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Finite Element Solution for Thermal Analysis of NiTiNOL- 60 Ball Bearing

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ABSTRACT:
Ball bearing produce heat due to frictional forces and this causes the temperature to rise and induce thermal stresses in the bearing. Finite element method was used to investigative the thermal behaviour the ball bearing. Bearing was analyzed for the resulting thermal stresses and deformation and compares to the strength of the NiTinol – 60 material used for construction of ball bearing . the stresses were found to be in the safe resigon.

1. INTRODUCTION:
Hard, Corrosion-Proof NickelTitanium Material for Use in Mechanical Components Shock-resistant material eliminates corrosion and polishes to a smooth surface finish NASA's Glenn Research Center has developed a new method for producing a shock- and corrosion-proof superelastic intermetallic material — NiTiNOL 60 (60NiTi) — for use in ball bearings and other mechanical components. These superelastic materials can withstand tremendous loads and stresses without permanent deformation or denting. At the same time, the nickel-titanium alloy is immune to corrosion and rust, unlike mechanical components made from iron or steel. In addition, the material does not chemically degrade or break down lubricants, a common problem with existing bearing materials. This material is best suited for oil lubricated rolling and sliding contact applications requiring superior and intrinsic corrosion resistance, electrical conductivity, and non-magnetic properties. The goal was to develop high-temperature, nonmagnetic alloys for missile cone applications. The research identified two compositions: NiTiNOL 55 (55Ni-45Ti) And NiTiNOL 60 (60Ni-40Ti), In weight percent. Because and NiTiNOL 60 was difficult to cast and hot work with NiTiNOL 60 was suspended. Puris developed patent-pending processing methods and chemistry control around the base NiTiNOL alloy to create SM-100 – the only commercially available variant of NiTiNOL 60

2. MATERIALS AND METHODS:
Based on the literature review, the design and process parameters were selected. The steady state thermal solution was obtained. The static structural solutions were obtained. Figure 2 shows the flow chart for thermal and structural solutions.

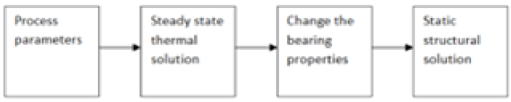

Figure 2. Flow chart for thermal and structural solution.

2.1 NiTiNOL -60 (2215EkN9) LIFE CALCULATE
2.1.1 Basic Rating Life and SKF Rating Life
An SKF Self – Aligning ball Bearing – 2215EKN9 is to Operate at 1000 R/Min under a constant radial Load F_r =15KN. Oil Lubrication is to be used, the oil has an actual kinematic viscosity(V) = 250 MM^2 /s at normal operating Temperature.

The desired reliability is 90% and it is assumed that the operating conditions are very clean.

a). The basic rating life 90% reliability is

$$L_{10} = (c/p)^3$$

From the product table for bearing 2215EKN9, C = 58.5 KN since the load is purely radial, $P=F_r=15$ kN(Equivalents dynamics bearing load) .

* In million revolutions

$$L_{10} = (58.5/15)^3$$
$$=59.3$$

* Or in operating hours

$$L_{10h} = (10^6/60xn)x\ L_{10}$$
$$= (1000000/60000)X59.3$$
$$= 988.3$$

Table 1. Life adjustment factor a_1

s.no.	Reliability(%)	Failure Probability (η)	SKF rating life (L_{nm})	Factor(a_1)
1.	90	10	L_{10m}	1
2.	95	5	L_{5m}	0,64
3.	96	4	L_{4m}	0,55
4.	97	3	L_{3m}	0,47
5.	98	2	L_{2m}	0,37
6.	99	1	L_{1m}	0,25

b). The SKF rating life for 90%reliability is

$$L_{10}m = a_1 a_{skf} L_{10}$$

As a reliability of 90% is required, the L10m life is to be calculated and a_1=1(table1) from the product table for bearing2215EKN9,

