

Fatigue, Modal and Transient Analysis on the Tapered-Roller Bearing

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ABSTRACT: The troubled bearings in service are primarily identified using wayside hot-box detectors (HBDs). Several set-out bearings have no visible damage but reveals discoloration of rollers within a cone inner race assembly. Subsequent laboratory experiments were performed to determine a minimum temperature and environment necessary to reproduce these discolorations and concluded that the discoloration is most likely due to roller temperatures greater than 232°C (450°F) for periods of at least 4hours due to fatigue. The purpose of the present work is to determine fatigue life, vibration characteristics and the dynamic response of a structure under the action of time-dependent loads.

KEYWORDS: Taper Roller Bearing, Fatigue, Transient Analysis, HBD.

I. INTRODUCTION

Tapered-roller bearings are the most widely used bearings in automobile and locomotives. When operated under satisfactory load, alignment and contaminant free conditions, the bearing service life is extremely long. As a general rule, bearings will continue the wheel life, and survive several reconditioning cycles prior to being discharged. 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°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 is expected 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°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

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tapered roller bearing, CATIA V5 is used. The FE model is utilized to 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 circumstances, 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 within 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. Moreover, 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. FINITE ELEMENT ANALYSIS

The main objective of the work presented here is to study how roller heating affects the temperature of the bearing assembly components, especially the external surface of the cup, which is the part of the bearing scanned by the infrared wayside hot-box detectors. To this end, a FE model was created and utilized to run several cup heating simulations in order to attain solutions. This study would be very challenging to perform experimentally due to the constraints imposed by placing conventional thermal sensors inside a rotating assembly. The following two subsections, Bearing Modeling and Boundary Conditions, provide a detailed description of the developed FE model including the boundary conditions used.

IV. BEARING MODELING

A solid model of the bearing-axle assembly is produced using the three-dimensional graphing software CATIA V5 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 that simplified 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 was done because the thermal resistances of the polyamide cages and grease are large compared to the rest of the bearing

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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 this 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 was 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 was obtained by first estimating the initial contact area of the rollers with the cup and cones using the Hertzian line contact theory (Brandlein et al. [13] and Harris and Kotzalas. [14]), 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 was then used to calculate the amount of roller compression.

V. BOUNDARY CONDITIONS

The validity of the FE model depends significantly on the correctness of the Boundary Conditions applied while running the simulations. In this regard, the Boundary Conditions used for this study were derived from previously conducted experimental work by Tarawneh et al. [2] and [3] and from material specifications provided by the bearing manufacturer. Four major Boundary Conditions namely conduction, convection, radiation, and heat flux are utilized.

The heat conduction coefficients for the bearing assembly were provided by the bearing manufacturer. The bearing steel, AISI 8620 with a thermal conductivity of $46.6 \text{ Wm}^{-1}\text{K}^{-1}$ is used. A sensitivity analysis was 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 were changed by $\pm 10\%$. Note that the thermal conductivity values for AISI 8620 do not change by more than 7% over the range of temperatures reported in this study.

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

Finally, to simulate heat generation within the bearing assembly, heat flux was applied to the circumferential surface of the rollers. The appropriate heat flux value was 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 was achieved. The acquired heat input per roller was 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). To illustrate this, consider an abnormal operation condition in which debris gets wedged between a roller and an adjacent cage bar causing the roller to become fully or partial jammed and resulting in excessive sliding friction between the roller and the cup and cone raceways. Since the mass of the roller is very small compared to the mass of the cup and cone, it is safe to assume that it will heat at a much faster rate than the other two components, thus, becoming the heat source.

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VI. FE MODEL VALIDATION

Dynamic tests were performed in a laboratory with an ambient temperature of 25°C utilizing a dynamic bearing tester that maintained a speed setting of 536 rpm corresponding to a train traveling with a velocity of approximately 91.7 km/h (57 mph) at full load (which corresponds to a force of 159 KN (35,750 lb) per bearing).

Normal operation simulates a bearing running under normal operating conditions. This model represents a healthy bearing with an average cup temperature of approximately 50°C (this temperature is the average of six thermocouple readings positioned 60° around the circumference of the cup at the middle of the bearing). The aforementioned temperature was acquired from the dynamic testing.

