Study of the Effect of Axial Grooves on the Thermal Behavior and Pressure Distribution in a Hydrodynamic Bearing Under the Influence of Thermal Elastic Deformation

Atheel Rafea Sultan[1] , Ibrahem Ail Muhsin[2]
Department of Mechanical Engineering, University of Tikrit, Salah al-Din, Iraq

Abstract:- This paper deals with the study of the effect of axial grooves on thermal behavior and pressure distribution in a hydrodynamic bearing under the influence of thermal elastic deformation. The two-dimensional Reynolds equation has been numerically solved using a computer program (Matlab) to find the values of pressure on the bearing and calculate the net force, taking into account the effect of both thermal and elastic deformation on the values of permittivity (clearance) and then pressure, and their effect on the Thermal behavior and pressure curve. The results showed that the axial grooves in the hydrodynamic bearing contribute to a decrease in the maximum values of pressure and temperature rates of the bearing and a decrease in the values of the absorbed power.

Keywords:- *Reynolds, Hydrodynamic Bearing, Finite Difference Method, Stiffness Modulus.*

I. INTRODUCTION

Hydrodynamic bearings with axial grooves generally consist of a number of grooves on the inner surface of the bearing. One of the most important necessities that called for the operation of these grooves is to treat the oil shortage in the advanced areas of the bearing in bearings of large diameters (diameter (D) and length (L)), and this has a significant impact on the performance of the bearing and its loadability.

The authors (Costa, L., Fillon, M., Miranda, A. S., and Claro, J. C.P.)[1] carried out an experimental investigation of the effect of groove location and supply pressure on the performance of a static load bearing where the hydrodynamic pressure and temperature distribution on the inner bore surface were measured. and datum temperature (THD) at a variable supply pressure, using a single groove datum located in three different positions. A series of tests were carried out for the used load and rotating speed with variable values. Experimental evidence shows that some datum properties are highly sensitive to changes in groove location and supply pressure that contribute to reductions in maximum temperature and maximum hydrodynamic pressure, with a moderate increase in oil flow rate, researchers (Brito, F. P. and Bouyer, J. and Fillon) , M. and Miranda, A. S.) [2]) conducted an empirical study of the effect of external load and oil supply temperature on the performance of a hydrodynamic bearing with a diameter of (100) mm containing two axial grooves, and they concluded that the presence of the axial groove significantly affects the temperature The heat of the lubricating fluid and therefore affects the thickness of the oil layer, especially at high speeds, where the axial groove contributes to reducing the average temperature of the lubricating fluid in the bracket due to the increase in the oil flow rate, while its effect is less in cases of low load and low speeds. The researchers (M. Vijaya Kini, R. S. Pai, D. Srikanth Rao, Satish Shenoy B and R.Pai) [3] concluded that the use of axial grooves is one of the best ways to supply lubricating fluid and contribute to oil cooling, as the flow rate increases The oil has an important role in maintaining the thickness of the oil film and removing most of the friction heat and contributes to increasing the stability of the support and reducing the applied load. The researchers (U. Singh a, L. Roy a, M. Sahu) [4] found when doing a thermodynamic theoretical analysis of the steady-state axial groove bearing that oil is supplied at a constant pressure. Where they did a thermodynamic analysis using each of the Reynolds equations and the energy equation and equations of heat conduction in the inner surface of the datum and column. And the most prominent finding is that the temperature of the inner surface of the support decreases at night near the entrance, followed by a rapid rise in the circumferential direction and a decrease in the cavity region. The role of the axial groove on the performance of the bearing is to reduce the maximum temperature and maximum pressure values of the hydrodynamic bearing. An increase in the oil supply pressure leads to a decrease in operating temperatures, which is more important for lower loads.

II. ENGINEERING ANALYSIS BEARING

The bearing consists of two main parts, the inner part, which is the rotating part (Journal), which rotates at a rotational speed of (N) around its center (Jc), and the fixed part called (Bearing) has its center (Bc) that contains axial grooves, and the distance between the center of the rotor and the outer part is called (e) (eccentricity) and the connecting line between them is inclined at an angle from the direction of the external load called the attitude angle. The inner and outer parts of the bearing are separated by a layer of oil, which

ISSN No:-2456-2165

is calculated from the end of the groove (Tα) to the angle of the beginning of the next groove (Lα) Figure (1) includes a hydrodynamic bearing with axial grooves and the effect of (EHL) and (THL) on the bearing.