$$D_m =0.5\ [d+D]$$
$$=0.5[75+130]$$
$$=102.5$$

The Rated oil viscosity at operating temperature for a speed of 1000 r/min,

$$V_1 = 11.5 mm^2/s$$
$$K = v/v$$
$$= 250/11.5$$
$$= 21.73$$

TABLE:2 Guideline values for factor η_c for different level of contamination

Conditions	Factor η_c for bearings with diameter dm < 100 dm ≥ 100 mm
Extreme cleanliness Particle size of the order of the lubricant film thickness Laboratory conditions	1 1
High cleanliness Oil filtered through an extremely fine filter Typical conditions: sealed bearings that are greased for life	0,8 ... 0,6 0,9 ... 0,8
Normal cleanliness Oil filtered through a fine filter Typical conditions: shielded bearings that are greased for life	0,6 ... 0,5 0,8 ... 0,6
Slight contamination Typical conditions: bearings without integral seals, coarse filtering, wear particles and slight ingress of contaminants	0,5 ... 0,3 0,6 ... 0,4

From the Product table p_u = 1.1K.N and P_u/p = 1.1/15 0.073.As the conditions are very clean, N_c = 0.85 (table 2) and N_c (P_u/P) = 0.85 (0.073) With K = 21.73 and ising the S K F Explorer Scale in Table 2, the value of a_{SKF} = 2.15 is obtained. Then according to the SKF rating life Eqation, in million revolutions

$$L_{10m} = 1\times 2.15x59.3$$
$$= 127.4$$

Or in operating hours using,

$$L_{10MH} = (10^6/60n)xL_{10m}$$
$$L_{10MH} = (1000000/60000\ )\ X127.4$$
$$= 2123.3$$

Same as calculated 1100,1200,1300,1400,1500 .

TABLE:3 Bearing Life Calculate for 90% Reliability

S. No	Parameters	1000	1100	1200	1300	1400	1500
1	Basic Rating Life in Million Resolutions	59.3	59.3	59.3	59.3	59.3	59.3
2	Basic Rating in Hours	988.3	898.4	823.96	760.2	705.9	658.8
3	SKF Rating Life in million resolution	127.4	127.4	127.4	127.4	127.4	127.4
4	Basic Rating life in Hours	2123.3	1930.3	1769.4	1633.3	1516.6	1415.4

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1. HEAT GENERATION IN THE BEARING

The major source of heat generation is the machining process and the friction between the balls and the races. The major portion of the heat is taken away by the coolant and the chips. In ball bearings heat is generated by three sources. First is the load related heat generation, second source is the viscous shear of lubricants between the solid bodies, known as viscous heat dissipation. The third source of heat is known as spin related heat generation. Considering this, analytical formulation for heat generation in a bearing was developed. The heat generated in a bearing is given as

$$H_f = 1x10\text{-}4.\ n.\ M$$

where, Hf is the heat generation due to friction in Watts, n is the rotational speed (rpm), M is the total frictional torque (N mm). Rotational speeds of 1000 ,1100, 1200, 1300,1400,1500, were taken and the total frictional torque as 100 N-mm. The internal heat generation can be calculated by using the formula, Internal Heat Generation = H_f / V. The volume of the Ball bearing was calculated as 2095mm^3. The values were tabulated and the internal heat generation was calculate The internal heat generation for different speeds is shown in Table 4.

TABLE : 4 Internal Heat Generation for Different Speed

S. No	Speed (RPM)	Heat Generation (W)	Volume of Bearings MM3	Heat Generated (NXMX10-4)	Heat Generation Unit Volume
1	1000	280 N-MM	191647	28	1.461X10-4
2	1100	286 N-MM	191647	31.46	1.619X10-4
3	1200	292 N –MM	191647	35.04	1.828X10-4
4	1300	298 N –MM	191647	37.96	1.980X10-4
5	1400	304 N-MM	191647	42.56	2.220X10-4
6	1500	310 N-MM	191647	46.5	2.42X10-4

4 . RESULTS AND DISCUSSION

4.1 Finite Element Simulation:

The bearing consists of Nitinol 60 (2215EKTN9) balls of Volume 2095mm^3.

