# Analysis of the Ball Bearing considering the Thermal (Temperature) and Friction Effects

AMBEPRASAD.S.KUSHWAHA[1], ATUL B WANKHADE [2], DINESH E MAHAJAN [3], DARSHAN K THAKUR [4]

1(Department of Thermal Engineering, RGPV University, Bhopal
Email: ambeprasad88@gmail.com )
2 (Department of Mechanical Engineering ,SGBA University, Amravati
Email: wankhadeatul33@gmail.com)
3(Department of Mechanical Engineering, SSBT`S COET, Bambhori, NMU University, Jalgaon
Email: dineshmahajan5@gmail.com )
4 (Department of Mechanical Engineering , SSBT`S COET, Bambhori, NMU University, Jalgaon
Email: darshanthakur@yahoo.co.in )

## ABSTRACT

**A ball bearing is a type of rolling-element bearing that uses balls to maintain the separation between the bearing races. The purpose of a ball bearing is to reduce rotational friction and support radial and axial loads Friction in bearings causes an increase of the temperature inside the bearing. If the heat produced cannot be adequately removed from the bearing, the temperature might exceed a certain limit, and as a result the bearing would fail. To analyze the heat flow in a bearing system, a typical ball bearing and its environment has been modeled and analyzed using the finite element method. The maximum temperature in the bearing has been calculated as a function of time with the rotational speed as a parameter. The goal of this analysis was to see how fast the temperature changes in the bearing system and if a given maximum temperature (e.g. maximum temperature of the lubricant or bearing metal) is reached. The simulation showed that the higher the rotational speed is, the faster the system reaches a steady state. The bearing did not reach a critical temperature in any of the examined rotational speeds. Scuffing a failure phenomenon i.e. is observed as a result of thermal phenomenon**

*Keywords* - ***Bearing, Scuffing, Simulation, Steady state.***

## I. INTRODUCTION

Bearings are important elements in many machines. To ensure that they reach their calculated lifetime it is important that the temperature conditions in the bearing are adequate. After turning on a machine, the maximum temperature in the bearing system rises up to a steady state temperature. The question was whether or not this steady state temperature is reached gradually or if the maximum temperature reaches a peak before going down to the steady state temperature. Such a peak could destroy the bearing if no special cooling is provided. Since the temperature development in a bearing strongly depends on its environment (thickness of the housing, cooling conditions etc.), the bearing has been put in a typical housing situation in this simulation. Another geometry would require a different model. The only heat source in the model is caused by friction. Convection was applied to the surfaces of the housing and the shaft. The results show that in the examined range of rotational speed this type of temperature peak does not exist, and therefore a special cooling system to start a machine is not necessary. It also can be observed that it takes about one to two hours to reach the steady state temperature.

Scuffing is a complex phenomenon of severe adhesive wear generated under particular combinations including contact pressure, lubrication, speed and friction. Scuffing involves the sudden collapse of the lubricant film and is generally regarded as resulting from thermal phenomena.

Under particular combinations including contact pressure, lubrication, speed and friction, a critical temperature is reached in the vicinity of the contact. At this temperature, the breakdown film with local welding or adhesion of the contacting surfaces can appear.

## II. DESCRIPTION OF SYSTEM

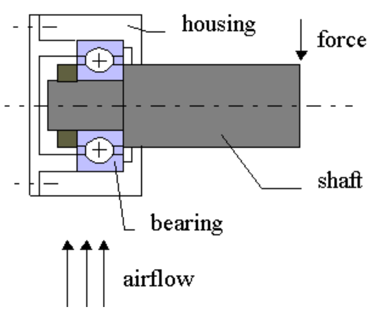


*Fig 1 :- Section of Bearing Housing*

A shaft is situated between two bearings with two housings. A force is applied in the middle of the shaft. Only half of the shaft and only one bearing with housing is shown and modeled for reasons of symmetry. For this geometry the temperature development in the bearing within a range of rpm between 1000 and 5000 has been calculated. Before modeling the geometry as an FEM model, the applied loads (the heat generation rate and the convection coefficients) were calculated

## III. THERMAL MODEL

After each calculation of bearing generated heat rates, either steady state or time transient temperature analysis may be performed. The computations are terminated in the following manner:

1. The steady state case terminates when each system node temperature is within h Centigrade of its previously predicted value. The value for A is specified by the user (typically 2'c).

2. The transient calculation terminates when the user specified transient time interval is reached or when one of the system temperature nodes exceeds 6OO0C (1112'F).

