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# Dynamic Models and Design of Spindle-Bearing Systems of Machine Tools: A Review

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*Machine tools are vital to modern industries. The performance of a machine tool is assessed according to the dimensional accuracy and surface finish of the machined work-pieces, which are closely related to the dynamic characteristics of the spindle-bearing system. The main purpose of this study is to review relevant studies on dynamic models and the design of spindle-bearing systems of machine tools. These studies related to dynamic models are categorized into two types according to the physical components: spindle shafts and bearings. The dynamic models in each category are comprehensively discussed and coordinated. Guidelines condensed from the sampled papers are provided, followed by an exploration of design-related papers. Finally, a conclusion on the analyzed works is drawn and future lines of research are identified.*

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## 1. Introduction

Machine tools, such as lathes and milling machines, are vital to modern industries. Currently, most products and the machines used to produce such products consist of components manufactured using a machine tool. The performance of a machine tool is usually assessed according to the dimensional accuracy and surface finish of its machined work-pieces. Deformations and vibrations, which are mostly induced in the machining system by cutting forces, mass unbalance, and chatter, may cause poor machining quality. Therefore, to avoid unacceptable machining outcomes, machine tools must be operated conservatively while accounting for their limited applications and benefits. Consequently, to realize the full potential of machine tools, recognizing the quality-deteriorated mechanism is necessary, depending primarily on the dynamic characteristics of the machine tools involved. A spindle-bearing system, which involves rotating operations of cutting tools or work-pieces for completing cutting jobs and transmitting the required power to cutting metal, is generally the most flexible part of a machine-tool structure; therefore, engineers must thoroughly understand the dynamic characteristics of a spindle-bearing system and design it accordingly.

Currently, to enhance productivity and machining quality, the spindle-bearing systems equipped in most machine tools are composed of motorized spindle shafts and angular contact ball bearings. A motorized spindle is assembled with a built-in motor as an integrated part of the spindle shaft, thus eliminating the need for conventional power transmission devices and reducing vibration.[42] Angular contact ball bearings are used most frequently in machine tools because of their favorable stiffness properties, low friction losses, long lives, and high cost-effectiveness ratio.[1] A typical type of spindle-bearing system used in a milling machine is depicted in Fig. 1, in which two sets of angular contact ball bearings are installed in a motorized spindle shaft with a cutting tool assembled at the left end.

The dynamic properties of spindle shafts and bearings, such as mass, stiffness, and damping, can be predicted individually by using theoretical models or by measuring the properties in experiments. Subsequently, the dynamic properties of the spindle-bearing system can be obtained by aggregating all of the properties of its components (i.e., the spindle shaft and all of the bearings). The resulting dynamics can be applied in assessing the response of the system with respect to


Fig. 1 A typical spindle-bearing system of a milling machine

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particular external loads. For example, in analyzing the chatter phenomenon in machining processes, the dynamics of spindle-bearing systems can be used to predict system stability with respect to the cutting forces.[5,6] According to the required dynamic properties, the spindle shafts, bearings, and entire spindle-bearing systems are designed to optimize the focused responses in a presumed operating environment.

Numerous academic studies have theoretically and experimentally investigated the dynamic characteristics of machine tool spindle shafts, bearings, and spindle-bearing systems. Most results reported in these papers can be applied in the design of spindle-bearing systems. The purpose of this paper is to present a structured review that provides a guide to recent research on the subject of dynamic models of spindle-bearing systems, which can be adopted in machine tool designs, and to identify research implications for future investigation in this field. To analyze the related literature, a search of bibliographic references was conducted through database libraries that contain the most outstanding journals in this particular area. The following section introduces the mechanism of the spindle-bearing system dynamic model and describes a classification framework for the remaining review. Section 3 comprehensively summarizes selected research on the spindle shaft dynamic model and bearing dynamic model. Section 4 lists the results extracted from literature review and introduces papers related to optimal design. Finally, in Section 5, we draw a conclusion and suggest future research directions.