The outer and inner diameter are D = 130 mm, d = 75 mm respectively, B = 15.8mm, a_0 = 100^O and Z = 34 balls. The bearing operates under dynamic load rating of C= 58.5kN and static load rating of C_0 = 15 kN and at a rotational speed of 1000-**1**500rpm The bearing CAD model creates in SOLID WORKS was imported in geometry part of ANSYS workbench .The study state thermal analysis was formed on the bearing .The mechanical and thermal properties of NITINOL – 60 material was generated in the Engineering Data part of anysis work bench. The model was mashed and the detiels of mash created as under. Number of nodes :73225 and number elements:21098 the mashed model as in given following figure 3 shows the imported data in ANSYS and figure 4 shows the meshing of the assembly.

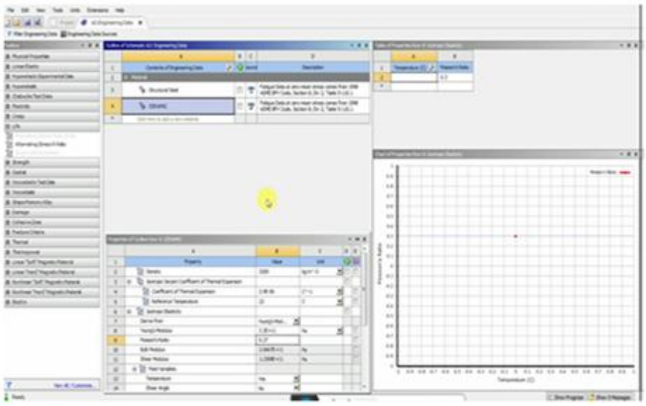

Figure 3. Importing the data in ANSYS

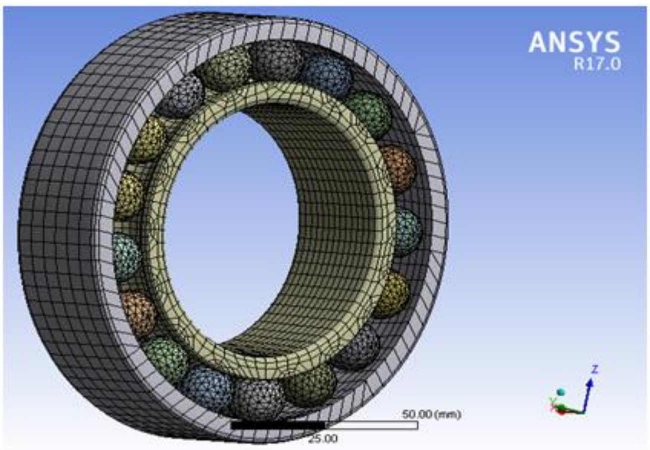

Figure 4. Meshing of the assembly.

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4.1.2Thermal Inputs:

The internal heat generated the bearing due to frictional forces was completed at different speeds of raising from 1000- 1500 rpm . The heat generated was applied on the inner ring of the bearing which ultimately transport to the balls and outer ring. The heat generated due to frictional process as discussed above is removed partial by convection, heat transfer through the all the surface exposes to the environment. Hence convection coefficient of 15 W/M^2 at 20^{0}c was applied on the outer ring bearing which same for all speeds.

4.1.3 Thermal Outputs:

The resulting temperatures distbustion in the bearing due to frictional heating and convection heat transfer are analyzed in the ANSYS work bench at different speeds raising from 1000-1500 rpm .The total heat flux generated due to above heat transfer phenomenon in the bearing the total heat flux and directional heat flux values (X ,Y,Z) generated due to above heat transfer phenomenon are also analyzed.

4.1.4 Structural Analysis Output:

The resulting temperature profile accress the bearing surfaces generated study state thermal analysis was improted into static structural analysis in ANSYS workbench .The resulting thermal stree, strain , deformation, were analyzed at different speed..

