



Sensitivity analysis of rotating auxiliary components of a high-speed spindle shaft assembly

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Abstract

Machine tool spindles, complex dynamic systems under thermal growth, are significantly influenced by the shaft assembly's dimensions, including the spindle shaft and various press-fit auxiliary components. Despite the importance of these dimensions, their effects on the spindle's dynamic and thermo-mechanical behavior have yet to be sufficiently explored. This study employed a detailed sensitivity analysis for numerous auxiliary components to identify dimensions that most effectively improve the frequency response function (FRF) and minimize thermal growth. The dynamic and thermal analyses were conducted using the receptance coupling substructure analysis (RCSA) and a reduced-order model of finite element model (ROM-FEM), respectively. The findings revealed that, for the considered design, the front stepped sleeve, spindle tail components, and the shaft's central hole are the most sensitive elements that offer an opportunity to improve spindle performance after bearing optimization.

Keywords Spindle · Sensitivity · Dynamics · Thermal · Structural modification

1 Introduction

High-speed spindles that achieve speeds higher than 20,000 RPM generally comprise a built-in electric motor, super precision angular contact ball bearing (ACBB) sets, an oil-air lubrication system, and motor and bearing cooling systems, together with an automatized drawbar system with through or shaft cooling [1]. Although they are costly,

these systems are necessary as high-speed spindles require torque, precision, and robustness simultaneously [2]. These requirements dictate that high-speed spindles need to have a well-balanced trade-off between dynamic and thermal behaviors. Therefore, it is necessary to use these systems in spindle design to meet desired thermal and dynamic criteria. High-speed applications introduce several challenges in spindle design, such as cutting forces and chatter, centrifugal forces and gyroscopic moments in rotating components, bearing configurations in ACBB-mounted spindles, bearing life and seizure risks, initial and thermally induced preloads in bearings, and excessive heat generation [1–3]. Therefore, using spindle dynamics and thermal models to predict system behavior becomes essential. In the literature, one of the most common techniques to predict spindle dynamics includes RCSA [1, 4] and spindle-specific finite element (FE) models [1, 5]. Generally, these models simplify the spindle assemblies as shaft and bearing systems together with tool-holders, which results in acceptable predictions. According to these metrics, numerous publications [6–8] have reported that the tooltip FRF of a given spindle assembly can be made more rigid by tuning the tool overhang and tool holder geometry. Erturk et al. [6] also reported on the dynamic absorber effect observed when the length of the spindle's tail section is increased. The

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dynamic absorber effect is characterized as when multiple dynamic modes merge and lessen each other's response. Schmitz et al. [7] provided further context by noting that the dynamic absorber effect occurs when the fundamental tool natural frequency matches with the natural frequency of the spindle-holder natural frequency. Tong et al. [9] used the particle swarm optimization (PSO) algorithm to optimize a spindle's static stiffness, natural frequencies, and friction torque. They treated the bearing preloads and positions on the shaft as design variables in their optimization problem. However, the interaction of thermomechanical and dynamic behaviors becomes more dominant in high-speed spindle applications, requiring a holistic spindle modeling approach. Li and Shin [10], in their benchmark study on the thermo-mechanical modeling of high-speed spindles, developed a modular approach to characterize the interactions between various thermal and dynamic behaviors occurring in a high-speed spindle by considering several spindle components, such as housings and motor cooling rather than only the shaft-bearing system. Their work also differentiated the effects of different bearing configurations and preloading mechanisms. The model employed in [10] was based on the thermal model of Hong et al. [11]. Before these studies, the heat generation characterization was made by Bossman and Tu for high-speed applications [12]. They also suggested a thermal model for high-speed spindles with an in-depth investigation of structural and operational details in a spindle [13]. Additionally, Jorgensen and Shin [14] demonstrated how to map bearing configurations in high-speed spindle dynamics and thermal predictions and proved that the "O" configuration increases spindle rigidity for ACBBs with drastic effects on dynamic and thermal behaviors. Regarding high-speed spindle bearings, Cao and Altintas employed a dynamic model and experimentally verified different bearing configurations and preloading mechanism effects [15]. They demonstrated that, for spring/constant preloading mechanisms, extra axial preload from the spindle tip causes an increase in front bearings' stiffness while maintaining rear bearings' stiffness until a threshold point, which causes modal frequencies to shift to higher frequencies. This dynamic response is in agreement with the thermal and dynamic findings from [10, 16, 17], for cases where thermally induced preloading is considered for spring/constant preloading mechanisms. Consider a spindle with a spring/constant preloading mechanism that operates at high speeds, where the springs are connected to the rear bearing set. After a while, due to heat generation, the front bearings expand while being positionally locked, generating thermally induced preload in the front bearing. At the same time, the shaft also expands due to the elevated temperatures, carrying the rear bearing set with them. The rear bearing set, preloaded by soft springs, can move axially [10, 16]. The springs absorb the axial expansion of

the shaft due to thermally induced preload and temperature rise until a threshold value, thus maintaining relatively constant preload and avoiding over-preload during high-speed applications where the speed exceeds 20,000 RPM. At the end of this thermal expansion loop, front bearings' stiffness increases due to thermally induced preload, causing excessive heat generation and worsening bearing life. The stiffness of rear bearings stays preserved until a threshold value or speed. Some high-speed spindle manufacturers like Franz Kessler use spring/constant preloading at the front bearings to avoid such cases in high-speed spindles as a solution. A very limited number of studies focus on both thermal and dynamic interaction-based optimization studies in the literature since they require consideration of many details [17]. However, these studies only focus on altering spindle shaft dimensions and bearing positions rather than spindle auxiliary components such as stepped sleeves, bearing spacers, and labyrinth seals, which are generally employed in high-speed spindle design. This work aims to provide a spindle sensitivity analysis that includes the possible rotating auxiliary components that alter the dimensions of the spindle shaft assembly. A time-efficient and comprehensive methodology, combining RCSA dynamic analysis and FEM-ROM thermal analysis, is utilized to perform this sensitivity analysis. Additionally, to the best of the authors' knowledge, this study will be the first attempt to demonstrate the sensitivity of spindle dynamics and thermal behavior to the central shaft cooling holes. The procedure used in this study can serve as an intermediary step in the spindle design process, helping spindle designers fine-tune the shaft-bearing system through sensitive auxiliary component dimensions, especially after the gross dimensions of the shaft and bearing locations have been finalized, and before final artifacts like the tool holder, tool, and drawbar are introduced into the modeling scheme. This article is structured as follows: Section 2 introduces various auxiliary shaft components. Section 3 discusses this study's methods to predict dynamic and thermal performance. In Section 4, the spindle design and findings of the sensitivity analysis are shared. Section 5 presents the conclusions based on the sensitivity analysis results.

2 Rotating auxiliary components and shaft design

The shaft assembly comprises numerous components affecting both dynamic and thermal performance. The most significant of these are the ones with high mass or stiffness contributions to the structure or in close vicinity to heat sources. This section explains the functional purpose of the main rotating auxiliary components in high-speed spindles.

