Dynamic Analysis of High Speed Spindle

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Abstract. Assessment of the thermo-mechanical behaviour of machine-tool spindles is a requisite for the reliable operation of high-speed CNC machine tools. The thermal behaviour and applied loads affect the performance of these high-speed spindles. The primary source of heat generation in the spindle is due to friction torque in angular contact ball bearings, cutting forces, centrifugal force, gyroratory effects and bearing preload. The spindle shaft is treated as beam element with one translational and one rotational degree-of-freedom at each node. Angular contact ball bearings are modelled to predict heat generation due to load, viscous, and spin-related effects. Mutual interlinkage between the thermal and structural behaviour of both spindle shaft and bearings is modelled using the effects caused due to the thermal expansion and rate of heat generation. The bearing preload is pushed up due to centrifugal force and temperature rise due to high spindle speed. Components matrices are assembled to configure a finite element model for the thermo-mechanical analysis of spindle-bearing systems using Euler-Bernoulli beam theory. Further this work presents ANSYS simulations on CNC spindle system with angular contact ball bearings using 103 CrIsteel material and hybrid type to predict thermal stress. Static and dynamic analysis is carried out on the spindle with and without thermal load and cutting load to predict the displacement and slope.

Keywords: CNC spindle, thermo-mechanical, finite element model, gyroratory effects, Hybrid bearing.

1. Introduction
High speed machining (HSS) is a promising advanced manufacturing technology for increasing productivity and reducing production costs dramatically. High speed machine tool and its spindle rigidity are crucial for high speed machining which is of considerable significance in the fields of research and development.

The spindle is the principal mechanical rotating component in milling /turning centers. The machine productivity and finish quality of the workpiece depends upon the static and dynamic stiffness of the spindle. The constitutional properties of the spindle are based on the geometry of the shaft, bearings, motor, type of tool holder, and the design configuration of the spindle assembly. The bearing arrangements are governed by type of cutting operation, applied cutting load, spindle stiffness
and limiting speed of the bearings. The main focus of HSS design is to attain long-term precision and high machining productivity for minimizing the operating costs of the machine. In high speed machine spindle assembly’s bearings plays a major role when it comes to high speeds. Generally angular contact ball bearings are used in spindle assemblies as they can take both axial loads and radial loads. Hence a study is made in this paper on the behaviour of these angular contact ball bearings at different speeds and also its effects on the spindle and machining. Forces acting on the high speed spindle during HSM are shown in the Figure 1.

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 the nose end. In order to provide structural stability, the spindle design and bearing selection are critical to producing 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, a large amount of heat generates due to the cutting force and viscous spin torque, which leads to excessive thermal preload on the bearings inducing stresses and deformation.

![Figure 1: Factors influencing spindle bearing system (SBS)](image)

| Symbol | Description |
|--------|-------------|
| I      | Beam elemental length (mm) |
| E      | Modulus of elasticity of Spindle material (N/mm²) |
| F      | Cutting force (N) |
| K      | Stiffness matrix (N/mm) |
| C⁰      | Static equivalent load rating (from SKF table) |
| d_m     | Mean diameter of bearing (mm) |
| F_r     | Radial force acting on bearing (N) |
| H_b     | Heat generated in bearings (W) |
| n       | Rotational speed in RPM |
| M_f     | Total frictional torque (N-mm) |
| M_l     | Frictional torque because of applied load (N-mm) |
| M_s     | Viscous torque owing to lubricant (N-mm) |
| M_s     | Spin torque (N-mm) |
| f_i     | Factor based on bearing design and
### 3. LITERATURE SURVEY:

Abele et al. (2010) presented in the review article, a hollow spindle shaft is preferred to meet the objectives of the machine tool spindle and mounted on angular contact ball bearing in headstock housing. The angular contact ball bearings possess both axial and radial stiffness at high external load at high rotational speeds. Bearing models are used for obtaining preload, and load due to thermal expansion during the running of the spindle.

Lin et al. (2013) presented review on the dynamic analysis of high speed spindle and discussed the governing equation of motion which includes gyroscopic and centrifugal force and its effects on the spindle performance. The hollow spindle shaft is analyzed by modeling in the FEA environment by using element models namely beam/brick/pipe.