Table:5 Simulation Results

S. No	Rotational Speed (RPM)	Temperature (K)		Total Heat Flux (W/mm2)		Total Deformation (mm)		Equivalent Stress (H bar)		Equivalent Elastic Strain (mm/mm)	
		Inner Ring	Outer Ring	Inner Ring	Outer Ring	Inner Ring	Outer Ring	Inner Ring	Outer Ring	Inner Ring	Outer Ring
1	*1000*	415	412	0.019	0.0022	0.040	0.031	77.84	9.46	0.003	0.0005
2	*1100*	436	433	0.023	0.0026	0.033	0.030	26.64	3.23	0.004	0.0007
3	*1200*	454	449	0.025	0.0031	0.038	0.003	33.21	4.53	0.005	0.0008
4	*1300*	470	465	0.031	0.0035	0.042	0.001	36.76	5.01	0.006	0.0016
5	*1400*	488	482	0.034	0.0038	0.046	0.015	45.30	5.50	0.007	0.0014
6	*1500*	507	502	0.035	0.0039	0.058	0.045	98.3	13.44	0.010	0.0015

Figuers 5-9 Indicates the temperature profile, heat flux, total deformation, equivalent stress and equivalent strain respectively at 1000 rpm .Figures 10-14 temperature profile, heat flux, total deformation, equivalent stress and equivalent strain respectively at 1500 rpm. The temperature, total heat flux, total deformation, equivalent stress and equivalent elastic strain values were taken from the simulated models with respect to rotational speed and were tabulated for obtaining the heat generation rate in the ball bearing. The heat generation was calculated. The simulation results are shown in Table 5. Figure 15 shows the rotational speed (rpm) vs. Temperature plot for various range of speeds. Figure 16 represents the rotational speed (rpm) vs. total heat flux (W/mm^2) plot for different speeds. Figures 17 and 18 shows the plot of rotational speed (rpm) vs. equivalent stress (MPa) and equivalent strain.Figure 19 indicates the rotational speed (rpm)vs . total deformation (mm) plot. respectively.

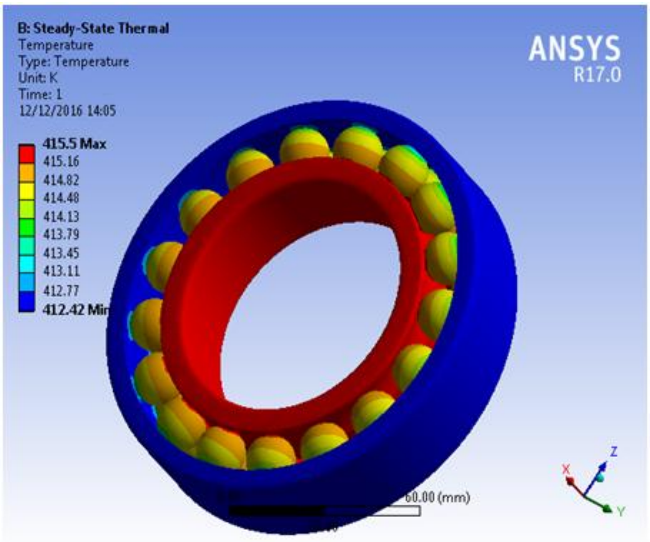

Figure 5. Temperature distribution(1000rpm).

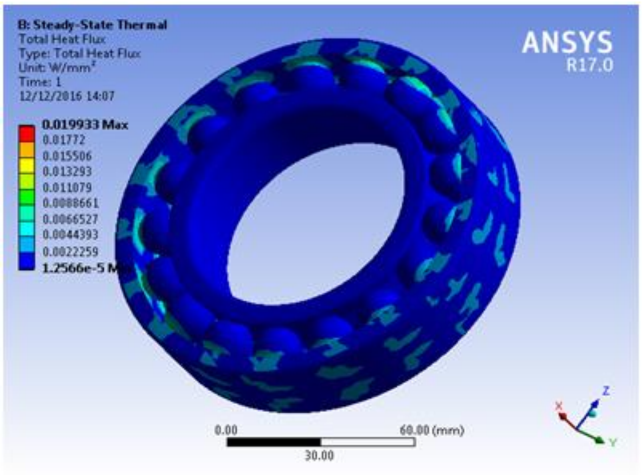

Figure 6 .Total heat flux(1000rpm).

