DESIGN GUIDELINES FOR HIGH SPEED CNC MACHINE TOOL SPINDLE BEARINGS

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ABSTRACT: *The manufacturing industries are facing challenges of designing high speed CNC machine tools to meet the demands of machining modern materials for different applications with wide range of higher spindle speeds. The spindle and bearing elements in headstock unit are subjected to structural loads which include cutting force, tool clamping force and bearing preload, thus producing reaction forces on the bearings during machining process. At higher speeds, large amount of heat is generated due to applied torque because of cutting load on bearings and viscous friction in the spindle bearing zone. This causes structural deformations in the bearing with the effect of thermal loads. The structural and thermal loads are calculated using appropriate theories for applying inputs during analysis. In the present study, an integrated thermo-structural couple field for super precision angular contact ball bearings, B7009ETP4S & HCB7009ETP4S in tandem arrangement is modelled, and design of experiments is simulated with varying boundary conditions in ANSYS Workbench to determine deformation, stresses and temperature variation induced in bearing elements. The temperature variation is also validated in numerical simulation using finite difference method, resistance network and experimental setup. The simulation results are in good agreement with experimental outputs and are within 10% variation, for reliable bearing performance. This approach has provided guidelines for system design to establish accurate prediction of bearing heat transfer and selection method.*

Keywords: *High Speed Spindle, Tandem Arrangement Of Angular Contact Bearings, Thermo-Structural Integration, Design Of Experiments, Finite Difference Method, Thermal Image.*

Nomenclature

δ	*Total spindle deflection in mm*	R_2	*Rear Bearing reaction force in N*
$\delta 1$	*Deflection due to radial yielding of the bearings in mm*	Δ_{max}	*Spindle nose deflection in µm*
$\delta 2$	*Deflection due to elastic bending of the spindle in mm*	Δ_r	*Bearing radial deflection in µm*
a	*Length of overhang in mm*	Δ_t	*Tangential deflection in µm*
L	*Bearing span in mm*	H_f	*Heat generated in watts*
E	*Young's modulus of Spindle material in N/mm²*	n	*rotational speed in rpm*
F_z	*Cutting force in N*	M	*Total frictional torque in Watts(W)*
K_1	*Stiffness of the front bearing in N/mm*	M_1	*Frictional torque due to applied load in W*
K_2	*Stiffness of the rear bearing in N/mm*	M_2	*viscous torque due to lubricant in W*
I_L	*Mass moment of inertia of the shaft at the bearing span in mm⁴*	f_1	*Factor depending on bearing design and relative load*
I_a	*Mass moment of inertia of the Nose end of the spindle in mm⁴*	Z	*Factor for angular contact ball bearing with 25° Contact angle (from SKF table)*
C_s	*Static equivalent load rating (from SKF table)*		
d_m	*Mean diameter of bearing in mm*	Y	*factor for angular contact ball bearing with 25° Contact angle (from SKF table)*
F_r	*Radial force acting on bearing in N*		
F_a	*axial force acting on bearing in N*	F_β	*Dynamic equivalent load in N*
X_0, Y_0	*Bearing coefficient values in tandem arrangement (from SKF table)*	F_s	*Static equivalent load in N*
R_1	*Front Bearing reaction force in N*	v_0	*Kinematic Viscosity in centipoises*

1. INTRODUCTION

The Machine tool Spindle is subjected to combined torsion, bending, axial and impact loads during metal removal work cycles producing reaction forces at bearing supports and deflection at nose end. In order to provide structural stability, the spindle design and bearing selection are critical to produce high quality components and also for reliable operation. . Today CNC machines are used for machining different work materials and operations at high cutting speeds and feed rates to achieve enhanced performance. At this very high spindle speeds large amount of heat is generated due to applied load and viscous torque which will lead to excessive thermal preload on the bearings inducing stresses and deformation.

ZhenhuanYe.et.al[1],attempted to establish the thermo mechanical coupling effect on the angular contact ball bearings with respect to geometrical orientation of the ball angle .R.SathiyaMoorthy .et.al [2] proposed a model for determining the heat generated in the bearing 2MMV99120angularcontact ball bearing by friction torque, gyro and centrifugal force effects in the bearings and results verified with SKF bearing data. Yanfang et.al [3] has developed mathematical model for estimating the bearing internal loads and then calculated heat generation in the bearing using the empirical formulae out lined in the standard bearing manufacturers. Xiaoping Li et.al [10] conducted study on the effect of variable preload and contact angle for predicting the temperature rise in the inner race using ANSYS and numerical simulation methods.

Md .F. Ghanati et.al [5] studied the non linear stiffness of the milling spindle with angular contact ball bearings based on geometrical and physical parameters. Wang yan –shuang et.al[6] estimated heat generation in the bearing primarily based on rolling velocity of inner ring and axial load effect. While it is observed that radial load influence on heat generation is not important. Balamurugan et al.[7] computed heat generation in ball bearing and analyzed stress , deformations in the bearing elements, A major observation to note that the inner race way deformation is on higher side than outer race at high rotational speeds. Comparative study has been carried out for induced stress in the steel and hybrid bearings.

Viorelpaleu [8,21] used alternative medium, kerosene mist for bearing lubrication in steel, hybrid and ceramic ball bearings to run up to 45000rpm experimentally . Further Simulated in ADAMS software to establish the effect of rotational speed on increase of frictional moment. S.Bharath Subramanian et.al [9] studied on the thermal analysis of ceramic conventional ball bearings also confirms with the increase of rotational speed increases deformation and stresses up to bearing failure: R.Nathan Kartz et.al [12] have reviewed the possibility of ceramic materials for rolling elements in bearings and compared its mechanical properties with steel for specific applications. Gupta[13] had taken ADORE computer code as reference line and improved the modal by numerical stepwise averaged manner for estimating heat generation and geometrical variations in the dimensions of the bearing elements. K.Tanimoto et al[16] have developed KOYO hybrid ball bearing to turbocharger applications of automotive engines for low frictional torque, less vibration and low heat generation at higher rotational speeds. Further these bearings found applications in high speed spindles in CNC machines. Hirotoshi Aramaki et al [17 ,18] experimentally demonstrated that hybrid bearings of NNK type at front and rear of a spindle unit have advantage of low gyroscopic moment ,low temperature rise and higher speed capability at low cost lubrication using either grease /oil- air mechanism. I wu et al [19] analyzed theoretically and experimentally to establish a mathematical model for heat transfer in spindle –bearing subassembly applying grease lubrication in a study state condition. And also indicated that failure of heat dissipation (cooling & lubrication) lead to thermally induced pre load causing thermal failure of bearings. Haitong Wang et al[20] proposed generalized model to predict structural and thermal characteristics like bearing pre load ,temperature variation and thermal displacements in the spindle bearing –system during transient thermal and static structural analysis. The results are verified experimentally in a test ring on NSK 7220 C bearing. The outcome of literature review reveals that the research has been carried out only on single bearing to evaluate performance for critical applications.

Present study has been carried out on tandem arrangement with super precision angular contact bearings up to 6000rpm with steel bearings and 10000 rpm with hybrid bearings with 25° contact angle. This arrangement enhances the axial load carrying capacity of bearing sets. Pre load is applied as shown in

Fig.3 (b) and Fig.4 (b). Key challenges for attaining higher speeds are Thermal deformation/spindle growth in the system design and also precisely compute heat generation. The heat generation is computed at different speeds and applied as basic input parameter (heat flow) for design of experiments in ANSYS in the steel as well as Hybrid bearings without/with grease lubrication. Research work in this article is focussed on the thermo-structural behaviour of spindle bearings running at different speeds with 10% variation in heat flow and outer race boundary temperature. Further, design guide lines are set for steel and hybrid bearing selections for high speed machining spindles.

2. BEARINGS FOR HIGH SPEED MACHINES

The mechanical properties for bearing materials are shown table 1. The geometry with boundary dimensions along with load and preload limits are shown in the figure 1 and table 2 [24]. The general arrangement of the vertical machining centre and cartridge housing is shown in figure 2(a) and figure 2 (b). Different bearing arrangements are preferred for specific applications. In the present case tandem arrangement is adopted for axial loading of the bearing pairs for high speed spindle application as shown in the figures3 (a), (b) and (c).