Arotaritei and Constantin (2016) presented review on the modelling and simulation of high speed spindle and dynamics of spindle bearing system to study the deformations and temperature profile in the assembly. The spindle analysis is conducted using Euler-Bernoulli beam theory with gyroscopic and centrifugal effects along with spindle inertia and bending moment in the governing equation.

Long and Su (2017) conducted virtual orthogonal simulation for optimization in the ANSYS software for a CNC machine tool spindle to determine the spindle deflections using input variables as size of the spindle, bearing stiffness and cutting force.

Alfares et al. (2019) studied thermal effects of the spindle bearing system of a grinding machine wheel head mounted in the angular contact ball bearings. The simple steady and transient heat transfer thermal resistance network discretization is implemented for the spindle bearing system to predict the thermal deformations through MATLAB ODE solver. Thermal deformations are directly proportional to spindle speed and influencing the thermal preload in the dynamic model.

Liu et al. (2019) modelled the thermo-structural integration of the spindle bearing system by thermal contact resistance network. The heat generation in the bearings is predicted by combined viscous friction and equivalent external load due to thermal deformations with external load as well as interference contact at solid joints. Li and Shin (2004) presented the bearing arrangement has a major influence on spindle stiffness. The tandem arrangement of bearing results in a higher stiffness and natural frequency of the system compared to DF arrangement (X configuration).

Zahedi and Movahhedy (2012) developed a comprehensive model for the thermomechanical performance of high-speed spindle units considering bearings, shaft, headstock housing. The temperature variation along the length of the spindle is plotted at 10000 RPM. The highest temperature rise is 60°C in the bearing at 400 mm and 40°C on the housing. The bearing stiffness reduces as the spindle speed increases.

| Symbol | Description |
|--------|-------------|
| F_{ax} | Axial force acting on bearing (N) |
| X_0, Y_0 | Bearing coefficient values in a tandem arrangement (from SKF table) |
| F_{th} | Thermal preload |
| \varepsilon_2 | Thermal displacement (\mu m) |
| K_r | Bearing elastic constant |
| I | Constant based on the type of bearing, 1.5 for angular contact ball bearings |
| C | Elliptical area integration |
| M_d | Mass matrix |
| HSS | Global displacement vector |
| E | High speed spindle |
| \nu | Young’s Modulus |
| Poisson’s Ratio | |

Relative load:
- Z: Factor for angular contact ball bearing with 25° attitude angle (from SKF table)
- Y: the factor for angular contact ball bearing with 25° attitude angle (from SKF table)
- F_\beta: Dynamic equivalent load (N)
- F_s: Static equivalent load (N)
- \nu_0: Kinematic Viscosity (centi-Poise)
- \Theta: Bearing attitude angle
- F_0: Factor-based on type of bearing
- \mu: Coefficient of friction
- a: Major axis of the elliptical contact zone in bearing
- HSM: High speed machining
Uhlmann and Hu(2012) performed a Transient heat transfer analysis is carried at an ambient temperature of 19°C to predict the deformations w.r.t tool canter point in VMC at a spindle speed of 12000 RPM for a time period of 480 minutes. The drift of tool canter points is 2 µm, 10 µm and 50 µm in X, Y and Z axes respectively from the simulations. Experimental temperature measurements are carried out under similar operating conditions using thermal imaging and thermocouples. The experimental axes deformations are 5 µm, 10 µm and 50 µm in X, Y, and Z axes, respectively.

Sathiya Moorthy and Prabhu Raja(2014) presented a model for the heat generation in the bearings for low as well as for higher rotational speeds. The proposed model is validated with three sets of experimental results at various speeds. The analytical model has upper deviation of 20% when the rotational speed is less than 2000 RPM, while the deviation is reduced to 5% in the rotational speed range of 2000-11000 RPM. 

Lu et al.(2007b) studied the dynamic characteristics of the high-speed integral spindle in FEM using ANSYS software. The integral spindle is subjected to static and dynamic loads, including centrifugal force and gyroscopic moments at high rotational speeds. Using the standard bearing equations, the axial and radial stiffness is calculated for hybrid angular contact bearings with specification HCB 7011 and HCB7008CDGA/HCP4 in tandem arrangement using spacer and locknuts for initial bearing preloading. The bearings are life time grease lubricated. Experimental temperature measurements are verified with the ANSYS model at 2000, 4000, and 6000 RPM spindle speeds. Further, the thermal displacement at the spindle nose is 3.51, 11.11, and 11.02 µm in the X, Y and Z directions respectively. 