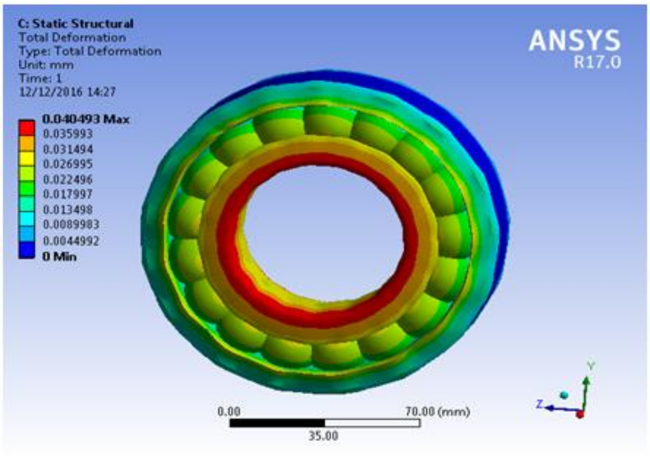

Figure 7. Total deformation (1000rpm).

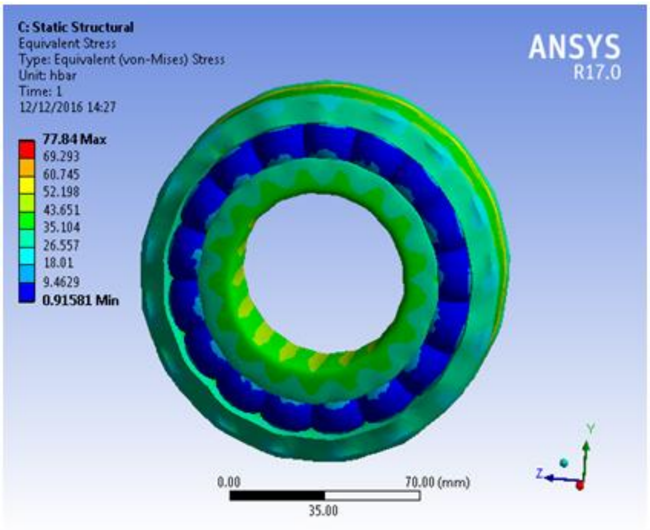

Figure 8. Equivalent stress (1000rpm).

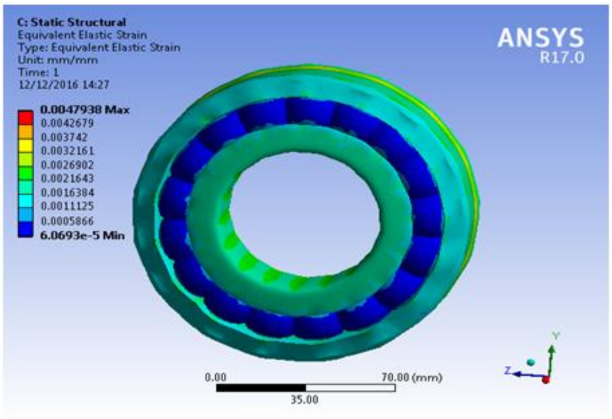

Figure 9 . Equivalent strain(1000rpm).

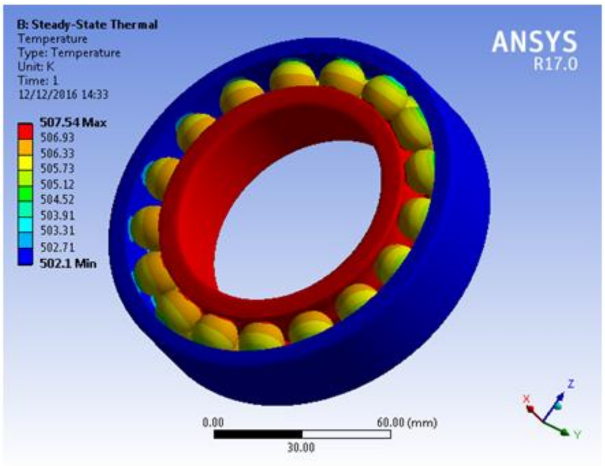

Figure 10. Temperature distribution(1500rpm)