III. PHYLOGENETIC ANALYSIS

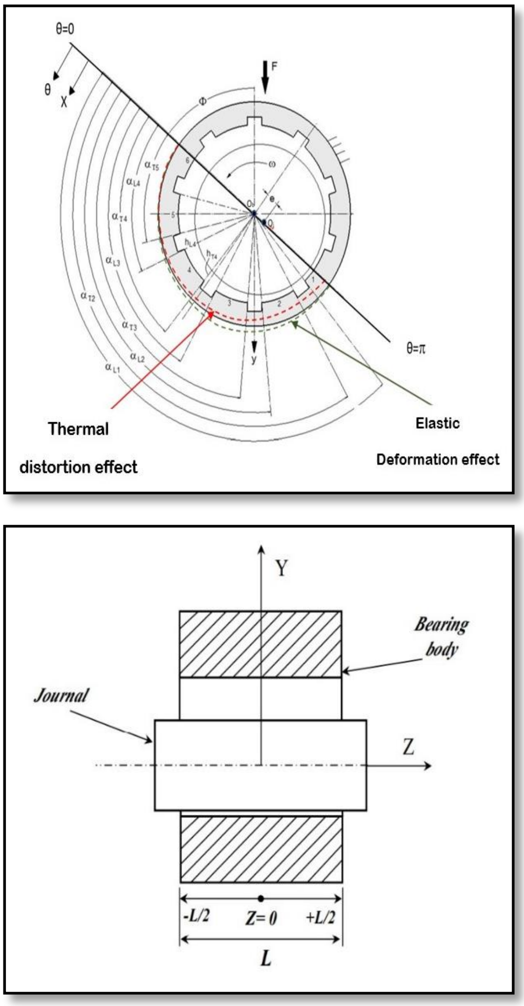

Figure (1) Hydrodynamic bearing with axial grooves and effect appears (EHL), and the (THL) on the bearing

The purpose of using axial grooves is to treat the oil shortage in the advanced areas of the bearing in bearings with large dimensions (diameter and length) and to control the bearing temperature. The increase in temperature leads to a decrease in the viscosity of the fluid and this affects the performance significantly.

In this research, the focus is on the pressure outputs generated under the influence of load, shaft rotation speed and oil specifications, where the Reynolds equation has been solved numerically using the method of finite differences. And the effect of the number of axial grooves on the generated pressure values[6].

$$\frac{\partial}{\partial x}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial x}\right)+\frac{\partial}{\partial z}\left(\frac{h^3}{\mu}\frac{\partial P}{\partial z}\right)=6U\frac{\partial h}{\partial x}+12\frac{\partial h}{\partial t}$$

IV. NUMERICAL SOLUTION

The Reynolds equation was solved using the finite difference method [7] and using a computer program (matlab). This method is used to solve the Reynolds equation to find the pressure values of the oil layer at each point (P(i,j)) by dividing the inner surface of the bearing into a grid with the same number in both directions, and the direction (M) is in the direction (X). In the axial direction (Z) and the number of its divisions was (k), The pressure values of the oil layer (P(i,j)) at each node were calculated using (central finite difference), meaning that each node is related to the four-node pressure. In Figure (2) it is solved by considering the simultaneous thermodynamic and elastic effect (TEHL). It was studied in the range of rotational speed of (rpm4000), the viscosity of the lubricating fluid varies with temperature and pressure, and this in turn affects the performance and stability of the bearing.

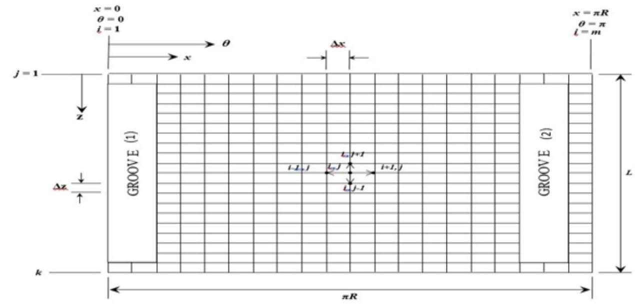

Figure (2) The surface of the support is divided into longitudinal and transverse lines, showing the distribution of grooves in the axial direction.

And after simplification using the finite difference method, it becomes as follows

$$P_{i,j}-A_1P_{i+1,j}-A_2P_{i-1,j}-A_3P_{i,j+1}-A_4P_{i,j-1}=-A_5$$

As the factors (A1, A2,A3, A4, A5). They are coefficients related to the geometrical shape and operational variables (such as rotational speed, oil viscosity, and others).

