Structural and Thermal Analysis on the Tapered-Roller Bearing

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Abstract

In the rail road industry damaged bearings in service are primarily identified by using wayside hot-box detectors (HBDs). In general if the temperature is greater than 105.5^{o}C the wayside hot-box detectors (HBDs) would trigger the alarm. The present work aims at enhancing the Hot Box Detector temperature of Taper Roller Bearing which is subjected to lot of cyclic loads and suggested to increase the temperature to 120^{o}C instead of 105^{o}C as designed earlier. Finite Element Analysis is carried out to find out the temperature distribution between the cup and roller surface of the taper roller bearing and also stresses at different temperatures are determined to know the safe limit of the taper roller bearing in order to amplify the Hot Box Detectors (HBD) temperature. Fatigue life is also estimated at the corresponding loading conditions.

Keywords: *Taper Roller Bearing, Roller Temperature, FEA on Taper Roller Bearing, Hot box detector, Fatigue.*

I. INTRODUCTION

Tapered-roller bearings arethe most widely used bearings in automobile and railroad industry. When operated under satisfactory load, alignment and contaminant free conditions, the bearing service life is exceptionally long. As a general rule, bearings will outlast the wheel life, and survive several reconditioning cycles prior to being retired. At the end of their life, bearings will initiate fatigue, particularly sub-surface fatigue, rather than wearing out due to surface abrasion. Fatigue failures, or sapling, can lead to material removal at the raceway surface which in turn will cause grease contamination and increased friction that manifests itself as heat within the bearing. The most common method of monitoring bearing health is by conventional wayside Hot-Box Detectors, located to record bearing cup temperatures as the train passes. These devices are designed to identify those bearings which are operating at temperatures greater than 105.5^{o}C above ambient conditions.

Upon closer disassembly and inspection, it has been observed that many of these non-verified bearings contain discolored rollers in an otherwise normal bearing. The discoloration of the steel is visual evidence that these rollers have been exposed to temperatures greater than what isexpected during normal operating conditions. Hence, initial work performed by the Tarawneh et al. [3], [7] focused on determining conditions that would replicate the discoloration observed in the rollers. A laboratory furnace was used to heat numerous rollers to elevated temperatures in various environments. Results indicated that the visual discoloration which best matched that observed in bearings removed from service were rollers heated in grease to temperatures over 232^{o}C for periods of at least 4 hours. To this end, a number of dynamic laboratory tests were carried out to explore several defects and hypothetical scenarios that can lead to bearing overheating.

Only external bearing cup temperatures could be recorded in the dynamic experiments since it is not feasible to measure the temperatures of the internal components of the bearing while it is rotating. In the static tests, monitoring the temperature of the rollers was possible, but the case studies were limited to a setup utilizing two cartridge heaters embedded in two rollers to provide the heating source. Additional embedded heaters could have been used, but it would have added greatly to the complexity of the experimental setup and instrumentation, not to mention the time and effort involved in conducting these experiments.

Numerical simulations tend to be a more economical means of obtaining prompt results and an efficient way to overcome the experimental challenges and limitations. Hence to develop a Finite Element (FE) model for a class K tapered roller bearing, CATIA V5 is used. The FE model is utilizedto run several different bearing heating simulations that provided definitive answers as to whether it is possible for rollers to heat to high temperatures without heating the cup surface to a sufficient temperature necessary to trigger any HBD alarms.

II. LITERATURE REVIEW

In most cases, experimentally acquired data is used to validate the accuracy of the derived FE models. Demirhan and Kanber [1] used ANSYS to investigate the stress and displacement distributions on cylindrical roller bearing rings. The study concluded that the stresses and displacements have different distribution characteristics on the inner and outer faces of the rings and are not uniformly distributed along the height of the rings because of large stresses at the contact points.

The thermal and dynamic behavior of railroad tapered-roller bearings has been explored extensively through several experimental and theoretical studies conducted by Tarawneh et al. First, in a series of papers, Tarawneh et al. [3, 4 and 5] and Cole et al. [5 and 6] examined the heat transfer paths with in tapered-roller bearings, and heat transfer to the bearing from an adjacent hot railroad wheel. Experimentally validated analytical expressions were developed to describe the surface temperature of the bearing cup, the temperature at the cone-axle interface, and the temperature along the wheel web. Additionally, the aforementioned studies resulted in the determination of heat transfer coefficients for heat dissipation from the bearing and wheel surfaces which can be used to devise reliable FE models. Second, in a series of four papers, Tarawneh et al. [2, 8 and 9] investigated the warm bearing temperature trending problem and were able to identify the root cause of this troubling phenomenon. It was concluded that vibration induced roller misalignment is the likely cause for the bearing temperature trending phenomenon seen in service (Tarawneh et al. [8]), and an on-track field test conducted and validated the results obtained from the laboratory testing. Furthermore, vibration signatures of temperature trended bearings in field and laboratory testing are also provided in [9].