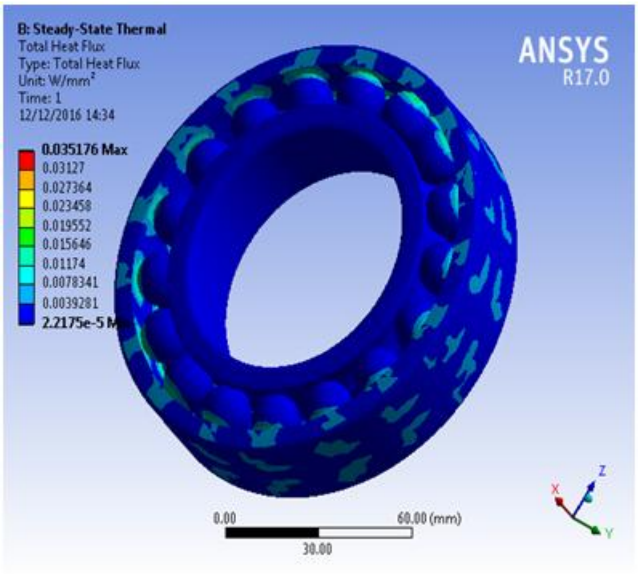

Figure 11 .Total heat flux(1500rpm).

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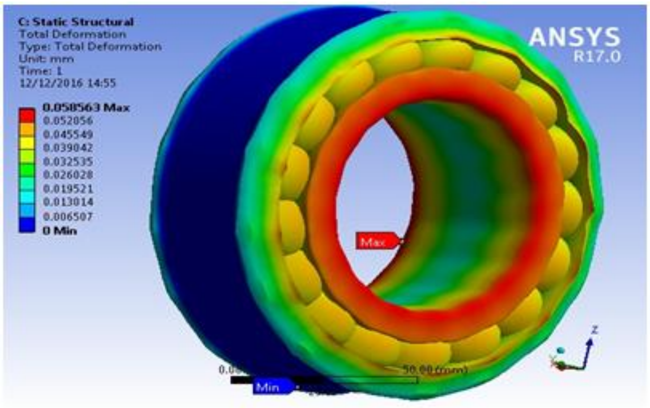

Figure 12 .Total deformation(1500rpm).

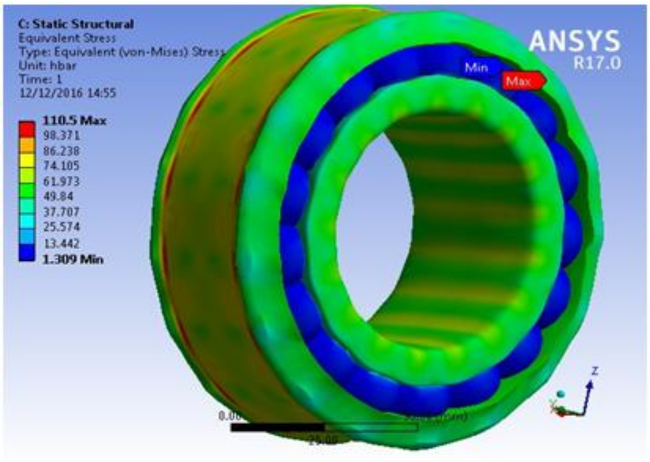

Figure 13.Equivalent stress (1500rpm).

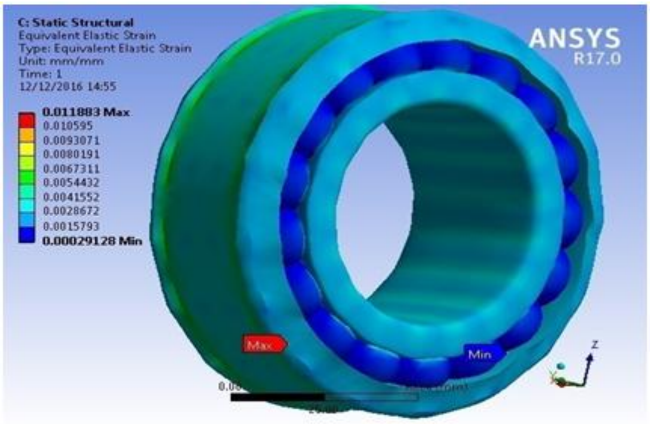

Figure 14 .Equivalent strain (1500rpm).