## 2. Classification scheme of literature review

The dynamics of spindle-bearing systems are highly complex because several nonlinear and closed-loop phenomena interact with each other thermally and mechanically. A concise mechanism of the dynamic model of spindle-bearing systems is shown in 2, including the essential physical models required to describe the dynamic properties of spindle-bearing systems. In addition, Fig. 2 shows the major factors, which include the design variables that are primarily determined at the design stages and the operation parameters that are usually established on shop floors.[42] The relationships among these components are as follows:

1. The spindle-bearing system dynamic model consists of a spindle shaft dynamic model and a bearing dynamic model.
2. The spindle-bearing system dynamic model is directly affected by the number and location of the bearings.
3. The spindle shaft dynamic model is assessed according to spindle shaft specifications such as materials and dimensions.

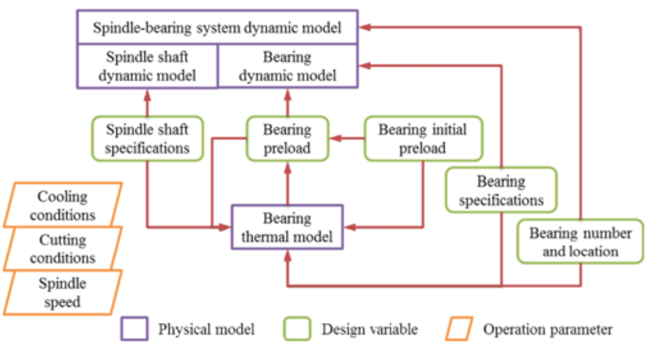


Fig. 2 Mechanism of a general spindle-bearing system dynamic model

4. The bearing dynamic model is assessed according to the bearing specifications, such as the number of balls, diameter of the balls, contact angle of the bearings, and the bearing preload. The bearing preload consists of the initial preload, a design variable, and thermally induced preload.
5. The thermally induced preload of bearings is controlled by a bearing thermal model, which can be represented as a function of the specifications of the spindle shaft and bearings, bearing initial preload, and number and location of bearings.
6. All dynamic models are generally influenced by operation parameters such as cooling conditions, cutting conditions, and spindle speed.

Based on the proposed modeling mechanism, an exploration of bibliographic references was performed. The literature sample comprises the English language, peer-reviewed journal papers on the area of interest, largely covering the period from 1992 to 2012. In particular, to avoid loss to divergence, the papers on bearing dynamic models focus on angular contact ball bearings because they are the bearings most frequently used in modern machine tools.[14]

Because a spindle-bearing system can apparently be divided into two physical parts (i.e., a spindle shaft and bearings), the screened research articles are classified into two portions of dynamic models in the following review according to the physical elements, as indicated in Table 1. Under such a classification, a paper may appear in both categories if its contents cover both fields. In Section 3, the sampled papers are sequentially reviewed as spindle shaft dynamic models and bearing dynamic models, with respect to the recommended classification.

## 3. Review of spindle-bearing system dynamic models

In this review, we focus on the dynamic models, regardless of their applications. To maintain conciseness, the contents are organized comprehensively. Instead of commenting on each paper individually, the dynamic models of spindle shafts and bearings are described generally and represented conceptually as functions of design variables and operation parameters by aggregating the results of related papers. However, for exceptional cases, detailed descriptions are provided. These models are reviewed in the following subsections with respect to the mentioned classification, as indicated in Table 1.

### 3.1 Dynamic model of spindle shafts

In general, spindle shafts are the major resources of mass for spindle-bearing systems. Therefore, the mass property of spindle shaft cannot be neglected in the dynamic model of spindle-bearing systems. In the papers indicated as “Rigid” under the category “Spindle shaft”

Table 1 Classification of the reviewed papers

| Category | Spindle shaft | Bearing |
|---|---|---|
| Ref. no. | Rigid: 3, 4, 14, 16, 21, 34<br>FE model: 6, 9-12, 18-20, 40-44, 47, 51, 57, 63, 68<br>Other models: 5, 13, 17, 28, 29, 31 | Stiffness model: 3-5, 9-14, 16, 22, 23, 25, 27, 31, 33, 34, 36, 40-42, 47, 57, 59<br>Thermal model: 7, 8, 20, 26, 28, 35, 37, 39-41, 48, 52, 62, 64-66 |

in Table 1, the spindle shafts were modeled as rigid beams, whereas the flexibility and damping effects were all assumed to have resulted from bearings. This type of assumption was especially common in works focusing on bearings. In these spindle shaft dynamic models, the spindle shaft merely contributes mass to the spindle-bearing system dynamic model.