2.1 Press-fitted sleeves

Press-fitted sleeves on a spindle shaft can be categorized as motor and stepped sleeves. The additional mass and stiffness press-fitted sleeves change the dynamic response of the employed spindle. Since the locations of press-fitted sleeves are generally close to heat sources such as bearings and motor, they also contribute to thermal dissipation via enhancing forced convection heat transfer. This is because their assembly on the shaft enlarges the rotating surface area. Moreover, their press-fit interface also adds a new thermal resistance medium between the heat sources and the spindle shaft, which is not the case when the same geometry is achieved by designing a shoulder in the shaft.

2.1.1 Motor sleeves

Motor sleeves are employed in electro-spindles to assemble the motor to the shaft. The spindle housing contains the stator while the rotor is mounted on the shaft over a press-fitted sleeve. Generally, these sleeves are mounted on the shaft by creating a temperature difference via nitrogen cooling to cause sufficient shrinkage of the shaft. The dismantling operation is performed by injecting pressurized oil at the sleeve-shaft interface, causing the sleeve to expand and slide out. Pockets for accommodating this oil are a part of the motor sleeve design. As the rotor occupies a significant portion along the length of the shaft and the assembling and dismantling technique alters a portion of a spindle shaft that has a non-negligible mass and is located under a primary source of heat, it is a crucial auxiliary component in the context of thermal and dynamic behaviors.

2.1.2 Stepped sleeves

Stepped sleeves are designed to lock or position the spindle bearings or built-in motors in their correct axial positions. These sleeves are designed with an internal step and are press-fitted onto the shaft using high interference fits to provide extra stiffness against bending moments and axial displacements. Depending on the design, the interference fit may occur on either one or both portions of the internal step. Moreover, they can serve as labyrinth seal rotors for oil leakage prevention and contamination. Although they can be complex to manufacture and assemble, stepped sleeves make it easy to access critical spindle components during maintenance and breakdowns while providing the necessary stiffness for high-speed spindles to operate smoothly [18].

2.2 Bearing spacers

Bearing spacers are ring-like sub-assemblies that sit between the bearing sets. They have a rotor on the spindle shaft and

a stator part assembled on the spindle housing. They can be employed as part of the bearing preloading mechanism by adjusting their width via grinding [18]. A spacer stator might have lubrication/air cooling channels, while the rotor has drainage channels in some high-speed spindles. This study focuses on spacer rotors since the sensitivity of the rotating auxiliary components is investigated.

2.3 Locknuts

Locknuts are precision accessories used to locate and fix the spindle shaft components accurately. The locknuts can locate, fix, and preload spindle bearings [18]. The rigid preloading is applied by tightening the locknut against the inner part of the bearing while the other ring remains in its initial position. This creates a relative axial displacement between both bearing rings, causing the rolling elements to be loaded, inducing the preload. Generally, constant or spring preload is applied in high-speed spindles to maintain a smooth high-speed operation. Precision locknuts are also employed to adjust the spring position to adjust constant/spring preloads. This study employs spring preload with an “O” bearing configuration. Thus, the shaft assembly design in Fig. 7 features a locknut at the rear side of the assembly.

2.4 Shaft geometry

The spindle shaft has various geometric features that are altered by auxiliary components that affect spindle thermal and dynamic performance. The dimensions of some of these features are adjustable at the design stage. The main shaft features are spindle tail, shaft central hole, and shaft shoulders. The spindle tail comprises all parts of the shaft assembly extending beyond the last bearing after the rear bearing set. This can include components like locknuts, sleeves, rotary union, and the extended portion of the shaft itself. Considerable research on the dynamic absorber effect created by tuning the tail mass has been reported in the literature [4, 6].

The shaft also features a hole at its center to accommodate the drawbar mechanism with a toolholder and cooling channels for through- and shaft cooling. These holes considerably alter the dynamic response of the shaft due to the change in mass and stiffness of the shaft. Additionally, since cooling fluids also flow through these holes, they also influence bearing/motor temperatures and thermal growth of the shaft. In practice, the central hole size may not be very flexible to change due to spindle designers being limited to standard drawbar and toolholder size. However, the central hole size has been investigated in this study to serve as a guideline for selecting drawbar mechanisms and designing shaft cooling channels. Shaft shoulders are also considered

design variables in this study. Shoulders are useful in locating bearings and auxiliary components along the shaft axis.

3 Dynamic and thermal modeling

Bearing modeling is the primary building block for both calculations. A five degrees-of-freedom quasi-static model for angular contact ball bearings (ACBBs) is employed in dynamic and thermal models in this study. The model assumes a fixed outer ring, and the inner ring is displaced under loading. RCSA is a practical tool to predict spindle dynamics with composite Timoshenko beam theory that allows using different materials for the auxiliary components. A quasi-static FE model is employed to predict thermal behavior.

3.1 Bearing modeling

Figure 1 shows the global coordinate system, the local coordinate system, and the free-body diagram for an arbitrary rotating element of an ACBB. In Fig. 1a, the external load vector, $\{F\}$, and the displacement vector, $\{\delta\}$, are defined, respectively.

$$\{F\} = \{F_x, F_y, F_z, M_x, M_y\}^T, \{\delta\} = \{\delta_x, \delta_y, \delta_z, \gamma_x, \gamma_y\}^T \quad (1)$$

Figure 1b shows the displacement vector of the inner ring cross-section and the corresponding inner ring contact load vector defined at the local ball.

$$\{u\}^T = \{u_r, u_z, \theta\}, \{Q\}^T = \{Q_r, Q_z, T\}, \quad (2)$$

In Eq. 2, u_r, u_z , and θ denote as the radial displacement, axial displacement, and rotation of the inner ring while

Q_r, Q_z , and T correspond to the radial and axial contact load and the contact moment at the local ball. Figure 1c displays the free-body diagram for the bottom ball of an ACBB. Here, Q_i and Q_e denote the contact loads between the inner and outer rings, respectively. Their corresponding contact angles can be expressed as α_i and α_e . The centrifugal force F_c of the ball is expressed in Eq. 3. m , d_m , and ω_c are the mass of the ball, bearing pitch diameter, and ball orbital speed, respectively. The gyroscopic moment M_g is expressed by Eq. 4.

$$F_c = 1/2 m d_m \omega_c^2 \quad (3)$$

$$m_g = 1/10 m D_B^2 \omega_c \omega_r \sin \beta \quad (4)$$

In Eq. 4, D_B is the diameter of the ball, and ω_r and β are the ball's rotational speed and the pitch angle, respectively. The contact loads at ball-inner ring contact and the ball-outer ring contact are obtained from the Hertz contact theory as shown in Eq. 2.

