C$
$T_{av} = 40 + 0.5 * 19.88 = 49.94°C$
Again for $T_{av} = 49.94°C$, $\mu = 45\ cP$, $S = 0.3298$, $T_{var} = 34$
$h_0 = 0.492\ x\ c$, $h_0 = 0.492\ x\ 0.145$, $h_0 = 0.07134\ mm$
$\frac{r}{c} f = 8.3$
Than $f = 8.3\ x\ \frac{c}{r} = 8.3\ x\ 2.2\ x\ 10^{-3} = 0.01826$
$\frac{P}{P_{max}} = 0.42$
$\emptyset = 54.8°$, $\theta_{po} = 76°$, $\theta_{pmax} = 17.6°$
$\frac{Q_s}{Q} = 0.6$, $\frac{Q}{rncl} = 4.39$
vii. $Q = 4.39\ x\ rcnl = 4.39\ x\ 0.06\ x\ 0.145\ x\ 30\ x\ .08 = 0.09166\ {m^3}/{s}$
$Q_s = 0.6\ x\ 0.09166 = 0.055\ {m^3}/{s}$
viii. $P_{max} = \frac{P}{0.42} = \frac{0.926}{0.42} = 2.21MPa$
ix. Heat Generated:
Hg = f. F.V = 0.01876 x 10000 x 11.31 = 2121.756 W
Heat Dissipation $H_D = CA\ (T_H - T_i) =$
C = Combine heat transfer Coefficient = 15.3 ${W}/{m^2}$K (from average design criteria)
A = Surface Area = 20 x D x L = 20 x 0.1 x 0.09 = 0.18 m²
$H_D = 15.3\ x\ 0.18\ (57.7 - 40)$ = 62.14 W

Table:- 6. Final Details of the design bearing

S.No.	Details of Design Parameters	Notation	For L/ D = 0.75	For L/ D = 1.00	For L/D = 1.25
1	Diameter of Bearing (mm)	D	120	120	120
2	Length of Bearing (mm)	L	90	120	150
3	Radial Clearance	C	0.145	0.145	0.145
4	Load on bearing (MPa)	P	0.926	0.69	0.56
5	Maximum Load on Bearing (MPa)	P_{max}	2.21	1.33	0.68
6	Final Average Temperature (ºC)	T_{av}	51.1	49.5	51
7	Initial Temperature (ºC)	T_i	40	40	40
8	Flow Rate (m³/sec)	Q	0.09166	0.1159	0.1214
9	Maximum Flow Rate (m³/sec)	Q_S	0.055	0.039	0.02914
10	Viscosity (cP)	μ	45	47	49
11	Final Temperature (ºC)	T_H	62.2	59	62
12	Heat Generated (W)	Hg	2121.75	2488.2	3958.5
13	Heat Dissipation (W)	H_D	65.22	83.72	96.94

V. Numerical Analysis Hydro-Dynamic Journal Bearing Using CFD

The solutions obtained from pressure distribution, i.e. dimensionless pressure for different L/D ratios. When increased fine mesh size is chosen, the results are much more precise. Later figures show that the variation of pressure distribution for same eccentricity ratios and for different L/D ratio. The generation of pressure inside the fluid film depends upon many things such as eccentricity ratio, L/D ratio, over relaxation factor and grid size. It is observed that with increase in L/D ratio, frictional force increases and more friction power is generated on the bottom part of the bearing. The variation in frictional power to variation in the L/D ratio is also presented. It is observed that eccentricity ratio has no significant effect on frictional power; however it increases with L/D ratio. With the help of an algorithm, design chart is developed between frictional coefficient and Somerfield Number of various values of L/D ratio. For a given data of Somerfield Number and eccentricity, frictional co-efficient is obtained to calculate the frictional power. It is observed that friction co-efficient is directly proportional to L/D ratio and that too considerably at larger L/D ratio. It is found out that with a hike in average temperature the heat generation reduces and heat dissipation increases. Both the types of heat get equalized at a particular value of temperature, known as equivalent temperature. It is observed that heat dissipation is almost for lesser values of L/D ratio. Heat dissipation increases, but simultaneously the friction-coefficient decreases. This behavior is more optimized for a L/D ratio of 1.25. Hence, if heat transfer is to be controlled a better combination of different L/D ratio.