In the numerical analysis temperature distribution was measured for a series of rotational speeds. The heat generation is due to the torque developed. Here two types of torques were considered, one is the load torque and other is the viscous torque. The heat generation value was inputted with the required conditions and the temperature profile; total heat flux of the entire model was measured. The obtained temperature from the thermal analysis was inputted by updating the conditions. The deformation of the model and the maximum stress distribution at the contact were measured. At higher speeds the dynamic response was significant and the thermal effects have to be considered. The thermal load affects the stiffness. The heat generation in the ball bearing is a major cause of thermal expansion.

Rotational Speed (RPM)	Inner Ring Temperature (K)	Outer Ring Temperature (K)
1000	415	412
1100	436	433
1200	454	449
1300	470	465
1400	488	482
1500	507	502

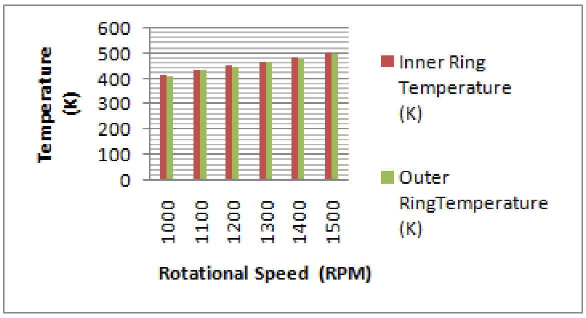

Figure 15. Rotational Speed Vs Temperature

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Rotational Speed (RPM)	Inner Ring Total Heat Flux (W/mm2)	Outer Ring Total Heat Flux (W/mm2)
1000	0.019	0.0022
1100	0.023	0.0026
1200	0.025	0.0031
1300	0.031	0.0035
1400	0.034	0.0038
1500	0.035	0.0039

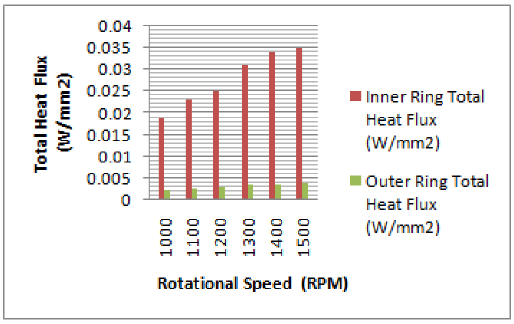

Figure 16.Rotational Speed Vs Total heat flux

Rotational Speed (RPM)	Inner Ring Equivalent Stress (H bar)	Outer Ring Equivalent Stress (H bar)
1000	77.84	9.46
1100	26.64	3.23
1200	33.21	4.53
1300	36.76	5.01
1400	45.3	5.5
1500	98.3	13.44

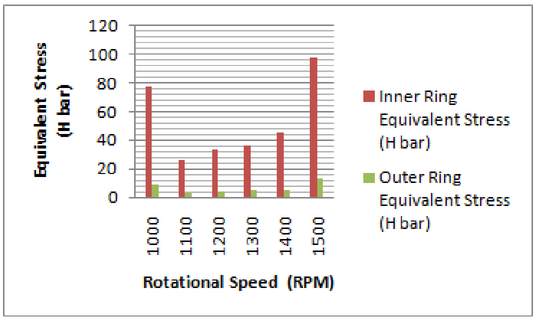

Figure 17. Rotational Speed Vs Equivalent Stress

Rotational Speed (RPM)	Inner Ring Equivalent Elastic Strain (mm/mm)	Outer Ring Equivalent Elastic Strain (mm/mm)
1000	0.003	0.0005
1100	0.004	0.0007
1200	0.005	0.0008
1300	0.006	0.0016
1400	0.007	0.0014
1500	0.01	0.0015

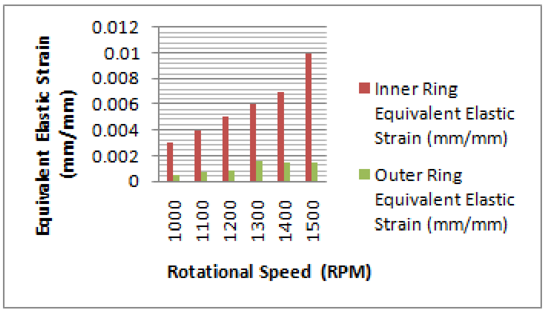

Figure 18. Rotational Speed Vs Equivalent Elastic Strain

Rotational Speed (RPM)	Inner Ring Total Deformation (mm)	Outer Ring Total Deformation (mm)
1000	0.04	0.031
1100	0.033	0.03
1200	0.038	0.003
1300	0.042	0.001
1400	0.046	0.015
1500	0.058	0.045

