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Predicting time-varying preload of spindle bearing based on two measured temperatures

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Abstract: This paper presents a method to predict time-varying preload of a spindle based on two measured temperature. Transient temperature in the spindle is estimated using inverse method. Increment of temperature resulting in thermal expansion is then calculated. Extra preload from thermal expansion at the time step is further added to initial value for giving final magnitude of the preload.

Key words: *time-varying preload, inverse method, spindle*

INTRODUCTION

Spindle with complex structure is an importance component in machine tools. To create high stiffness for the bearing as well as spindle, a preload is usually applied on these bearings. There are three main methods for applying preload which are fixed position, constant and variable preloads. Among them, the fixed position preload is often employed on the spindle worked at low speed and heavy load. However, the preload using fixed position may vary due to growth up temperature in the spindle. The variation of the preload is caused by non-uniform distribution of the temperature which induces thermal expansion of the spindle parts. Thus, to determining variation of the preload, the temperature in entire spindle must be examined. Bosmanns and Tu [1] based on heat flow of a motorized spindle in [2] and the finite difference method (FDM) to established a thermal model for high speed motorized spindle. The temperature in whole spindle was obtained. The finite element method (FEM) is applied to analyze thermal and thermal-structure coupling in [3]. Zivkovic et al. [4] presented a model based on bearing model and FEM to consider non-stationary change of temperature and preload. A monitoring thermally induced preload in spindle bearing was presented in [5]. The induced thermal preload calculated by thermal expansion was compared to experimental results. Than and Huang [6] presented an algorithm to analyze nonlinear thermal effects in spindle bearings subjected to preload. They employed FDM to establish thermal model and then acquire temperature distribution. Besides, applying above method to find temperature, inverse method is widely applied to estimate temperature, heat generation, parameters, etc. in recent decades. Ngo et al. [7] proposed an inverse method to predict interface temperature, heat generation and convection coefficient in welding process. The heat flux and transient temperature on a turning cutting tool were carried using a nonlinear inverse technique combined with COMSOL [8]. Through the literature review, it can be seen that non researches study on time-varying preload based on inverse heat transfer method. This paper apply the inverse method to obtain temperature distribution in the spindle which then utilizes for examining thermal expansion and its resulting in change of the preload.

INVERSE HEAT TRANSFER METHOD

The inverse method combines the finite element (FE) thermal model and the conjugate gradient method. Based on two measured temperature taken on spindle housing, the two unknown heat generation at front bearing q_1 and rear bearing q_2 as shown in Fig. 1(a) are estimated and then the temperature in entire spindle is acquired. The heat sources for a period time Δt are written as:

$$\vec{\mathbf{w}}_n = \begin{bmatrix} q_1 & q_2 \end{bmatrix}_n \text{ for } t_n \le t \le t_{n+1}, t_n = n\Delta t, \text{ and } n = 1,2,3... \quad (1)$$

The inverse solution $\vec{\mathbf{w}}_n = \begin{bmatrix} q_1 & q_2 \end{bmatrix}_n$ for period n^{th} is obtained when the objective function $\mathbf{J}(\vec{\mathbf{w}}_n)$ in Eq. (2) is minimized.

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$$J\left(\vec{\mathbf{w}}\right)=\int_{t=t_n}^{t_{n+1}}\sum_{i=1}^{2}\left[T(x_i,z_i,t)-T_m(x_i,z_i,t)\right]^2 dt \tag{2}$$

where $T_m(x_i,z_i,t)$ is the measurement temperature and $T(x_i,z_i,t)$ is the temperature of solving direct problem. In order to minimize $J\left(\vec{\mathbf{w}}\right)$, the conjugate gradient method (CGM) is applied in this study. Iteration of CGM is given as follows:

$$\vec{\mathbf{w}}_n^{k+1}=\vec{\mathbf{w}}_n^{k}-\beta^k\mathbf{P}^{k+1} \text{ with } k=0,1,2... \tag{3}$$

and

$$\beta^k=\int_{t=t_n}^{t_{n+1}}\sum_{i=1}^{2}\left[T(t,x_i,z_i)-T_m(t,x_i,z_i)\right]\Delta T(t,x_i,z_i)dt \Bigg/ \int_{t=t_n}^{t_{n+1}}\sum_{i=1}^{2}\Delta T^2(t,x_i,z_i)dt \tag{4}$$