Lu et al.(2007a) conducted the experimental work on the high speed spindle to measure the temperature raise at the front, and rear bearing which is found to be 60°C and 67°C respectively when the spindle speed varies from 5000-20000 RPM. Hence, the external cooling of the bearing is done. 

Chen and Hsu(2003) implemented an auto regression dynamic thermal error model in place of the static regression model for prediction of thermal deformation of the rotating spindle. The new thermal model proposed is found to be accurate compared to the temperature-based model and validated experimentally. 

Lin et al.(2003) presented their work on high-speed rotational effects and softening of the spindle due to centrifugal force. Theoretical and experimental studies are carried on high-speed spindle including gyroscopic, centrifugal and cutting loads. Angular contact ball bearings with hybrid material are used for high-speed spindles. The increased radial stiffness compensates for the spindle softening effect.

Choi and Lee(1997) investigated the thermal behavior of the spindle bearing system with a bevel gear located on the bearing span. Since the drive is taking place form the front bearing zone, the temperature rise is higher than rear bearing for 30 minutes operation. The experimental work on the B7020C and B7016C bearings is carried out to measure the temperatures using J type thermocouples up to spindle speeds of 3500 RPM. ANSYS simulation and experimental results are compared for validation. The heat generated for B7020C at 3500 RPM is 216 W for duration of 1200 s with a temperature rise of 46°C.

4. MODELLING OF HIGH-SPEED SPINDLE
In the present work, a CNC vertical machining Centre (VMC) model spark-E (ACE Micromatic) is taken for the study of the thermo-mechanical functioning of the high-speed spindle bearing system under static and dynamic conditions. The spindle is mounted in the headstock housing supported by DB (O configuration) bearing arrangement. The abutment dimensions and bearing specifications are shown in Table 1 and Figure 2 respectively. The SOLIDWORKS software version, 2016 (Dassault Systems(2016)), is used for the CAD drawings, presented in Figure 2and ANSYS work bench version 16.0 for analysis. The maximum operating spindle speed is up to 6000 RPM. The bearings are preloaded by mechanical locknut through cylindrical spacers under medium preload conditions.

The spindle material is EN36C which is case-carburized and hardened steel up to 58HRC. The standard bearing materials are as per SAE 103Cr1 series, which are used up to 6000 RPM. Hybrid
bearings with ceramic balls and steel rings are used for higher spindle speeds beyond 6000 RPM for the better thermomechanical performance of the spindle bearing system.

**Figure 2**: Geometrical dimensions of the Spindle

### 5. THERMAL MODELLING

The structural and thermal processes in a bearing are combined. This integration demonstrates itself in the thermal deformation of structural components and the change of thermal properties due to heat transfer through spindle assembly mechanical parts. The temperature at which a bearing operates based on factors, such as operating speeds, applied load, bearing arrangement, lubricant properties, housing design, and working environment. All these factors affect either heat generation or heat transfer. The thermal analysis includes heat generation and heat transfer through different modes within the bearing and spindle.

**Table 1**: Abutment dimensions and load ratings Schaeffler group(2008a)

| Bearing Designation | Boundary dimensions (mm) | Abutment Dimensions (mm) | Basic load rating (kN) | Preload (N) | Number of balls | Ball diameter (mm) |
|---------------------|--------------------------|--------------------------|------------------------|-------------|-----------------|-------------------|
|                     |                          |                          | Dynamic               | Static      |                 |                   |
| B7009ETP4S          | 45 75 16 1 1             | 53.6 64.2 0.6            | 26.5                   | 20.0        | 202             | 7.144             |
| HCB7009ETP4S        | 45 75 16 1 1             | 53.6 64.2 0.6            | 18.0                   | 14.0        | 90              | 7.144             |

**Table 2**: Material property CMTI Machine tool design handbook(2015)

| Component           | Material     | ρ (Kg/m³) | K (W/m K) | E (GPa) | ν   | α (× 10⁻⁶) | σᵧ (MPa) |
|---------------------|--------------|-----------|-----------|---------|-----|------------|----------|
| Spindle             | 21Cr1Mo28    | 7870      | 45        | 210     | 0.3 | 11.9       | 770      |
| Bearing race        | SAE103Cr1    | 7860      | 46        | 208     | 0.3 | 11.9       | 2500     |
| Steel bearing ball  | SAE103Cr1    | 7860      | 46        | 208     | 0.3 | 11.9       | 2500     |
| Ceramic bearing ball| Si₃N₄        | 3200      | 32        | 314     | 0.27| 3.2        | 700      |
| Free cutting steel  | 14Mn1S14     | 7750      | 45        | 210     | 0.3 | 11.9       | 490      |
The heat generated in the steel bearing is estimated due to the effects of cutting load, viscous friction and spin friction.