From the literature review, the need for reliable numerical models that can predict internal bearing temperatures becomes apparent. With this motivation, the authors utilized their earlier experimental results and analytical models to develop a FE model that can be used to simulate numerous normal and abnormal operating scenarios, that are otherwise very complex and time consuming to duplicate in a laboratory setting, and acquire internal and external bearing temperatures. The developed model can prove to be a very useful tool in future thermal research of railroad tapered-roller bearings. Knowing the temperature distribution within the internal components of the bearing during normal and abnormal operating conditions can help bearing manufacturers explore possible design modifications to their bearings to dissipate the heat more efficiently or select appropriate lubricants that can withstand the internal temperatures experienced by the bearings.

III. DESCRIPTION OF BEARING MATERIAL USED FOR ANALYSIS

Bearing steel has a density of 7.85 gm per cubic-cm, coefficient of linear expansion of 0.00001 per K and thermal conductivity of 30-40 W/mK. Bearing steel is magnetic in nature and good for thermal and electrical conduction. The Table 1 shows the chemical composition of AISI 8620 alloy steel. The mechanical and thermal properties of annealed AISI 8620 alloy steel are outlined in the Table 2.

TABLE 1 CHEMICAL COMPOSITION

Element	Content
Iron, Fe	96.895-98.02
Manganese, Mn	0.700-0.900
Nickel, Ni	0.400-0.700
Chromium, Cr	0.400-0.600
Carbon, C	0.180-0.230
Silicon, Si	0.150-0.350
Molybdenum, Mo	0.150-0.250
Sulfur, S	$\leq$ 0.0400
Phosphorous, P	$\leq$ 0.0350

TABLE 2 MECHANICAL PROPERTIES AND THERMAL PROPERTIES

Properties	Metric
Ultimate strength	530 MPa
Yield strength	385 MPa
Elastic modulus	190-210 GPa
Bulk modulus (typical for steel)	140 GPa
Shear modulus (typical for steel)	80 GPa
Poisson’s ratio	0.27-0.30
Izod Impact	115 J
Hardness, Brinell	149
Thermal expansion co-efficient	11 µm/m°C
Thermal conductivity	46.6 W/Mk

IV. THREE DIMENSIONAL MODELING OF TAPER ROLLER BEARING

A solid model of the bearing is generated utilizing CAT1A V5 to develop the FE model for this study. The axle is rendered as a simple cylinder with a 0.1572 m diameter and a 2.2 m length, which is sufficient in size to act as a heat sink. The bearing is modeled after a class K railroad tapered-roller bearing with some minor modifications thatsimplified the geometry and resulted in a significant reduction to the computational time.

For the first assumption, no cages, seals, wear rings, or grease were included in the model; this is done because the thermal resistances of the polyamide cages and grease are large compared to the rest of the bearing assembly, so the majority of the heat will flow from the rollers to the bearing cup and inner cones. Recent advances in bearing seal technology have resulted in minimal contact low friction seals. Furthermore, these seals constitute a small fraction of the total weight of the bearing and are separated from the rest of the internal bearing components by a combination of air and grease which both have high thermal resistances. The wear rings are in contact with the axle, which constitutes a very large body of metal and acts as a heat sink. Hence, the omission of the aforementioned components from the FE model will not have a significant effect on the results acquired from the present study.

The second assumption is concerned with the contact area between the rollers and the cup and cone raceways. Under normal operating conditions, only the upper hemisphere of a railroad bearing is loaded; therefore, larger contact areas exist between the rollers and the cup and cones in this region. However, since the rollers of a bearing enter and exit the loaded zone continuously as they execute multiple revolutions in a second, an average contact area is applied to all 46 rollers in the bearing. Hence, in the model, the contact area between each roller and the cup and cone raceways is 123.96 mm^2, which represents 3.68% of the total surface area of the roller. The contact area is obtained by first estimating the initial contact area of the rollers with the cup and cones using the Hertzian line contact theory and then using FE analysis to determine the load distribution on the upper hemisphere of a fully-loaded bearing (full load corresponds to a force of 159 KN (35,750 lb) per bearing applied through the bearing adapter), which is then used to calculate the amount of roller compression.