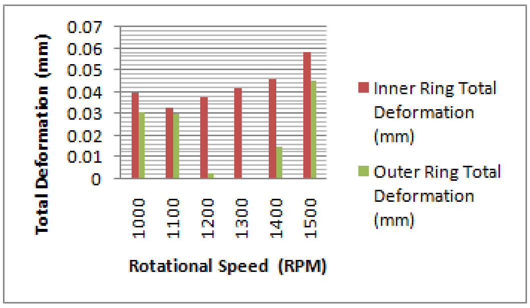

Figure 19 .Rotational Speed Vs Total Deformation (mm)

5. CONCLUSION:

Skf standard ball bearing 2215EKTN9 has been considered in the present study to investigate the thermal behavior and also the stresses induced due to thermal loads under speeds raging from 1000 to 1500 rpm with a radial loading of 15 KN NiTinol - 60 a nickel titanium alloy has been considered for the above bearing . 60NiTi, which contains 60% nickel and 40% titanium, is a super elastic inter metallic material for use in bearings, gears, and other mechanical systems. When properly processed, 60NiTi is hard, lightweight, electrically conductive, highly corrosion resistant, readily machined prior to final heat treatment, non-galling, and non-magnetic. Bearing-grade 60NiTi is manufactured via a patented, high-temperature powder metallurgy (PM) process. To Make 60NiTi balls, the powder is hiped into rough, spherical ball blanks that are Then ground, polished, and lapped. This material is best suited for oil lubricated rolling and sliding contact applications: Aerospace bearings, gears, drives, actuators, and other mechanical systems, rotorcraft engine bearings, rotor mechanisms, and drive systems. The frictional heat produced in the bearing causes the temperature to rise in the bearing surfaces and the bearing partly gets cooled due to convection currents operating the bearing surfaces . The temperature raises from 415 k to 507 k when speed increased from 1000 to 1500 rpm . The resulting thermal stress were minimum 12 Mpa at 1000rpm and maximum 61 Mpa at 1500 rpm . These stresses are much lower than the yield strength of the material which is greater than the deformations were very much less .when the speed has 1000 to 1500 rpm maximum value being less than o.1 mm considering the above the bearing is safe under the 15 K N load and at speeds reaming from 1000 to 1500rpm.

6. REFERENCES:

[1]Vidyasagar R. Bajaj . Finite Element Analysis of Integral Shaft Bearing .ijeert ; vol 3; 2105; p 28-36.

[2]Prasanna Subbarao Bhamidipati, FEA Analysis of Novel Design of Cylindrical Roller Bearing, R.C.E, affiliated to Jawaharlal Nehru technological university Hyderabad, India May, 2006 .

[3]Ambeprasad.S.Kushwaha, Atul B Wankhade, Dinesh E Mahajan, Darshan K Thakur, " Analysis of the Ball Bearing considering the Thermal (Temperature) and Friction Effects", National Conference on Emerging Trends in Engineering & Technology (VNCET-30 Mar'12)

[4] Akkudas Chennakesavulu, Structural Analysis of Ball Bearings in ANSYS , ijsetr, ISSN 2319-8885 Vol.04,Issue.33, August-2015, Pages:6719-6723.

[5]M. Chandra Sekhar Reddy, Thermal Stress Analysis Of A Ball Bearing By Finite Element Method , (IJARET) ,Volume 6, Issue 11, Nov 2015, pp. 80-90.

[6]Takeo Koyama, Applying FEM to the Design of Automotive Bearings, Automotive Bearing Technology Department 1997.

[7]Viramgama Parth D. Analysis of Single Row Deep Groove Ball Bearing ,IJERT, ISSN: 2278-0181 Vol. 3 Issue 5, May – 2014.