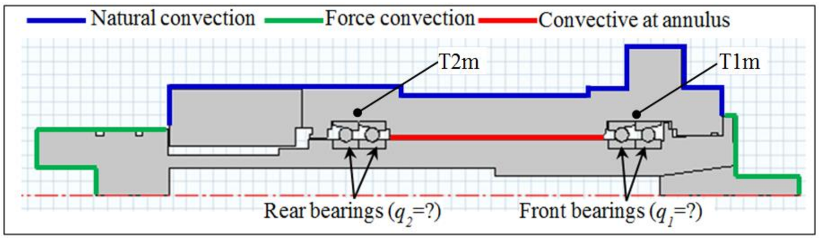
$$\Delta T(t,x_i,z_i)=T\left(t,x_i,z_i;\vec{\mathbf{w}}_n+\Delta\vec{\mathbf{w}}_n\right)-T\left(t,x_i,z_i;\vec{\mathbf{w}}_n\right) \tag{5}$$

$$\mathbf{P}^{k+1}=\nabla\mathbf{J}^k+r^k\mathbf{P}^k \tag{6}$$

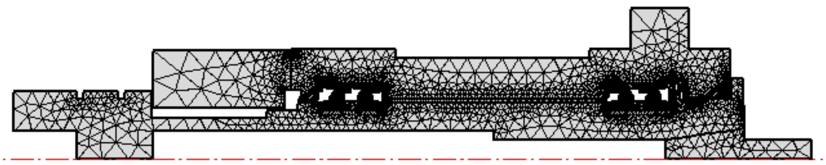
$$r^k=\sum_{i=1}^{2}\left[(\nabla\mathbf{J})^k\right]^2 \Bigg/ \sum_{i=1}^{2}\left[(\nabla\mathbf{J})^{k-1}\right]^2 \tag{7}$$

$$\nabla J^k=\left[\partial\mathbf{J}(\mathbf{w})/\partial q_1 \;\; \partial\mathbf{J}(\mathbf{w})/\partial q_2\right]^T \tag{8}$$

in which the superscript k stands for the number of iterations; β represents the search step size; $\mathbf{P}$ is the direction of descent; $\nabla\mathbf{J}$ is the gradient of objective function and r is the conjugation coefficient. These above equations reveal that all parameters for CGM iteration are computed through temperature of direct problem which will be introduced following.



(a) Structure and convective condition



(b) Meshed

Fig. 1: The finite element model of the spindle

In this study, the direct problem is established using FE analysis COMSOL software. Fig. 1 shows the structure and convective condition of the spindle. Because temperature in the spindle is not too high, the radiation is ignored in this study. Hence, the heat will be dissipated through convective phenomenon. The convective coefficient can calculated as [9]:

$$h=\overline{Nu}k_{air}/d \tag{9}$$

where d is the equivalent diameter, k_{air} is the thermal conductivity of air, and $\overline{Nu}$ is the average Nusselt number. The convection in the spindle is classified into three kinds which are natural, force and annulus convections; and the $\overline{Nu}$ of each kind is determined by:

$$\overline{Nu} = \left\{0.6 + 0.387(Ra_D)^{1/6} \Big/ \left[1 + \left(0.559 / \Pr\right)^{9/16}\right]^{8/27}\right\}^2 \quad \textit{for natural convection, } [10] \tag{10}$$

$$\overline{Nu} = 0.6366(\mathrm{Re}\,\mathrm{Pr})^{1/2} \quad \textit{for force convection around rotational shaft, } [11] \tag{11}$$

$$\overline{Nu} = \begin{cases} 2(\Delta r / r_i) / \ln\left(1 + \Delta r / r_i\right) & \text{for } Ta^2 / F_g < 1700 \\ 0.128\left(Ta^2 / F_g\right)^{0.367} & \text{for } 1700 \le Ta^2 / F_g < 10^4 \\ 0.409\left(Ta^2 / F_g\right)^{0.241} & \text{for } 10^4 \le Ta^2 / F_g \le 10^7 \end{cases} \quad \textit{for convective at annulus, } [12] \tag{12}$$

