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Vibration Response Analysis on Spindle System of Milling Machine

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Abstract: The dynamic characteristics and dynamics parameters of rolling bearings is very important to dynamics and vibration response of rotate machine such as rotor systems, gear systems and Spindle systems. The frequencies of rotate machine are affected by dynamics parameters of rolling bearings at different places. The purpose in the work presented is to research a new approach and multibody model of Spindle systems with equivalent dynamics parameters of rolling bearings. The flexible Spindle body has been constructed by the fixed interface component mode method. The four different models of rolling bearings for Spindle systems have been developed using equivalent spring and damper elements. The experiments of Spindle body and Spindle system have been carried out. Experimental modal frequencies have been got by impulse vibration test and sweep frequency vibration test. The frequencies and vibration response of Spindle systems have been calculated by adjusting equivalent spring and damper elements to minimizing errors between the calculated and experiment frequencies. The results show the errors of frequencies of linear equal and unequal spring and damper models are large except the first frequency. However, the errors of nonlinear equal and unequal spring and damper models are small. The predicted frequencies of nonlinear unequal spring and damper model are the most accurate and agree well with the experiment results. The presented method can be applied to calculate the nonlinear equivalent stiffness parameters accurately for rolling bearings in multibody systems.

Introduction

The performance of rotating mechanics has been affected by the dynamic characteristics of rolling bearings. The dynamic characteristics of the joint faces of rolling bearings are nonlinear and time-varying. The dynamics of rolling bearings and the dynamics parameters of Spindle system have been researched by many researchers. The research on single rolling bearing has been considered much more factors such as lubrication, cage, waveness, roughness. The dynamics of multibody systems with rolling bearings has been an important research area recently. The dynamics parameters of rolling bearings are usually equivalent to linear spring and damper elements for dynamics analysis of Spindle system. The method of modal experiment and parameter identification are generally applied to calculated the equivalent stiffness and damper parameters[1].

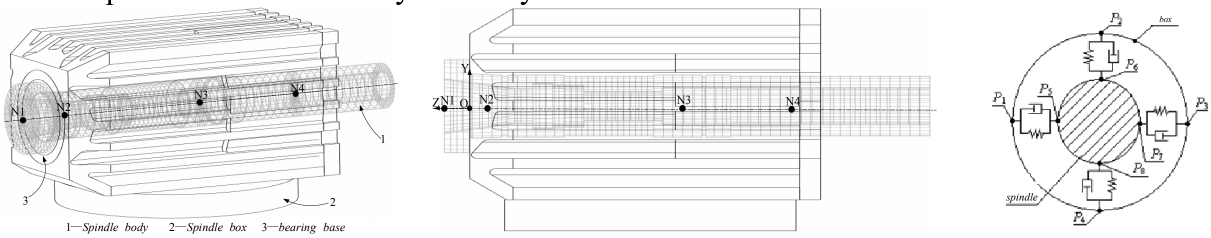
The linear equivalent stiffness parameters and dynamics model of spindle system was constructed and the modal frequencies were calculated by A.M.Sharan[2] , W.R.Wang and C.N.Chang[3], K.W.Wang and C.H.Chen[4].. The linear equivalent model of spindle-gear system was constructed and dynamics analysis by T.Q.Yao[5]. The modeling approach for interface stiffness of spindle-tool holder has been discussed and the FEA of spindle systems been carried out by X.S.Gao[6]. The finite element modeling of high-speed spindle system dynamics with spindle-holder-tool joints and Effect of interfaces on dynamic characteristics of a spindle system have been discussed by B.WANG[7,8]. The FEM model of three-layer structure spindle systems of boring machine has been constructed and the linear stiffness and damper parameters have been identified using parameter identification method by X.Han[9].

The method of equivalent dynamics model of spindle systems with different linear and nonlinear joint parameters has been discussed and verified in the presented work.

The model of Spindle system

2.1 Flexible Spindle Model

In the Figure.1, The cutter is fixed on the N1 location, the double row roller bearing and is fixed on the N2 equivalent location, two angular ball bearings is fixed on the N3 equivalent location by face to face, the driven gear is assembled to N4 equivalent location by multiple spline. The FEM model of Spindle is constructed by modal synthesis method.



a three-dimensional model b two-dimensional projection c spring-damper model

Fig.1 The model of Spindle component

In this investigation, fixed interface component mode method was used to model the flexible Spindle. This approach is based on the Rayleigh-Ritz method as developed by Craig and Bampton. In this approach a complex structure is divided into components whose modes are subdivided into normal and constraint modes. The normal modes are obtained by solving the eigenvalues problem.

$$K - \omega^2 M = 0 \tag{1}$$

where K and M are stiffness and mass matrices. The boundaries of the structure were fixed at the bearing locations represented by interface points. The number of normal modes of interest depends on the application and frequency band of interest.