However, when more reliable estimation is required or the spindle shaft cannot be treated as a rigid body, excepting mass, the stiffness of the spindle shafts must also be considered in analyzing system dynamics. Several methods can be used to model the dynamic response of a flexible shaft. The choice of modeling method must be based on the features of the shaft. Currently, the most popular approach for modeling the mechanical behavior of a spindle shaft is the finite element (FE) method because of its capability to manage complex geometry and boundary conditions, and the calculation approaches for solving the finite element system equations are no more critical when using modern powerful computer processors. Therefore, as shown in Table 1, the number of papers that report using the “FE model” is far beyond that of the sampled papers that report using “Other models.” Most of the “FE model” papers, such as that written by Nelson,[54] follow the FE model that was developed in the rotor-dynamics field.

Before building the FE model of a spindle shaft, the modeling method for each element, which is usually modeled as a beam, must be determined first. Three beam models are most commonly used: Euler-Bernoulli, Rayleigh, and Timoshenko. The differences among these theories are the assumptions they make. Euler-Bernoulli beam theory, the simplest of the three, assumes that no rotatory inertia or shear deformation is on the spindle shaft. Rayleigh beam theory accounts for rotatory inertia but not shear deformation. In Timoshenko beam theory, both effects are considered in modeling; therefore, it is the most complex and precise of the theories. Most of the papers reported using the Timoshenko beam model,[6,11,17-19,31,40-42,51] and two reported using the Rayleigh.[13,68]

To construct the spindle shaft FE model, the spindle shaft is first discretized into a finite number of beam elements, as shown in Fig. 3, and each node of elements is assigned certain degrees of freedom as required. Associated with several specific shape functions, the kinetic and potential energy of each element can be obtained by integrating those of the cross-sections along the axis of the spindle shaft, and expressed as functions of the physical and geometrical properties of the element. By summing the kinetic and potential energy of all elements, the total of the spindle shaft can be obtained; and by using the Lagrange equation, the equation of motion (EOM) of the spindle shaft can finally be deduced. For example, a simple free vibration EOM for a motorized spindle shaft can be represented as[43]

$$\boldsymbol{M}_s\ddot{\boldsymbol{q}}+(\boldsymbol{D}_s-\Omega\boldsymbol{G})\dot{\boldsymbol{q}}+(\boldsymbol{K}_s-\Omega^2\boldsymbol{N})\boldsymbol{q}=\boldsymbol{0} \tag{1}$$

where $\boldsymbol{q}$ is the global node displacement vector of the system; $\Omega$ is the spindle speed; $\boldsymbol{M}_s$, $\boldsymbol{K}_s$, and $\boldsymbol{D}_s$ are the mass, stiffness, and damping matrices of the spindle shaft, respectively; $\boldsymbol{G}$ is the gyroscopic matrix; and $\Omega^2\boldsymbol{Nq}$ represents the softening effects of centrifugal forces. Usually, the damping of the spindle shaft is neglected or estimated empirically.[6] The gyroscopic effects arise from 3D rotation and change with the rotating speed and mass of the objects, whereas the centrifugal forces are caused by rotation. The high-speed effect terms, including $\Omega\boldsymbol{G}\dot{\boldsymbol{q}}$ and $\Omega^2\boldsymbol{Nq}$ in Eq. (1), often appear in the EOM of the papers related to high-speed machining.[6,9-12,18-20,29,40-42,47,51,57] The matrices in Eq. (1) are all determined based on the specifications of the spindle shaft, and the details of these matrices can be found in Refs.[11,20,41] Aside from the FE method, other methods can be used to derive the dynamic behavior of spindle shafts; for example, the analytical model for continuous shafts,[5,13,17] the influence coefficient method of discrete lumped masses,[31] and the transfer matrix method.[28,29] However, these methods are not as robust as the FE method.