To validate and quantify the accuracy of the results acquired from the FE model, both an experimental and a theoretical technique were employed. The experimental method required the use of a dynamic bearing tester powered by a 22 kW (30 hp) motor that is managed by a smart controller. The controller has outputs that can be monitored via a data acquisition system to record the axle speed and the power required to run the bearings. In an experiment aimed at replicating the normal operation of bearings, four fully assembled healthy class K bearings were run utilizing the aforementioned dynamic bearing tester at a speed of 91.7 km/h (57 mph) with a full load.

VII. BEARING MATERIAL

PHYSICAL PROPERTIES

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 following tables 1, 2 and 3 show the chemical composition, thermal properties and mechanical properties of AISI 8620 alloy steel.

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	≤ 0.0400
Phosphorous, P	≤ 0.0350

TABLE 2 THERMAL PROPERTIES OF TAPER ROLLER BEARING

PROPERTIES	VALUE
Thermal expansion co-efficient	11 μm/m°C
Thermal conductivity	46.6 W/Mk

TABLE 3 MECHANICAL PROPERTIES OF TAPER ROLLER BEARING

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

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VIII. FINITE ELEMENT MODELING AND ANALYSIS OF TAPER ROLLER BEARING

The three dimensional model for the Taper Roller Bearing assembly as per the specifications is generated by using CATIA V5 for the Finite Element analysis in ANSYS 13.0. An axle is rendered as a simple cylinder with a 0.1572 m diameter and 2.2 m length, which is sufficient in size to act as a heat sink as per the designer or manufacturer specifications. The contact area between each roller and the cup and cone raceways is 123.96 mm², which represents 3.68% of the total surface area of the roller. The Taper Roller Bearing is modeled as a class K railroad tapered-roller bearing with some minor modifications that simplified the geometry and results in a significant reduction to the computational time. An assumption, cages, seals, wear rings, or grease are not included in the model is made 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 of Taper Roller Bearing.

The contact low seals constitute 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 an axle, which constitutes a very large metal body and acts as a heat sink. Hence, the omission of the aforementioned components from the Finite Element model will not have a significant effect on the results obtained from the present work.

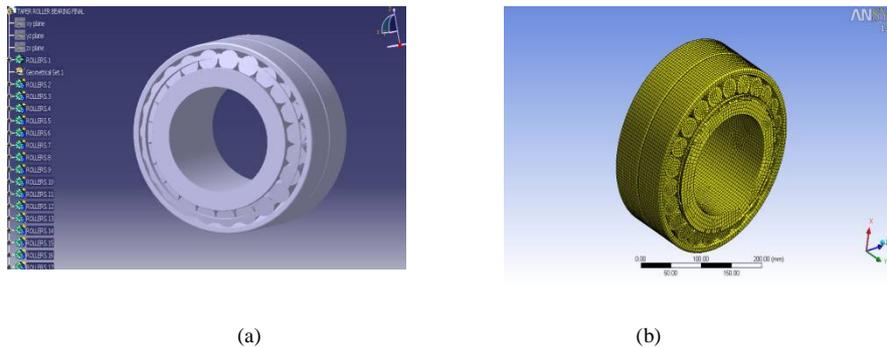


Fig.1(a)Taper Roller Bearing model inCATIA V5, (b) Meshed model of Taper Roller Bearing

The finite element method is to divide the structure or solution region into finite subdivisions or elements. Hence the structure is to be modeled with suitable finite elements. The number, type, size and arrangement of the elements are to be decided by the designer (Table 4).

TABLE 4 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

Solid model of the Taper Roller Bearing assembly for the Finite Element analysis conducted for this study is depicted in Fig. 1(a) and (b) shows the meshed model of the taper Roller bearing. The major boundary conditions are utilized namely: conduction, convection, radiation, and heat flux. Note: Heat flux is applied to the circumferential surface of the rollers only, Bearing overall heat transfer coefficient is applied to the bearing external surfaces (i.e., bearing cup external surface and side walls, and bearing cone side walls).

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IX. FATIGUE LIFE ANALYSIS OF THE TAPER ROLLER BEARING

Fatigue failure is caused by repeated cyclic loading. In practice, loads significantly below static limits can cause failure if the load is repeated for sufficient number of cycles. Characterizing the capability of a material to survive that many cycles a component may experience during its lifetime is the aim of fatigue analysis.

