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HDL MODEL OF MACHINE TOOL SLIDEWAY

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Abstract: *The paper describes a modelling technique of tool rest of modern machining centre with focus at damping of dovetail groove slideway. Damping of tool rest structure strongly affects the stability of machining process, which must be maintained at wide range of machining condition, thus the adequate model of such damping should respect its physical origin to enable realistic estimation of boundaries between stable regions and self-excited vibrations at design stage of a new machine. Presented model of tool rest comprises damping characteristics obtained by HDL analysis of oil film of dovetail groove.*

Keywords: ***Hydrodynamic Lubrication, Dovetail Groove, Damping, Tool Rest.***

1. Introduction

Stability of machining is a critical parameter of development process of new machine tool. Potential instability in form of self-excited vibrations of system: tool—workpiece is an inadmissible event, which profoundly degrades geometric precision and surface finish of work surface, accelerates tool wear and creates excessive noise. This phenomenon, caused by nonlinearity of the system of tool—workpiece—machine, restricts applicable cutting conditions in given case of machining process. It is understandable that the ability to predict regions of stable cutting conditions of future products is one of machine tool manufacturers' key initiatives, assisting analysis lead development (Altintas et al., 2005). Damping of the machine structures affects instability thresholds and plays important role in preventing self-excited vibrations. Major source of damping of tool rest structure has its origin in slideways, where the oil film present between the slide and the saddle acts as a squeeze film damper. See fig.1 for the tool rest geometry. This paper deals with hydrodynamic lubrication (HDL) analysis of the oil film. Results of the HDL analysis are used for calculation of damping coefficients that were used in the model of tool rest built by Craig-Bampton substructuring technique; such linear model can be treated in frequency domain, which is convenient for incorporating it into complex models of machine tool, built in order to analyse stability of cutting process (Turkes et al., 2010). More complex models respecting interaction between the slide and the saddle via squeeze film and mechanical contacts are strongly nonlinear, what leaves the option of time domain analysis only.

2. Model of the oil film

Clearances between dovetail groove surfaces of the slide and the saddle are adjustable and maintained at order of magnitude of 0.01 mm. With respect to the other two spatial dimensions of overlapped portion of groove surfaces, which are both at least thousand times the order of magnitude, the oil flow inside gaps can be expected laminar with negligible inertial volume forces. Such situation is ensured provided that Reynolds squeeze number is significantly smaller than one (Bilkay & Anlagan, 2004):

$$Re_S = \frac{\rho\omega\, d^2}{\mu}, \tag{1}$$

where symbols μ, ρ, ω and d mean oil viscosity, oil density, angular frequency of vibration and clearance respectively. On such condition, the pressure within the oil film satisfies Reynolds equation.

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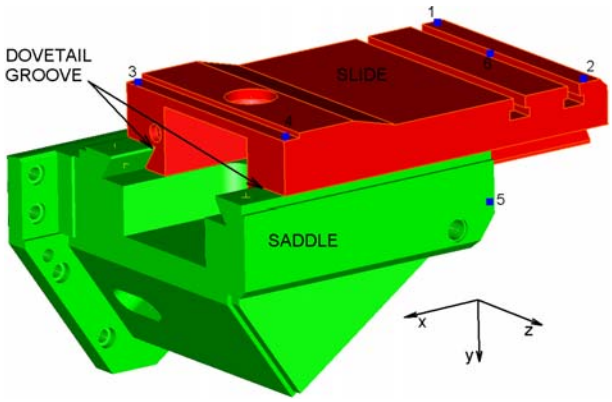

Fig. 1: Geometry of the tool rest.