V. BOUNDARY CONDITIONS CONSIDERED FOR THE ANALYSIS

The validity of the FE model depends greatly on the correctness of the Boundary Conditions applied when running the simulations. With this in mind, the Boundary Conditions used for this study are derived from previously conducted experimental efforts Tarawneh et al. [2, 3 and 8] and from material specifications provided by the bearing manufacturer. Four major Boundary Conditions, which are described in this section, are utilized; namely: conduction, convection, radiation, and heat flux.

The heat conduction coefficients for the bearing assembly are provided by the bearing manufacturer. For the bearing steel, AISI 8620 with a thermal conductivity of 46.6 $Wm^{-1}K^{-1}$ is used. A sensitivity analysis is performed to ensure that variations in the thermal conductivity values due to temperature have a marginal effect on the reported results; the results differed by less than 1% when both thermal conductivity values are changed by +/- 10%. Note that the thermal conductivity values for AISI 8620 do not change by more than 7% over the range of temperatures reported.

Convection and radiation Boundary Conditions for the bearing are acquired from the results of previous experimental testing conducted by the authors. Tarawneh et al. [3] provides an overall heat transfer coefficient Ho = 8.32 W $K^{-1}$for the bearing cup, which takes into account forced convection generated by a 5 m/ s airstream and radiation to an ambient at a temperature of 25^{O}C. However, since the software used for the FE simulations requires convection coefficients to be entered in units of W m-$^{2}K^{-1}$, the external surface area of the bearing cup A_{cup}= 0.1262 m^2 is used to obtain the heat transfer coefficient in the appropriate units (h_o = 65.9 W $m^{-2}K^{-1}$). The latter overall heat transfer coefficient is applied to the external (exposed) surface of the bearing only. A convection coefficient value of h_{axle}= 25 Wm^{-2} K^{-1}corresponding to a 6 m/s airstream and a 25 ^{o}C ambient temperature. The characteristic length, Lc, is equal to the diameter, D, of the axle.

Finally, to simulate heat generation within the bearing assembly, heat flux is applied to the circumferential surface of the rollers. The appropriate heat flux value is determined through a trial-and-error process starting with an overall heat input of 11.5 W per roller (normal operation conditions) and increasing this input until the desired external cup temperature is achieved. The acquired heat input per roller is then divided by the surface area of the roller to obtain the heat flux value. Here, it is assumed that the rollers are the source of heat within the bearing which is justified considering the mass of the roller (0.145 kg) relative to the mass of the bearing cup (11.53 kg) and cone (3.9 kg).

VI. FINITE ELEMENT MODELING AND ANALYSIS OF TAPER ROLLER BEARING

Solid model of the Taper Roller Bearing assembly for the Finite Element analysis conducted for this study is depicted in Fig.1. Fig. 2 shows the meshed model of the taper roller bearing.The “bricks and tetrahedral” solid mesh type option is used since it generates the most accurate mesh utilizing the less number of elements. The mesh description is given in the Table 3. The major boundary conditions are utilized namely: conduction, convection, radiation, and heat flux.

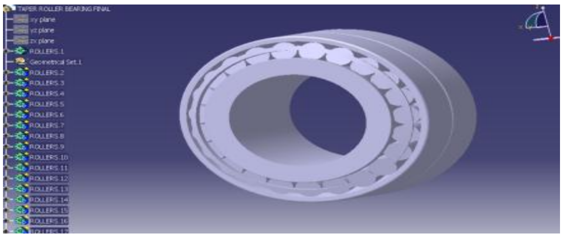


Fig. 1 Taper Roller Bearing model in CATIA V5

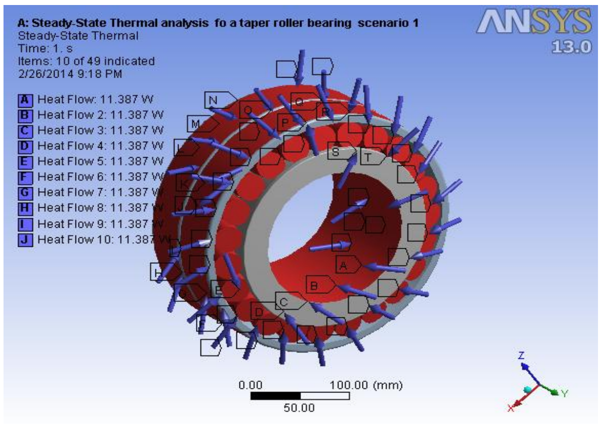


Fig. 2 Meshed model of Taper Roller Bearing

TABLE 3 FINITE ELEMENT MODEL SUMMARY OF TAPER ROLLER BEARING

S.No.	Description	Quantity
1	Total Nodes	355693
2	Total Elements	195809
3	Total Body Elements	94856
4	Total Contact Elements	100953
5	Element Types	4
6	Contacts	138

A. STEADY-STATE THERMAL ANALYSIS

In the present problem, the static load applied on the Taper Roller Bearing is 159 KN as per designer specification, the temperatures acting on rollers and cup are 55°C and 50.2°C respectively as per the calculations obtained from the literature [3].