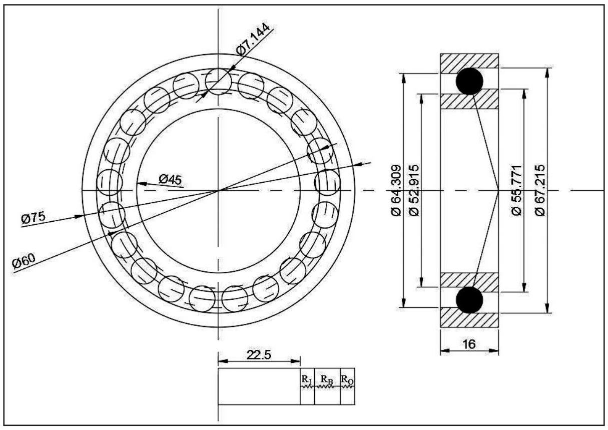


Fig 1. CAD Model of Angular Contact Ball Bearing

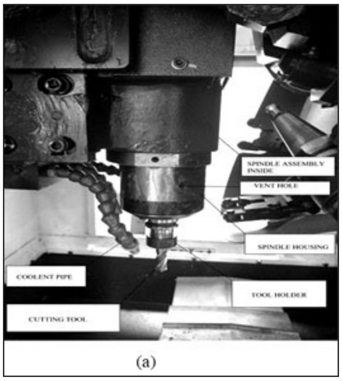


Fig 2 (a): General Arrangement of the VMC

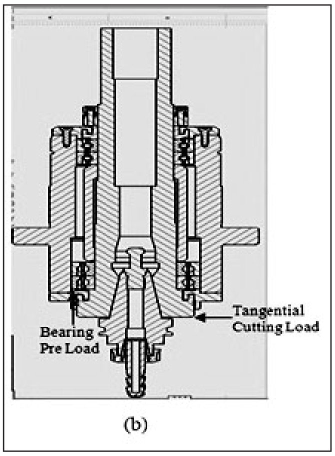


Fig 2(b): Bearing Arrangement and Applied Loads

Table 1

Properties at ambient temperature	units	Ceramic ball ($si_3 n_4$)	Bearing steel (103cr1)
Density	g/cm^3	3.2	7.8
Coefficient of expansion	10^{-6}/k	3.2	11.5
Young's modulus	GPa	3.15	216
Poission's ratio	-	0.26	0.3
Thermal conductivity	W/mk	30-35	40-45
Tensile yield strength	MPa	700	2500

Table 2

FAG Designation	Boundary dimensions Dynamic Static					Abutment dimensions			Basic load rating (kN)		Pre load in N	Number of balls	Ball dia in
B7009ETP4S	45	75	16	1	1	53.6	64.2	0.6	26.5	20.0	202	21	7.144
HCB7009ETP4S	45	75	16	1	1	53.6	64.2	0.6	18.0	14.0	90	21	7.144

Machine model: AMS spark- Vertical Machining Centre, Make: ACE Micrometric group, Spindle taper: 7/24 No. 30, Spindle speed: 60-6000 rpm, Spindle speed (opt): 80-8000 rpm, Spindle power: 5.5/3.75 kW.

Rapid traverse X/Y/Z: 20/20/15 m/min, Maximum tool weight: 2.5kgf.

The cutting load is estimated by,

$$F_Z = \frac{6120 * P_W * 1000}{\pi * D * N(r.p.m)} \qquad [23]$$

Power at the spindle (P_w) = UK_hK_rQ kW

Material removal rate, $Q = \frac{btS_m}{1000} cm^3/min$

U=Unit power kW/cc/min (from tables), K_h =correction factor for flank wear(from tables), K_f= correction factor for rake angle(from tables),b = width of cut (mm), t = depth of cut (mm), S_m = feed rate mm/min.

The tangential cutting load (F_z) is applied for determining the static deflection of the spindle nose end using the Macaulay's method considering stiffness(elastic support) of the bearings. F_z is basic input for computing heat generation in the bearings [11] and resulting deflections are shown in table 3.

$$\delta = F_Z \left[\frac{1}{S_1}\left(\frac{a+L}{L}\right)^2 + \frac{1}{S_2}\left(\frac{a}{L}\right)^2 + \frac{a^2}{3E}\left(\frac{L}{I_L} + \frac{a}{I_a}\right)\right]$$

Calculation of Spindle and bearing structural deflections:

The spindle deflections are calculated by using closed form solutions and is given by

$\Delta_{max} = [F a^2(l+a)] / 3EI = 0.0011$ mm at nose end.

Front bearing radial deflections = $R_1/k_{Bearing}$ = 1.432 µm

Rear bearing radial deflections = $R_2/k_{Bearing}$ = 0.259 µm

Heat Generation: The major heat generation

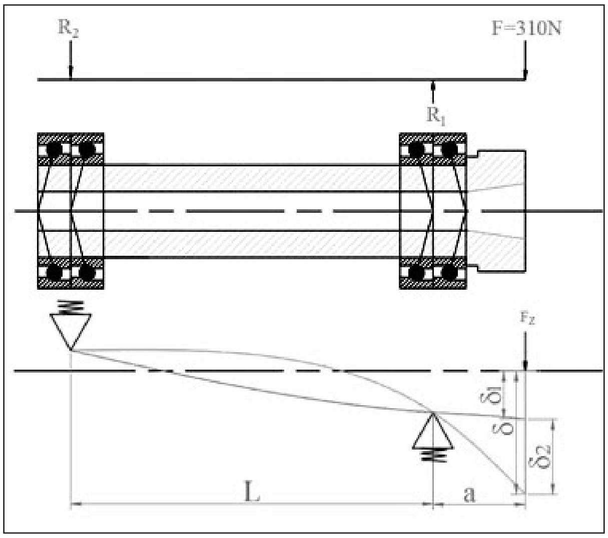
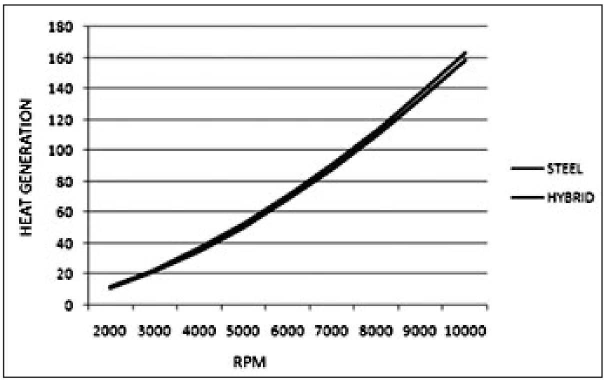


Fig. 3 (a) Represents Spindle Loading and Bearing Reaction Forces, (b) Represents Bearing Tandem Arrangements at Front and Rear and (c) Represents Spindle Nose Deflection



Graph 1: Heat Generated in Steel and Hybrid Bearings

Table 3

SKF Empirical and Macaulay's formulae for determining the Radial and Tangential Deflection[4]		
Type of Bearing	Radial deflection when Δ_t=0	Tangential Deflection when Δ_r=0
Angular Contact Ball Bearing	$\Delta_r = \frac{43.64 \times 10^{-9}}{\cos\beta}\sqrt[3]{\frac{F^2}{d}}$ when $\Delta_t = 0$	$\Delta_t = \frac{43.64 \times 10^{-9}}{\sin\beta}\sqrt[3]{\frac{F^2}{d}}$ when $\Delta_r = 0$
Empirical value(SKF)	0.001 mm	0.00245mm
Macaulay's	0.0011	0.002833

Table - 4

Typical result at 2000 rpm	
Measurement Technique	Output temp in °C
1.ANSYS WORK -BENCH	34
2.FDM	34.5
3.Infrared temperature gun	34.2
4.Thermal image	34.1