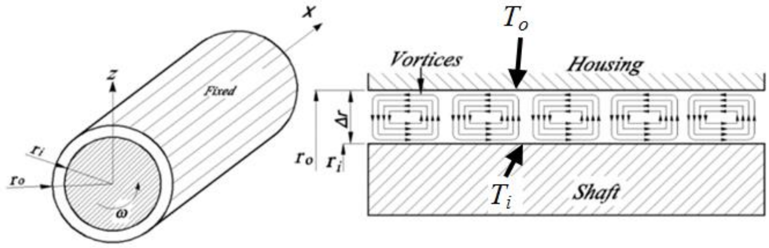


Fig. 2: Taylor vortices at annulus of spindle

Because the temperature of air at annulus depends time, the convective in there is difficult to model. Fig. 2 describes schematic representation of Taylor vortices in an annulus with shaft rotation. Based on convection theory, the heat dissipated by convection q_{out} is given as:

$$q_{out} = h(T_i - T_{air}) \approx k_e \,\partial T / \partial r \tag{13}$$

$$h(T_i - T_o) \approx k_e \left(T_i - T_o\right) / \Delta r$$

$$k_e \approx h\Delta r = \overline{Nu} k_{air}\, \Delta r / d \tag{14}$$

in which k_e is the equivalent thermal conductivity of the air; T_i and T_o are the temperatures on surfaces of shaft and housing, respectively. Therefore, the convective at the annulus is replaced by an air having k_e. The meshed model is displayed in Fig. 1(b). Detail of calculation procedure is drawn in Fig. 3.

THERMAL INDUCED PRELOAD

The initial preload of bearing, F_{p0}, is created by fixing the relation of the bearing through suitable for dimension of shaft, housing and space. However, different increment of temperature of spindle parts due to its operation can induce dissimilar amount of thermal expansion in them. This is the reason caused change of the preload. To determine amount variation of the preload, the thermal expansions of shaft and housing spaces must be calculated based on the temperature results which were obtained from above inverse heat transfer problem.

Fig. 4 represents schematic of bearing arrangement in the spindle using tandem (DT) configuration. From this figure, the thermal expansion of spindle components is calculated as:

$$\varepsilon_s = \alpha \cdot Ls \cdot (T_s - T_0) \tag{15}$$

$$\varepsilon_h = \alpha \cdot Lh \cdot (T_h - T_0) \tag{16}$$

$$\varepsilon_b = \alpha_b D(T_b - T_0) \tag{17}$$

$$\varepsilon_{rings} = 0.5\alpha\left[D_{io}(T_{ir} - T_0) - D_{oi}(T_{or} - T_0)\right] \tag{18}$$

where α is the thermal expansion coefficient. ε_s, ε_h and ε_b are thermal expansion of shaft, housing and ball, respectively. ε_{rings} is the different expansion of the inner ring and outer rings. T_s, T_h, T_b, T_{ir} and T_{or} are sequentially average temperature of shaft, housing, ball, inner ring and outer ring. T_0 stands for initial temperature. Other parameters can refer in Fig. 4. Contact angle ϕ is assumed as a constant. Thus, the total deformation of each bearing along contact line can be expressed as:

$$\delta = \varepsilon_b + \varepsilon_{rings} \cos\phi - (\varepsilon_s - \varepsilon_h)/4 \tag{19}$$