In order to validate the model, Steady-state thermal analysis is carried out to determine temperatures, heat flow rates and heat fluxes in the Taper Roller Bearing that are caused by thermal loads that do not vary with time. The temperature distribution and heat flux of the taper roller bearing from the Finite Element Analysis is given in the Table 4.

The FE model simulation replicating the latter normal heating scenario indicates that the total heat input, Q_{total} , needed to attain the 50°C cup temperature is about 529 W (within 3% of the experimentally obtained value), which translates into a roller heat input, Qroller, of 11.5 W (for each of the 46 rollers), and a maximum average roller temperature of 56.135°C.Thus, under normal operating conditions, the temperature of the rollers is only about 5°C hotter than the average cup temperature.

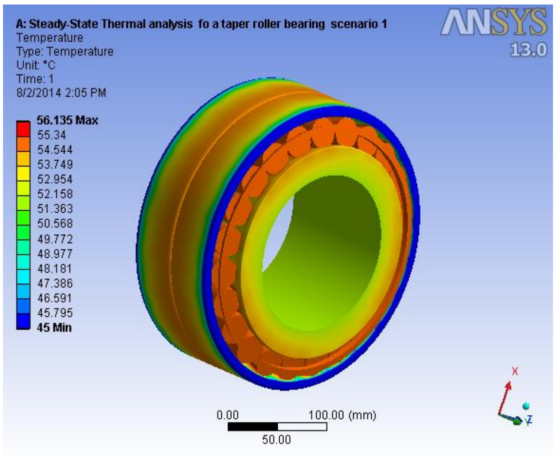


Fig. 3(a) Heat flowing directions of Taper Roller Bearing

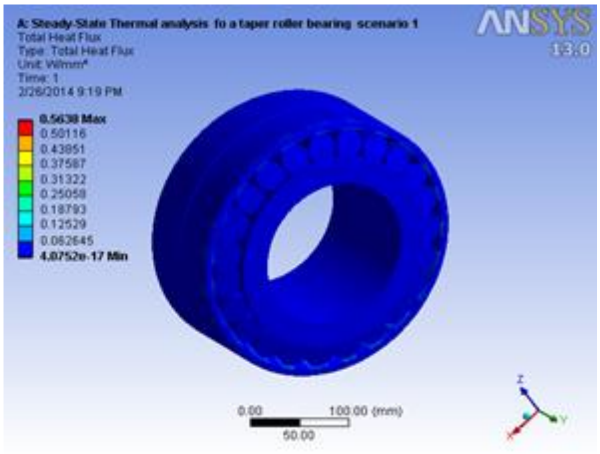


Fig. 3(b) Temperatures of Taper Roller Bearing

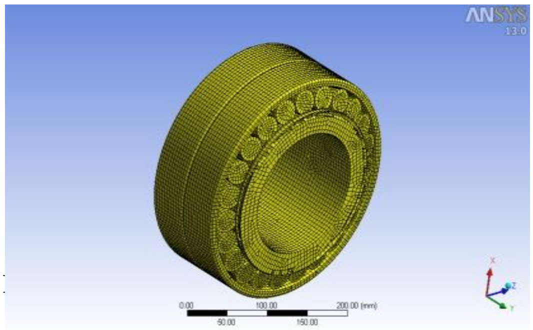


Fig. 3(c) Total heat flux of Taper Roller Bearing

TABLE 4 FEA RESULT OF STEADY STATE THERMAL ANALYSIS

S.No.	Quantity	Maximum value
1	Temperature of cup (°C)	51.363
2	Temperature of rollers (°C)	56.135
3	Temperature of cone (°C)	52.158
4	Heat flux W/mm2	0.5638

TABLE 5 COMPARISON OF FEA TEMPERATURE VALUES AT CUP TEMPERATURE (50.2 °C) WITHEXPERIMENTALTEMPERATURE VALUES

S.No.	Description ofPart	FEA Analysis Temperature (°C)	Experimental Temperature(°C)
1	Cup	51.363	50.2
2	Cone	52.954	51.4
3	Rollers	56.135	55

Table 5 gives the comparison of FEA temperature values at cup temperature (50.2 oC) with experimental temperature values.The difference between these two results is very low, which indicates the validity the finite element model simulation.