5.1. Heat generation due to load:
The primary heat generation 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 the bearing is computed as Harris and Kotzalas (1996)

\[ H_b = 1.047 \times 10^{-4} n M_f \]  

Where \( H_b \), \( n \) and \( M_f \) are heat generated, rotational speed and total frictional torque respectively

The total frictional torque \( M \) consists of three components, torque \( M_1 \) which is due to cutting force and the other torque \( M_2 \) due to viscosity of the lubricant, and \( M_s \) is the spin-torque, \( M_f = M_1 + M_2 + M_s \).

5.2. Frictional torque due to cutting force (\( M_1 \)):
The torque due to cutting force is calculated as Harris and Kotzalas (1996)

\[ M_1 = f_1 \times F_\beta \times d_m \]  

Where \( f_1 \) denotes the factor based upon bearing design and corresponding load. For angular contact ball bearings is given as Schaeffler group (2008b)

\[ f_1 = z (F_s C_s) Y \]  

The static equivalent load, \( F_s \) is given as (Schaeffler group (2008b))

\[ F_s = X_o F_r + Y_o F_a \]  

For angular contact ball bearings, the dynamically equivalent load \( F_\beta \) depends upon the magnitude and orientation of the applied load, and is given as Schaeffler group (2008b)

\[ F_\beta = \max (0.9F_a \cot \theta - 0.1F_t) \]  

Solving the equation (2) by taking pitch diameter of bearing \( d_m = 60 \) mm, the torque developed due to applied load is given as \( M_1 = 7.4 \) N mm.

5.3. Viscous Friction Torque (\( M_2 \)):
For bearings, the viscous friction torque is expressed empirically as, Harris and Kotzalas (1996)

\[ M_2 = 10^{-7} f_o d_m^3 (v_o n)^{2/3} \]  

The value of \( v_o = 20 \) is taken at the uniform temperature of 40\(^\circ\)C. \( f_o \) is the factor depending upon the type of bearing and method of lubrication. \( f_o = 2 \) (adapted from Harris and Kotzalas (1996)), \( M_2 = 48.02 \) N mm, Total \( M = 55.42 \) N mm.

The heat generated in the bearing is computed as \( H_b = 12.128 \) W at 2000 RPM. Similarly, the heat generated at various speeds is calculated for steel and hybrid bearings, as shown in Figure 3. The heat generated in the hybrid bearing is lower due to parameters such as \( F_a, F_s, f_1 \) and \( f_o \) in comparison with steel bearing. The heat generated in hybrid bearing is 10.18 w at 2000 RPM.
5.4. Heat generation due to spin:

\[ M_s = \frac{3\mu a}{{8}} \varepsilon F_p = 35.43 \times 10^{-3} \text{ (N-mm)} \]  

(7)

Where \( \mu \), \( a \), \( \varepsilon \), and \( F \) are the friction factor, the major axis of the elliptical contact region, the elliptical integral of 2\textsuperscript{nd} kind over the contact zone and preload respectively as presented by \( \mu = 0.01 \), \( a=3 \), \( \varepsilon = 0.7 \), and \( F_p = 300 \text{ N} \).

The thermal preload \( F_{th} \) on the bearings due to the thermal deformations is given as Takabi and Khonsari(2013)

\[ F_{th} = K_r \varepsilon t^{n_1} \]  

(8)

Where \( K_r \), \( \varepsilon t \) and \( n_1 \) denote the radial elastic constant (adapted from Harris and Kotzalas(1996)), thermal displacement (based on contact stress theory) of the bearing, and constant based on the type of bearing respectively.