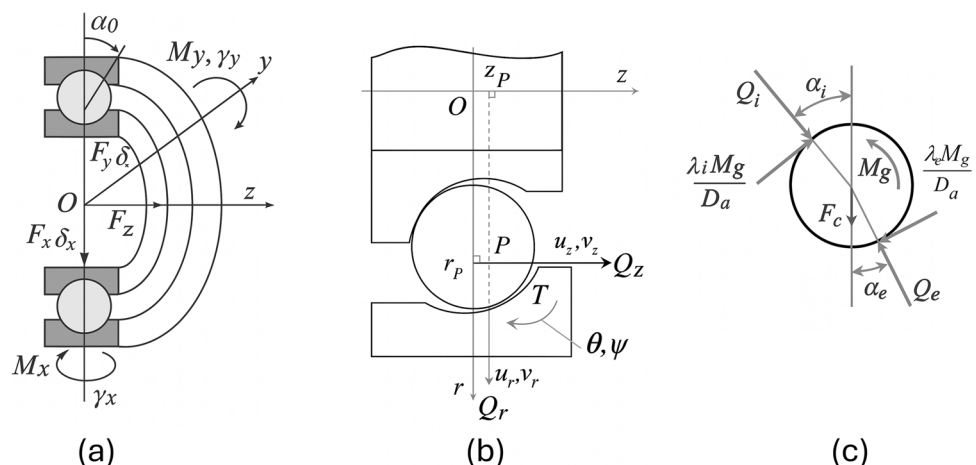
$$Q_i = K_i \delta_i^{1.5}, Q_e = K_e \delta_e^{1.5} \quad (5)$$

Here, δ_i and δ_e stand for deformation between the inner ring and the ball and the outer ring and the ball, and K_i and K_e are the corresponding load–displacement proportional coefficients. The free-body diagram in Fig. 1c is used to derive Eq. 3 as follows:

$$Q_i \cos(\alpha_i) - Q_e \cos(\alpha_e) + F_c - \frac{M_g}{D} (\lambda_i \sin(\alpha_i) - \lambda_e \sin(\alpha_e)) = 0 \quad (6)$$

In Eq. 3, λ_i and λ_e are the support ratios of the inner and outer rings to the gyroscopic moment. In a high-speed spindle, it can be assumed that the outer race control mode

Fig. 1 Bearing coordinate systems and ball free-body diagram. **a** Global coordinate system, **b** local coordinate system, **c** free body diagram of the bottom ball



is a rolling motion based on the outer ring, leaving $\lambda_e = 2F$; otherwise $\lambda_i = \lambda_e = 1$. The equilibrium equation for the entire bearing in the global coordinate system is indicated in Eq. 4.

$$\{F\} + \sum_{j=1}^Z [T_\psi]_j^T \{Q\}_j = 0 \quad (7)$$

Here, Z is the number of balls, and $[T_\psi]_j^T$ is a transformation matrix from the center coordinates of the inner ring curvature radius to the global coordinates. Equations 3 and 4 are non-linear, and Eq. 3 is obtained while solving Eq. 4. In this process, all displacements of the bearing center coordinates are calculated.

The bearing stiffness is obtained from Eq. 7 as follows:

$$K = \left[\frac{\partial \{F\}}{\partial \{\delta\}} \right] = -\sum_{j=1}^Z [T_\psi]_j^T \left[\frac{\partial \{Q\}}{\partial \{u\}} \right] [T_\psi]_j \quad (8)$$

3.2 Dynamic modeling

3.2.1 Receptance coupling substructure analysis (RCSA)

RCSA is an analytical method to predict the dynamic response, or the frequency response function (FRF), of mechanical assemblies by coupling their individual substructures receptances. After it was first applied to machine tool spindles by Schmitz et al. [19] to predict tooltip FRF, it has been adapted as one of the primary spindle dynamics analysis tools.

To implement the RCSA method, the spindle shaft was divided into discrete segments at key locations. These locations are defined at points along the shaft axis with major geometric changes and at bearing locations. The frequency-dependent endpoint receptance functions, $R_{a,ij}$, were then computed by solving Eq. 5 for each shaft segment using the Timoshenko beam theory with free-free boundary conditions.

$$\begin{bmatrix} X_i \\ \theta_i \end{bmatrix} = \begin{bmatrix} h_{a,ij} & l_{a,ij} \\ n_{a,ij} & p_{a,ij} \end{bmatrix} \begin{bmatrix} f_j \\ m_j \end{bmatrix}, \text{ where } R_{a,ij} = \begin{bmatrix} h_{a,ij} & l_{a,ij} \\ n_{a,ij} & p_{a,ij} \end{bmatrix} \quad (9)$$

Here, X and θ denote translational and rotational displacements at the endpoint i , and f and m are the applied force and moment at endpoint j , respectively. Subscripts $i, j \in \{1, 2\}$ refer to the response and excitation ends of each substructure, respectively. Subscript a denotes the identity of the shaft segment. The matrix components, h_{ij} , l_{ij} , n_{ij} , and p_{ij} , are the different receptance forms of the beam, where h , l , n , and p represent

the displacement-force, displacement-moment, rotation-force, and rotation-moment receptance functions of the substructure. Thus, h_{12} , for instance, is the transverse vibration response of the beam at endpoint 1 when a linear harmonic force is applied at endpoint 2.

The RCSA method is then used to iteratively couple these individual substructure receptances. The receptance matrix for the two-segment beam, C, formed by rigidly coupling the beams A and B in Fig. 2, is given by Eq. 10 as interpreted from [20]:

$$\begin{bmatrix} [R_{C,11}] & [R_{C,21}] \\ [R_{C,21}] & [R_{C,22}] \end{bmatrix} = \begin{bmatrix} R_{A,11} - R_{A,12}(R_2)^{-1}R_{A,21} & R_{A,21} - R_{A,12}(R_2)^{-1}R_{A,22} \\ R_{A,21} - R_{A,22}(R_2)^{-1}R_{A,21} & R_{A,22} - R_{A,22}(R_2)^{-1}R_{A,22} \end{bmatrix} \quad (10)$$

The form of the matrix R_2 in Eq. 10 depends on whether the joint is rigid or non-rigid. For rigidly coupled shaft sections, as in Fig. 2, R_2 is represented by Eq. 11.

$$R_2 = R_{A,22} - R_{B,22} \quad (11)$$

For non-rigid connections with flexible components like bearings, the matrix R_2 is modified to include the bearing stiffness and damping matrix, K . The modified R_2 and K matrices are given in Eqs. 12 and 13, respectively. Figure 3 illustrates a non-rigid joint. The subscripts y and θ represent translation and rotation, respectively, in Eq. 13 while m indicates applied moments, and f is for applied linear force.