Considering incompressibility of the oil, the Reynolds equation can be put into dimensionless form:

$$\frac{\partial}{\partial \xi}\left(H^3 \frac{\partial P}{\partial \xi}\right)+\frac{\partial}{\partial \eta}\left(H^3 \frac{\partial P}{\partial \eta}\right)=120\,\mu_R \frac{\partial H}{\partial \tau}, \tag{2}$$

using following quantities:

$$\xi=\frac{x}{1\cdot \mathrm{mm}},\quad \eta=\frac{y}{1\cdot \mathrm{mm}},\quad H=\frac{h}{1\cdot \mu\mathrm{m}},\quad P=\frac{p}{1\cdot 10^5\,\mathrm{Pa}},\quad \mu_R=\frac{\mu}{1\cdot \mathrm{Pa\ s}},\quad \tau=\frac{t}{1\cdot \mathrm{s}}, \tag{3}$$

where x,y are coordinates on the groove surface, p is the oil film pressure, h means oil film thickness and t is time. The equation (2) permits the oil pressure to become negative, i.e. lower than ambient pressure, when the relative motion of slide and saddle causes growth of the oil film thickness. In reality, the oil contains dissolved air, which cavitates when pressure drops slightly below atmospheric pressure. The air can also enter the volume of the oil film from the boundaries, also causing fluid film rupture. The simplest way to account for cavitation and air entrance is to restrict the oil film pressure to positive values with respect to the atmospheric pressure.

3. Numerical analysis of the oil film

The Reynolds equation (2) was solved by means of finite elements method. The mesh of the single surface of the dovetail groove was created using quadrilateral isoparametric elements; see fig. 2. The elements displayed in red belong to the oil feed groove. This groove is very deep (~1 mm) compared to the film thickness of the rest of the surface plotted in blue. The volume of the oil feed groove serves as the lubricant reservoir delivering the oil to the entire surface. With regards to the big volume of the oil feed groove, the boundary conditions prescribing ambient pressure to all feed groove nodes as well as to the surface boundary nodes were applied. The oil supplied to the tool rest is delivered periodically in small amounts of oil compared to the total volume of the feed grooves and oil ducts inside tool rest bodies.

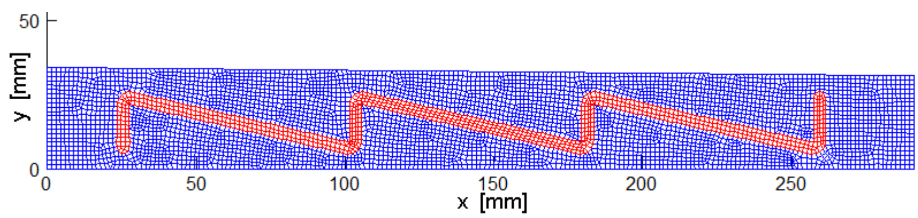

Fig. 2: Mesh of the single surface of the dovetail groove geometry.

Weak solution of the pressure is obtained by solving set of linear equations $\mathbf{K}\cdot\mathbf{P}=\mathbf{F}$, where matrix $\mathbf{K}$ and right-hand side vector $\mathbf{F}$ are composed of elemental *stiffness* matrix and elemental forcing vector calculated as

$$K_{ij}^{\{e\}}=-\int_\Omega H^3\left(\frac{\partial e_i}{\partial \xi}\frac{\partial e_j}{\partial \xi}+\frac{\partial e_i}{\partial \eta}\frac{\partial e_j}{\partial \eta}\right)d\Omega,\quad F_j^{\{e\}}=120\mu_R\int_\Omega \frac{\partial H}{\partial \tau}e_j\,d\Omega. \tag{4}$$

Example of calculated weak solution of the pressure profile is depicted in Fig. 3.