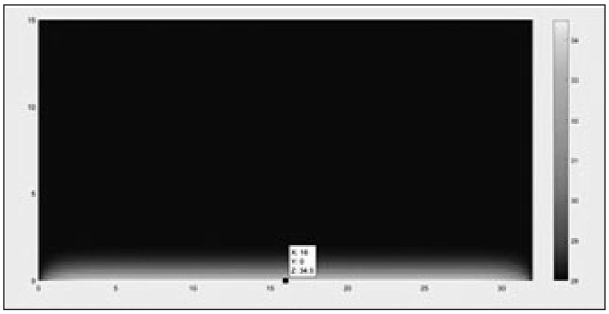


Fig 4. Heat Dissipation in Tandem Bearing Arrangement Through FDM

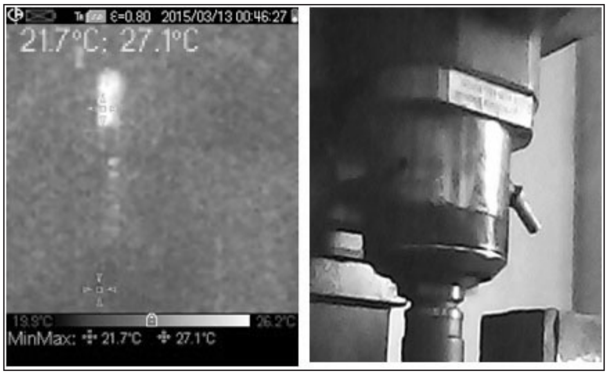


Fig 5. Thermal Imaging of Spindle Bearing System of CNC VMC

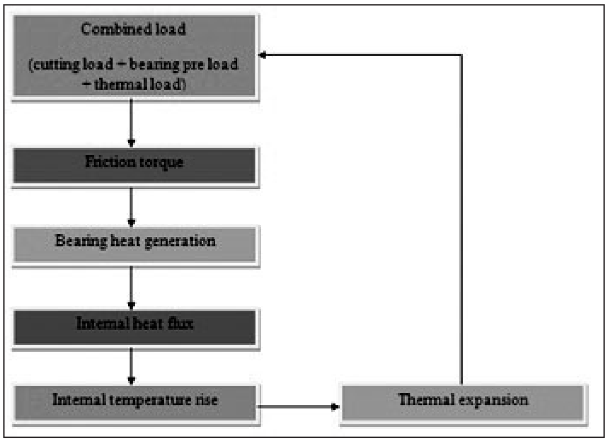


Fig 6. Flow Chart for Thermally Induced Preload

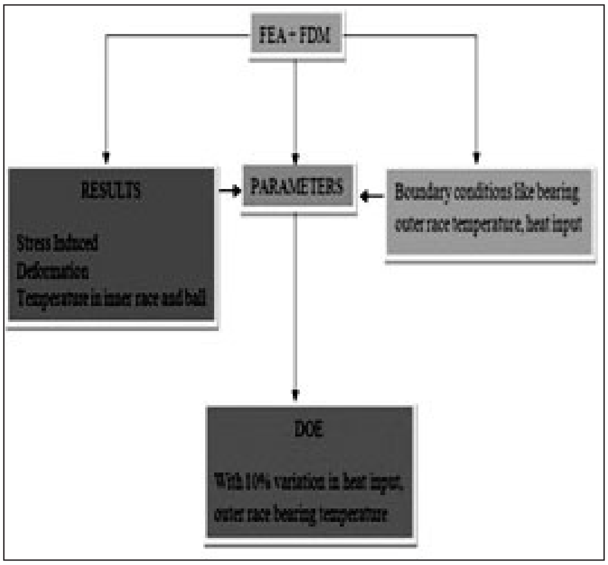


Fig 7. Methodology for TSI Based DOE

of the system is mainly caused by two sources i.e. cutting load and friction between balls and races of bearings. The heat generated in bearings is the primary cause of temperature change. The heat generated in bearing is computed by the relation [12, 14]

$$H_f = 1.047X10^{-4}\ n\ M \quad \text{.......... (i)}$$

Where H_f, n and M are heat generated, rotational speed & total frictional torque respectively

The total frictional torque M consists of two parts, torque M_1 which is due to applied load and the other torque M_2 due to viscosity of lubricant i.e. $M = M_1 + M_2$

Frictional torque due to applied load (M_1):

The torque due to applied load can be empirically approximated by the following:

$$M_1 = f1 \times F_\beta \times d_m \quad \text{.......... (ii)}$$

In which f_1 is the factor depending upon bearing design and relative load. For angular contact ball bearings

$$f_1 = z\,(F_s/C_s)^Y \quad \text{........... (iii)}$$

Where F_s is the static equivalent load and is given by

$$F_s = X_oF_r + Y_oF_a \quad \text{............. (iv)}$$

F_β is dynamical equivalent load, for angular contact ball bearings generally depends on

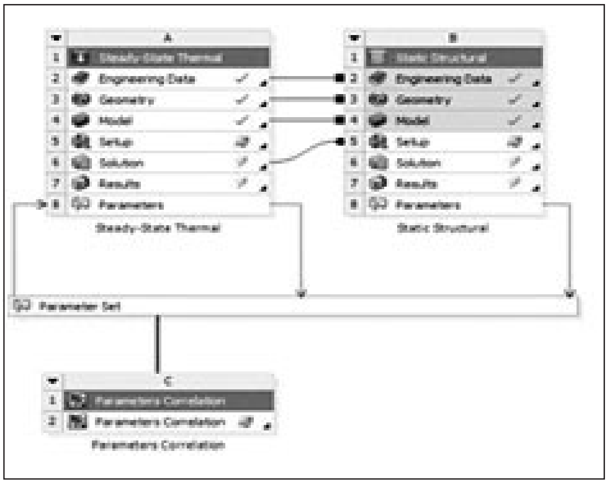


Fig 8. TSI Based DOE Integration

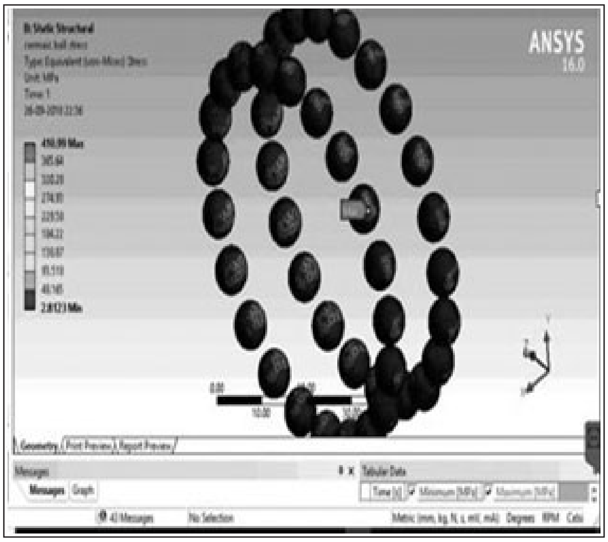


Fig 9. Von Mises Stress in Inner Race

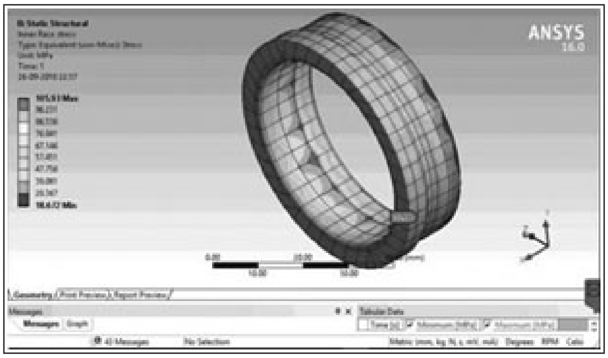


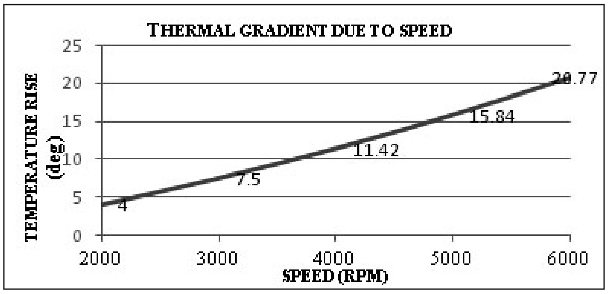
Fig 10. Von Mises Stress in Inner Race

the magnitude and direction of the applied load, F_β is given as

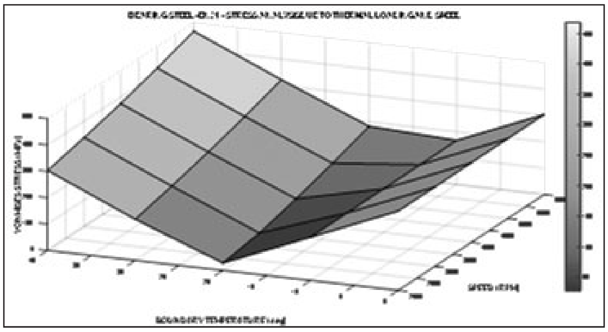
$F_\beta = \max(0.9F_a \cot\theta - 0.1F_r)$ ……… (v)

Solving the equation (ii) by taking pitch diameter

Fig 11. Von Mises Stress in Outer Race



Graph 2: Plot Showing Variation in Thermal Gradient at Different Speeds



Graph 3: 3D MAT LAB TOOL. Plot Showing Von Mises Stresses Induced at Different Fixed Outer Race Temperatures at Different Speeds

of bearing d_m = 60 mm, the torque developed due to applied load is given as M_1 = 7.4 N mm.

Viscous Friction Torque (M_2):

For bearings, the viscous friction torque is expressed empirically as, $M_2 = 10^{-7} f_o (v_o n)^{2/3}$

The value of v_o = 20 is taken, at the uniform temperature of 40^0C.

f_o is the factor depending upon the type of bearing and method of lubrication. The value is taken from harries [4]. It is taken as f_o = 2, M_2 = 48.02 N mm, Total M=55.42 N mm.