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 the boundary conditions are heat conduction coefficient and overall heat transfer coefficients are considered. For the taper roller bearing, AISI 8620 steel with a thermal conductivity of $46.6 \text{ Wm}^{-1} \text{ K}^{-1}$ is used. Convection and radiation boundary conditions for the bearing are the overall heat transfer coefficient in the appropriate units ($h_o = 65.9 \text{ W m}^{-2} \text{ K}^{-1}$) 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 coupled field result stresses are input load for Fatigue analysis. The fatigue life and damage obtained in the ANSYS are depicted in the figure 2 (a) and (b).

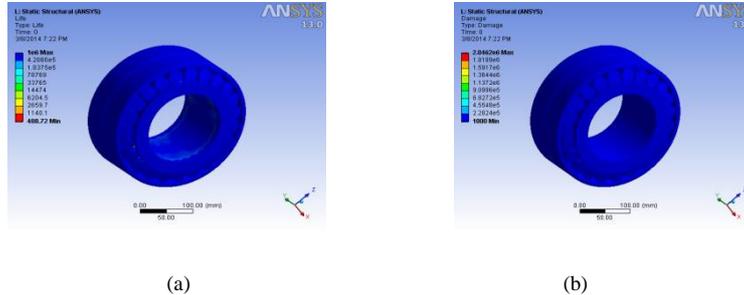


Fig. 2 (a) Fatigue life (b) Damage of the Taper Roller Bearing

Table 5 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 106 cycles. Hence the Taper Roller bearing is safe for infinite number of cycles.

TABLE 5 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

Under normal working conditions, Taper Roller Bearings are subjected to vibrations. In order to avoid damage due to resonance, Modal analysis is carried out on Taper Roller Bearing which is discussed in the next section.

X. MODAL ANALYSIS OF THE TAPER ROLLER BEARING

Modal analysis is used to determine the natural frequencies and mode shapes of a structure. The natural frequencies and mode shapes are important parameters in the design of a structure for dynamic loading conditions. A structure can have any number of modes up to the number of degree of freedom (DOF) in the model. Mode means the pair of one natural frequency and corresponding mode shape. Mode shape means in what shape the structure would tend to vibrate at each frequency.

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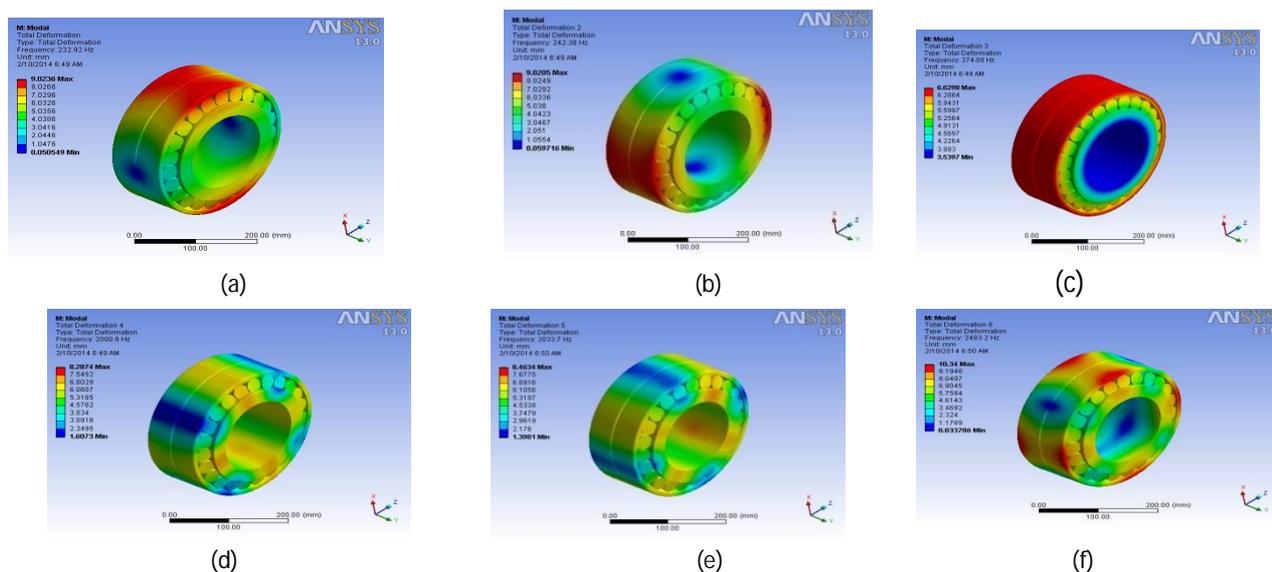


Fig. 3 Mode shapes of the Taper Roller Bearing (a) Mode 1 (b) Mode 2 (c) Mode 3 (d) Mode 4 (e) Mode 5 (f) Mode 6

The modal analysis of the taper roller bearing, different mode shapes are shown from Fig. 3(a) to 3(f). The boundary condition for modal analysis is cylindrical support (means Axial movement is free, Radial and Tangential movements are fixed); that is the rolling effect is apply for this analysis.