V. BOUNDARY CONDITIONS

To solve the Reynolds equation The boundary condition (half Summerfield boundary where the positive portion of the pressure on the surface of the boundary is in the boundary (π>θ>0)[8].
And the pressure limits are in the direction (X):
P=0 When θ=(αT)
And the
P=0 When θ=(αL)
And the pressure limits are in the direction (Z):

$$P=0 \text{ at } z=\pm\frac{L}{2}=0$$

Also, the maximum pressure occurs in the middle, that is,
$\frac{\partial P}{\partial z}=0$ at $z=0$

ISSN No:-2456-2165

VI. THERMAL BEHAVIOR

The temperature change has a significant impact on the performance and stability of the bracket because the increase in temperature will lead to a decrease in viscosity and the occurrence of thermal deformations, the rise in temperature occurs as a result of shearing the oil layers, the shear stress is affected by several factors, the most important of which are the speed of rotation and External load and oil viscosity, find (Pinkus)[9]The following formula for calculating temperature.

$$T_i = \left[T_o + \frac{1}{\beta_\circ} ln(1 + \lambda I_{\theta i})\right]$$

and can be obtained (λ) from the equation:

$$\lambda = 2\omega \left[\left(\frac{r}{C_r}\right)^2 \left(\frac{\beta_o \mu_o}{Cp\rho}\right)\right]$$

The value ($I_{\theta i}$) represents the integral of the angle between the oil inlet temperature (T_o) and the location of the point whose temperature is to be calculated (T_i)

$$I_{\theta i} = \int \left[\frac{C_r}{h}\right]^2 d\theta_i$$

$$I_{\theta i} = \int \left[\frac{C_r}{C_r + e\cos(\theta_i)}\right]^2 d\theta_i$$

The average temperature is calculated as follows:

Average Temp. $=\frac{\sum Ti}{n}$

VII. E LASTIC THERMAL DEFORMATION (TEHL)

The theory of elastic thermal deformation is one of the important theories that help to more accurately predict the performance of the bearing, and show its effect, so the solution will consist of four steps: the first step, calculating the pressure and temperature on the surface of the bearing, and the second step for each pair, pair, and pair Calculate the effect of elastic stress on the bearing surface, and the fourth is the effect of thermal deformation on the thickness of the oil layer, as follows:[10]

$$h_{TEHL(i,j)} = h_{(i,j)} + \delta_{e(i,j)} - \delta_{thi}$$

VIII. RESULTS AND DISCUSSION

The results were obtained using a computer program (Matlab) . Where by, the thickness of the oil layer was calculated, and the Reynolds equation was solved numerically to find the values of pressure on the bearing, and then the resultant forces were calculated, taking into account the effect of both the elastic and thermal deformations on the values of the permittivity (clearance) of the strut and then the pressure, And a statement of their effect on the resultant forces and the values of the stiffness coefficients (basic and auxiliary) This study included six models of the bearing (under study) using the numbers of axial grooves (two,, three , four , five , six, seven, regular grooves). The bearing, using a rotational velocity of the axis of (4000rpm) and of an eccentricity ratio of (n=0.1-0.9), for the purpose of studying the effect of the axial grooves on the performance of the datum.

Figure (4) shows the temperature distribution for a conventional bracket and a bracket that contains (two, three, four, five, six, seven) axial grooves and a rotational speed of (4000)rpm) and a decentralization rate of (n=0.7) where we notice a decrease in temperature as the number of axial grooves increases Which agreed with the search results (3).

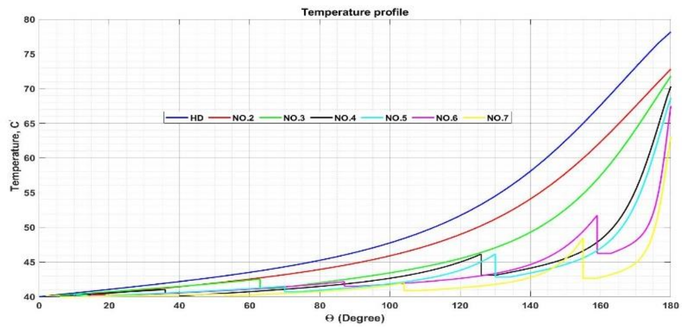

Figure (4) circumferential temperature distribution for a conventional bearing and bearings containing axial grooves

Figure (5) shows the temperature averages of bearing without grooves and axial grooves at the same speed and for the same eccentricity ratio Above, where we notice a decrease in the temperature rates of the axial grooves as their number increases compared to the traditional grooves.