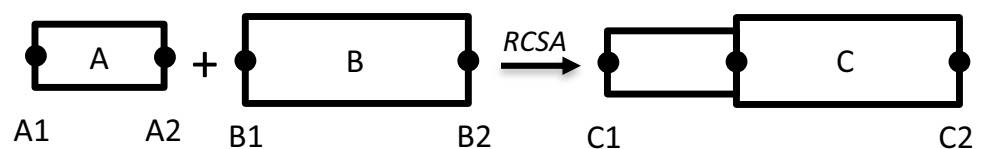
$$R_2 = R_{A,22} - R_{B,22} + K^{-1} \quad (12)$$

$$K = \begin{bmatrix} k_{yf} + i\omega c_{yf} & k_{\theta f} + i\omega c_{\theta f} \\ k_{ym} + i\omega c_{ym} & k_{\theta m} + i\omega c_{\theta m} \end{bmatrix} \quad (13)$$

In this work, the shaft was divided into 27 segments (refer to full geometric representation in Fig. 7). Figure 4 shows a schematic representation of the final assembly, namely sp , with the spindle tip on the left side and the tail on the right. To create the FRF plots presented in subsequent sections, the absolute value of the linear response at the spindle tip due to force excitation at the spindle tip, h_{sp11} , is extracted from the matrix $R_{C,11}$ in Eq. 10.

The four stiffness values for populating the K matrices of each bearing are extracted from the standard 5×5 bearing stiffness matrix in Eq. 8. The damping values were obtained from experimental results from a high-speed spindle with similar design features, dimensions, and bearings. These values were further verified to be in the same range as those shared in

Fig. 2 Rigid substructure coupling



the literature [21–23]. Table 1 enlists the stiffness and damping values used in this work. The remaining terms were taken as zero due to their insignificant contribution.

3.3 Thermal modeling

A detailed high-speed spindle FE model consists of heat sources: cutting process related, bearings, and a built-in motor. The heat sinks in the internal environment of a high-speed spindle comprise the spindle shaft, shaft cooling water, ambient air, motor cooling air, and air cooling on press-fitted sleeves. Usually, viscous air shear should be included as a heat source since the air gap between rotating and static parts is tight, and the forced air convection by motor cooling air and oil-air lubricant should be added as heat sinks into the FE model for high-speed spindles [12]. Additionally, conduction analysis should be done based on clearance and press fit conditions for outer bearing

rings and housing components on a spindle [13]. However, in this study, only rotating components are simulated in the FE model by focusing on the shaft cooling system in a high-speed spindle. Therefore, the housing and outer bearing ring interaction, oil-air lubrication system effects, and viscous shear losses due to heat generation are neglected in the FE model. The considered heat sources are bearings and built-in motors in the FE model. Since the high-speed spindle considered in this study includes a shaft cooling system, one of the heat sinks is shaft cooling water flow. The shaft cooling water flow is represented via simplified beam elements in the FE model, which includes shaft rotating effects and shaft coolant flow rate with inlet temperature. Additionally, the spindle FE model includes motor cooling air and stepped sleeves cooling air for forced convection and ambient air for natural convection as the other heat sinks. This representative FE simulation focuses on demonstrating the structural effects of the auxiliary components of

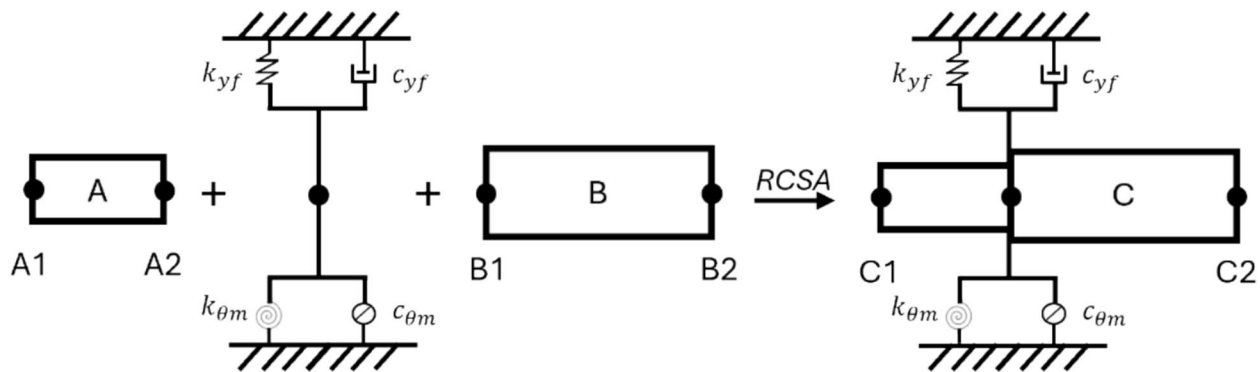


Fig. 3 Flexible substructure coupling

Fig. 4 Schematic representation of final spindle assembly

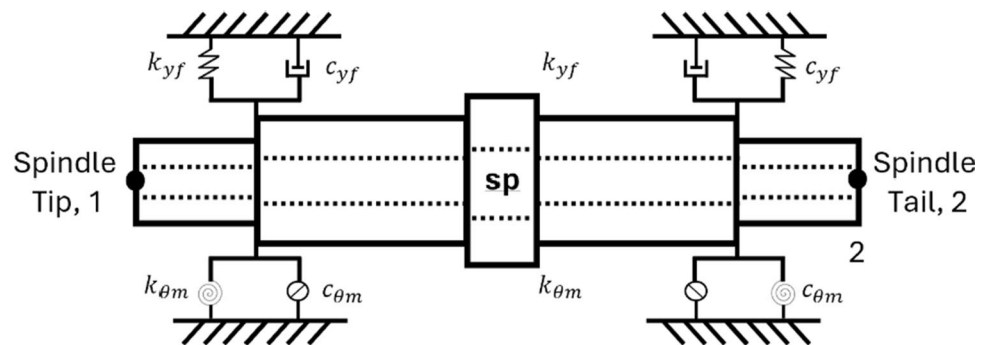


Table 1 Bearing stiffness and damping values

	K_{yf} , translational stiffness (N/m)	$K_{\theta m}$, rotational stiffness (N.m/rad)	$K_{ym}, K_{\theta f}$, cross stiffness (N/rad)	C_{yf} , translational damping (N.s/rad)	$C_{\theta m}$, rotational damping (N.m.s/rad)
Front bearings	6.27e7	8.54e3	– 6.936e5	650	1.8
Rear bearings	4.73e7	3.70e3	3.91e5	400	1

a high-speed shaft that generally undergo significant design changes due to the implementation of shaft cooling and draw-bar systems. The simulation was designed to specifically understand the impact of these changes on the spindle shaft's overall structural integrity. The simulation results will help to optimize the design of high-speed spindle shafts for improved

performance and longevity. Bearing frictional moments of inner (M_i) and outer (M_e) rings are calculated for thermal modeling as indicated in Eq. 8 and in [10]. f_0 and f_1 change with respect to bearing parameters and lubrication conditions while D is the bearing diameter, γ is the kinematic viscosity, and ω_c is the rolling velocity.