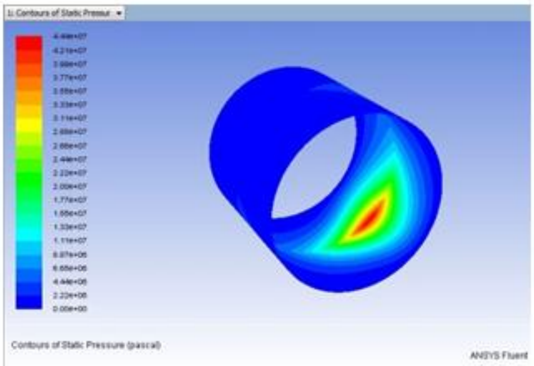

Fig:- 2. Pressure profile of (L/D=1.00)

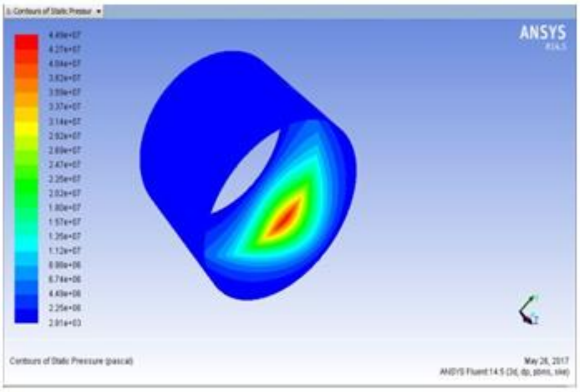

Fig:- 3. Pressure profile of (L/D=1.25)

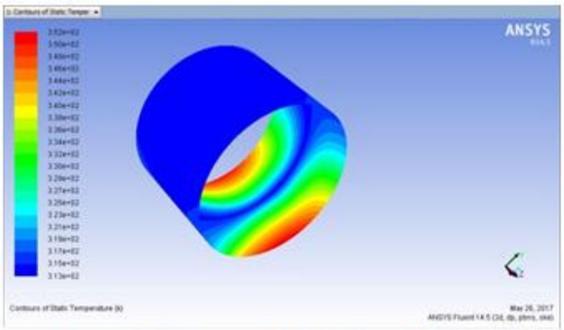

Fig:- 4. Pressure profile of (L/D=.75)

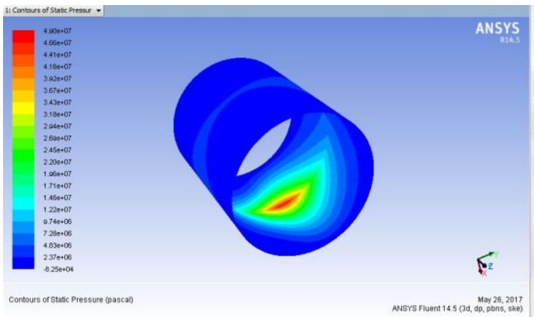

Fig:- 5 Temperature profile of (L/D=0.75)

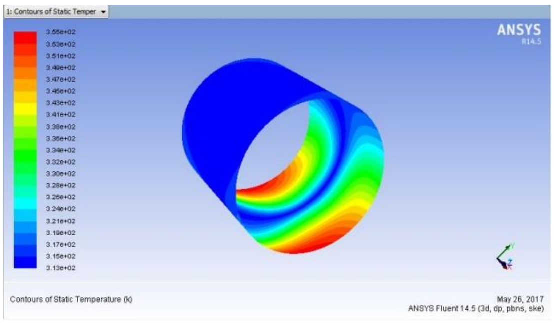

Fig:- 6 Temperature profile of (L/D=1.00)

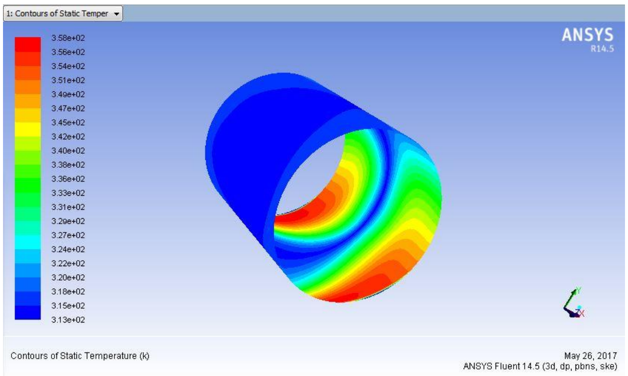

Fig:- 7 Temperature profile of (L/D=1.25)

VI. Conclusion

From this research, I have concluded that as length to diameter ratio is increased, the value of temperature and pressure also increases. To avoid the danger of fluid film rupture in the clearance space, the limit is imposed on peak pressure allowable. As the percentage error in journal bearing is minimum for a L/D ratio of 0.75, this analysis method is best suitable for designing the hydrodynamic bearing having a L/D ratio. We found that pressure distribution in the different L/D ratio journal bearings having quite higher from 1.00 but lower from 0.75 l/d ratio. But in higher l/d ratio is having maximum temperature resisting capacity and there pressure capacity is higher as compare to l/d ratio of 1.00 but lower from 0.75. The pressure generated inside the lubricating film of hydrodynamic journal bearing can be easily shown in three dimensional forms representing circumferential and axial pressure distribution of the figure 2, figure 3 and figure 4. The temperature generated inside the lubricating film of hydrodynamic journal bearing can be easily shown in three dimensional forms representing circumferential and axial pressure distribution of the figure 5, figure 6 and figure 7. Finally we concluded that the higher l/d ratio hydrodynamic journal bearing with Bingham fluid film structure is much better than other l/d ratio of the journal bearings.

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