The constraint modes are static deformation shapes obtained by applying a unit displacement to each boundary nodes while constraining the degrees of freedom of other boundary nodes. The number of constraint modes in flexible bodies depends on number of interface points. Each interface point is associated with six constraint modes.

The degrees of freedom of each component are reduced by using modal substitution l

$$\{u_c\} = \begin{bmatrix} u_1 \\ u_B \end{bmatrix} \begin{bmatrix} \varphi_N & \varphi_C \\ 0 & 1 \end{bmatrix}_c \begin{bmatrix} q_N \\ u_B \end{bmatrix}_c \tag{2}$$

where u_I is displacement of nodes inside the component (internal nodes), u_B is the displacement of nodes on the boundary of component (boundary nodes), u_N are the natural modes, u_C are the constraint modes, and q_N are the generalized co-ordinates of the natural modes. The behavior of the entire model is analyzed by assembling the response of each component.

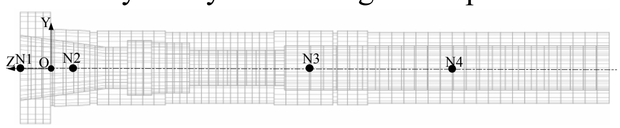
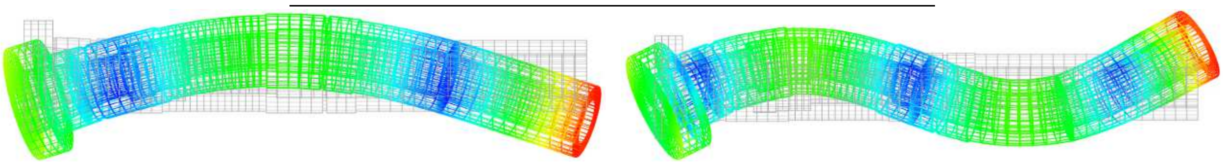


Fig.2 The FEM of Spindle

Tab.1 The relative error of the calculated and experiment results

order/frequency	calculated	experiment	relative error
1	1350.4	1345.8	0.34%
2	3486.1	3366.5	3.55%



a the first bend frequency, f_1=1350.4Hz b the second bend frequency, f_2=3486.1Hz

Fig.3 The frequencies of Spindle

The relative errors of the calculated and experiment frequencies are showed in Tab.1. The vibration modes of the first and second frequencies are showed in Fig.3. The relative errors are small between the calculated and experiment frequencies.

2.2 Equivalent Dynamic Bearing Model

A key aspect of modeling the bearing dynamics with the spring-damper element method is obtaining the total forces and moments acting on the whole rolling bearing. In the current model, the linear and nonlinear spring-damper element are considered as a part of the analysis respectively.

In the reference[11], the linear and equivalent stiffness on the location N2 and N3 has been calculated by the modal experiment and parameter identification method. The spring-damper element model is showed in the Fig.1c. In order to get the relatively accurate support stiffness parameters of rolling bearings, the relative errors between the calculated and experimented frequencies should be minimum by adjusting the linear and equivalent spring-damper parameters. However, this method can only achieved the first bend frequency, the error is too large for the other frequencies in the reference.

2.3 Dynamic Spindle System Model

The shaft are modeled by the FEM and each rolling bearings are modeled by spring-damper elements. Interface points are established at points where the bearing supports on the Spindle. These interface points are made coincident with the bearing inner race center of gravity. Thus the dynamic response is passed from one model to other. Figure 1 depicts these interface point interactions as two headed arrows indicating that the exchange of dynamic response occurs from both the sides, namely, rolling bearings and Spindle. As mentioned previously, the rolling bearings have single piece outer races. Therefore, the outer races of the two spring-damper arrangement, each representing one of the bearing, are rigidly linked to the bearing base. Finally the bearing base is attached to the Spindle box through a fixed constraint.

$$M\ddot{q} + C\dot{q} + K_{\mathrm{L}}q + K_{\mathrm{N}}q^{n} = F \tag{3}$$

where M is total mass matrix， C is total damp matrix, K_{L} and K_{N} are linear and nonlinear stiffness matrix, q is generalized coordinate vector, F is generalized loads， n is nonlinear factor, when roller bearing linear contact, n=10/9 and ball bearing point contact, n=3/2.