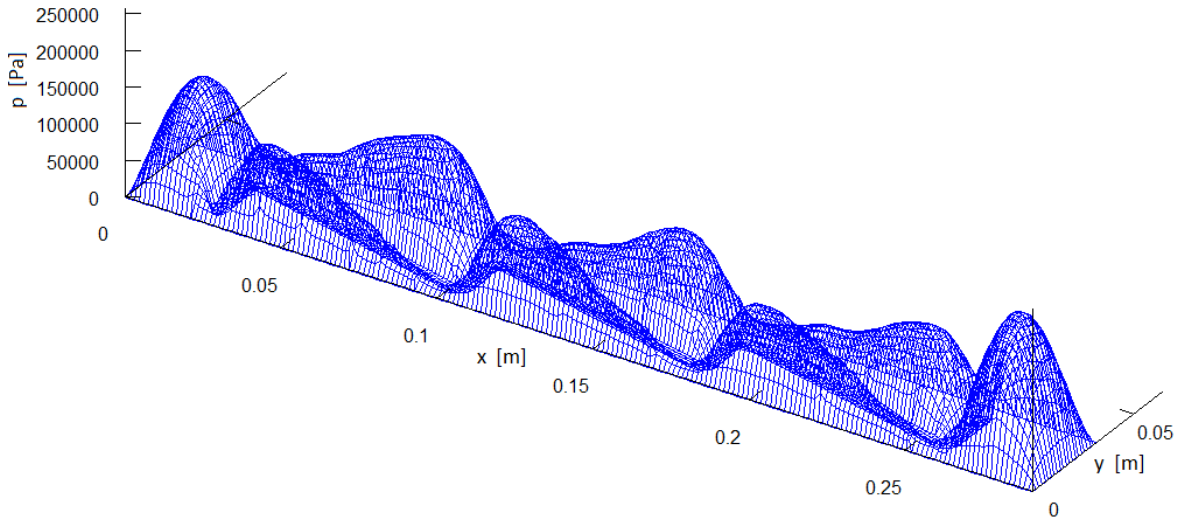

Fig. 3: Example of the weak solution of pressure profile of dovetail groove surface.

Integral of the pressure distribution over groove surface provides resultant of the hydrodynamic force, by which the slide and the saddle interact during defined relative motion. The equation (2) is linear, so the force interaction can be described by means of linear damping coefficients dependant on clearance and lubricant viscosity, provided that the amplitudes of relative motion are small with respect to the actual clearance. Damping coefficients evaluated from series of analyses are plotted in the Fig. 4.

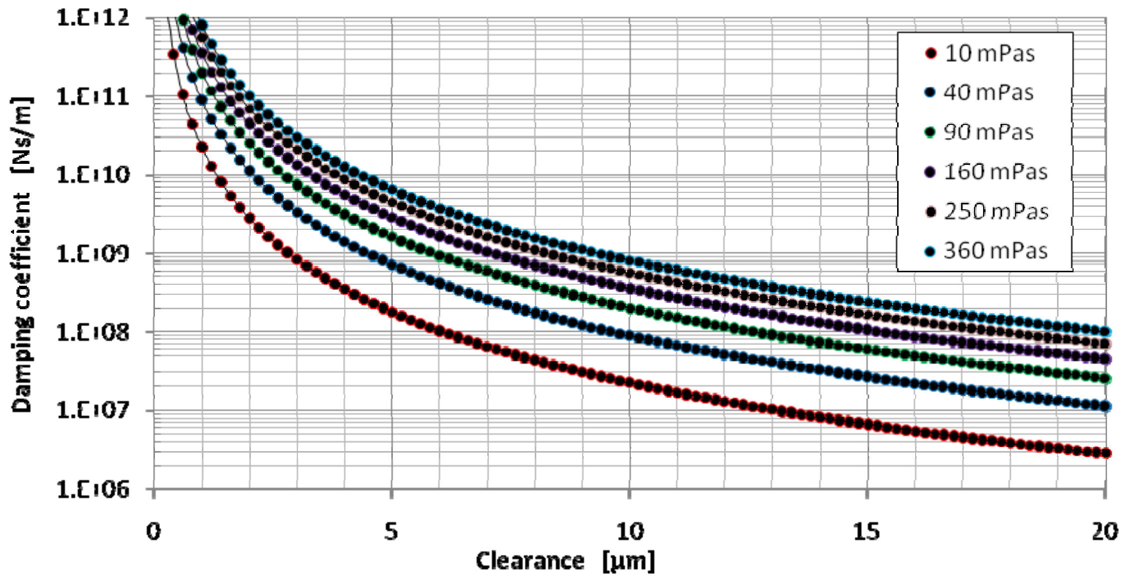

Fig. 4: Damping coefficients of single surface of dovetail groove.