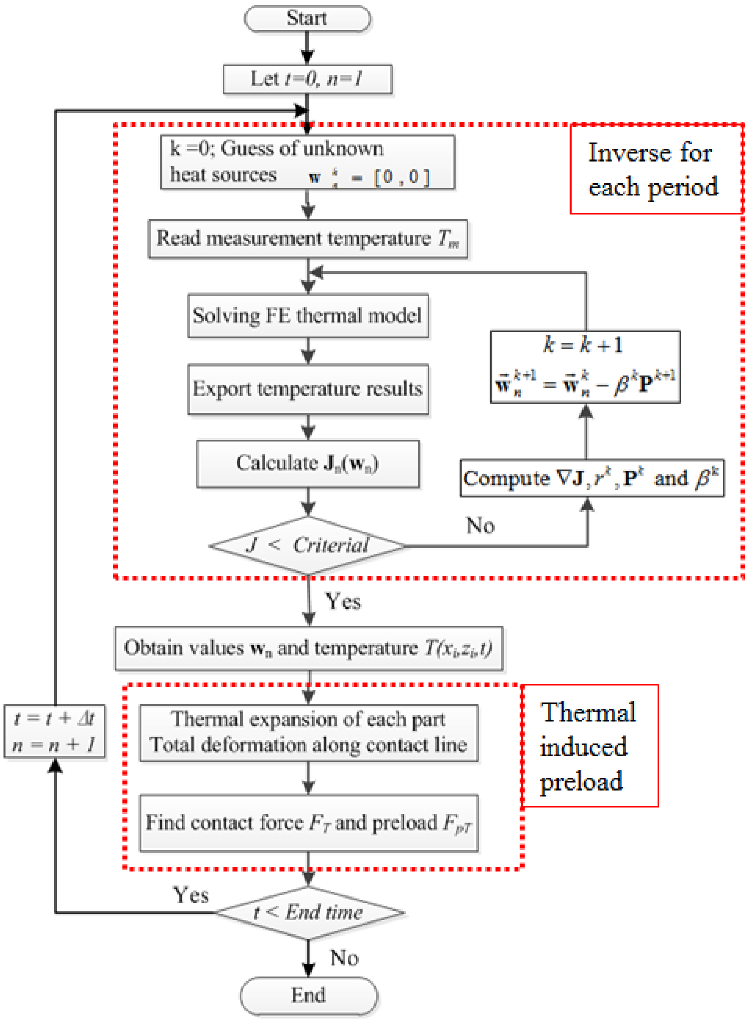


Fig. 3: Flowchart of the computational procedure

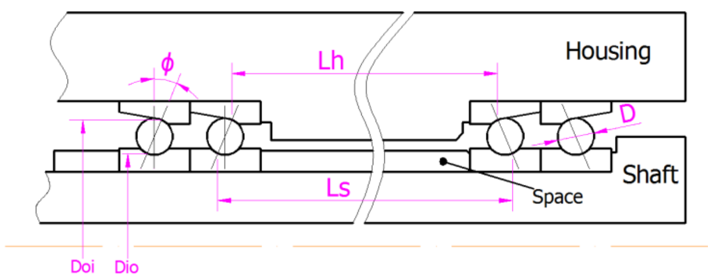


Fig. 4: Bearing arrangement in the spindle

Based on the Hertzian contact principle, the contact force due to the change of interference is determined as:

$$F_T = k_t \delta^{1.5} \tag{20}$$

Finally, the thermal induced preload is computed by

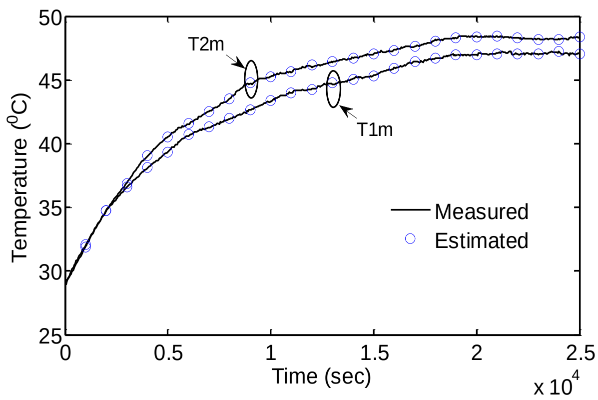
$$F_{pT} = N \cdot F_T \sin\phi \tag{21}$$

In Eqs. (20) and (21), k_t is the elastic constant of the bearing and N is the number of rolling elements. Thus, the thermal induced preload can be easily acquired when knowing temperature of the spindle. By combining with the initial preload, the total preload can be easily acquired. Calculation of preload at each time step is placed in inverse algorithm as shown in Fig. 3.