Table 5: Allowable Stresses in the Bearing Elements

S. No	Bearing type	Allowable stress in inner race in Mpa	Allowable stress in ball in Mpa	Allowable stress in outer race in Mpa
1.	B7009 TP4S(steel 103cr1)	485	485	485
2	HCB7009 ETP4S(hybrid with ceramic balls)	485	1980	485

Table 6: Temperatures and Stresses - Steel Bearing, B 7009ETP4S: by ANSYS- TSI of DOE at 2000 rpm

S. No	Speed in rpm	Heat input in watts, H_f	Outer race boundary temperature (in ºC)	Inner race output temperature(in ºC)	Von-Mises Stress in inner race (in Mpa)	Von-Mises Stress in steel ball (in Mpa)	Von-Mises Stress in outer race (in Mpa)
1	2000	12.27	0	4	75	260	310
2			10	14	105	350	250
3			20	24	20	50	30
4			30	34	60	180	100
5			40	44	100	325	250
6			50	54	140	475	350

Table 7: Temperatures and Stresses in Hybrid Bearing, HCB 7009ETP4S - by ANSYS- TSI of DOE, at 2000rpm

S. No	Speed in rpm	Heat input in Watts, H_f	Outer race boundary temperature(in ºC)	Inner race output temperature (in ºC)	Stress in inner race in Mpa	Stress in ceramic ball in Mpa	Stress in outer race in Mpa
1	2000	11.24	0	8	69	294	253
2			10	18	29	155	105
3			20	28	14	45	60
4			30	38	57	93	227
5			40	48	106	410	247
6			50	58	140	579	383

Table 8: Temperatures and stresses in Steel bearing B 7009ETP4S- by ANSYS- TSI of DOE, 6000rpm

S No	Speed (Rpm)	Thermal Load (W)	Outer Race Temperature (°C)	Inner Race Temperature (°C)	Stress Induced (Mpa)		
					Outer Race	Ball	Inner Race
1	6000	70.66	0	43.81	195.77	302.53	40.95
2			10	52.87	281.33	177.85	81.05
3			20	65.22	436.37	186.211	133.27
4			30	72.91	575.76	213.18	174.7
5			40	78.44	652.75	235.97	203.62
6			50	94.68	853.1	385.3	269.8

Table 9: Temperatures and Stresses in Hybrid Bearing HCB 7009ETP4S - by ANSYS- TSI of DOE, 6000rpm

S No	Speed (Rpm)	Thermal Load (W)	Outer Race Temperature (°C)	Inner Race Temperature (°C)	Stress Induced (Mpa)		
					Outer Race	Ball	Inner Race(P6)
1	6000	66.67	0	49.17	299.81	307.33	63.56
2			10	58.08	427.01	184.28	102.71
3			20	70.94	645.51	200.43	156.00
4			30	78.29	809.24	220.52	194
5			40	83.28	882.47	230.37	220.77
6			50	100.06	1126.58	369.73	288.04

Table 10: Typical Thermal Gradient Results at 2000RPM

S. No	Method	Heat input At 2000rpm	OR fixed boundary	IR temperature	Variation In ^{0}C	Remarks %variation
1	ANSYS -DOE	12.7	30	34	4	13.33
2	FDM MAT LAB	12.7	30	34.5	4.5	15.0
3	Infrared Temperature gun	Study state thermal after 30 mins	28	32.3	4.2	15.0

Table 11

SPEED	2000	3000	4000	5000	6000
THERMAL GRADIENT by DOI ANSYS	4	7.5	11.2	15.84	20.77

The heat generated in bearing is computed as H_f = 12.128 W at 2000rpm.Similarly the heat generated at various speeds is calculated for steel and hybrid bearings as shown in graph 1.

HEAT TRANSFER IN THE BEARING: MATLAB code is written for estimating heat transfer using Numerical. FDM Technique to assess temperature gradient operating with 10% heat input variation in the inner race at different outer race boundary conditions ranging from 0-50^{0}C. Temperature measurement is also carried out at different speeds from 2000 to 6000rpm on the CNC machine at front bearing (spindle nose end) near inner race in the hollow portion of spindle, running for 30 min to attain steady state heat transfer taking into account of ambient conditions. Typical sample results are shown in fig.4 and summary of results by different approaches in the table 4.

Thermal imaging of spindle bearing system has been done to study the temperature distribution at 2000 rpm using KRYKARD thermal imaging equipment model no: CA 1950-DIACAM 2 as shown in fig 5. It works on the principle of emissivity of the material being imaged.

3. THERMO-STRUCTURAL INTEGRATION includes Analysis framework, parameter correlation (ANSYS work bench multi-physics coupling).

METHODOLOGY The methodology implemented for conduct of experiment and analysis of tandem arranged angular contact ball bearings studied for their increase of preload, stresses induced in the inner race, ball and outer race, the deformation in bearings and temperature gradient between inner race and outer race in the steady state spindle bearing housing assembly as shown in fig 6&7

4. RESULTS AND DISCUSSION

1. In this present study, TSI based DOE ,FDM in MATLAB and experimental thermal imaging, thermal infra ray gun work has been carried out to analyze thermally induced stresses for steel and hybrid (B7009ETP4S & HCB7009ETP4S) super precision angular contact ball bearings in tandem setup till fail safe mode as shown in the table 6,7,8 and 9. The allowable stresses in the two types of bearing materials are provided in table 5 for reference.

   The drive coupling at the rear bearing side rear flange, spindle and bearing housing, the timer pulley etc with mutual interaction of the assembly components will be sharing the stress flow and the induced stress will be less when compared with table values 6,7,8 & 9. The reference values are as per table 5.

2. As the rotational speed increases which causes higher heat generation leading to thermal displacement and inducing stresses both in inner, outer race and balls. The research results provide a reference to the temperature distribution, magnitude of deformation and induced stresses in bearing elements and data points for high speed applications.

3. Using SKF Empirical relations, the calculated radial and tangential deflections are 0.001 & 0.00245 mm respectively in steel bearing running at 6000 speed. The Macaulay's deflection formulae (considering bearing stiffness) estimate the deflections to be 0.001432 &0.002833mm. The SKF empirical relation and Macaulay's deflection formulae are in close agreement. As per the JIS STD the allowable limits are 0.005mm. However due to manufacturing and assembly practices (errors) the deflections are found within the allowable limits.

4. Temperature rise by different approaches are summarized as follows:

   The results show that the methods 1 and 2 are in agreement with experimental validation of method 3. It is to note that the thermal gradient at 2000 rpm is found to be within 15% (having error 1.67% compared with method 1) as per the table 10.

5. It is observed that the outer race temperature should be maintained in the range of 10 to 30^{0}C to ensure the minimal induced stresses in the inner, ball, outer races for reliable operation of the bearings. Too low temperature is also not recommended as it will introduce contraction thermal stress. Hence the design guide line is to maintain in between 10-30 ^{0}C.

6. Maximum temperature attained at 6000 rpm is 95^{0}C at inner race .The bearing operates at grease boundary which can with stand up to 120^{0}C without breakdown /seizure.

7. Total deformation observed is 2.5x10^{-6}m at 6000 rpm at operating temperature of 25^{0}C indicating stable structural and thermal phenomena for high speed spindle providing reliable machining operations.

8. The heat generation in steel bearing is higher compared to hybrid bearing at the same rotational speed as per the graph 1(At 10,000rpm steel 163w and hybrid 159w.)

9. Thermal gradient observed between IR to OR for the steel bearing and also shown in the graph 2.

10. 3D MATLAB tool is developed for major input parameters which include rotational speed, outer race boundary temperature to the stress induced in the outer race. This tool is very valuable as design guide line for monitoring the bearing temperature during the machining operation

11. It is observed that from the graphs 12,13,14,15 and 16 (a), (b) and (c) trend similarities are observed in stress distribution, deformation and temperature variation in the two types of bearings. Details are provided in the legend. The notations (a), (b), (c) refers to steel bearing and (d), (e), (f) refers to hybrid bearing.

    - At 2000rpm the stressess are within the permissible limits (refer table 5), deformation is observed as 20 µm and temperature gradient of inner race w.r.t outer race boundary is found to be 4^{o}C.
    - At 3000 rpm the stress induced in the ceramic ball is 700 Mpa. The stress induced in ceramic ball is higher than that of steel ball due to lower thermal conduction co-efficient.
    - Similar phenomena is observed at 4000, 5000 and 6000rpm.

12. COMPARATIVE STUDY OF CRITICAL PARAMETERS:

- At 2000 RPM for steel bearing:

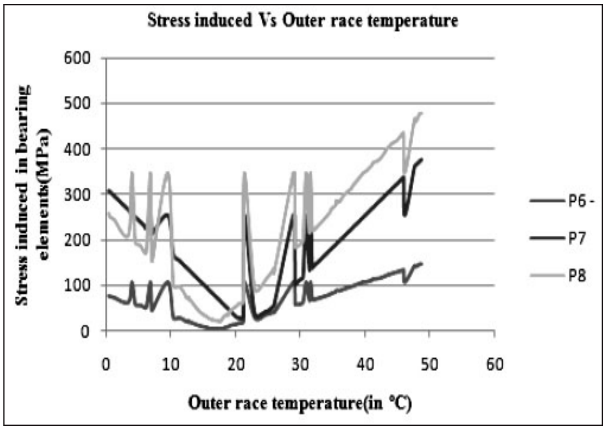


Fig 12 (a). Plot Showing Stress Induced in Bearing Elements (in MPa)Vs Outer Race Temperature(in ºC)

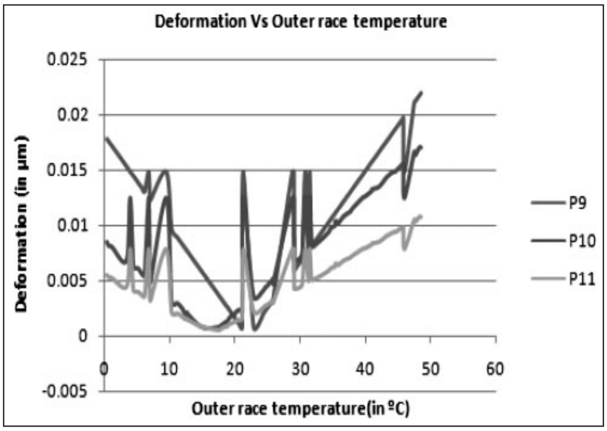


Fig 12 (b). Plot Showing Deformation (in µm) Vs Outer Race Temperature(in ºC)