$$M_i = \left[0.675f_0(\gamma\omega_c)^{2/3} D^3 + f_1 \left(Q_i / Q_{i,\max} \right)^{1/3} Q_i D \right], M_e = \left[0.675f_0(\gamma\omega_c)^{2/3} D^3 + f_1 \left(Q_e / Q_{e,\max} \right)^{1/3} Q_e D \right] \quad (14)$$

Additionally, spin moments of the bearings at inner and outer rings (M_{si}) and (M_{se}) are calculated as shown in Eq. 7. μ_s is the friction coefficient, a_i and a_e are the major axes of the contact at bearing rings. ζ is based on the contact geometry.

$$M_{si} = 3\mu_s Q_i a_i \zeta / 8, M_{se} = 3\mu_s Q_e a_e \zeta / 8 \quad (15)$$

The heat generation from bearings according to Eqs. 14 and 15 is given by Eq. 16. The calculated heat values are applied as boundary conditions in the FE model. ω_{ri} and ω_{re} are spin velocities.

$$H_i = \omega_c \cdot M_i + \omega_{ri} \cdot M_{si}, H_e = \omega_c \cdot M_e + \omega_{re} \cdot M_{se} \quad (16)$$

Another major heat source for spindle thermal growth is the heat produced by the built-in motor. Since there is no well-established calculation method for motor heat prediction in the literature, an empirical heat generation formula provided by the motor manufacturer of our spindle is used.

For the FE analysis, two types of heat sinks are considered. The first is the heat dissipation due to forced convection at shaft cooling regions. Convection coefficient (h_{liquid}) inside the shaft cooling hole is calculated by Eqs. 17 and 18. In Eq. 17, Nu is Nusselt number [10], k_{liquid} is fluid conductivity, and D is flow cross-section of a tube. The Reynolds number is calculated by considering u (average speed of fluid) and η (fluid kinematic viscosity).

$$h_{liquid} = Nu \cdot k_{liquid} / D \quad (17)$$

$$Re = \frac{u \cdot D}{\eta_{liquid}}, \text{ where } u = \sqrt{u_{axial}^2 + u_{radial}^2} \quad (18)$$

During spindle rotation, another heat dissipative mechanism: forced convection, occurs between the motors' rotor and stator. A similar effect of forced convection can also be witnessed in press-fitted sleeves when the spindle is in use. The forced convection coefficient outside of the shaft (h_{air}) is calculated by Eqs. 19–21. l_{shaft} is the perimeter

of the shaft assembly with auxiliary components. The Reynolds number is predicted by employing u_{air} (air speed around shaft) which depends on the spindle speed n and the kinematic viscosity ϑ_{air} [24]. These predicted forced convection coefficients are applied as boundary conditions to the FE model as well.

$$h_{air} = 0.664 \frac{\lambda}{l} Re^{1/2} Pr^{1/3}, \text{ where } l_{shaft} = \pi d_{shaft} \quad (19)$$

$$Re = \frac{u_{air} l_{shaft}}{\vartheta_{air}}, \text{ where } u_{air} = \frac{l_{shaft} \cdot n}{60} \quad (20)$$

$$h_{air} = 0.664 \lambda \left(\frac{n}{60 \vartheta_{air}} \right)^{1/2} Pr^{1/3} \quad (21)$$

These predicted forced convection coefficients are also applied to the FE model as boundary conditions. The FE model is parameterized, and a reduced-order model is created to accelerate FE simulations to match the speed of RCSA simulations. The shaft hole and auxiliary components' outer perimeters are parameterized as geometrical inputs, while the FE model's forced convection coefficients are parameterized as operational inputs.

The parameterized outputs from FE analysis are bearing and motor temperatures, thermal growth, and the normal and shear stress safety factors. The shaft's temperature distribution from FE analyses for a 10.5 mm shaft hole radius is illustrated in Figs. 5 and 6 for 40,000 rpm. The thermal growth results revealed that an enlarged shaft hole increases the heat dissipation capacity most by reducing temperature gradients and thermal distortions. According to the results, a larger shaft hole for cooling provides better thermal stability for a spindle. Thus, the central shaft hole dimension is selected for sensitivity analysis. However, a larger shaft hole also weakens the structure dynamically. In this study, this tradeoff between hole radius and shaft strength is assessed by the shear and normal stress safety factors.

Fig. 5 Parameterized FE analysis results for temperature distribution

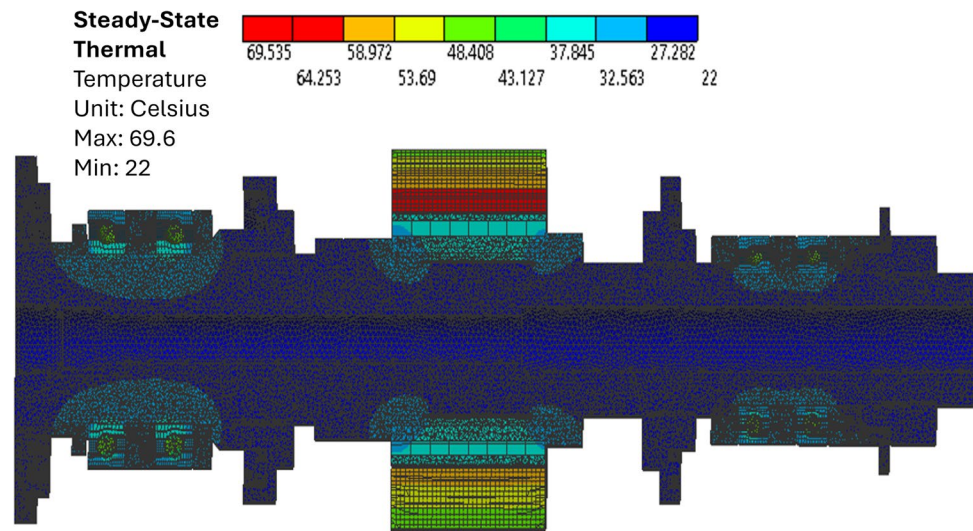
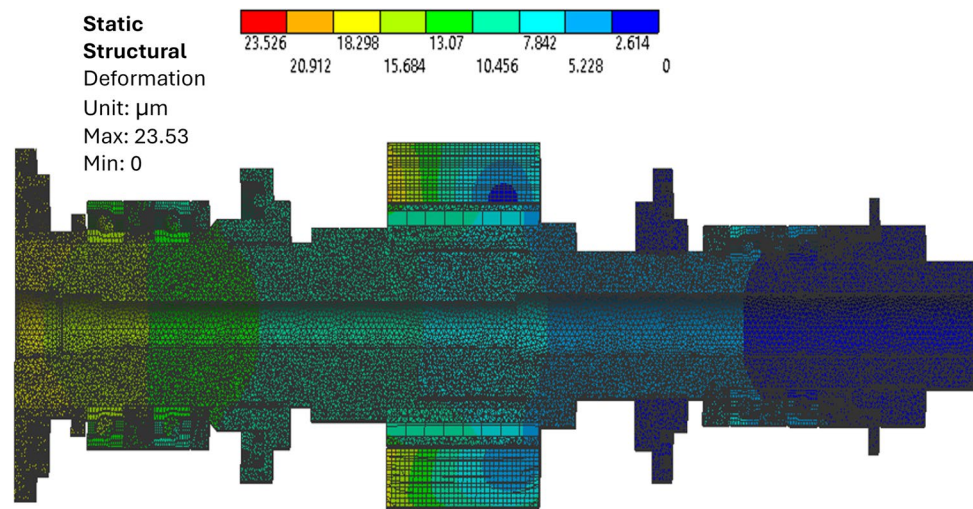


Fig. 6 Parameterized FE analysis results for structural deformations



4 Sensitivity analysis

An in-house spindle design (see Fig. 7) was used for this study. Although the nominal dimensions were chosen based on operational requirements and general design sense, 11 auxiliary and shaft dimensions were identified to be adjustable within certain bounds based on practical assembly constraints, as shown in Table 2.