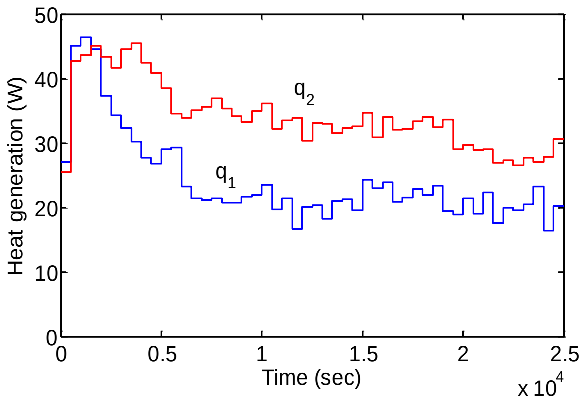
RESULTS AND DISCUSSIONS

Figs. 5 and 6 show estimated heat sources and temperature for speeds of 8,000 and 10,000 rpm, respectively. Results show that the estimated are in good agreement with measured temperatures. Trend of heat generations from bearings consists with previous result [5]. Because of measurement error effects, there is oscillation on the heat source results, refer to Figs. 5(b) and 6(b). In addition, temperature distribution in whole spindle is also acquired as display in Figs. 5(c) and 6(c). Temperature variation of bearing parts and average temperature within bearing span respect to time are extracted and given in Figs. 5(d) and 6(d).

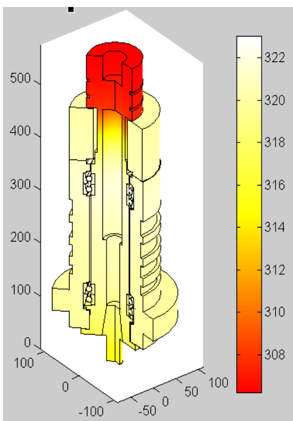
After having the transient temperature, preload can be updated based on adding preload induced by thermal into initial preload for each time step. Fig.7 draws time-varying preload for two speeds. It can be seen that preload is decreased in the first time stage, next is increased and finally is become steady-state. It is similar to results reported in [6, 13]. It is clear that the preload of 10,000 rpm is higher than that of 8,000rpm at stationary; and the preload sequentially reaches 504.1 N and 741.1 N at 8,000 rpm and 10,000 rpm. One can conclude that the preload nonlinear varies with time under thermal effects.