The free vibration of bearing without loading condition is 232.92 Hz. Consider the model is cylindrical. Modal analysis follows the Block Lanczos method. The modes are taken into the account is six. After coupling the modal analysis the frequencies are respectively 232.92, 242.38, 274.09, 2000.6, 2033.7, 2483.2 Hz. The bearing working frequency is 9Hz (536/60). The obtained natural frequency of bearing is 232.92Hz. So there is no chance to occur resonance because working frequency is less than the natural frequency. The component is safe for dynamic loading.

TABLE 6 RESULTS OF MODAL ANALYSIS OF THE TAPER ROLLER BEARING

MODE NO.	FREQUENCY(Hz)	MAXIMUM DEFORMATION IN MM
1	232.98	9.0236
2	242.38	9.0205
3	274.89	6.6298
4	2000.6	8.2874
5	2033.7	8.4634
6	2483.2	10.34

Table 6 shows the natural frequencies and maximum deformation corresponding to six mode shapes of Taper Roller bearing. To determine the dynamic response of the system under any type of time-varying temperature load transient thermal analysis is needed to evaluate the response of deformable bodies and is discussed in the next section.

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XI. TRANSIENT THERMAL ANALYSIS OF THE TAPER ROLLER BEARING

Transient dynamic analysis is a technique used to determine the dynamic response of a structure under the action of any general time-dependent loads. This type of analysis is to determine the time-varying displacements, strains, stresses and forces in a structure as it responds to any combination of static, transient and harmonic loads.

The boundary conditions considered for Transient thermal analysis already mentioned earlier. The total time considered for the transient thermal analysis is 12 hours with a step size of an hour, assuming ambient temperature to be 25°C. Table 7 shows the variation of temperature loads of cup and rollers of the Taper Roller Bearing with respect to time for Transient thermal analysis. Fig. 4(a),4(b) and 4(c) represents the failure temperature, total heat flux and directional heat flux of the taper roller bearing.

TABLE 7 TRANSIENT THERMAL INPUT TEMPERATURE OF CUP AND ROLLER

S.No.	STEPS	TIME [SEC]	TEMPERATURE OF CUP [°C]	TEMPERATURE ROLLERS [°C]
	1	0	25	25
1		1800	48.474	130.1
2	2	3600	48.51	130.3
3	3	7200	48.546	130.5
4	4	10800	48.582	130.7
5	5	14400	48.618	130.9
6	6	18000	48.654	131.1
7	7	21600	48.69	131.3
8	8	25200	48.726	131.5
9	9	28800	48.761	131.7
10	10	32400	48.797	131.9
11	11	36000	48.833	132.1
12	12	39600	48.868	132.3
13	13	43200	48.95	134.4

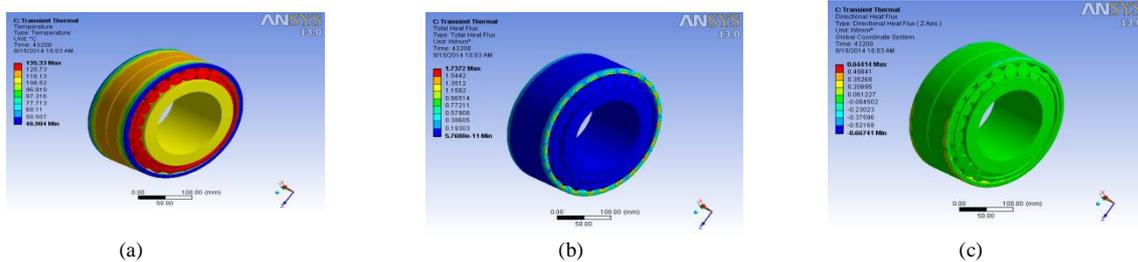


Fig. 4(a)Temperature (b) Total Heat Flux (c) Directional Heat Fluxof the Taper Roller Bearing