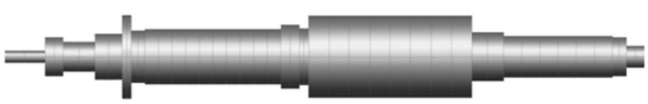
Fig. 3 A motorized spindle shaft with finite elements

Except for the dynamic model of the spindle shaft, some previous studies have concentrated on the joints of the spindle shaft. The joints include the interfaces between the shaft and tool holder and between the tool holder and cutting tool. Li and Shin[40] added the stiffness and damping terms resulting from the interfaces to their FEM model. Smith et al.[60] examined the effect of drawbar force on the dynamic stiffness of milling machines. They concluded that an increase in drawbar force raised the static stiffness while reducing the damping of tool-holder-spindle interfaces. Regarding the drawbar mechanism, Chen and Hwang[14] investigated the effects of centrifugal forces, also showing that the stiffness of the face/tapper contact interface of the ball track-type mechanism outperforms the wedge-type mechanism. To include the stiffness of interfaces in predicting the tool point dynamic response, Schmitz and Donalson[58] developed a receptance coupling (RC) method to analyze the substructures. The RC method was also adopted in Refs. [17, 55]. Movehhedy and Gerami[50] improved the RC method by using two linear joint elements and employed the genetic algorithm (GA) to determine the joint parameters. Namazi et al.[53] proposed a model of contact stiffness and damping at the tool holder and spindle interface, where the contact is modeled using uniformly distributed translational and rotational springs. Based on these previous studies, stiffness and damping arise further if the effects of interfaces are considered, thus making the model more practical for real-world scenarios. The damping of interfaces is usually more significant than the internal damping of a spindle shaft. Therefore, for research that must account for damping, such as that analyzing chatter, the damping property of the joints is recommended to be included in the dynamic model.

### 3.2. Dynamic model of bearings

In this subsection, the literature related to the dynamic properties mass ($\boldsymbol{M}_b$), damping ($\boldsymbol{D}_b$), and stiffness ($\boldsymbol{K}_b$) of bearings is reviewed. The mass of bearings contributing to the spindle-bearing system dynamic model is usually neglected or combined with the mass of the spindle shaft. Similar to the spindle shaft, the damping of a bearing is generally obtained by conducting experiments.[21,23,51,68] Based on the sampled papers listed in Table 1, we found that most research on angular contact ball bearings and the bearing dynamic model focused on the stiffness. Moreover, most stiffness models of angular contact ball bearings were built based on those developed by Jones[30] and/or Harris,[24] who modeled a bearing as a massless nonlinear contact spring.

A typical angular contact ball bearing is shown in Fig. 4. To calculate the stiffness of the bearings, the deformation behaviors of the contacts between the balls and inner/outer rings must be established first. Using Hertzian contact theory, the contact force ($W$) and deflection ($\delta$) relationship of two elastic bodies are represented as[11]

$$W = k_h \delta^{2/3} \tag{2}$$

where the stiffness coefficient $k_h$ is a function of the materials of the balls and rings, the diameter of the balls, pitch diameter of the bearing, contact angle, curvatures of rings, and loads.[3,11]

If the bearing is stationary, then the contact angles ($\theta$) between the balls and the inner/outer rings are identical, even when a certain external load, such as the axial preload, exists. With an external load applied, the centers of the balls and curvatures are moved because of the deflection caused by the load, but remain collinear, as shown in Fig. 5(a). This implies that the contact angles change according to the amplitudes of the external loads. Under the assumption that the bearing is static, the load equilibrium equations can be developed.[3-5,11,16,22,23,25,34] However, because the contact angles are functions of forces, numerical iteration techniques may be required to solve the equations. After the equations are solved, by taking derivatives of forces with respect to the displacements, the stiffness of each direction can be obtained. When the bearing rotates, as shown in Fig. 5(b), the inner and outer contact angles become dissimilar because of the centrifugal force and gyroscopic moment caused by bearing rotation.[6,9,10,12,13,27,31,39,40,59] To calculate bearing stiffness, an approach similar to that used in the stationary situation can be applied, but the contact angles of the inner and outer rings must be considered separately. In summary, the general nonlinear stiffness equation of angular contact ball bearings can be expressed as[12,41,59]

$$\boldsymbol{K}_b = f(P, Z, D, \theta_C, \Omega) \tag{3}$$

where $P$ is the bearing preload, which consists of the initial preload ($P_i$) and thermally induced preload ($P_t$), as indicated in Fig. 1; $Z$ is the number of balls; $D$ is the ball diameter; and $\theta_c$ is the initial contact angles. To enhance the stiffness of the angular contact ball bearings integrated in the FE model, the bearing stiffness can also be represented in matrix forms.[11,22,25]