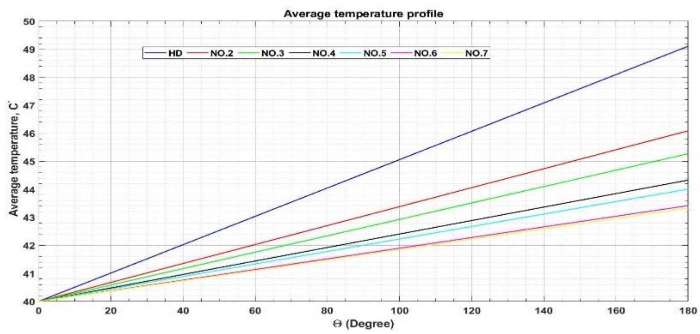

Figure (5) Average temperature of a conventional bracket and brackets containing axial grooves

IX. OIL FILM PRESSURE

Figure (6) shows a comparison between the circumferential pressure distribution for a conventional bearing and a bearing containing axial grooves of numbers (two, three, four, five, six, seven) and at a rotational speed of (rpm4000). Where we note the difference in the shape of the pressure curve due to the effect of the axial grooves, so the curve appears divided, because the pressure values are divided between each two grooves, where the pressure starts to increase from zero at the angle of the end of the previous groove (αT) The pressure values continue to accumulate until they reach their highest value and then decrease until their value becomes zero at the starting angle of the subsequent groove(Lα) And thus, sequentially according to the number of grooves, and we note that the maximum pressure values in the axial grooves bearing were lower than the traditional bearing, which agreed with the results of the research(4) It decreased in the two-groove bearing by (9%) and in the three-groove bearing (17%), and decreased by (21%) for the four-groove bearing, and in the five-groove bearing (22%) and the decrease was by (25%)) for the six-groove bearing, and in the saddle with seven axial grooves, the value decreased by 27%. And we notice that the greater the number of grooves in the bracket, the lower the maximum pressure values.

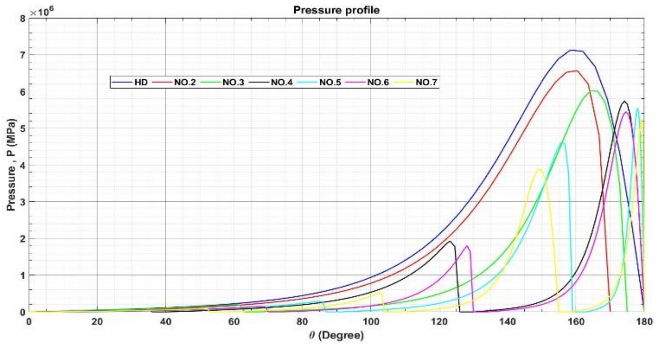

Figure (6)change in oil layer pressure(P) circumferential for conventional bearing and bearing pivot grooves

X. LOAD NUMBER

Figure shows (7)Relationship between loading number (LN)with decentralized ratio(n), where we notice a large difference in the loading number (LN)The greater the number of axial grooves in the bearing compared to the traditional bearing(HD) . And the results show when eccentricity ratio (n=0.8) With a decrease in the loading number of the axial grooves bearing from the traditional bearing, the decrease was in the two-groove bearing by(17%), and in the three-groove bearing (29%), and it decreased in the four-groove bearing by (38%), and it was in May Grooves (52%), and the decrease amounted to (77%) in the six-groove dash, while in the seven-groove dash, the loading number decreased by (85%).

The above results show that the bearing that contains axial grooves has a lower loading number than the traditional dash, due to the low load capacity of the axial grooves cradle.

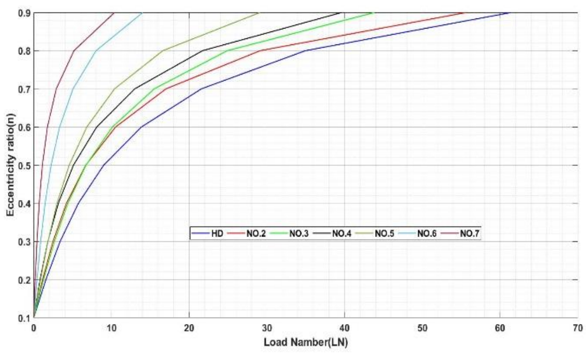

Figure (7)relationship Load number(LN) and the eccentricity ratio (n)

XI. CONCLUSIONS

Based on the current results, the following can be concluded:

1. The use of axial grooves contributes to reducing the average temperature of the lubricating fluid, as well as the support as a whole.
2. Increasing the thickness of the oil layer in the axial grooves' support due to the low-temperature rates.
3. The axial grooves contributed to the decrease in the maximum values of hydraulic and dynamic pressures as the number of grooves increased.