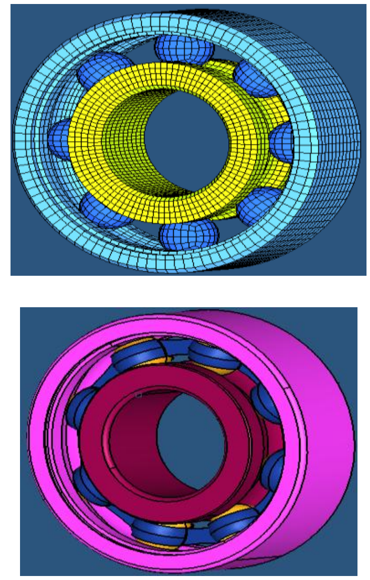

*Fig 2 :- Meshing and Analysis of Beaing*

### III.I STEADY STATE TEMPERATURE MAP

The physical structure is considered to be divided into a number of elements represented by nodes. Heat flow node i from surrounding nodes j, plus the heat generated at. node i, must equal zero to satisfy the definition of steady state conditions.[1]

After each calculation of bearing generated heat, which results from a solution of the bearing portion of the program, a set of system temperatures is determined which satisfy:

q i = qoi+qgi = o for all nodes i

where,

qoi is the heat flow from all neighboring nodes to node i

qgi is the heat generated at node i.

Values are calculated to represent heat created by bearing friction.

### III.II CALCULATION OF HEAT TRANSFER RATE

- Heat transfer mechanisms which occur in a bearing application are:
- Conduction between inner ring and shaft and between outer ring and housing.
- Convection between the surface of the housing and the surrounding air.
- Radiation between the shaft and the housing
- Forced convection between the bearing and circulating oil.

## III.IIIGENERATED HEAT

A heat source may exist at node i. The quantity representing the source magnitude must be added to the net heat flowing from neighboring nodes. When the heat source is other than a spherical roller bearing, it may be considered to produce known amounts of power, in which case constant numbers are entered as input to the Program

### A CONDUCTION

The heat flow qci,j which is transferred by conduction from node i to node j, is

$$q_{ci,j} = \frac{\lambda A}{\ell} (t_i - t_j)$$

Where, X = the thermal conductivity of the medium
L = length between i and j

### B FREE CONVECTION

Free convection between a solid medium and air, the heat flow qvi,j transferred between nodes i and j can be calculated from the equation

$$q_{vi,j} = \alpha_v A \ |t_i - t_j|^d \ . \ \mathrm{SIGN}\ (t_i - t_j)$$

where
a = t h e film c o e f f i c i e n t of heat t r a n s f e r by V f r e e convection
A = t h e surface area of thermal contact between the media
d = is an exponent , usually = 1.25, but any value can be s p e c i f i e d as imput to the program

$$\mathrm{SIGN} = \begin{cases} 1 \text{ if } t_i > t_j \\ -1 \text{ if } t_i < t_j \end{cases}$$

### C FORCED CONVECTION

Heat flow qwi , t r a n s f e r r e d by forced convection can be obtained from the following equation.

$$q_{wi,j} = \alpha_w A(t_i - t_j)$$

where $\alpha_w$ is the film c o e f f i c i e n t of heat t r a n s f e r during forced convection. This value is dependent on t h e actual shape, t h e s u r f a c e condition of the body, the difference i n speed, as we l l t h e properties of the liquid or the gas[1]
In most cases, it is possible t o c a l c u l a t e the c o e f f i c i e n t of forced convection from a general r e l a t i o n s h i p of the form,

$$N_u = aR_e^b P_r^c$$

where a, b, and c are constants obtained from handbooks
Re and Pr are dimensionless numbers defined by
Nu = Nusselt number = $\alpha_w L / \lambda$
L = characteristic length
h = conductivity of the fluid
U = characteristic speed
P = density of the fluid
n = dynamic viscosity of the fluid
Pr = Prandtl's number = rlCp/X
Re= Reynold's number = ULp/rl
Cp= specific heat