B. COUPLED FIELD ANALYSIS (STATIC + THERMAL)

However Taper Roller Bearings are subjected to both structural loads as well as thermal loads while it is in operation. Hence coupled field analysis is to carry out for analysis Taper Roller Bearing to find out various stresses and deformations.

Table 6 shows that the stresses obtained from thermal and static structural coupled field analysis are below the yield strength (385 MPa) and ultimate strength (530 MPa) of the material mentioned in the Table 2 and the bearing stresses are in safe limit.

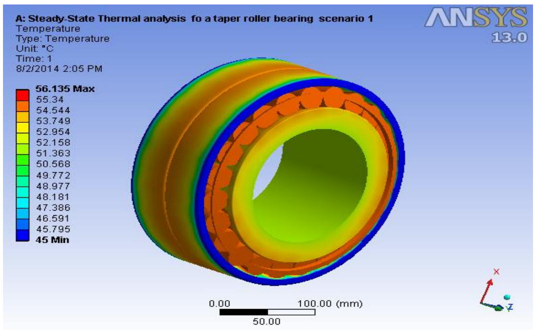


Fig. 4.a Temperature of the Taper RollerBearing

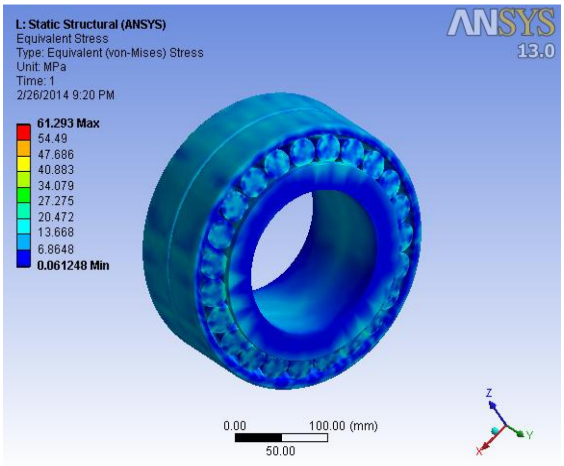


Fig. 4.b Equivalent stress of the Taper Roller Bearing

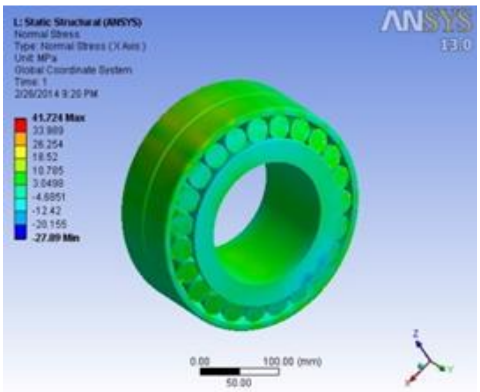


Fig. 4.c Normal Stress of the Taper Roller Bearing

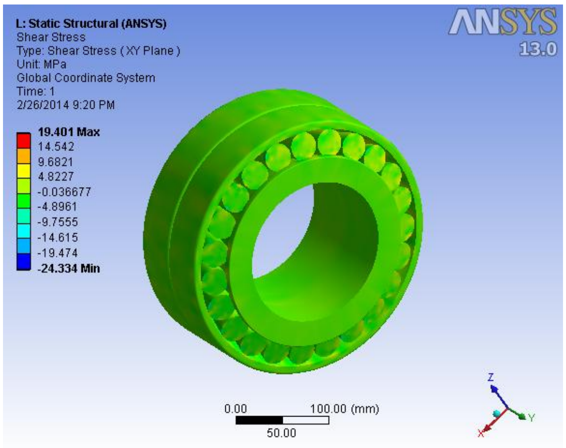


Fig. 4.d Shear Stress of the Taper Roller Bearing

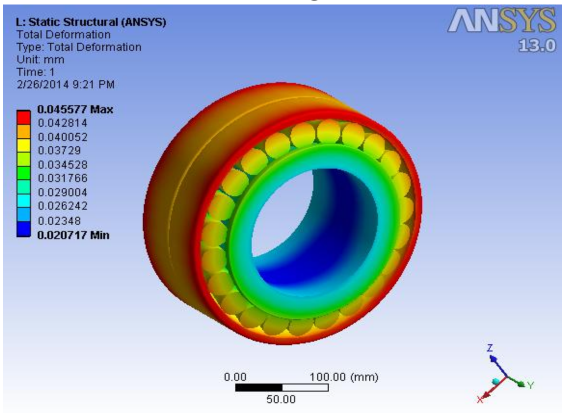


Fig. 4.e Total Deformation of the Taper Roller Bearing

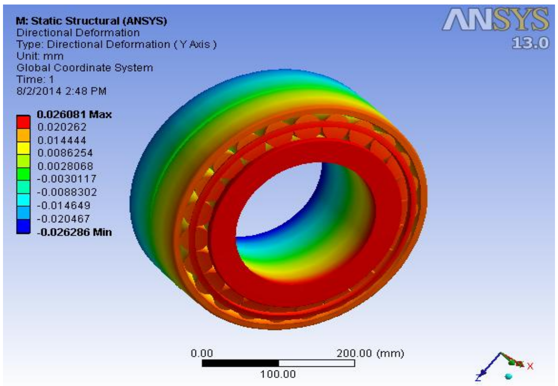


Fig. 4.f Directional deformation of the Taper Roller Bearing

TABLE 6 RESULTS OF COUPLED FIELD (THERMAL + STATIC) ANALYSIS OF THE TAPER ROLLER BEARING

S.No.	Type of stress	Maximum value obtained
1	Equivalent stress	61.293 MPa
2	Normal stress	41.724 MPa
3	Shear stress	19.401 MPa
4	Bearing Surface Temperature	50.69°C
5	Total deformation	0.045577 mm
6	Directional deformation	0.026081mm

C. COUPLED FIELD ANALYSIS RESULTS AT VARIOUS TEMPERATURES OF TAPER ROLLER BEARING

Analysis is further extended to different cup temperatures to determine the failure temperature of the Taper Roller Bearing is depicted from the Fig. 5(a) to Fig. 5(f), six simulations are made on the Taper Roller Bearing starting with the cup temperature of 130.5°C with an increment of 10°C in each step.