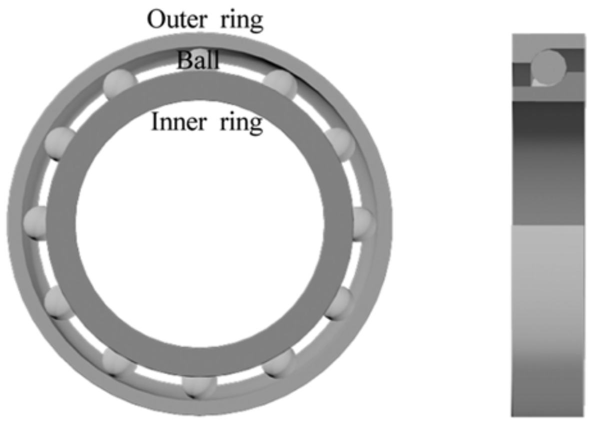


Fig. 4 An angular contact ball bearing

As shown in Table 1, the papers under the category "Stiffness models" of bearings are further classified according to the effects of factors on which they are focused. The results shown in Table 2 indicate that most of the papers are related to the bearing initial preload and spindle speed, whereas fewer address the specifications of bearings, such as the ball number and contact angle. This may be because the bearings have become commercially standardized parts.

The bearing preload mechanisms, which greatly affect the properties of the spindle-bearing system, can generally be classified as rigid and non-rigid.[1,39] Usually, the rigid preload is applied by using lock nuts, whereas the non-rigid preload is applied by using hydraulic or spring kits. When certain uneven thermal expansions of the spindle shaft and bearing occur, the induced preload may not be significant for the non-rigid bearings with springs. However, if the bearings are rigidly preloaded, a thermally induced preload may appear. To access the thermally induced preload of high-speed spindle bearings, Tu and Stein proposed a model relating the temperatures of spindle shaft, housing and bearing balls.[62,64] They revised the model to predict the temperatures of spindle shaft and bearing balls with the temperature of housing;[65] and the model was used to actively regulate the temperature of housing to control bearing preloads.[66]

In General, calculating the thermally induced preload requires using a bearing thermal model, as indicated in Fig. 1. The bearing thermal model typically consists of a heat transfer model, a thermal expanding model, and a thermal preload model, as shown in Fig. 6.[42]

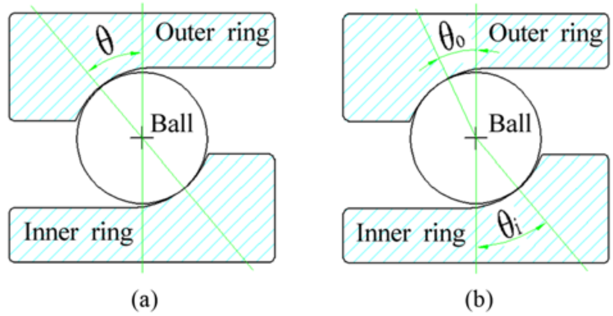


Fig. 5 Contact angles of bearings under (a) station (b) rotation

Table 2 Categories of bearing stiffness models related papers

| | Initial preload | Spindle speed | Ball no. | Angle |
|---|---|---|---|---|
| Ref. no. | 3-5, 9, 11, 16, 22, 23, 25, 31, 36, 40, 41 | 9-11, 13, 14, 31, 34, 40, 47, 57, 59 | 3, 4 | 40 |