From the thermal model of the bearing, the cutting load-induced friction torque is lower, whereas the viscous friction torque is higher at the same spindle speed. Hence, as the spindle speed increases, the heat generation increases, this causes the thermal deformation of the bearings thus leading to an increase in the thermal preload. The temperature increases in the bearing by thermal conductive heat transfer network and temperature is controlled by grease packed in bearings(Jørgensen and Shin(1997)). The thermal preload is added to the initial preload (through lock nut), and the spindle speed is 6000 RPM as the FEM boundary conditions.

6. Finite element formulation of the spindle:

The spindle is discretized as one-dimensional beam elements by applying Euler –Bernoulli beam theory. The spindle is idealized as a five node beam element having two degrees of freedom (one translational and one rotational) at each node, as shown in Figure 4 and Figure 6. The FEM dynamic model for the spindle bearing system constitutes discretized shaft elements with coupling effects. The model is similar to be a rotor-dynamic system. Both static and dynamic analysis is performed by imposing appropriate loading and boundary conditions. The cutting load is applied on the node 1 and thermal load is applied at nodes 2, 3 which are bearing supports.
Global Nodal displacement vector, \( \mathbf{d} = [\mathbf{x}_1 \mathbf{\theta}_1 \mathbf{x}_2 \mathbf{\theta}_2 \mathbf{x}_3 \mathbf{\theta}_3 \mathbf{x}_4 \mathbf{\theta}_4 \mathbf{x}_5 \mathbf{\theta}_5]^T \) (9)

Where \( \mathbf{x}_i \) and \( \mathbf{\theta}_i \) are respective node deflection and slope.

The governing equation of motion for the spindle deformations under static and dynamic conditions is given by,

\[
[M][\ddot{\mathbf{d}}] + \omega [G] [\dot{\mathbf{d}}] + (\mathbf{K} - \omega^2 [M_\omega])\mathbf{d} = \mathbf{F}(t)
\] (10)

Where, \([M] \), \([G] \), \([K] \) and \([F] \) denote assembled mass, skew symmetric gyroscopic, centrifugal, stiffness and force matrices respectively. The term \( \omega^2 M_\omega d \) represent the softening effect of centrifugal force which reduces the stiffness of the spindle bearing system and the term \( \omega [G] [\dot{\mathbf{d}}] \) represent gyroscopic moment. The matrices are calculated based on the geometry, speed and mass of the spindle shaft.

The stiffness matrix for \( i_{th} \) element of the beam is given as,

\[
[K_i] = \frac{E I}{l^3} \begin{bmatrix}
12 & 6l & -12 & 6l \\
6l & 4l^2 & -6l & 2l^2 \\
-12 & -6l & 12 & -6l \\
6l & 2l^2 & -6l & -4l^2
\end{bmatrix}
\] (11)

Global stiffness matrix \([K] = [K_1] + [K_2] + [K_3] + [K_4] \)
Elemental mass matrix, \([M_i] = \frac{\rho A l_i}{2} \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}\) (12)

Global mass matrix \([M] = [M_1] + [M_2] + [M_3] + [M_4]\) (Jorgensen and Shin(1998))

Elemental gyroscopic matrix \([G_i] = \frac{\rho l_x}{30(i + \omega)^2} \begin{bmatrix} -36 & 3l(1 - 5\omega) & 36 & 3l(l - \omega) \\ -3l(1 - 5\omega) & l^2(4 + 5\omega + 10\omega^2) & -36 & -l^2(1 + 5\omega - 5\omega^2) \\ 36 & -3l(1 - 5\omega) & -36 & -3l(1 - 5\omega) \\ -3l(1 - 5\omega) & -l^2(1 + 5\omega - 5\omega^2) & 3l(1 - 5\omega) & l^2(4 + 5\omega + 10\omega^2) \end{bmatrix}\) (13)

Global gyroscopic matrix \([G] = [G_1] + [G_2] + [G_3] + [G_4]\)

Elemental centrifugal matrix \([K_{c,i}] = m \omega^2 \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix}\) (14)

Global centrifugal matrix \([K_c] = [K_{c,1}] + [K_{c,2}] + [K_{c,3}] + [K_{c,4}]\)

The centrifugal force, gyratory moments, thermal, and cutting loads are considered in the dynamic analysis. During HSM, the bearing ball experiences gyroscopic moment due to an increase in friction by the sliding motion. The gyratory moment, \(G\) on each bearing ball is calculated as Harris and Kotzalas(1996).

\[G = J \omega_o \omega_b \sin \beta\] (15)

Where, mass moment of inertia of each ball, \(J = \frac{1}{60} \times \rho \pi D^5\)

\(\omega_o, \omega_b, n_c, n_b\) denote the orbital speed of the ball, the rotational speed of bearing element, cage speed (RPM), and ball speed (RPM), respectively.