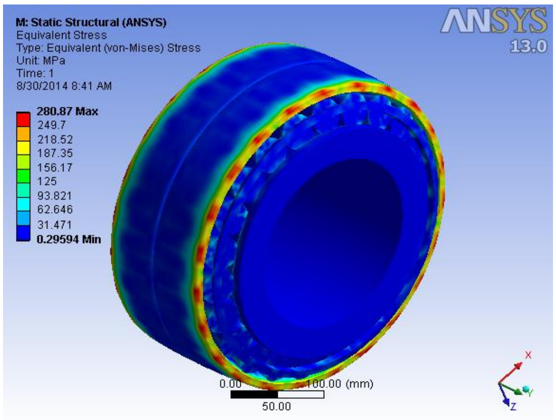


Fig.5.a Equivalent stress at 130.5°C

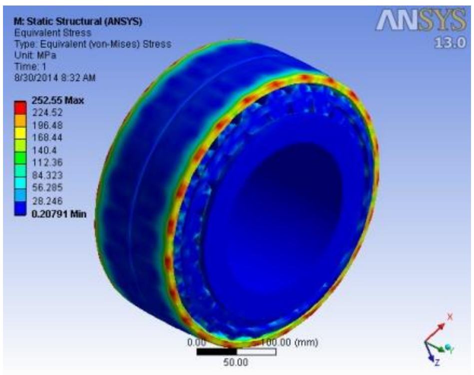


Fig. 5.b Equivalent stress at 140.5°C

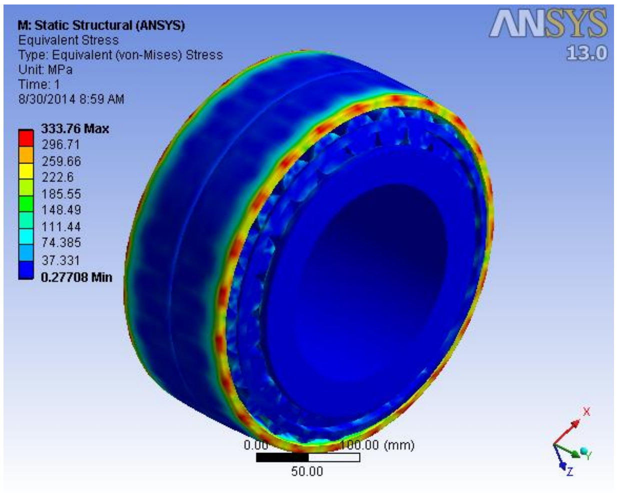


Fig. 5.c Equivalent stress at 150.5°C

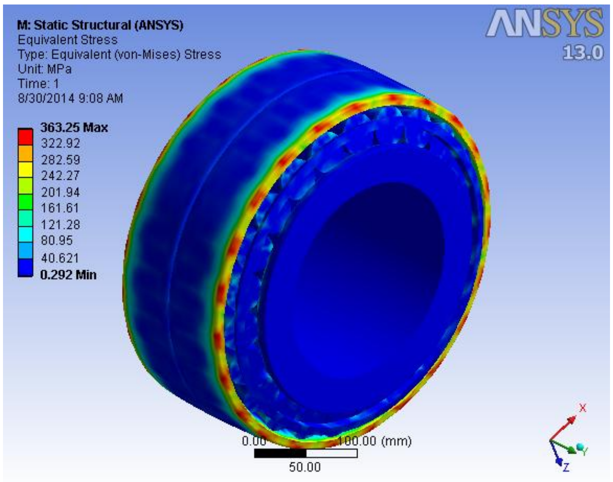


Fig. 5.d Equivalent stress at 160.5°C

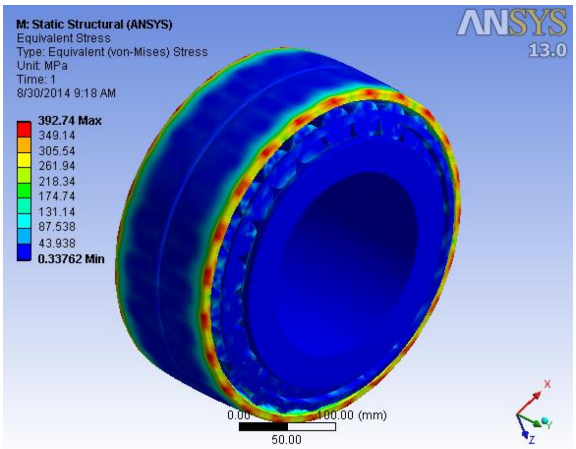


Fig. 5.e Equivalent stress at 169.5°C

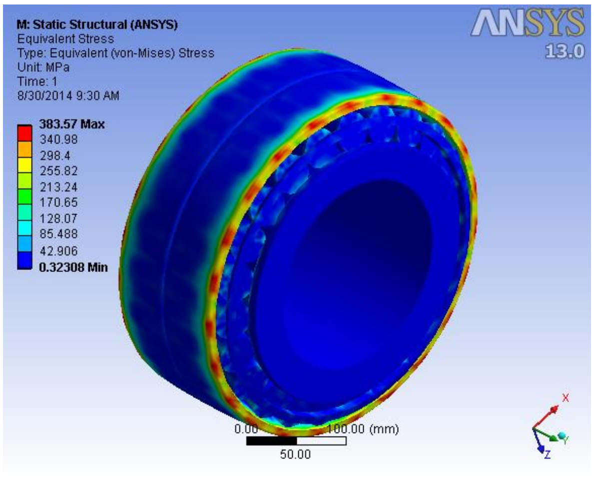


Fig. 5.f Equivalent stress at 170.5°C

TABLE 7 COUPLED FIELD ANALYSIS RESULTS AT VARIOUS TEMPERATURES OF TAPER ROLLER BEARING

S.No	Cup Temp. (°C)	Roller Temp. (°C)	Equivalent Stress MPa	Normal Stress MPa	Shear Stress MPa
1	105.5	116.98	119.68	76.254	62.487
2	130.5	149.67	252.55	258.68	116.10
3	140.5	161.36	280.87	273.74	133.75
4	150.5	172.50	333.76	325.54	159.07
5	160.5	183.63	363.25	354.54	173.24
6	169.0	190.87	383.57	375.34	183.49
7	170.5	194.77	392.74	383.53	187.42

The stresses obtained from analysis are safe up to cup temperature is 169°C. The Taper Roller Bearing may fail at 170.5°C because the equivalent stress is 392.74 MPa, more than the yield stress 385 MPa mentioned as per Table 2.

By considering the results obtained from the coupled field analysis the failure temperature of the Taper Roller Bearing is 170.5°C (Table 7), where equivalent stress exceeds the yield stress 385Mpa (Table 2). But the Transient Thermal analysis results reveal that the cup and all rollers (46 No.) in the bearing are safe even upto the temperature of 120°C. Considering the fact that most of the lubricants used in railroad bearings start to degrade when operated at temperatures above 125°C for prolonged periods. Hence the proposed HBD temperature is 120°C (earlier it is found to be 105°C from literature [3]) to remove the bearing from service.