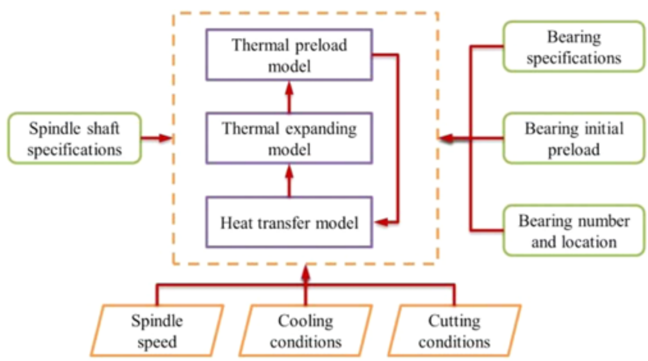


Fig. 6 Mechanism of bearing thermal model

In modeling the spindle-bearing system heat transfer, the heat sources and sinks must be recognized first. Three main heat sources in a spindle-bearing system during machining are: bearings, motor, and a viscous shear of air in the gap between the rotor and stator, as indicated in Fig. 7.[8] The heat generation of bearings ($\dot{Q}_b$) can be expressed as a function of preload, friction forces, and spindle rotating speed[8,28,35,39,40,48]

$$\dot{Q}_b = f(P, \Omega, \kappa) \tag{4}$$

where $\kappa$ is a friction coefficient of the balls. The heat generated by the motor ($\dot{Q}_m$) can be written as[8,40]

$$\dot{Q}_m = f(\Omega, T, \eta) \tag{5}$$

where $T$ is the motor torque and $\eta$ is an efficiency coefficient. The heat generated by the viscous shear of air ($\dot{Q}_a$) is[8]

$$\dot{Q}_a = f(\Omega, d_r, l_r, \mu, h) \tag{6}$$

where $d_r$ is the rotor diameter, $l_r$ is the rotor length, $\mu$ is the dynamic viscosity of air, and $h$ is the gap between the rotor and the stator. If the cutting process is considered, it should also be considered as a heat source. The heat generation-related models of the cutting process are discussed in Ref.[2] The heat sinks of a spindle-bearing system include the spindle base (housing), ambient air, motor cooling water, motor air, and lubrication flow.[7]

After the heat sources and sinks are identified, the heat transfer from the sources to the sinks can be constructed. The internal heat transfer mechanisms in the spindle-bearing system are versatile. The models required to describe the heat transfer inside a spindle-bearing system can be divided into two groups:[7,28,35,40,48,52]

1. Heat transfer through bearings: convection of lubrication air, conduction between the balls and the raceways, conduction between the outer bearing rings and the housing;
2. Heat transfer from the spindle structure into the fluids: convection of motor cooling air, convection of cooling water, convection of ambient air.

These conduction or convection models depend on the specifications of the spindle shaft and bearings, bearing preload, number and location of bearings, spindle speed, cutting conditions, and cooling conditions.

Based on the models of heat sources, sinks, and transfer, a finite difference or element model can be developed to calculate the temperature distribution of the spindle-bearing system by using governing equations.[7,8,28,40,48] After obtaining the temperature distributions, the thermal expansions of the spindle-bearing system can be found using the thermal expansion model,[20,35,40,41,48] and the thermally induced preload can be predicted accordingly. Because the thermally induced preload affects the heat generation of the spindle-bearing system, as shown in Figs. 1 and 6, iteration techniques may also be required to obtain the solutions.

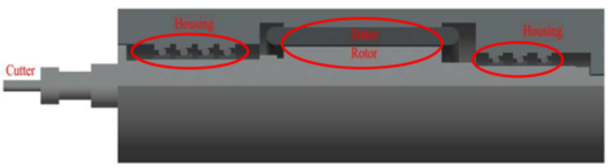


Fig. 7 Heat sources of a spindle-bearing system

**3.3 Dynamic model of spindle-bearing systems**

The spindle-bearing system dynamic model can be obtained by integrating the dynamic models of the spindle shaft and bearings. The resulting equations of motion can be expressed as[41]

$$(M_s + M_b)\ddot{q} + (D_s + D_b - \Omega G)\dot{q} + (K_s + K_b - \Omega^2 N)q = F(t) \tag{7}$$

where $\boldsymbol{F}(t)$ is the external load of interest. By solving Eq. (7), the system response can be predicted with respect to a specific input. Lin et al.[41] developed an integrated model of high-speed spindle-bearing systems to examine the high-speed effects on system dynamics. This comprehensive model was applied later in the studies of optimizing design parameters of spindle-bearing systems.[42-45] Li and Shin proposed an integrated dynamic thermo-mechanical model of high-speed spindles to provide solutions in terms of all the design parameters and operating conditions.[40] Altintas and Cao presented a general method to model spindle assemblies for evaluating the static and dynamic behaviors considering cutting forces.[6,11,12] A similar model was used to investigate the dynamics of high-speed spindles with respect to different bearing preloads and to analyze chatter stability of milling.[9,10,57] Mane et al. developed an integrated spindle-workpiece model to predict stability of high-speed milling.[47]

## 4. Design of spindle-bearing systems

This section first summarizes the research results of reviewed literature related to the design variables or operating parameters. These conclusions can be used as guidelines for design engineers. Then, the design-related papers are introduced to explore the possible optimal design approaches of spindle-bearing systems.