Eq.(3) is solved by a linearization method for constrainted multibody sytems with Recurdyn. If the stiffness of rolling bearings has been given, the frequencies of Spindle system will be calculated. Details are discussed in the reference[10]. Four models with flexible Spindle, rigid Spindle box, rigid bearing base and different spring-damper are discussed in the thesis. The four models are the linear and equal spring-damper model(①LESDM), the linear and unequal spring-damper model(②LUSDM), the nonlinear and equal spring-damper model (③NESDM), the nonlinear and unequal spring-damper model(④NUSDM). The target function

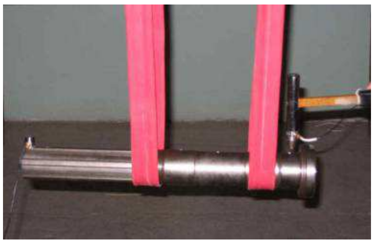
$$error_{\min} = \left\{ \sqrt{\sum_{i=1}^{3} \min\left(e_i^2\right)} \Big/ 3 \right\} \tag{4}$$

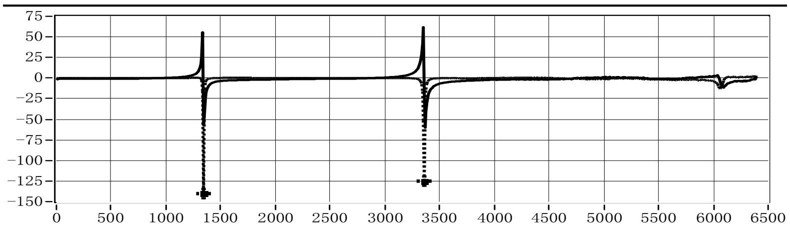
where $\min\left(e_i\right) = \min\left\{\left(f_{\mathrm{c}i} - f_{\mathrm{t}i}\right)/f_{\mathrm{t}i}\right\}$, $f_{\mathrm{c}i} = \sqrt{K^*/M^*}$ are the calculated frequencies from the Eq.(3). f_{ti} is the tested frequencies from the experiment. The constraint condition of variables K_{L}, K_{N} are $1.0\times10^4 \le K_L \le 9.0\times10^4$,$1.0\times10^5 \le K_N \le 9.0\times10^5$.The linear Unit of stiffness is N/mm and nonlinear Unit is $\mathrm{N/mm^{10/9}}$ or $\mathrm{N/mm^{3/2}}$.

Results and Discussion

Experimental and Analytical Results Corroboration.

The frequencies of Spindle and Spindle system predicted by the spring-damper Spindle model were corroborated with the results obtained from the Spindle system test rig showed in Fig.4 and Fig.5.

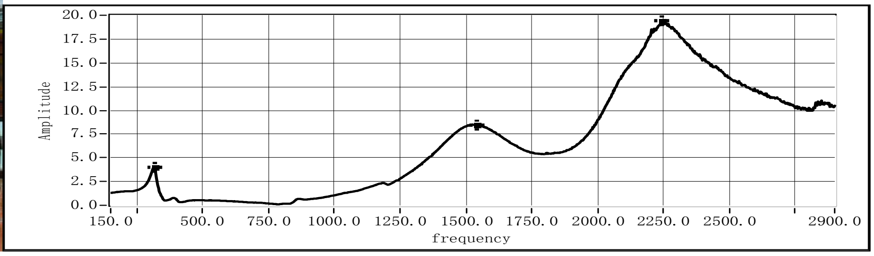




a the pulse excitation of Spindle b the experiment frequencies response results of Spindle

Fig.4 The frequency response of Spindle by pulse excitation

The sine sweep vibration test of Spindle component has been applied to test the frequencies response and research the equivalent dynamics paramers. The Spindle component is consisted of Spindle, Spindle box, rolling bearings and bearing base. To get accurate frequencies response, the sweep frequency range is 100Hz to 3000Hz and the sweep frequency increment has been selected as 0.5Hz. The first three orders frequencies of the Spindle component are f_1=317.4Hz, f_2=1540.4Hz, f_3=2244.0Hz.



a photograph of vibration test b frequency response results of 100-3000HZ

Fig.5 Sine sweep vibration test of spindle systems

The equivalent stiffness parameters at N2 and N3 location of the Spindle component has been calculated by Eq.(4) and optimization analysis in Recurdyn software. The calculated and experiment results and relative errors are showed in Tab.2.