![Graph](image)

**Figure 6:** Gyroscopic couple and Centrifugal effects on high-speed spindle

The centrifugal force \((F)\) at the bearings calculated as \(F = m R \omega^2\) (16)

Where \(m, \omega, \) and \(R\) denote the mass, spindle rotational speed, and pitch circle radius of the bearing, respectively.
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Table 3: Gyratory and Centrifugal force

| Speed in RPM | Gyratory moment (Nmm) | Centrifugal force (N) |
|--------------|-----------------------|----------------------|
|              | Steel bearing         | Hybrid bearing       | Steel bearing | Hybrid bearing |
| 6000         | 2.401                 | 1.113                | 17.760        | 7.231         |
| 12000        | 9.604                 | 4.451                | 71.043        | 28.923        |

The gyroscopic moment and centrifugal force are functions of rotational speed. At 6000 RPM, the gyroscopic moment is 2.401 N-mm and has little effect on the spindle bearing dynamics. But at higher spindle speeds, the frequency value beyond 1st order is getting lowered. The effect of centrifugal force is to produce a spin softening effect and reduces the bearing stiffness. Further, the reduced bearing stiffness lowers the natural frequency by 5% up to 12000 RPM, as presented by Wang et al. (2011)

Table 4: Tangential cutting force and bearing temperatures during high-speed machining

| S.No | Cutting velocity, m/min | Spindle speed, RPM | DOC, mm | Feed, mm/min | MRR, cc/min | Tangential cutting force, N | Power at cutter (kW) | Power at motor (kW) | Bearing temperature, °C |
|------|-------------------------|-------------------|--------|--------------|-------------|---------------------------|----------------------|----------------------|------------------------|
| 1    | 500                     | 1989.4            | 2      | 994.7        | 119.36      | 310                       | 2.6                  | 3.06                 | 26                     |
| 2    | 800                     | 3183              | 1.5    | 1591         | 143.24      | 232.5                     | 3.1                  | 3.65                 | 28.5                   |
| 3    | 1000                    | 3978              | 1      | 1989         | 119.37      | 155                       | 2.6                  | 3.06                 | 31.2                   |
| 4    | 1250                    | 4973.6            | 0.9    | 2486         | 134.29      | 139.4                     | 2.9                  | 3.41                 | 33.8                   |
| 5    | 1500                    | 5968              | 0.75   | 2984         | 134.29      | 116.3                     | 2.9                  | 3.41                 | 36.9                   |

DOC and MRR represent the depth of cut and material removal rates, respectively.

During machining of free cutting steel 14Mn1S14, the tangential cutting force decreases as the cutting velocity increases. However, the material removal rate increases with cutting velocity. The spindle power utilized also increases with the material removal rate. Experimentally, the temperature observed is 36.9 °C using an infrared temperature gun on the spindle inside at front bearing zone. From Figure 7, the bearing temperature steadily increases, although the tangential force is decreasing. This experimental result indicates that at higher cutting velocities, the viscous friction in the bearing is predominant to the applied tangential force in the increase of bearing temperature. Since feed rate is increasing with increase in cutting velocity and MRR is increasing (Altintas and Cao, 2005)

Tangential Cutting Force, \( P_z = \frac{6120 \times P_w}{V_c} \) (CMTI Machine tool design handbook(2015)).

As the spindle power is calculated based on the cutting parameters, the observations from the Table 3 are the depth of cut is reduced for chatter free cutting. Hence the tangential cutting force is decreasing.

\[
\text{Cutting velocity}, V_c = \frac{\pi D n}{1000} \text{ m/min} \quad \text{n is the spindle rotational speed (RPM)}.
\]

\[ D \text{ is the diameter of the work material and cutting tool for turning and milling respectively} \]

\[ f_r = Z \times f_z \text{ mm/rev}, f_m = f_r \times n \text{ mm/min} \]

\[ f_m, Z, f_z \text{ is the feed per minute, number of teeth, and feed in mm/tooth respectively.} \]

\[
\text{Power at tool tip}, P_w = UK_n K_r Q \quad \text{(CMTI Machine tool design handbook(2015))}
\]

\[
\text{Material removal rate}, Q = f_m d V_c \quad \text{(CMTI Machine tool design handbook(2015))}
\]
Material removal rate (Q), unit power (U), Correction factors for flank wear ($K_n$) and rake angle ($K_r$).