**4.1 Effects of design variables or operation parameters on system dynamics**

Based on the literature review, the design variables of spindle-bearing systems can be categorized into the following four groups: spindle shaft specifications, bearing specifications, number and location of bearings, and the initial bearing preload. Therefore, the sample results from previous studies can be synthesized with respect to the four sets of design variables and operation parameters:

1. Spindle shaft specifications
   - The first-mode natural frequency increases with an increase in the diameter of the spindle shaft.[28,29,41,42]
   - he first-mode natural frequency decreases with an increase in the length of the spindle shaft.[28,29,41,42]
   - The rear end of the spindle is the optimal position for the mass insert to improve the dynamic characteristics of the spindle-bearing system.[68]
   - The mass of the tool has been shown to change the dynamic response dramatically, and the speed and load also affect the frequency response.[31]
2. Bearing specifications
   - A damping added to the position near the rear bearing can obtain the most favorable effect for improving the dynamic characteristics of the spindle-bearing system.[68]

- The number of balls in the bearings can be of importance in the spindle-bearing dynamics and should be considered at the design stage.[3]
- As the number of balls is increased, the system becomes stiffer, because a large number of balls support the shaft.[4]

3. Number and location of bearings

- Bearing orientation significantly affects the spindle stiffness.[39]
- The key design variables of the system's first-mode and second-mode natural frequencies include the spacing between the front and rear bearing sets, and the spacing between the middle line of the two bearing sets and the free end of the cutter.[42]

4. Bearing initial preload

- The natural frequencies of the spindle-bearing system increase with an increase in the bearing preload because of the increase in the bearing stiffness element.[3-5,12,21,28,29,41,68]
- When the bearing preload is increased, the system damping capacity is decreased.[12,68]
- The larger the initial preload that is applied, the fewer vibration amplitudes that are generated.[5]
- As the preload increases to a certain value, the peak-to-peak amplitude decreases. Beyond this value, the reduction in vibration amplitude is insignificant.[5]
- The most appropriate combination of bearing material is the ceramic ball, steel inner race, and steel outer race with respect to the thermo-elastic behavior.[35]
- The spindle system with a rigid preload shows higher stiffness than does the system with a constant preload at high speed; therefore, this type of preload is preferred for high-speed spindles if the thermally induced preload is effectively controlled.[9]
- The overall axial rigidity of the spindle system is determined according to the front bearings in the constant preload case, whereas the radial rigidity is similar in both preload cases.[9]
- The bearing stiffness decreases substantially more at the lower preload than at the higher preload because of the rotation of the spindle.[11,12]
- The external axial/radial loads have a smaller effect on systems with rigid preloads than on systems with constant preloads.[9]

5. Operation parameters

- The stiffness of the bearing decreases as the rotational speed increases.[9,13,14,41,57,59]
- Cooling control is a highly effective method for compensating for the changes in dynamic behavior of the spindle-bearing system.[35]
- The coolant temperature, initial preload, and housing outer diameter are more significant in determining spindle characteristic behaviors.[37]
- The rotational speed effects of the spindle shaft have a greater influence on the lower frequencies. For higher modes, the rotational speed effects of the spindle shaft are negligible compared to the speed effects of the bearings.[6,11,12]

### 4.2 Design of spindle-bearing systems

This subsection reviews literature on the design of spindle-bearing systems. Because the papers reporting on design topics are relatively few and not closely related to each other, some papers that are not directly related to machine tools were also collected and are introduced separately.

Wang and Chen[67] applied a modified search method based on a GA to find optimal rigid and elastic supports for beams under three different boundary conditions. In their algorithm, a large probability of mutation in the GA search and elitist strategy were used. The effectiveness of the proposed approach has been also investigated.