4.1 Dynamic sensitivity

The most significant effect of altering the identified dimensions is on the dynamic performance as it changes the structure's mass and stiffness. A sensitivity analysis was conducted to determine the individual effects of each variable dimension on dynamic performance indicators, which

were the natural frequency and FRF amplitude of the first dominant mode, identified through RCSA.

These performance indicators were checked at numerous points within these bounds and the largest change in each indicator was recorded for each variable. For each analysis, only one dimension was changed while the rest of the geometry was kept constant. Figure 8 shows the sensitivity of each variable with respect to both the performance indicators. A 5% change in dynamic behaviors was set as the minimum threshold for a dimension to be classified as “sensitive.” The plot shows that variables 2, 4, 8, 9, 10, and 11 meet these criteria. Furthermore, the findings show that the natural frequency is much less sensitive to change compared to the FRF amplitude, which showed more than $\pm 10\%$ change in certain cases. The natural frequency, on the other hand, remained unchanged in many cases.

The most significant effect of altering the identified dimensions is on the dynamic performance as it changes

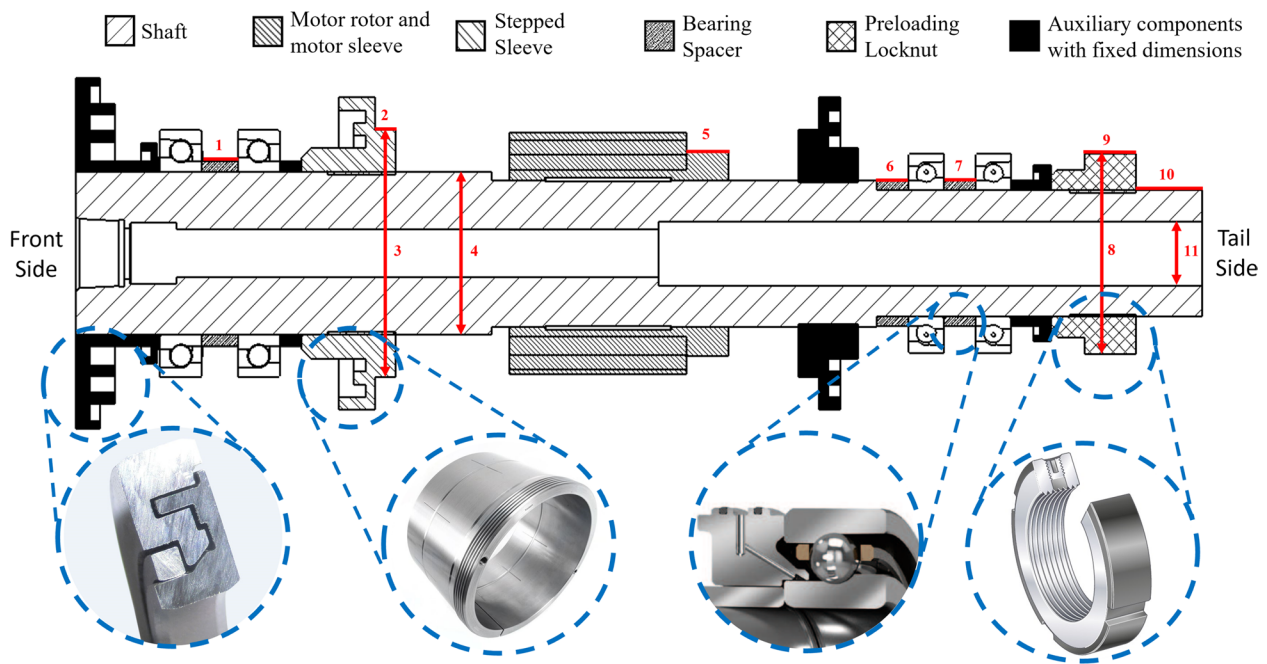


Fig. 7 Spindle shaft auxiliary components and adjustable dimensions

Table 2 Upper and lower bounds of variable dimensions

Variable no	Variable name	Lower bound (mm)	Upper bound (mm)
1	Front bearing spacer length	5	15
2	Front stepped sleeve length	5.75	25
3	Front stepped sleeve diameter	55	80
4	Motor shoulder diameter	45	65
5	Motor sleeve length	5	20
6	Frist rear spacer length	6	12
7	Second rear spacer length	6	12
8	Rear locknut diameter	40	70
9	Rear locknut length	15	35
10	Tail length	0	150
11	Shaft central hole radius	4	17

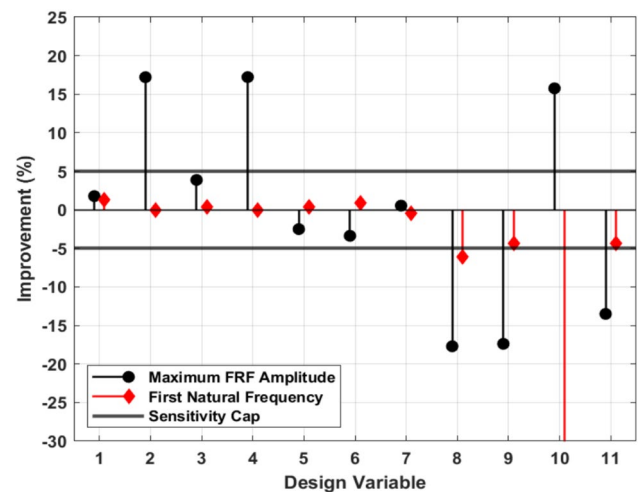


Fig. 8 Summary of sensitivity analysis

the structure's mass and stiffness. To demonstrate the aggregate effect of auxiliary rotating components, Fig. 9 shows the FRF of the same shaft assembly with and without all its auxiliary components. The structure's frequency response is clearly altered when the auxiliary components are included in the simulation, demonstrating the need for such an analysis.

The next sections discuss some of the identified sensitive parameters in detail. It is reported here that the remaining sensitive parameters showed similar effects on spindle-tip dynamics, thus each parameter is not discussed to avoid repetition.