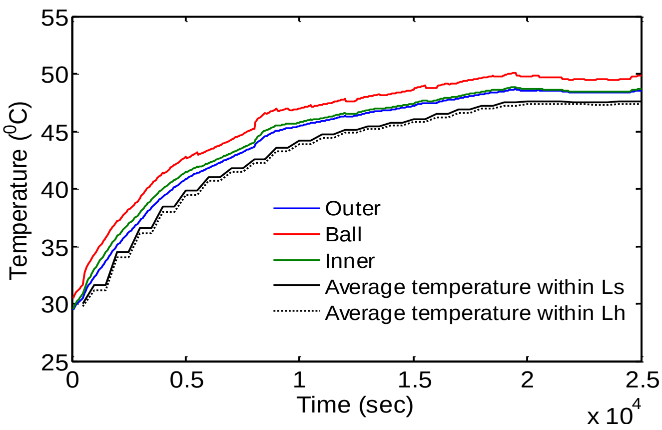
(a) Temperatures



(b) Heat sources

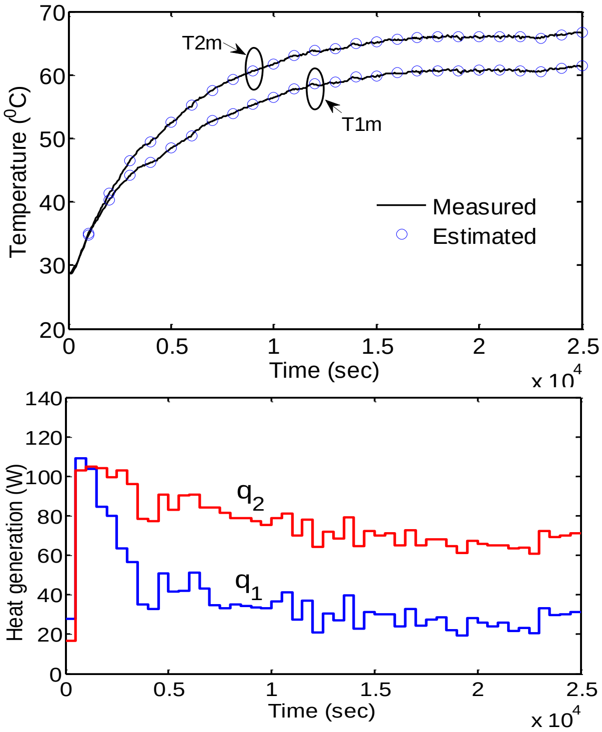


(c) Entire temperature

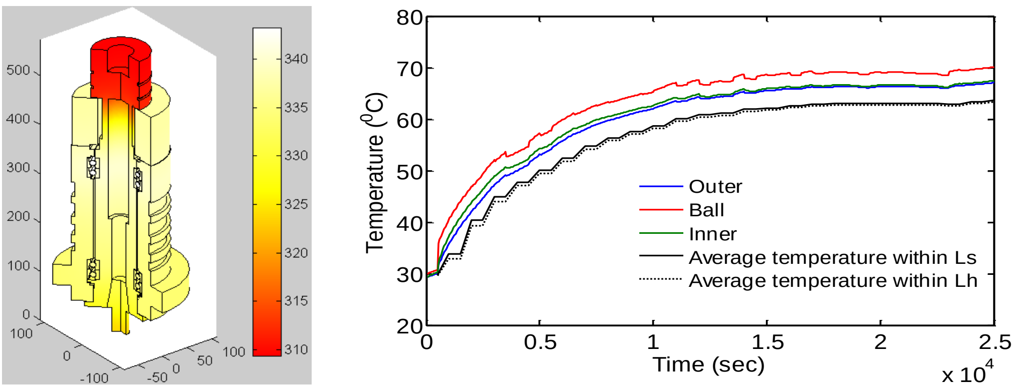


(d) Extract temperature

Fig. 5: Heat sources and temperature results for 8,000 rpm



(a) Temperatures (b) Heat sources



(c) Entire temperature (d) Extract temperature

Fig. 6: Heat sources and temperature results for 10,000 rpm

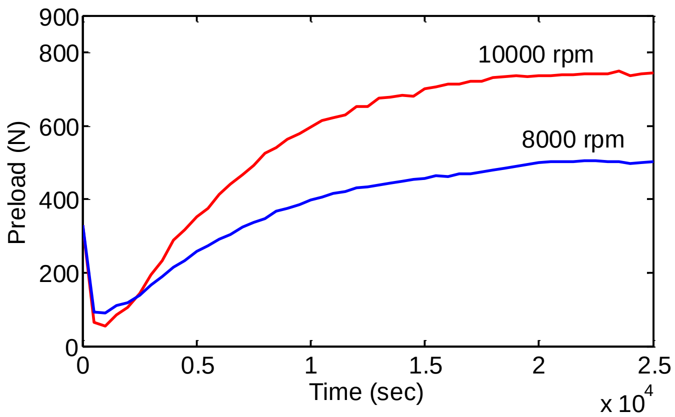


Fig. 7: Time-varying preload for different speeds

CONCLUSIONS

Time-varying preload of a spindle has successfully estimated base on two measurement temperature in this research. Transient heat sources and temperature are obtained through inverse method using these temperatures. For each time step, thermal expansion of bearing parts, shaft and housing is determined for treating effects of thermal on the preload. Results show that the preload nonlinearly changes with time due to temperature growth. At steady-state, the preload of all speeds is higher than initial preloads. Significant effects of the speed and thermal on preload is observed. These finding here can provide useful information for considering dynamic behaviours and life of bearings as well as the spindle.

ACKNOWLEDGEMENTS

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