Montusiewicz and Osycka[49] used a four-stage multi-criterion optimization strategy to obtain the optimal design of machine tool spindle systems. A computer-aided optimal design software package for the spindle systems of machine tools is developed based on this optimization strategy. Numerical results from applying the proposed approach to a particular example showed that the software was relatively helpful in the decision-making process.

Srinivasan et al.[61] presented a non-iterative, numerically robust method to rank candidate magnetic bearing configurations, which can be used to determine the optimal locations of bearings before the actual bearing design. The objective of the method was to improve the design process by separating the problem of determining the "best" bearing locations from that of determining the actual bearing design.

Lee and Choi[38] proposed a design approach for reducing the weight of a flexible rotor in ball bearings with rotational speed and load-dependent stiffness characteristics under constraints of the system eigenvalues and bearing fatigue life. The design variables they selected were inner radii of shaft elements, the positions of the ball bearings, and the preload on the bearings. An augmented Lagrange multiplier method was used to find the optimal solutions. A design of a multi-stepped rotor supported by two angular contact ball bearings was used as an example.

Wu et al.[69] developed a modified two-level optimization approach for the concept design of a machine tool. The lower level of optimization was applied to each structural part of the tool and the upper level of the machine tool as an integrated system. Examples presented in the paper showed that the weight of the machine was minimized while sufficient stiffness was maintained in the concept design.

Kang et al.[32] demonstrated integrated computer aided engineering (CAE) strategies for designing machine tool spindles. They used the FE method to model the spindle-bearing system for evaluating both static and dynamic characteristics. The computations of bearing dynamic coefficients and thermal performance were incorporated into the static and dynamic analyses of spindle-bearing systems.

Choi and Yang[15] used an immune-GA (IGA) to optimize the design of rotor-bearing systems. The design variables they considered were the shaft diameter, bearing length, and clearance. The results revealed that the IGA can reduce the weight of the shaft and improve the critical speed under dynamic constraints.

Lin and Lin[46] used the FE method to develop the minimal weight design of a rotor-supported oil-lubricated bearing subject to a frequency constraint by using a sequential quadratic programming technique. The effects of minimal weight design on the distribution of rotor shape, weight saving, and system stability were also examined.

Lin and Tu[42] conducted a study to investigate the effects of eight parameters on the natural frequencies of the spindle system. Based on the rule of maximum improvement first, a set of systematic design procedures was also proposed to develop the optimal design of a spindle-bearing system.

Lin[43] used the Taguchi method to determine the optimal design of a high-speed motorized spindle-bearing system by considering the high-speed effects. The optimized signal-to-noise ratio (SNR) enhanced the system's first-mode natural frequency (FMNF) and reduced the variations of the FMNF that occurred at high speeds because of high-speed effects.

Lin[45] developed an effective approach that integrated a GA and a Monte Carlo simulation to determine the optimal nominal and tolerance values of the location of bearings. To verify the effectiveness of the proposed approach, an actual design example of bearing location for a high-speed motorized spindle-bearing system was investigated. A sensitivity study on the weighted values was also conducted.

## 5. Conclusion and Future Research Possibilities

This paper reviews approximately 60 papers to investigate the progress of the modeling and design of spindle-bearing systems. The sampled literature was divided into two categories: spindle shafts and bearings. The dynamic models for these two main components are reviewed separately. Critical design guidelines are accumulated from these papers and optimal design approaches are briefly explored. Moreover, the following research topics are identified for future studies:

- Most of the reviewed papers are related to high-speed machining. Therefore, research that includes high-speed effects on the dynamic models of spindle shafts and bearings is recommended.
- Compared to other fields, the thermal models of bearings are still not thorough. More research should investigate thermally related complications to enhance temperature predictions.
- Because the bearing preload varies according to the operating conditions, a time domain simulation may be a more adequate approach for reflecting the real situations of a spindle-bearing system. Integrating all of the models and calculating the response with respect to time-varying input may be necessary.
- The number of papers related to the design approach is less than that related to the dynamic models. The application of classical or modern optimization techniques in the spindle-bearing system design is a promising topic for future research.

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