4.1.1 Front stepped sleeve and motor shoulder

The length of the front stepped sleeve and the diameter of the motor shoulder (variables 2 and 4 in Fig. 7) show high sensitivity for the dominant FRF amplitude. It is to be noted that both these variables add material to the same section of the shaft. Thus, they have a similar effect on dynamic performance. Figure 10 shows that increasing the sleeve's length from 5 to 25 mm decreased the amplitude of the dominant peak by 17%. The reason for this is the occurrence of dynamic absorber phenomena [6, 25]. The change

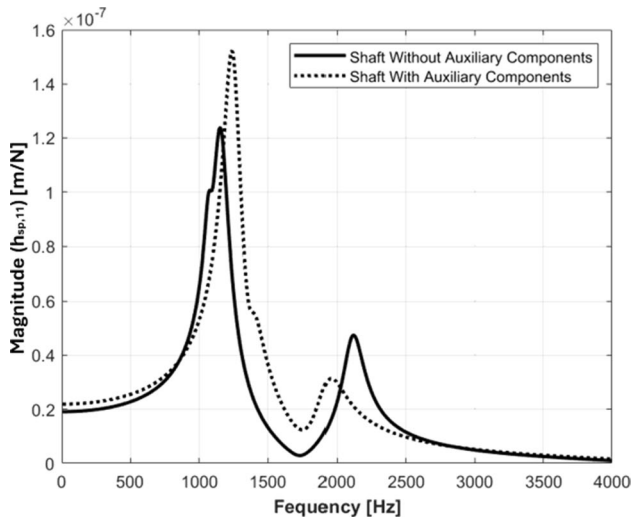


Fig. 9 Spindle-tip FRF with and without auxiliary components

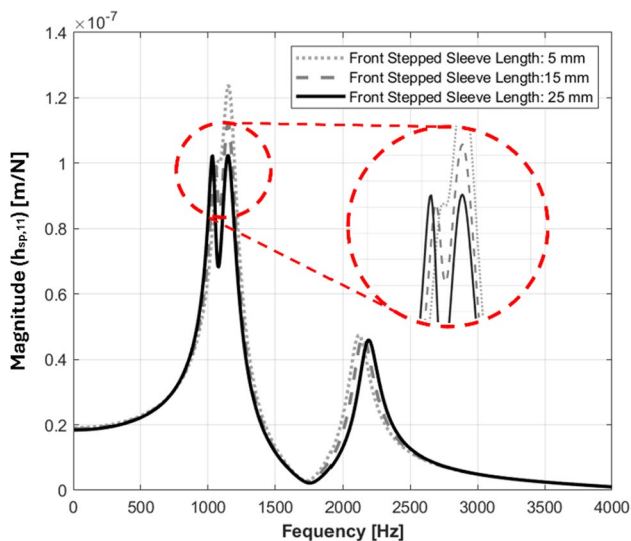


Fig. 10 Front stepped sleeve length sensitivity

in geometry caused the dominant mode to split into two smaller modes with roughly the same amplitudes. This is a beneficial outcome as lower FRF amplitudes decrease vibration response and increase chatter stability.

4.1.2 Spindle tail components (shaft tail and preloading locknut)

For the spindle tail, the diameter and length of the rear-side preloading locknut, as well as the tail length, were considered in the sensitivity analysis (variables 8, 9, and 10). The results were similar to those reported in the previous section. Dimensional adjustments to any of these can be used to increase spindle rigidity through the dynamic

absorber effect. The results of the tail length (variable 10) variation were particularly interesting, where decreasing the tail length from 20 to 0 mm led to a near-perfect occurrence of mode splitting (Fig. 11). This contrasts with previously reported results [25], where the tail was elongated to achieve the mode-splitting effect, which added significant mass to the assembly. By decreasing the tail length to 0 mm, a 2.5% weight reduction was also achieved. This will have a positive outcome on general spindle performance, like the bearings carrying less weight and thus generating less heat, and the motor being capable of providing higher acceleration due to the lower rotational inertia. It is important to clarify here that a tail length of 0 mm does not mean the entire tail section of the shaft is removed. The “spindle tail” comprises all masses that come after the rear bearing set. In Fig. 5, the tail includes a spacer, a preloading locknut, and a tail extension (variable 10). Therefore, even when the tail length is decreased to 0 mm, there is still significant mass in the tail section in the form of functional components that are necessary to achieve a robust and rigid assembly.

4.1.3 Shaft central hole

The hole at the center of the spindle shaft (variable 11) contains the drawbar mechanism that holds and actuates the tool holder. Moreover, it also serves as a channel for through-cooling, which delivers coolant to the cutting tool. Due to these systems, the diameter of the shaft hole is not completely free to change but can still be adjusted to a certain extent by selecting an appropriately sized tool holder. For instance, if a larger shaft hole is required, designers may select the larger HSK-E 32 tool holder instead of the currently used HSK-E 25. Figure 12 shows that changing the diameter of the central hole also exhibited improvements

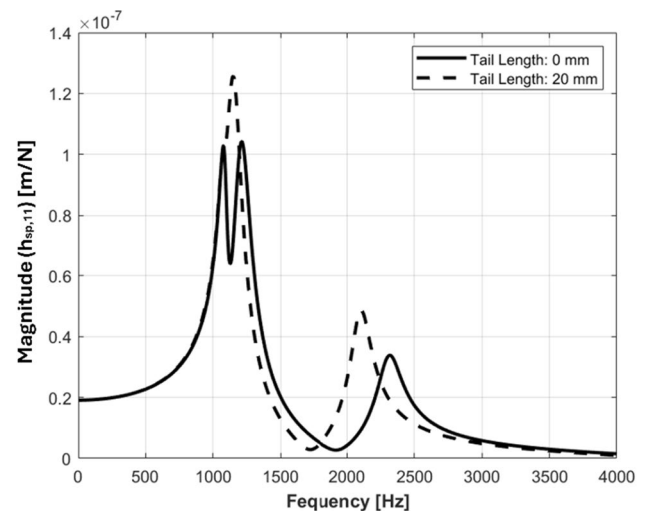


Fig. 11 Tail length sensitivity

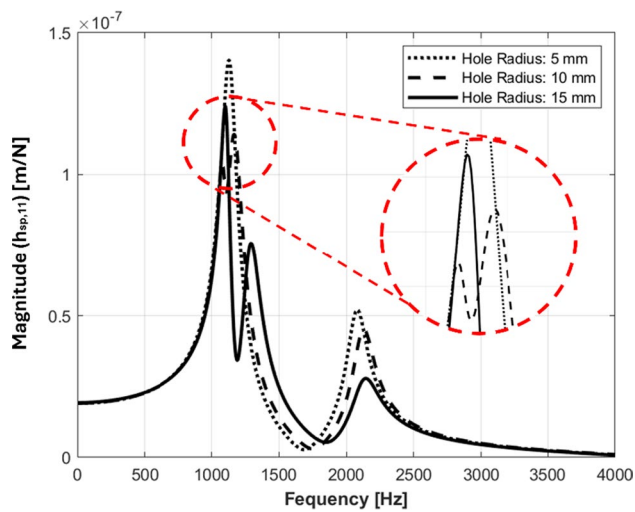


Fig. 12 Hole radius sensitivity

in dynamic performance indicators through mode splitting/modal interactions.