REFERENCES

[1]. P. G. Morton, J. H. Johnson, and M. H. Walton, “Paper XI(ii) The influence of grooves in bearings on the stability and response of rotating systems,” Tribol. Ser., vol. 11, no. C, 1987, doi: 10.1016/S0167-8922(08)70963-5.

[2]. F. P. Brito, J. Bouyer, M. Fillon, and A. S. Miranda, “Thermal behaviour and performance characteristics of a twin axial groove journal bearing as a function of applied load and oil supply temperature,” Tribologia, vol. 25, no. 3. pp. 24–33, 2006.

[3]. “Effect of Groove Location on the Dynamic Characteristics of Multiple Axial Groove Water Lubricated Journal Bearing.” .

[4]. U. Singh, L. Roy, and M. Sahu, “Steady-state thermo-hydrodynamic analysis of cylindrical fluid film journal bearing with an axial groove,” Tribol. Int., vol. 41, no. 12, pp. 1135–1144, 2008.

[5]. H. Feng, S. Jiang, and A. Ji, “Investigations of the static and dynamic characteristics of water-lubricated hydrodynamic journal bearing considering turbulent, thermohydrodynamic and misaligned effects,” Tribology International, vol. 130. pp. 245–260, 2019, doi: 10.1016/j.triboint.2018.09.007.

[6]. J. A. Norris, K. J. Stabile, and R. H. Jinnah, “An introduction to tribology.,” Journal of surgical orthopaedic advances, vol. 17, no. 1. pp. 2–5, 2008.

[7]. G. Wu, W. Zhou, X. Wei, and L. Wang, “A superlinear iteration method for calculation of finite length journal bearing’s static equilibrium position,” doi: 10.1098/rsos.161059.

[8]. Gwidon W. Stachowiak and A. W. Batchelor, “ENGINEERING TRIBOLOGY,” Antimicrob. Agents Chemother., vol. 58, no. 12, pp. 7250–7257, 2014, doi: 10.1128/AAC.03728-14.

[9]. O. Pinkus and S. S. Bupara, “Adiabatic Solutions for Finite Journal Bearings,” J. Lubr. Technol., vol. 101, no. 4, pp. 492–496, Oct. 1979, doi: 10.1115/1.3453400.

[10]. S. Bukovnik, N. Dörr, V. Čaika, W. J. Bartz, and B. Loibnegger, “Analysis of diverse simulation models for combustion engine journal bearings and the influence of oil condition,” Tribol. Int., vol. 39, no. 8, pp. 820–826, Aug. 2006, doi: 10.1016/j.triboint.2005.07.023.

[11]. A. Harnoy, Bearing design in machinery : engineering tribology and lubrication. 2003.

ISSN No:-2456-2165

ABBREVIATIONS

D	Bearing Diameter (m)	E_{AL}	Modules of Elasticity for the Light Metal (GPa)
D	Journal Diameter (m)	T_i	The temperature at any radial section on the bearing surface(℃)
L	Bearing Length (m)	$T_\circ$	Inlet oil temperature(℃)
N	Rotational speed (rpm)	$\beta_\circ$	Temperature coefficient of viscosity (calculated from the temperature/viscosity graph) (1/ ºC)
C_D	Diametric Clearance (m)	α_{AL}	Thermal expansion coefficient of the bearing surface m/(m. ºC)
C_r	Radial Clearance (m)	δ_{th}	Thermal distortion due to temperature rise effect (m)
H	Oil Film Thickness (m)	ΔT	The temperatures difference between any section and the inlet section.(ºC)
Ob	Bearing center	ρ	Density(Kg/m^3)
Oj	Journal center	μ	Kinematic Viscosity (m^2/Sec)
E	Eccentricity (m)	η	Dynamic Viscosity (Pas *Sec)
N	Eccentricity Ratio	h_{th}	Oil film thickness (considering the thermal distortion effect)(m)
Φ	Attitude Angle (Degree)	U	Linear Speed of the rotating Surface (m/sec)
P	Oil Film Pressure (Pas)	m	Number of nodes on x direction
T	Light metal Thickness (m)	k	Number of nodes on z direction
ω	Angular velocity (rad/sec)	Θ_{max}	The Angle of max Pressure (Degree)
h_{max}	Max Oil Film Thickness (m)	h_{min}	Min Oil Film Thickness (m)