4. Model of the tool rest

Model of the tool rest was built by means of Craig-Bampton substructuring technique. Normal modes along with boundary nodes were calculated from finite elements models of those parts. Boundary nodes had been chosen, such that each functional surface of dovetail groove contained 6 of them evenly distributed along the length (*x*) of the surface. Adjacent nodes of both bodies have been linked via linear dampers, whose damping coefficients have been selected according to results of HDL analysis (one twelfth of the values in the Fig. 4; Due to gaseous cavitation of the lubricant, the oil film damper works only when the oil film is being squeezed). Frequency response functions (FRF) were calculated from results of transient analysis of the model being excited by sine sweep signal at location 6 in direction y (Fig. 1).

5. Comparison to experimental data

FRF were also obtained experimentally on tool rest hardware in configuration corresponding to the analyzed case. Sine sweep, random signal and impact hammer signal were used. Some of the calculated and experimental frequency response functions are depicted in Fig. 5.

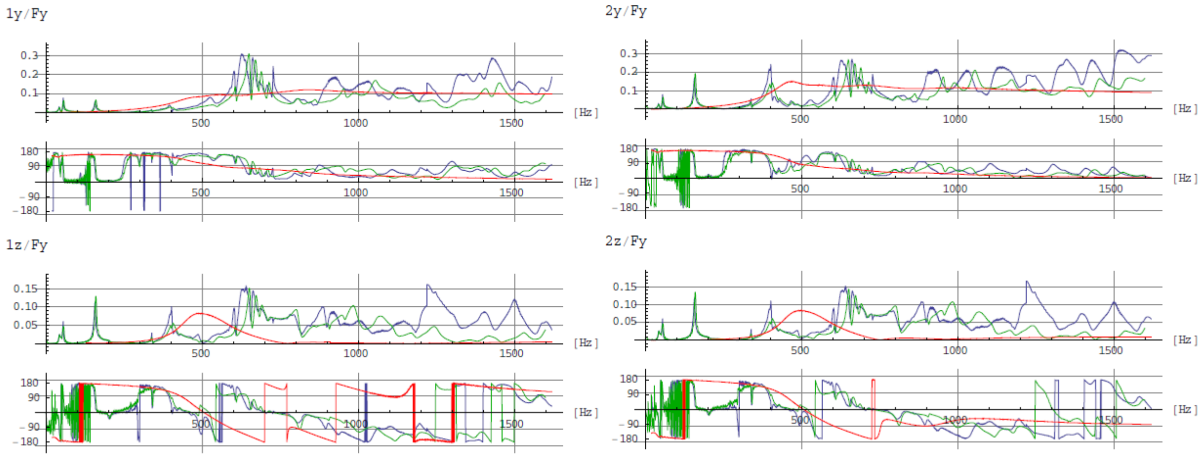

Fig. 5: Frequency response functions: Amplitudes [ms^{-2}/N] *and phases* [deg] *of inertance; Red: Numerical results (linear model), Blue: Experimental results (harmonic sweep), Green: Experimental results (impact hammer).*

The measured peaks in amplitude of inertance lower than 250 Hz have been proven to have origin in finite stiffness of saddle clamping to heavy cast iron block. In the numerical model, the saddle was clamped by rigid restraints. Amplitude peaks that occur at higher frequencies are influenced by nonlinearity of the system and its polyharmonic response to a harmonic excitation. Agreement of the experimental and calculated data is better for direction y at locations 1 and 2 than in the lateral direction z. In the former direction, the nonlinear system of tool rest is reasonably approximated by linear model. Potential cause of greater discrepancies in the latter case is the asymmetry of tested hardware, especially the presence of wedge at one side of saddle, by which the groove clearances are set up.

6. Conclusions

Analysed tool rest exhibited significant nonlinear behaviour confirmed by experiments. Presented linear model provides estimation of frequency characteristics for models of machining process for investigation of its stability formulated in the frequency domain; however the agreement of numerical and experimental results varies upon different locations and directions of the sensor. The outcomes foreshow that further improvement of the tool rest model should be made by taking into account nonlinear forces, originating in fluid film and mechanical contacts. Such complex models would not be accessible in frequency domain.

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