## IV. THE FINITE ELEMENT MODEL

In order to model the given geometry with the applied heat generation and the convection on the surfaces of the shaft and housing, several assumptions are made. They are listed below:
The geometry is modeled as an axisymmetric model. Since the rolling elements (balls) are not axisymmetric, they are simplified as a round tube, which again is axisymmetric. This should not have a big influence on the results, since it leads to an axisymmetric temperature distribution which is to be expected. Aside from the fact that the axisymmetric model has less elements than a corresponding 3-D model and therefore uses less CPU time to calculate, another advantage is, that the results are more accurate than in a 3-D solution.
The heat generation is applied to the tube of the 'rolling elements' as a volume load. In reality the elements are spinning around, and the heat is produced on their surface by friction. Therefore, the temperature of the rolling elements is relatively constant over their surface. To obtain this in the model, a high conduction coefficient and a low heat capacity is assumed for the material of the bearing balls. A conduction coefficient which is ten times as high and a heat capacity which is a tenth of the one of steel lead to this result.
The bolts to attach the cover to the housing are omitted and also a flange with bolt holes to mount the housing is omitted.
The sealing between the housing and the shaft was neglected. A heat conducting connection between the shaft and the housing is assumed. This should not have a big influence because the housing wall is very thin there and therefore, there is no major heatflow (see also figure 10).

Other small details such as the threads of the bearing nut and curves in the shapes of the geometry are also neglected.

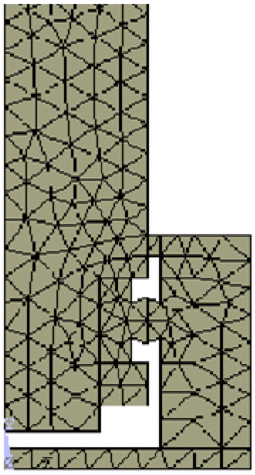

*Fig.3 Mesh diagram of the axisymmetric model, used for the transient analysis*

## IV.I THE ITERATION

The maximum temperature of the bearing with respect to time had to be calculated. The heat generation, which is not constant over the time (HG = HG(n,T)), is itself a function of the unknown temperature-time curve. Since the FEM code ANSYS that was used did not allow a heat generation as a function of temperature, the solution was obtained with iteration. For the first iteration step the heat generation at the starting temperature (293 K) (starting heat generation) was used and assumed to be constant over time. With this the first iteration of the temperature-time curve was obtained. After finding this temperature-time curve, a new heat generation rate as a function of time was calculated. With this new heat generation rate a new temperature function was calculated and so on. This iteration converged after several steps.

## V. RESULTS

For the rotational speed of 1000, 2000, 3000 and 5000 rpm the iterations of the maximum temperature and the heat generation are displayed. It can be observed that the curves converge. In figure 9 the final results of the iterations at different rpms are displayed. It can be seen, that with increasing rotational speed, the steady state temperature is reached faster and is situated at a higher level. The temperature does not reach a peak before going down again to the steady state temperature in any of the cases. So the system obviously is too inert for such fast temperature changes.[2]

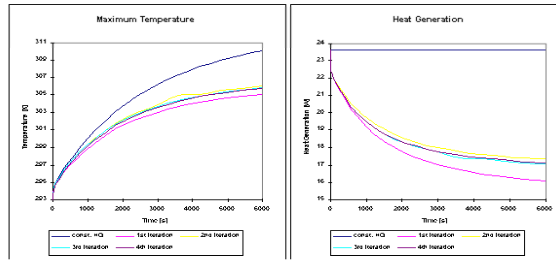


*Fig4 Temperature and Heat Generation at 1000 rpm*

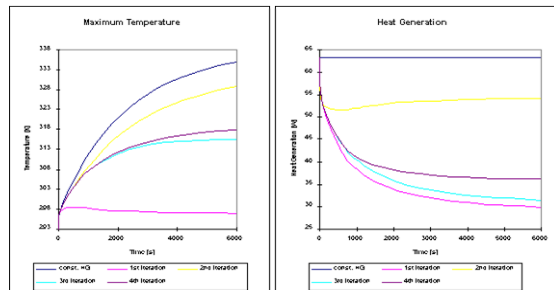


*Fig.5 Temperature and Heat Generation at 2000 rpm*

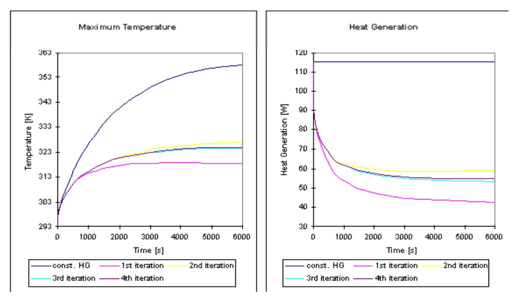


*Fig.6 Temperature and Heat Generation at 3000 rpm*

The manner in which the heat is removed from the bearing over the housing and the shaft can be observed in below figure . The maximum temperature, as in all cases, occurs in the rolling element. The temperature over the ball is almost constant as postulated.