![Figure 7: Effect of cutting speed on tangential cutting force and bearing temperature](image-url)

6.1. For static analysis:
An analysis is carried out by mechanical lock nut fixed bearing preload and interference effects at the support locations along with a static load of 300 N at node 1 (cutting load). The beam deflections and slopes are computed from static deflection equation of FEM. (17).

$$
[d] = [K]^{-1} ([F] + [F_c])
$$

$$
= 10^{-5} [3.022 58.9 1305 44.18 49.61 -10.32 -30.23 1.085 -24.97 5301]^T
$$

6.2. For dynamic analysis:
Initially, the analysis is carried out with beam inertia, gyroscopic, centrifugal, and cutting load effects as the inputs. The structural stiffness matrix is coupled with a centrifugal matrix. Hence,

$$
[K]_{\text{global}} = [K] + [K]_{c}
$$

The beam deflections and slopes are computed from (2).

$$
[d] = [Z]^{-1} ([F] + [F_c])
$$

Where,

$$
Z = -\omega^2 \text{Sin}(\omega t)[M] + \omega \text{Cos}(\omega t)[G] + \text{Sin}(\omega t)[K]_{\text{global}}
$$

$$
= 10^{-10} [-1836 -51.62 -9392.1 2506.8 -4310 4.219 -8164 -2.324 9422 18.41]^T
$$

6.3. For dynamic with thermal load:
A similar analysis is performed by incorporating the thermal load and the beam deflections and slopes are computed using (19).

$$
[d] = [Z]^{-1} ([F] + [F_c] + [F_{ch}])
$$

$$
= 10^{-10} [-4642 -236 -2.49 \times 10^6 2506.3 -1.55 \times 10^5 35.01 -2 \times 10^5 6.33 -9399 13.49]^T
$$
The spindle deflections and slopes with static and dynamic (with and without thermal load) analysis are calculated using equations (21), (22), and (23), the results are presented in Table 5 and Table 6 along with trend graphs shown in Figure 9 (a), (b), (c) and (d).

7. Analysis of the high speed spindle

The stress induced, displacement and strain are extracted using ANSYS finite element analysis at 2000, 3000, 4000, 5000 and 6000 RPM for thermo - Structural behavior by applying fixed bearing boundary conditions with and without external cutting load. Variation in the displacement, strain and stress w.r.t spindle rotational speeds for steel bearing and hybrid bearing are shown in Figure 10 (a) (b) (c) and (d). The maximum Von-Mises stress and deformation in steel bearing are 439.87 MPa, and 0.71 µm whereas the same in hybrid bearing are 460.67 and 0.69 respectively (Lu et al. (2007b) [18]. The stress in the hybrid bearing is higher than steel bearing due to low thermal conductivity and deformation is lower than steel bearing due low coefficient of expansion. Therefore hybrid bearings are preferred for high speed spindles and also lower centrifugal force and Gyratory moment due to lower density as shown in the Table 2.

![Figure 8: Analysis results of spindle with (a), (b) steel and (c), (d) hybrid bearings](image-url)
Table 5: Slope and Deflections without cutting load at 6000 RPM

| Node number | Deflections |   |   |
|-------------|-------------|---|---|
|             | Static      | Dynamic | Dynamic with thermal load |
|             | Translation (x, µm) | Slope, rad ($\theta \times 10^{-6}$) | Translation (x, µm) | Slope, rad ($\theta \times 10^{-6}$) | Translation (x, µm) | Slope, rad ($\theta \times 10^{-6}$) |
| 1           | 0.0133      | 258 | -1.818x10^{-3} | -2.278x10^{-3} | -4.635x10^{-2} | -2132 |
| 2           | 0.2066      | 211 | -0.955x10^{-2} | -2.018x10^{-4} | -0.241 | 0.927x10^{-3} |
| 3           | -0.0836     | 0.65 | -2.461x10^{-6} | 1.793x10^{-3} | 0.1946 | -4.727x10^{-3} |