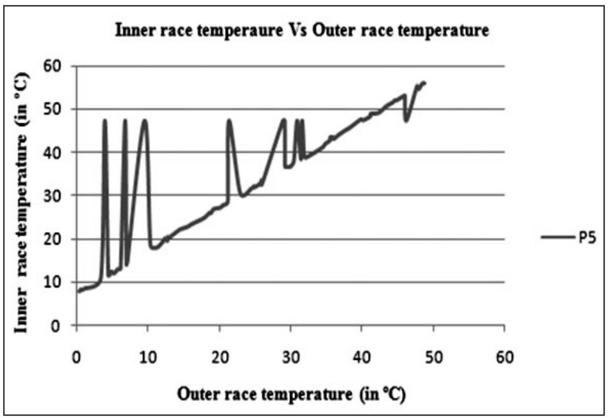


Fig 12 (C). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- AT 2000 RPM for hybrid bearing:

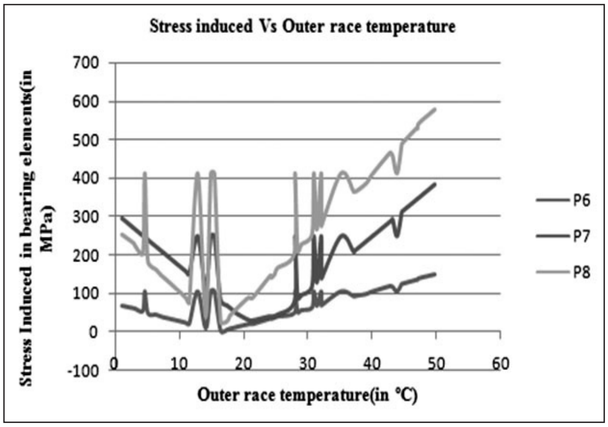


Fig 12 (D). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

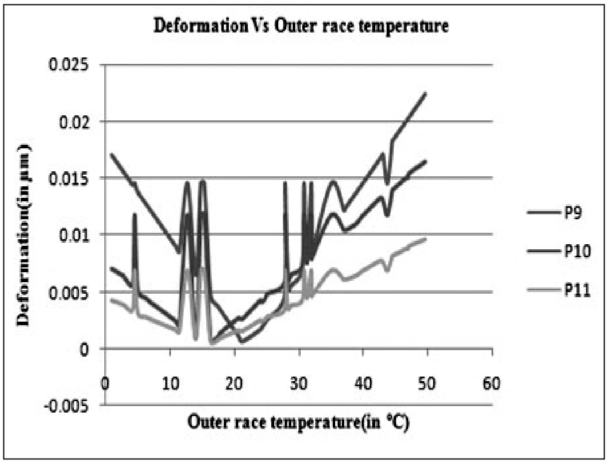


Fig 12 (E). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

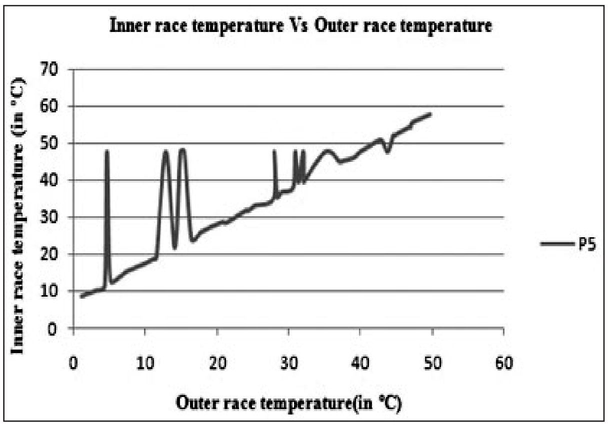


Fig 12 (f). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 3000 RPM for steel bearing:

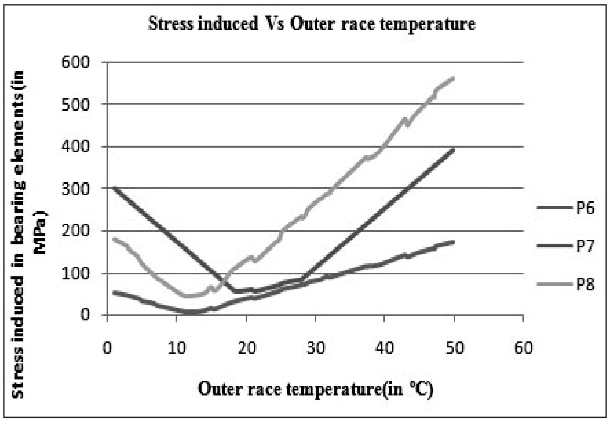


Fig 13 (A). Plot Showing Stress Induced In Bearing Elements (In Mpa)Vs Outer Race Temperature(In ºc)

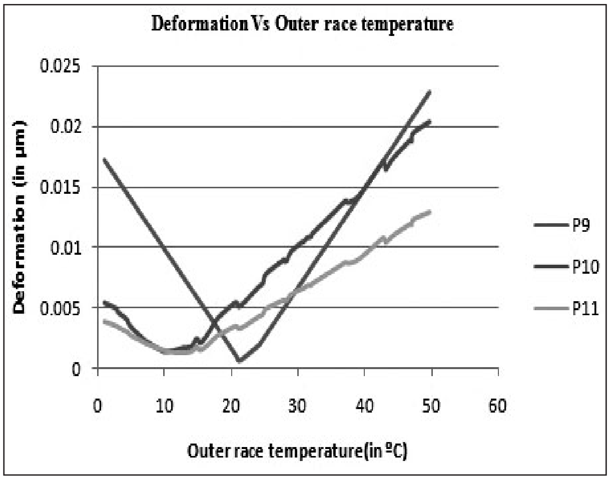


Fig 13 (B). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

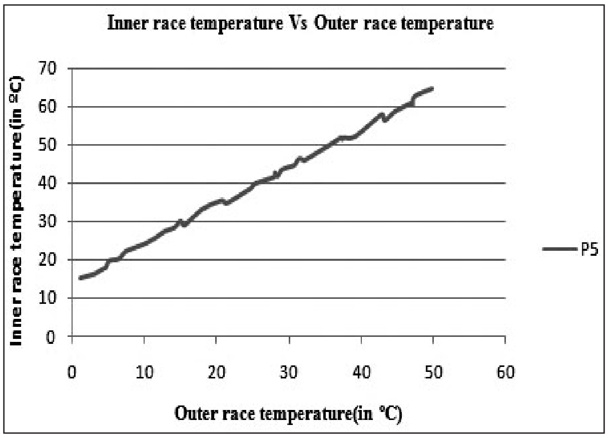


Fig 13 (C). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 3000RPM for hybrid bearing:

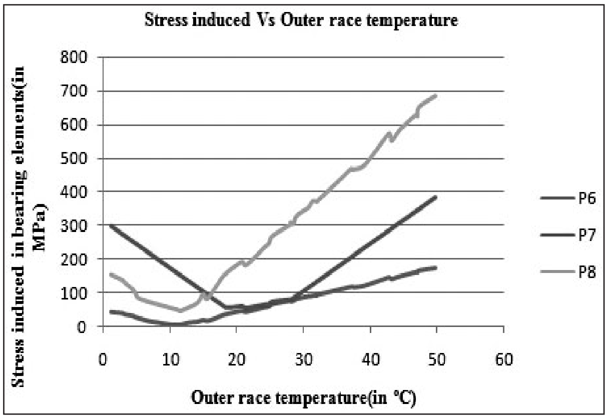


Fig 13(D). Plot Showing Stress Induced In Bearing Elements (In Mpa)Vs Outer Race Temperature(In ºc)

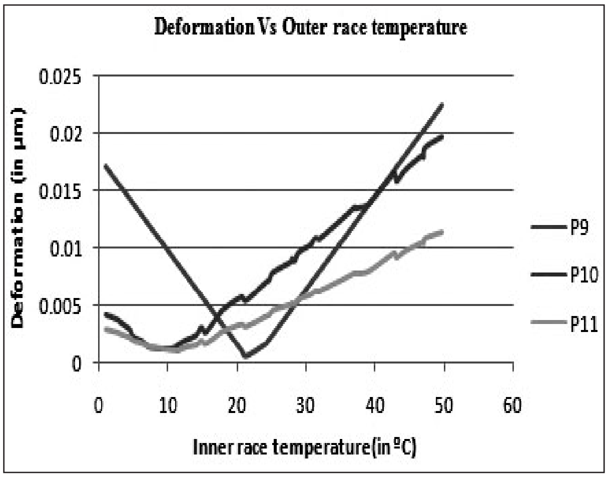


Fig 13(e). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

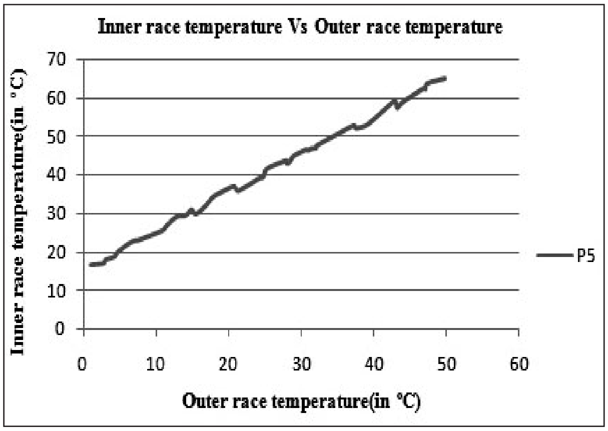


Fig 13 (f). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 4000 RPM for steel bearing:

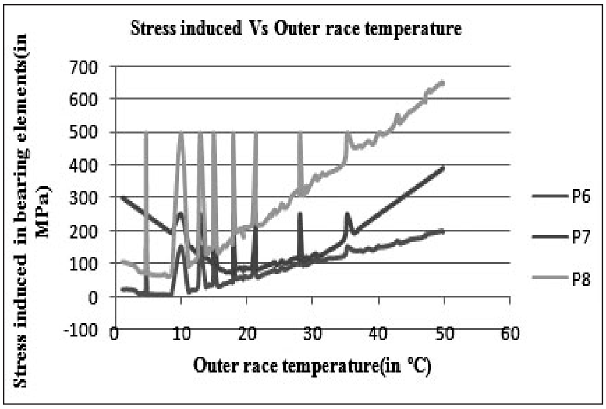


Fig 14 (A). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

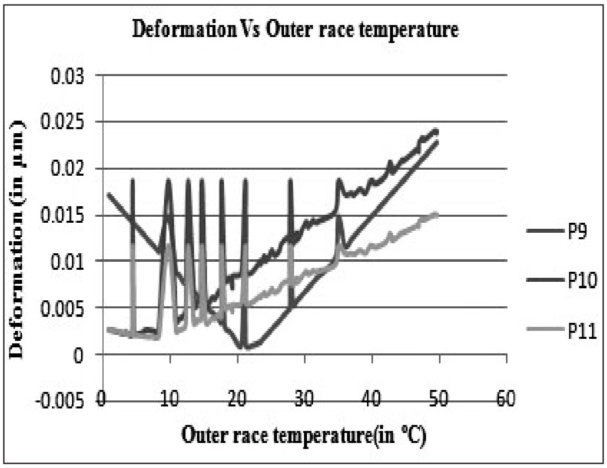


Fig 14(B). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

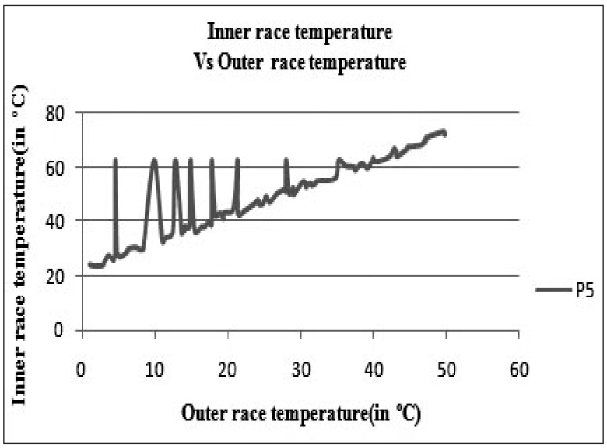


Fig 14(C). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 4000 RPM for hybrid bearing:

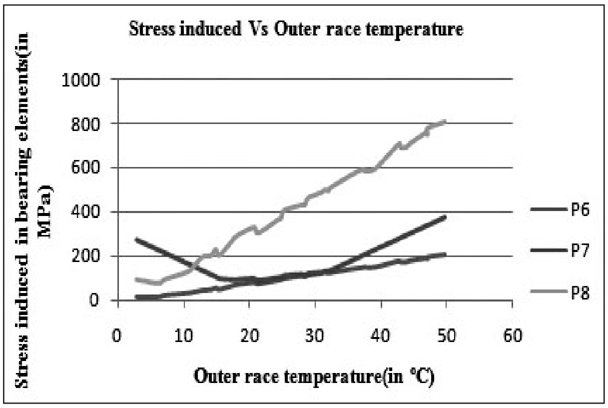


Fig 14 (D). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

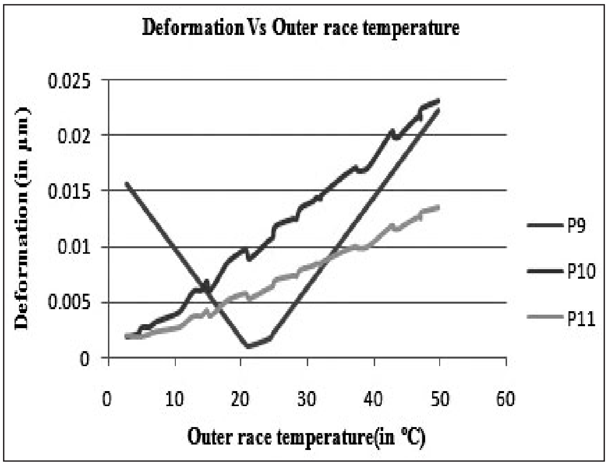


Fig 14 (E). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

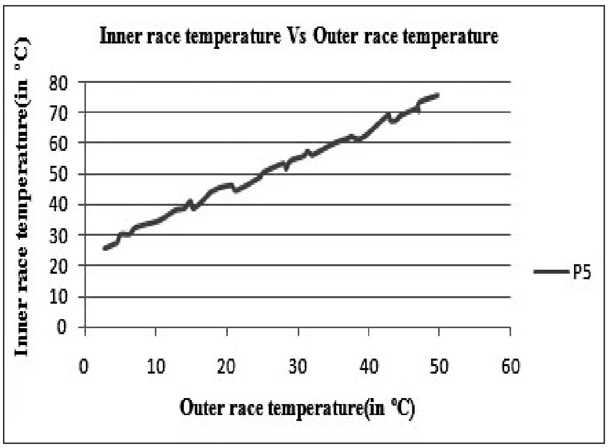


Fig 14 (F). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 5000 RPM for steel bearing:

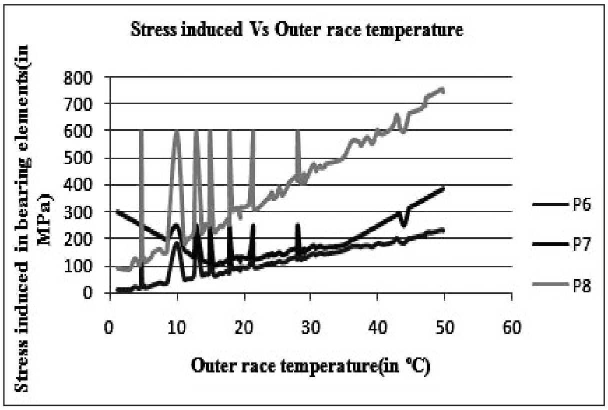


Fig 15(A). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

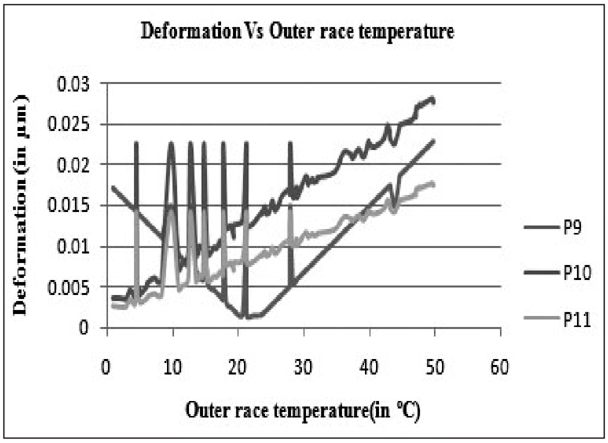


Fig 15(B). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

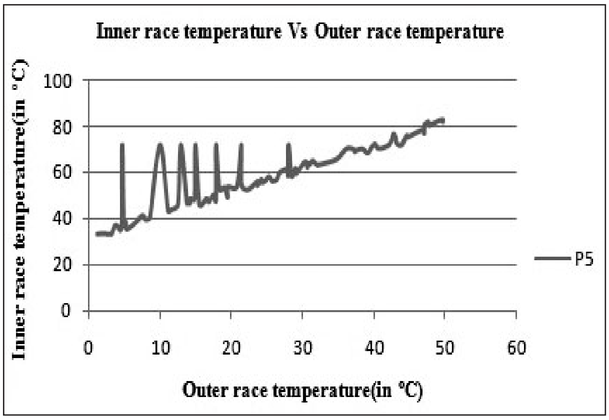


Fig 15(C). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 5000 RPM for hybrid bearing:

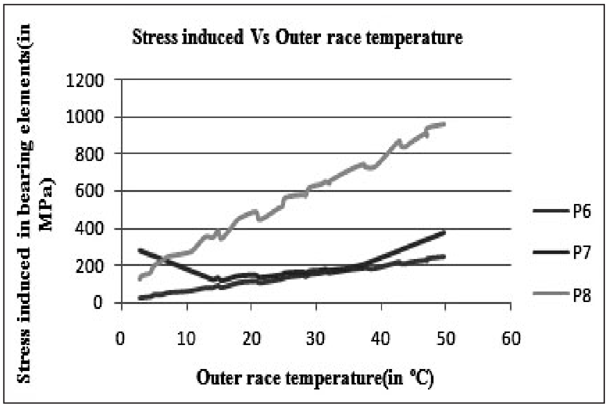


Fig 15 (D). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

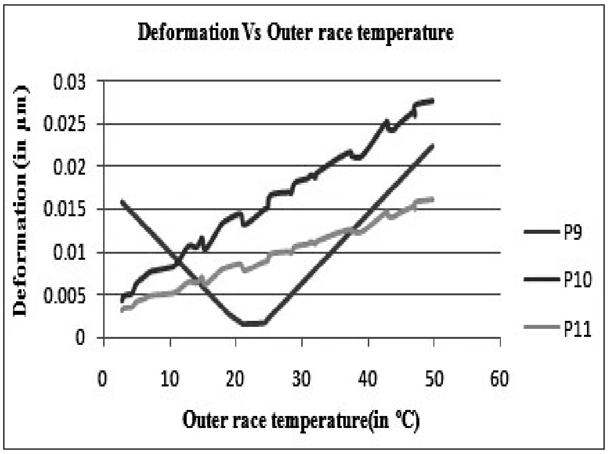


Fig 15 (E). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

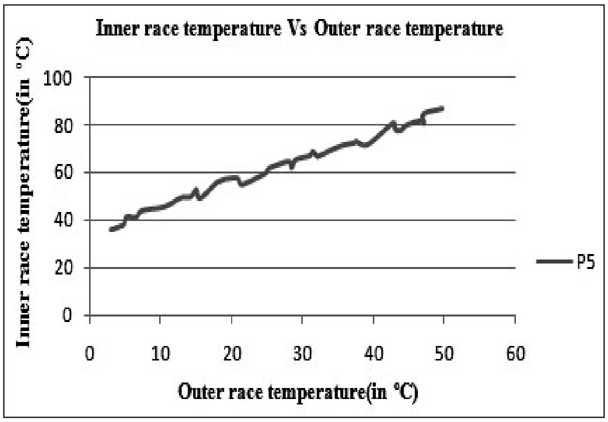


Fig 15 (F). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 6000 RPM for steel bearing:

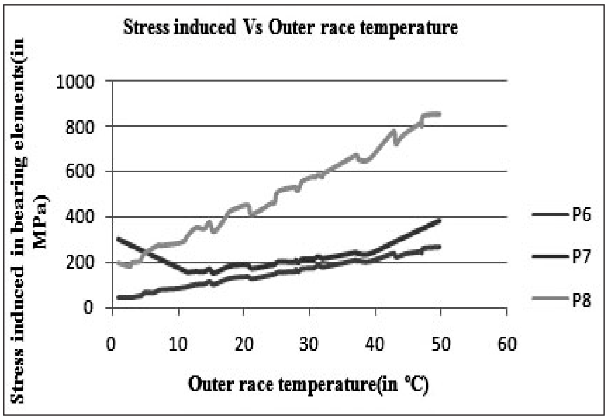


Fig 16(A). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

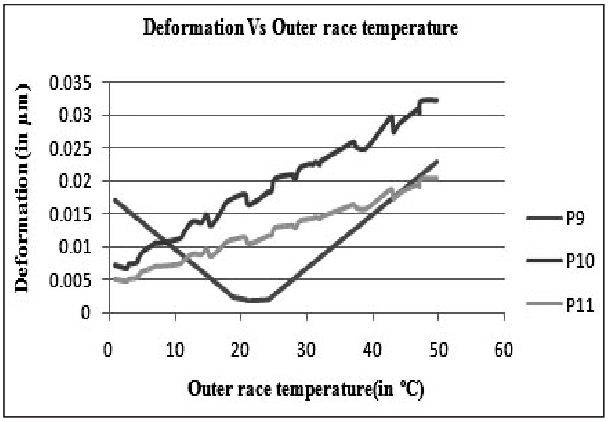


Fig 16 (B). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

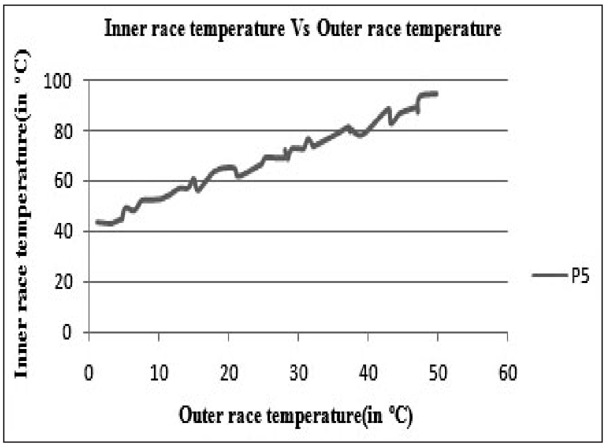


Fig 16(C). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

- At 6000 RPM for hybrid bearing:

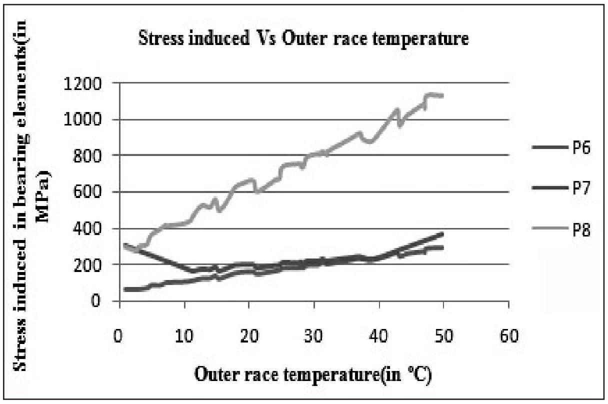


Fig 16 (D). Plot Showing Stress Induced In Bearing Elements(In Mpa)Vs Outer Race Temperature(In ºc)

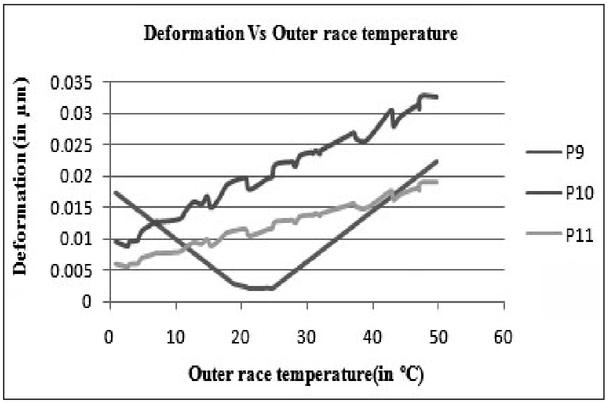


Fig 16 (E). Plot Showing Deformation (In Mm) Vs Outer Race Temperature(In ºc)

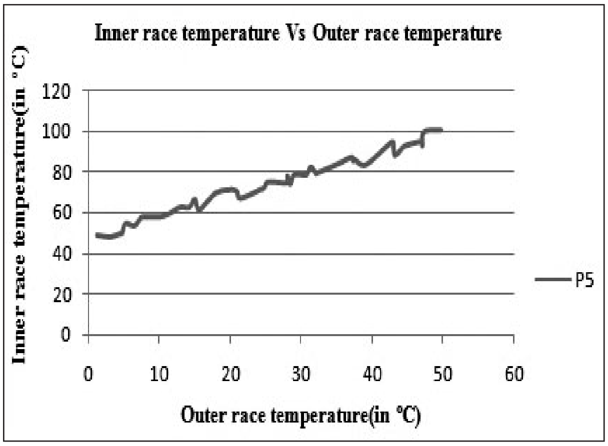


Fig 16 (F). Plot Showing Inner Race Temperature (In ºc) Vs Outer Race Temperature(In ºc)

Where,

P5 = rise in inner race temperature o C

P6= Stress induced in the inner race in Mpa

P7= Stress induced in the outer race in Mpa

P8= Stress induced in the outer race in Mpa

P9= inner race deformation in µm

P10= inner race deformation in µm

P11= inner race deformation in µm

13. Comparision plots have been drawn for highly stressed balls to visualize the von Mises stress induced at different speeds as shown in Fig.17(a) and Fig.17(b). It is observed that at certain point one ball is deforminig due to thermal expansion effect. Hence solid grease lubriction will ensure higher speeds for reliable operation.