Looking at the calculated steady state temperatures, they seem to be low. They might be calculated too low, since the convection coefficients were calculated with several assumptions and are probably to high. Despite this fact it can be seen, that there is no peak in the temperature-time curve. The next step at his point would be, to obtain the correct convection coefficients with experiments and apply them to the model[2]

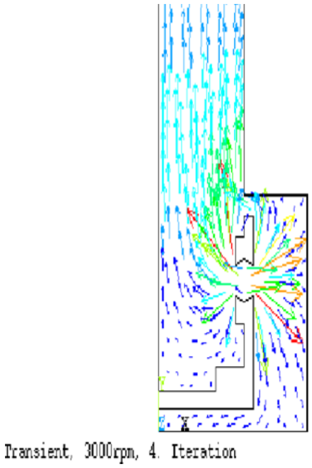


*Fig.7 Heat removal from the bearing*

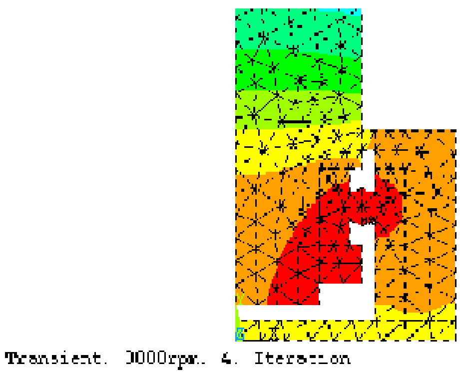


*Fig.8 Temperature distribution in the system (after 6000s, 4. iteration, 3000 rpm)*

## VI. TEMPERATURE INFLUENCE ON BEARING SCUFFING FAILURE

Scuffing involves the sudden collapse of the lubricant film and is generally regarded as resulting from thermal phenomena. Under particular combinations including contact pressure, lubrication, speed and friction, a critical temperature is reached in the vicinity of the contact. At this temperature, the breakdown film with local welding or adhesion of the contacting surfaces can appear. As result of local adhesion, the friction increases and causes a still higher temperature, and then the process can become catastrophic. Scuffing can take place either at low sliding velocity and high normal pressure, by softening and melting a thick surface layer of the contacting bodies, or at high sliding velocities and low normal pressure, by a process of abrasion and adhesion [4].

Nowadays, the skidding and/or scuffing become an important limiting factor for high-speed rolling bearings used in aircraft turbine engines

## VI.I TEMPERATURE DISTRIBUTION MODEL

The total contact temperature on the ball race contact ellipse is a sum of both the bulk and flash temperatures. The bulk temperature is easily measured, but the flash temperature must be calculated.

### ASSUMPTIONS

The calculation of flash temperature is based on the theory of a moving heat source over a semi-infinite solid, formulated by Jaeger. As the Peclet's number is greater than 10 for the ball races contacts [5] the heat flow in the direction perpendicular to the movement may be neglected. From point of view of scuffing failure, the sliding speeds on the contact ellipse (especially on the inner one) in the tested ball bearing (7206) are very important and similar to the rolling speed. The difference between the ball and inner race temperatures is significant. That justifies an unequal heat partition, $\Lambda$, on each two contacting surfaces, by: $\Lambda$ = Uball / Urace .

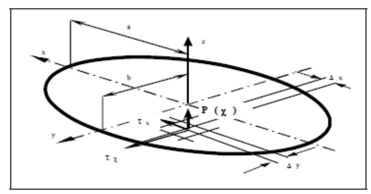

*Fig 9 Contact ellipse discretization*

For a point on a surface contact area P( $\chi$ ), the temperatures distribution was calculated using an iterative procedure on discretized ellipse as presented in figure.