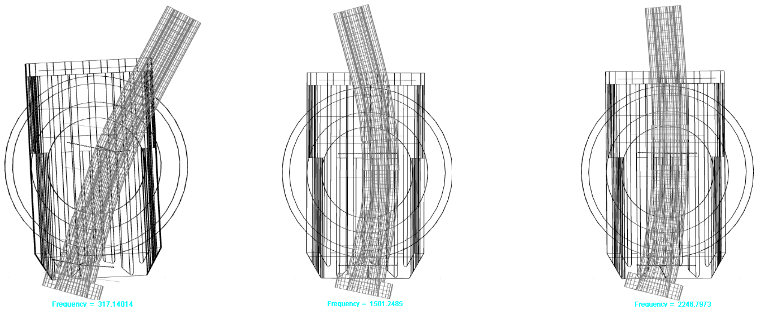
Tab.2 The calculated and experiment frequencies and errors

model	location		Frequencies/Hz			relative error(%)		
	N2	N3	first	second	third	first	second	third
①	1.45e4	1.45e4	316.5	714.9	1614.2	-0.28	-53.59	-28.04
	3.02e4	3.02e4	317.9	946.3	1553.8	0.16	-38.57	-30.77
②	4e4	2.8e4	317.3	1041.8	1551.6	-0.03	-32.37	-30.86
③	3.56e5	3.65e5	317.6	1455.7	2428.7	0.06	-5.50	8.22
④	6e5	3.24e5	317.1	1501.2	2246.8	-0.03	-2.49	0.09
experiment	--	--	317.4	1540.4	2244.3	--	--	--

The frequencies and vibration response of Spindle systems have been calculated by adjusting equivalent spring and damper elements to minimizing errors between the calculated and experiment frequencies. Then the equivalent stiffness parameters have been calculated by the presented method.

Generally, the radial stiffness at the Spindle head N2 is greater than at the rear-end N3 to achieve the milling quality. In Tab.2, the errors of frequencies of ①LESDM and ②LUSDM are large except the first frequency. The ①LESDM and ②LUSDM are accurate for the first frequency, but inaccurate for high order frequencies. The ①LESDM and ②LUSDM are sometimes may be applied to calculate the equivalent stiffness parameters inaccurately. The errors of frequencies of ③NESDM and ④NUSDM are small, the results of ④NUSDM are the most accurate.

According to Hertz Contact Theory, the contact dynamics and vibration of rolling bearings are nonlinear. Therefore, the load-deformation factor of radial stiffness of roller bearings at the Spindle head N2 is showed as linear contact factor 10/9. The load-deformation factor for angular ball bearings at the rear-end N3 is showed as point contact factor 3/2.



a first f_1=317.1Hz b second f_2=1501.2Hz c third f_3=2246.8Hz

Fig.6. Frequencies of Spindle component by the ④NUSDM

The frequencies of Spindle component supported by the ④NUSDM are showed in Fig.6. The first order frequency is 317.1Hz coupling rigid vibration of Spinlde, box and ④NUSDM. The second and third order frequencies are 1501.2Hz and 2246.8Hz. They are coupling flexible bending vibration of Spinlde and ④NUSDM.The ④NUSDM is typically strong nonlinear model and gets nonlinear nonequivalent spring and damper elements. The ④NUSDM and the dynamics model of Spindle have been verified by the calculated results. It can be applied to calculate the equivalent stiffness parameters accurately for rolling bearings in multibody systems.

Conclusions

The models of Spindle body and Spindle component have been constructed by FEM with equivalent spring damper elements. The simulations of vibration response have been carried out. The calculated frequencies are compared with the experiment results by the pulse excitation and the sine sweep vibration test. Therefore, the equivalent stiffness parameters have been calculated by minimizing errors between the calculated and experiment frequencies. It is believed that good agreements between the results obtained from NUSDM models and the test results will also be the reasonable evidence for the availability of the presented approach.

The presented method and the model are general and available to research the frequencies, vibration response and dynamics characteristics of Spindle systems.

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