4.1.4 Sensitivity to bearing damping

The bearing stiffness and damping are known to have a significant impact on spindle dynamics in addition to auxiliary component dimensions. While the impact of bearing stiffness has been widely reported in the literature [5, 9, 26], the effect of bearing damping is seldom discussed. To check the cross-sensitivity between structural changes and bearing damping, an additional sensitivity analysis was performed on the tuned split-mode FRF case shown in Fig. 10 (front stepped sleeve length: 5 mm). Three cases are presented in Fig. 13, where the damping values in Table 1 were varied by factors of 0.75, 1, and 1.25 for the front bearing set only, rear bearing set only, and both bearing sets. It was observed that the damping only affected the

magnitude of the FRF while the change in modal frequencies was negligible. Moreover, as expected, increasing the damping caused the vibration amplitudes to decrease, and vice versa. These results are in line with the observations reported by Erturk et al. [26]. The differences between the three cases, however, provide valuable information. It can be deduced from Figs. 13a, b that the damping of the front bearing set has a more uniform impact on the modal amplitudes, where both FRF peaks move up and down by similar ratios for up to 25% change in damping values. Figure 13c indicates that the damping of the rear bearing set, however, only interacts with the high-frequency mode.

While it is understood that controlling the bearing damping in machine tool spindles is a difficult task, these findings can serve as a generalized guideline for improving spindle dynamic performance in combination with structural changes.

4.2 Thermal sensitivity

According to the FE results, the most dominant variable for thermal performance is the shaft's central hole (variable 11) since it is a part of the cooling system. Figure 14 shows the variation of these parameters throughout the entire range of the central hole radius. The spindle-tip thermal growth and front-bearing temperature decrease with the hole radius. This result is expected since a larger hole accommodates more coolant and has a higher heat transfer rate. However, choosing the maximum shaft hole radius is impractical due to the accompanying loss of strength, as confirmed by the trends of shear stress and normal stress safety factors. Instead, a trade-off exists between the thermal performance factors and stress safety factors, indicating that an appropriate balance between the two is to be determined as a design decision.

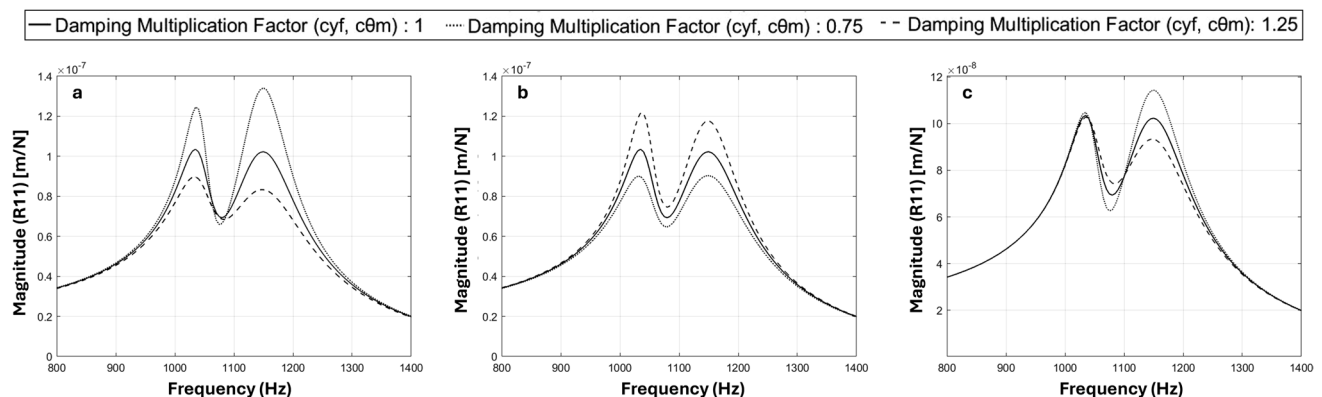


Fig. 13 Sensitivity for bearing damping: **a** front and rear bearing sets, **b** only front bearing set, **c** only rear bearing set

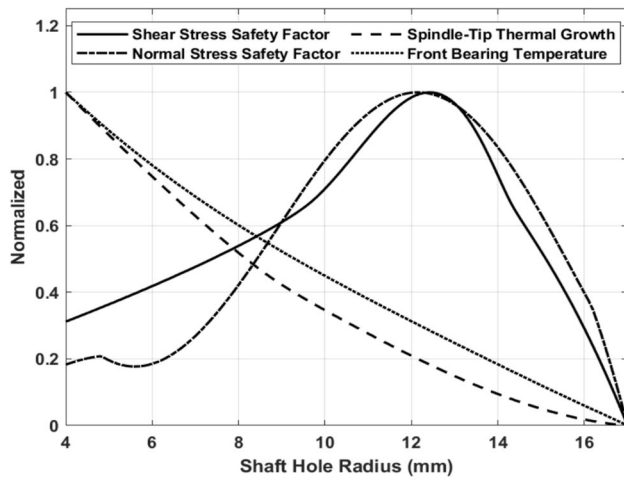


Fig. 14 Thermal sensitivity of shaft central hole

5 Conclusion

This study presented a spindle dynamic and thermal sensitivity analysis comprising various possible rotating auxiliary components on a high-speed spindle shaft. Primarily, the analysis focused on auxiliary components whose dimensions are alterable after the optimization of bearing configuration, preloads, and locations is done. Some of the results can serve as design takeaways for the design of spindle shaft and auxiliary components. The summary of the key results is as follows:

- In the literature, spindle tail length is shown to create a mode-splitting/absorber effect. However, sensitivity analysis revealed that front stepped sleeves and central shaft holes are also strong candidates to create a similar dynamic absorber effect, with the additional benefit of higher thermal stability.
- In high-speed spindles, a short yet thick shaft design is generally advised. Thus, the tail length should be kept short, resulting in lighter spindle shafts that cause less heat generation, avoid dynamic problems, and lead to a smooth operation experience at high speeds. The sensitivity results revealed that adjusting the tail length to zero is possible with better dynamic and thermal behavior via adjusting the auxiliary component dimensions.
- The auxiliary components alter the spindle shaft and can be used to select and optimize the drawbar dimensions and tool holder selection by adopting a similar approach in this study.

This findings can allow spindle designers to develop shorter, lightweight shaft designs while leveraging the benefits of effects like dynamic absorbers. However, the findings of this study are specific only to the spindle design

and bearings used in this study. Furthermore, the presented methodology can serve as a preliminary study to identify suitable design variables for a systematic structural optimization framework.

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Declarations

Conflict of interest The authors declare no competing interests.

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