Table 6: Slope and Deflections with cutting load at 6000 RPM

| Node number | Deflections |   |   |
|-------------|-------------|---|---|
|             | Static      | Dynamic | Dynamic with thermal load |
|             | Translation (x, µm) | Slope, rad ($\theta \times 10^{-6}$) | Translation (x, µm) | Slope, rad ($\theta \times 10^{-6}$) | Translation (x, µm) | Slope, rad ($\theta \times 10^{-6}$) |
| 1           | 0.0302      | 589 | -1.836x10^{-3} | -5.1623x10^{-3} | -4.642x10^{-3} | -2.36x10^{-3} |
| 2           | 13.05       | 441.8 | -9.39x10^{-3} | 0.2506 | -0.249 | 0.2506 |
| 3           | 0.4961      | -103.2 | -4.31x10^{-4} | 4.219x10^{-6} | -1.559x10^{-2} | 3.501x10^{-3} |
| 4           | -0.302      | 10.85 | -8.164x10^{-4} | 2.324x10^{-4} | -2.002x10^{-2} | 6.336x10^{-4} |
| 5           | -0.2497     | 5301 | -9.422x10^{-4} | 1.841x10^{-3} | -9.399x10^{-3} | 1.349x10^{-3} |

(a) Without cutting load

(b) With cutting load
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8. RESULTS AND DISCUSSION

1. The angular contact ball bearing with lifetime Li-based grease lubrication in a tandem arrangement is used. The heat generated at 6000 RPM is 70.66 W, and the bearing temperature is 41.7 °C at an ambient temperature of 28°C. Analytical and experimental studies are performed using the same bearing configuration and specifications. The variation in temperature of 4.8 °C is due to conductive and convective heat transfer in the experimental work. The bearings are provided with grease lubrication up to 6000 RPM and oil jacket cooling beyond 6000 RPM to ensure reliable thermal bearing performance during high-speed machining (Lu et al.(2007a)).
value is considered for estimating the thermal preload on the bearing and is computed 39.99N from the equation (12). The critical speed of the spindle is increased with increased thermal preload up to a certain extent beyond the bearing failure takes place.

2. The coupling of gyratory, centrifugal, and cutting load has significant effect on the spindle dynamic deflections. From the Figure 6(a) and (b) and Table 3, it is observed that the gyroscopic moment and centrifugal force in hybrid bearing are 46.36% and 40.72% of steel bearing respectively at all spindle speeds and effect are minimized by using hybrid bearing. A design guideline is to use hybrid bearings beyond 6000 RPM to minimize the coupling effects as presented in the table 3.

3. It is observed from Table 2, that the tangential cutting force decreases with increased speed. Further, it also depends on other cutting parameters and spindle-motor characteristics (power – torque-speed). The correlation of two variables (cutting force, bearing temperature) with cutting velocity is shown in figure 7. The spindle attains its full power utilization with a tangential cutting load of 116.3 N during machining, as shown in Table 4.

4. Table 5 represent the FEM solutions of spindle deflections and slope at nodal points with and without the application of cutting loads in the force vector, respectively. The nodal displacements and slopes are shown in Figure 9 (a), (b), (c), and (d). The maximum spindle deflection by FEM solution is 13.5 µm, whereas, the simulation result is 0.71 µm as shown in the Figure 8. The results are within the limit as per the CNC machine tool geometrical testing standards. O Klicay(1983) determined the deflection of the spindle using the beam theory for a simply supported with overhanging load condition.

5. The ANSYS simulations for steel and hybrid bearings are performed up to 6000 RPM to determine the Von-mises stress and displacement. The results are shown and plotted in figures 8 (a) (b) (c) and (d), 10(a) and (b). The results indicate that the high-speed spindle is stable with 2.5 µm deflection by attaining equilibrium state due to speed-related effects compared with the results of Uhlmann and Hu(2012) at 12000 RPM.

9. CONCLUSION
A comprehensive process is presented to predict the heat generation in the angular contact ball bearings at different operating speeds. The bearing oil cooling is required to be implemented in the high-speed spindle for reliable thermal performance during HSM. The high-speed spindle dynamic analysis is more complicated due to speed-dependent non-linear and time-varying effects when compared to the low-speed spindle. FEM solutions are implemented using the Euler- Bernoulli beam with coupling effects to obtain the nodal deflections and slopes of HSS up to 6000RPM.

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