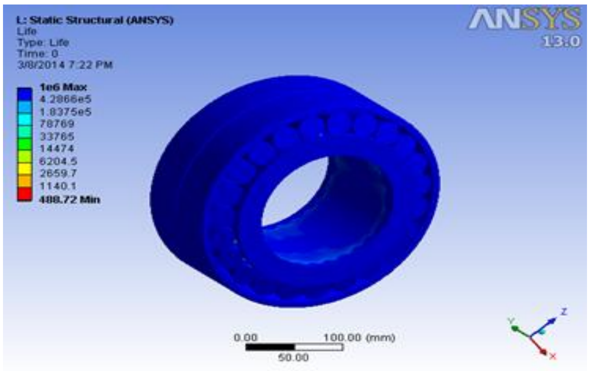
D. FATIGUE LIFE ANALYSIS OF THE TAPER ROLLER BEARING

The fatigue or endurance limit of a material is defined as the maximum amplitude of completely reversed stresses that the standard specimen can sustain for an unlimited number of cycles without fatigue failure. Since the fatigue test cannot be conducted for unlimited or infinite number of cycles, 10^6 cycles is considered as a sufficient number of cycles.

For Fatigue analysis heat conduction coefficient and overall heat transfer coefficients are considered as boundary conditions. For the taper roller bearing, AISI 8620 steel with a thermal conductivity of 46.6 $Wm^{-1}K^{-1}$ is used. Convection and radiation boundary conditions for the bearing are the overall heat transfer coefficient in the appropriate units (h_o = 65.9 W m^{-2} K^{-1}) and radiation to an ambient at a temperature of 25°C. Heat flow is applied to the circumferential surface of the roller is 11.5W.

The boundary conditions are taken at cylindrical support (means axial movement is free, radial and tangential movements are fixed) and fully reversal load 159 KN is taken at center of gravity of the bearing. The stresses obtained from the coupled field analysis are input load for Fatigue analysis.

Fig. 6.(a) Fatigue life of the Taper Roller Bearing



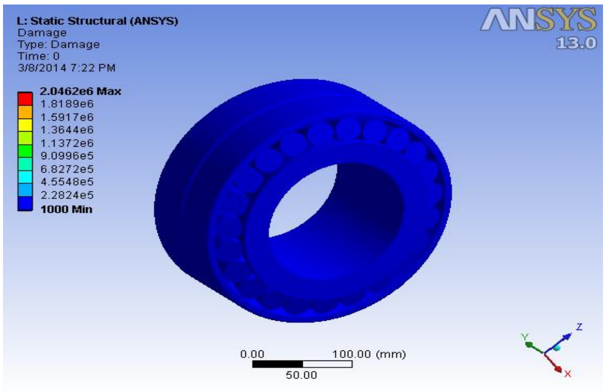


Fig. 6.(b) Damage of the Taper Roller Bearing

Table 8 shows the Fatigue analysis results of the Taper Roller bearing. The result shows that the bearing can withstand the cyclic loads up to 2.04e6 cycles which is beyond the infinite life of 10^6 cycles. Hence the Taper Roller bearing is safe for infinite number of cycles.

TABLE 8 FATIGUE ANALYSIS RESULTS OF TAPER ROLLER BEARING

S.No.	Parameters	Value
1	Maximum Design Life	1.e+006 cycles
2	Maximum Damage	2.0462e+006 cycles

VII. CONCLUSIONS

The model is validated by comparing the Finite Element results with the experimentally and theoretically obtained temperatures from the literature. The comparison of the temperature values revealed that the present results agreed with a difference of 3%.

Coupled field analysis (thermal + static) have been carried out on the Taper Roller bearing. The stresses obtained from thermal and static structural coupled field analysis are below the yield strength 385 MPa and ultimate strength 530 MPa as per designer specifications. So the Taper Roller bearing stresses are in safe limit.

The results obtained from Transient Thermal analysis of a taper roller bearing by varying temperature (25°C – 135.33°C) with respect to time 12 hours, the maximum load applied is 159 KN radially. The stresses obtained from Transient thermal and static structural coupled field analysis are below the yield strength 385 MPa and ultimate strength 530 MPa as per the designer specifications. So the bearing stresses are in safe limit. Though the temperatures are higher than the Hot Box Detector (HBD) temperature (105.5°C) the stresses are within the safe limit. Hence one can set the HBD temperature to 120°C (earlier it is 105°C).

Fatigue life analysis of the Taper Roller bearing shows that the bearing has infinite life when the stress is below 86.2 MPa. Since the maximum stress in Taper Roller Bearing is 61.293 MPa (equivalent stress from coupled field analysis), the Taper Roller bearing has infinite life.

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