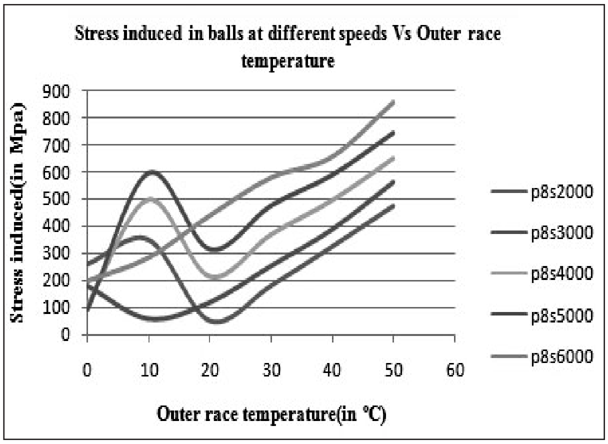


Fig 17(A). Plot Showing Stress Induced In Balls At Different Speeds(In Mpa) Vs Outer Race Temperature (In ºc) For Steel Bearings

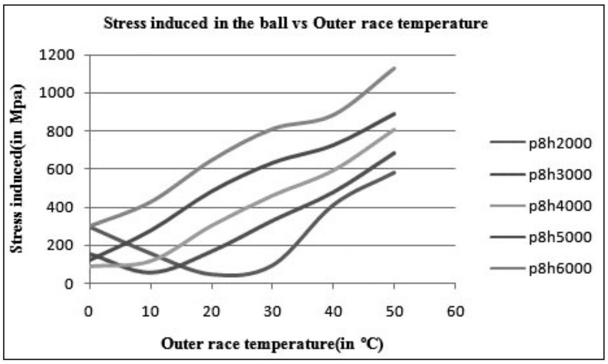


Fig 17(B). Plot Showing Stress Induced In Balls At Different Speeds(In Mpa) Vs Outer Race Temperature(In ºc) For Hybrid Bearings

14. Comparision plots have been drawn for deformation of inner race to understand the deflection phenomena at different speeds. It is concluded that at certain point the inner race is deforming due thermal expansion effect. Hence solid grease lubrication will ensure higher speeds for reliable operationas shown in fig 18 (a) &(b).

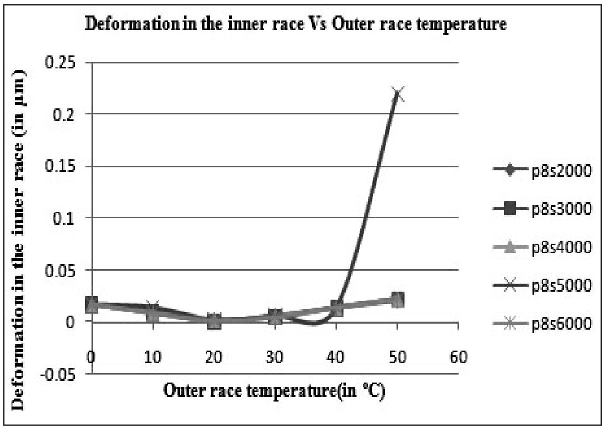


Fig 18 (A). Plot Showing Deformation In The Inner Race (In Mm) Vs Outer Race Temperature(In ºc) For Steel Bearings

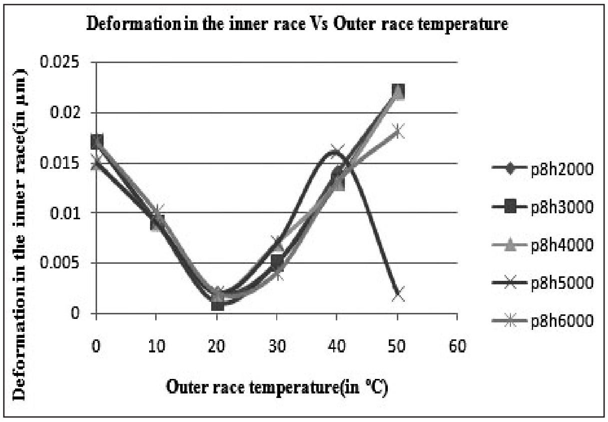


Fig 18 (b). Plot showing Deformation in the Inner Race (in µm) Vs Outer Race Temperature(in ºC) for Hybrid Bearings

15. The temaperature, stressess and deformations are beyond the allowable limits. Hence lithium based solid grease lubrication is essential for reducing friction to sustain the bearing performance. At 8000rpm observed von Misesstress is 1553 Mpa at 47^{o}C and inner race temperature is 131^{o}C in hybrid bearing as shown in the Fig.19(a), Fig.19(b) and Fig.19(c) for hybrid bearing.

- At 8000 RPM for hybrid bearing:

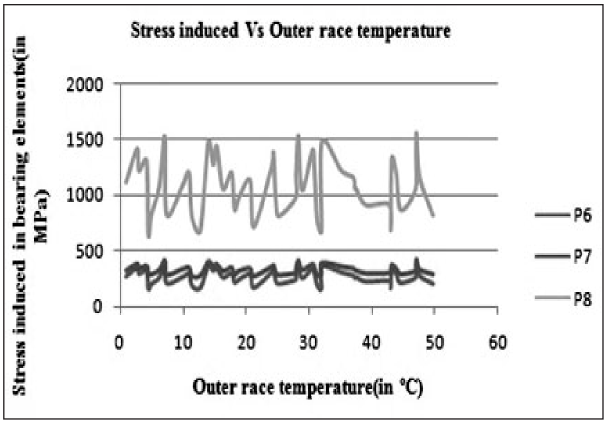


Fig 19(a). Plot showing Stress induced in bearing elements(in MPa)Vs Outer race temperature(in ºC)

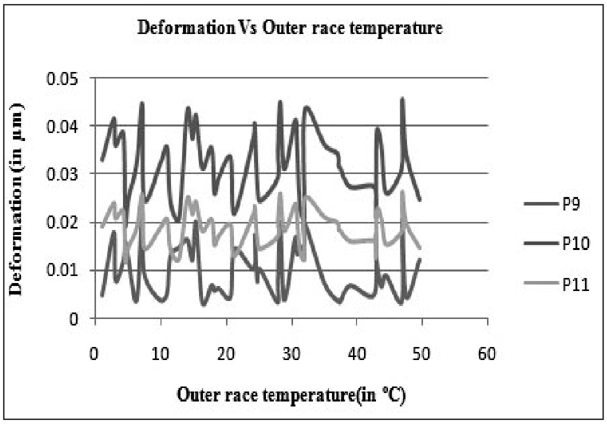


Fig 19(b). Plot showing deformation (in µm) Vs Outer race temperature(in ºC)

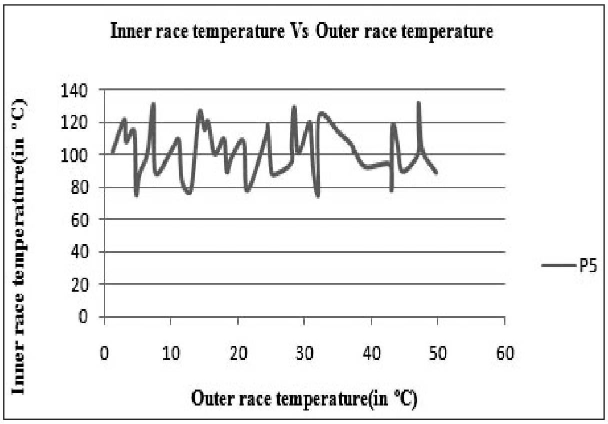


Fig 19(c). Plot showing Inner race temperature (in ºC) Vs Outer race temperature(in ºC)

5. CONCLUSION

1. Simulation results shall provide guide lines for amount of heat to be removed for reliable operation of the bearings while machining .With reference to table 6,7, 8 and 9 it is easy to specify attainable speed and comparing it to the induced stress in bearing elements at the operating speeds.
2. It is observed that the outer race temperature should be maintained in the range of 10 to 30^{0}C to ensure the minimal induced stresses in the inner, ball, outer races for reliable operation of the bearings. Too low temperature is also not recommended as it will introduce contraction thermal stress .Hence the design guide line is to maintain in between 10-30 ^{0}C.
3. Observed that maximum 10% variation is allowed in the heat input value beyond which the bearing encounters failure phenomena before attaining a rotational speed of 6000RPM.
4. The axial and radial deflections for angular ball bearing are computed and compared with SKF empirical formulae. These deflection values are found to be well within the allowable limits as per JIS (Japanese Industrial Standard) STD.
5. Bearing temperature monitoring should be part of machine tool building in the high-speed machining for condition monitoring, refer to graphs 4, 5, 6, 7 and 8 for steel as well as hybrid bearings.
6. The critical design parameters namely stress,deformation and thermal gradient are within the allowble limits. However there are peak values observed due to thermal expasion of bearings(deformation) and application of solid preload by spacers and locknuts.
7. The induced stress, deformation and inner race temperature in hybrid bearing are on higher side when compared to steel bearing .This is due to lower thermal conductivity of ceramic balls.
8. For Tandem bearing arrangement the speed limits as specified in the catalogue will be lower.
9. Accuracy class P4S is superior to ISO P4 with better running accuracy and a sealed grease lubricated bearing is used in the DOE for which results are computed.This is a selection criteria for high speed bearing. Future analysis may be carried on face to face & back to back bearing arrangement for specific applications.

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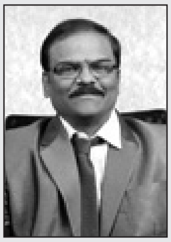

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