TABLE 8 RESULT OF TRANSIENT THERMAL ANALYSIS OF THE TAPER ROLLER BEARING

S.No.	PARAMETER	VALUE
1	Temperature of cup °C	120
2	Temperature of Roller °C	132.4
3	Temperature of cone °C	116
4	Total heat flux w/mm ²	1.7372
5	Directional Heat flux w/mm ²	0.64414

Table 8 shows the results of Transient Thermal analysis of Taper Roller bearing. The operating conditions that would lead to failure of the taper roller bearing is a bearing cup temperature of 120°C. But the HBD would trigger alarm at 105.5°C and the bearings are removed from the service. By considering the results obtained from the coupled

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field analysis the failure temperature of the Taper Roller Bearing is 170.5°C [15], where equivalent stress exceeds the yield stress 385Mpa (Table 3). But the Transient Thermal analysis results reveal that the cup and all rollers (46 No.) in the bearing are safe even up to 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 105°C) to remove the bearing from service.

XII. TRANSIENT THERMAL ANALYSIS WITH STATIC LOAD OF THE 159 KN ON TAPER ROLLER BEARING

The coupled field analysis (thermal transient and static structural analysis) of a taper roller bearing is carried out. The boundary conditions for transient thermal analysis with a static load are heat conduction coefficient and overall heat transfer coefficients are considered. For the bearing steel, AISI 8620 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 11.5 W and the varying temperature loads are shown in table 7. Ambient temperature is 25°C, Time in hours = 12, Step size = 1 hour. The boundary conditions are taken at cylindrical support (means axial movement is free, radial and Tangential movements are fixed) and structural load 159 KN is taken at center of gravity of the bearing. Fig5(a) to 5(d) shows the Transient Thermal Analysis results.

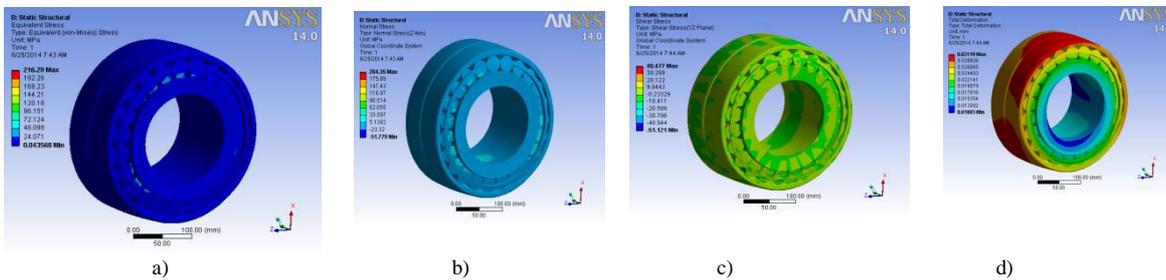


Fig. 5(a) Equivalent Stress (b) Normal Stress (c) Shear Stress (d) Total Deformation of the Taper Roller Bearing

TABLE 9 RESULT OF TRANSIENT COUPLED FIELD ANALYSIS OF TAPER ROLLER BEARING

S.No.	TYPE OF STRESS	VALUE OBTAINED
1	Equivalent stress MPa	216.29
2	Normal Stress MPa	204.35
3	Shear Stress MPa	40.477
4	Total Deformation mm	0.03119

Table 9 shows the results of Transient coupled field analysis of the Taper Roller bearing. 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 (Table 3). So the Taper Roller Bearing stresses are in the safe limit.

XIII. CONCLUSIONS

Taper Roller Bearings are aimed to withstand both axial and radial loads which are widely used in automobiles and rail road industry. Even one percent of life gaining leads to reduction in the maintenance cost of the component. The purpose of the present work is to evaluate the Hot Box Detector (HBD) temperature by demonstrating that the bearing is safe beyond the HBD temperature and finding the failure temperature and stresses under cyclic loading also. The conclusions derived from the present work are discussed.

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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.

Modal analysis of the Taper Roller bearing shows that the natural frequency of designed Taper Roller bearing is 232.92Hz. The bearing working frequency is 9Hz (536/60). So there is no chance to occur resonance because working frequency is less than the natural frequency. The component is safe for dynamic loading.

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).

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