## VI.II FLASH AND SURFACE TEMPERATURE COMPUTATION

The network on discretized ellipse uses the steps: $\Delta$x = a/10 and $\Delta$y = b/10 To evaluate flash temperatures, Tf, on rolling direction, we considered

$$T_{f\,ball} = \sqrt{\frac{1}{\pi \rho_s c_s u_{ball} k_s}} \cdot \int_{y_0}^{y_{i+1}} \left[ \frac{k_f}{h_c} \cdot \left( T_{c\,race(\chi)} - T_{c\,ball(\chi)} \right) + \Lambda \cdot q_{ball(\chi)} \right] \frac{d\chi}{\sqrt{y_{i+1} - \chi}}$$

$$T_{f\,race} = \sqrt{\frac{1}{\pi \rho_s c_s u_{race} k_s}} \cdot \int_{y_0}^{y_{i+1}} \left[ \frac{k_f}{h_c} \cdot \left( T_{c\,ball(\chi)} - T_{c\,race(\chi)} \right) + (1 - \Lambda) \cdot q_{race(\chi)} \right] \frac{d\chi}{\sqrt{y_{i+1} - \chi}}$$

These relations reflect the cumulative character of energy dissipated on ellipse contact To evaluate surface temperatures, Tc , we add the flash temperature as above computed to the average temperature Tm , measured by an infrared thermometer.[3]
We obtained:

$$T_{cball}(y) = T_{m_{ball}} + \sqrt{\frac{1}{\pi \rho_s c_s u_{ball} k_s}} \cdot \int_{-b}^{y} \left[ \frac{k_f}{h_c} \cdot \left( T_{crace(\chi)} - T_{cball(\chi)} \right) + \Lambda \cdot q_{ball(\chi)} \right] \frac{d\chi}{\sqrt{y-\chi}}$$

$$T_{crace}(y) = T_{m_{race}} + \sqrt{\frac{1}{\pi \rho_s c_s u_{race} k_s}} \cdot \int_{-b}^{y} \left[ \frac{k_f}{h_c} \cdot \left( T_{cball(\chi)} - T_{crace(\chi)} \right) + (1-\Lambda) \cdot q_{race_{(\chi)}} \right] \frac{d\chi}{\sqrt{y-\chi}}$$

where:
uball , urace - ball and inner race tangential speed, respectively;ks, kf - thermal conductivity of solid (ball and inner race an lubricant, respectively;
hc - central film thickness; cball ( ) T $\chi$ , crace ( ) T $\chi$ - ball and race surface temperatures in the point of coordinates (xj, yi - $\Delta$ y); qball( $\chi$ ), qrace( $\chi$ ) - total heat generated by viscous friction qf and on the contact asperities qa, when there is no complete separation through lubricant in the point of coordinates (xj, yi - $\Delta$ y).

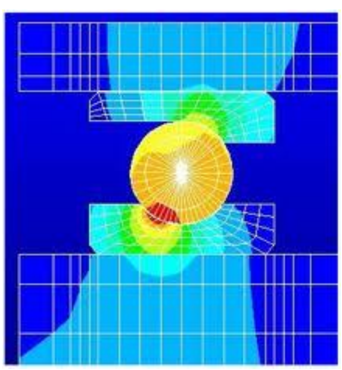

*Fig.10 temperature distribution in balls*

## VII. CONCLUSION

With this analysis it has been shown that the temperature in this bearing system increases gradually to a steady state temperature. To calculate steady state temperatures only a simple static analysis is necessary, which is far less CPU-time consuming. It is important to know the different conditions that affect how the heat is transferred. The coefficients have to be obtained with experiments. Once good reliable data for the heat transportation is available, it is sufficient to calculate the steady state temperature, which is the highest temperature reached.
If the so calculated static temperature then is below a critical temperature, it can be said, that this is a save case. No dangerous temperatures are reached after turning the machine on.
The scuffing results indicate that there is a complex interdependence between mechanical, physical and chemical effects in the rolling contacts
The temperature distribution model is a wide covering one because it includes the thermal transfer between ball and races and also the heat generated both by the viscous friction in lubricant film and the boundary fiction on the asperity contact.

## REFERENCES

[1]SPHERICAL ROLLER BEARING ANALYSIS by R. J. Kleckner J. Pirvics

[2] Temperature Analysis in a Ball Bearing under Heat Generation caused by Friction by Benjamin Rembold 1996

[3] Temperature Influence on Bearing Scuffing Failure byC. BUJOREANU, S. CRETU

[4]. Nélias, D., Seabra, J., Flamand, L., Dalmaz, G.,1996, "Power Loss Prediction in High-Speed Roller Bearings", Dissipative Processes in Tribology, D. Dowson et al., p.465-478.

[5]. Zhu, D., Cheng, H. S., 1989, "An Analysis and Computational Procedure for EHL Film Thickness, Friction and Flash Temperature in Line and Point Contacts", Trib. Trans., No. 32, p. 36