MACHINE TOOL VIBRATION REDUCTION USING HYDROSTATIC GUIDEWAY

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Utility properties of the machine tools, such as machining accuracy, surface quality, and productivity are affected by many factors that also include guideway properties. Guideways enable machine tool parts to move. It is generally understood that hydrostatic guideways exhibit better damping properties than linear guideways with rolling elements. However, quantitative expressions of better damping appear in the literature very sporadically. Therefore, this paper presents a model of hydrostatic guideway damping, that include squeeze damping of the thin layer of oil. Furthermore, hydrostatic and linear guideways damping properties are compared on a model example of a large machine tool. Results indicate that hydrostatic guideway reduces the forced oscillation amplitude of the first eigenfrequency 15 times in case of the modeled machine tool.

KEYWORDS

hydrostatic guideway, squeeze damping, vibrationattenuation, machine tool

1 INTRODUCTION

Utility properties of machine tools (MT), such as machining accuracy, surface quality, and productivity are also affected by damping of machine tool structure [Weck 1997]. Damping can be increased by manufacturing structural parts from cast iron or composite material [Chang 2001]. Damping improvement is also achieved by various part fillings, e.g., aluminum foam and glass balls [Sonawane 2017]. Another source of damping are guideways that movably connect machine tool parts. Linear guideways (guideways containing rolling elements) exhibit lower damping in comparison with hydrostatic (HS) guideways [Wardle 2015]. This article assesses improvement of dynamic properties of large vertical milling machine equipped with hydrostatic guideways. Forced oscillations amplitude of the ram tool center point is studied.A process of milling induces dynamic forces that lead to machine tool structure vibrations. Damping dissipates the energy of vibrations and reduces vibrations amplitude. The higher is damping the smaller vibration amplitude. This paper assesses whether hydrostatic guideways significantly reduce vibration amplitude. The paper also proposes a methodology to compare different kinds of guideways concerning damping.

2 MODEL DESCRIPTION

This chapter proposes a methodology to compare HS and linear guideways concerning damping. Furthermore, the chapter describes the damping model of HS guideways and FE model of studied machine tool ram.

2.1 Approach to guideway comparison

The operating principle of linear and HS guideways is rather different. Linear guideways make use of several rolling elements that recirculate in a guideway carriage to enable linear movement of machine parts. Rolling elements are small balls or rolls made of steel or ceramics. Rolling elements connect two sliding parts and are permanently in contact. Thus, vibrations are easily transferred thru linear guideways [Brecher 2013]. Rolling elements are elastic bodies with similar stiffness but the very low capability of damping.

The HS guideway comprises a rail (prism) and HS pocket. The pocket is shown in Fig. 1 consist of a cavity and a land. The cavity is supplied with externally pressurized oil that flows out of the cavity thru the narrow gap between the land and the rail. The pressure of oil over the pocket area provide load carrying capacity. HS pocket and the rail are permanently separated by a thin layer of oil. Sliding parts are not in contact and energy of vibrations is dissipated in the thin layer of oil. HS pocket and the opposing surface of the rail are referred to as an HS cell, and the narrow gap is also referred to as a throttling gap.

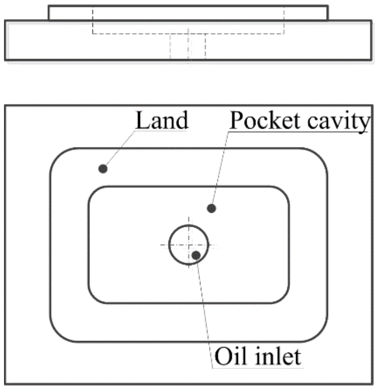

Figure 1. Hydrostatic pocket [Lazak 2016]

Next paragraph discusses significant design parameter that enables us to compare two guideway types. Operating life of linear guideways depends highly on guideway type, load, preloads, environment and lubrication and can vary largely. On the contrary, the operating life of HS guideways is almost not limited because the surfaces of rail and pocket are not in mechanical contact. Therefore, service life is not a suitable parameter for guideways comparison. Installation dimensions are not the convenient parameter since one carriage of linear guideway can carry both radial and lateral forces while one HS pocket can carry only radial force in one direction. So, the design requirement is very different and not suitable for comparing. Operation of HS guideways requires energy whereas linear guideways are passive components. The friction of HS guideways is approaching to zero at low speeds. On the other hand friction coefficient of linear guideways equal approximately $0{,}01$. Therefore, comparing guideways concerning energy is not suitable. Load carrying capacity appears to be sufficient parameter even though, load carrying capacity of linear guideways depends on service life. Stiffness is the beneficial parameter for evaluation of mathematical model results. Two guideways with the same stiffness have equal eigenfrequencies. Then resonance oscillation amplitudes can be compared and damping evaluated. Thus, it is beneficial to compare two guideways with equal stiffness and load carrying capacity and reasonable operating life.

2.2 Damping model of HS guideways

Damping of thin lands can be described by equation(1) [Rowe 2012], where dimensions of HS pocket are $a = 81\ mm, b = 81\ mm, l = 16{,}3\ mm$ and pump pressure equals $p_p = 50\ bar$. Dimensions are clear from Fig 2.

$$b_{HS} = \frac{\eta A_L l^2}{h^3} = \frac{\eta d\, l^3}{h^3} \qquad (1)$$

Where η means dynamic viscosity, A_L denotes area of land, l is the width of land and h means the thickness of oil layer. Computed damping of one HS pocket equals $5{,}610^5 \ \ Nsm^{-1}$.

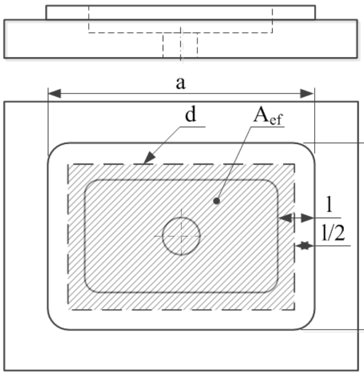

Figure 2. Dimensions of HS pocket [Lazak 2016]

To support radial loads in both directions, two HS pockets are required, and thus damping is also double.

2.3 Model of machine tool ram

The machine tool ram is three meters long with square cross-section $300 \times 300\ mm$ with wall thickness of $30\ mm$ (Fig 3). HS pockets or carriages are located at cross-slide in the distance of 800 mm. The tool is located at the lower end of the ram and, its vibrations in the direction of the Y-axis are examined. An excitation force is applied at the tool center point in the direction of Y-axis. The ram is mode modeled beams in 2D space and describes bending and axial displacement. Carriages and HS pockets are replaced by springs and dampers. A ball screw for positioning of ram is also replaced by the spring and the damper (k_3, b_3).

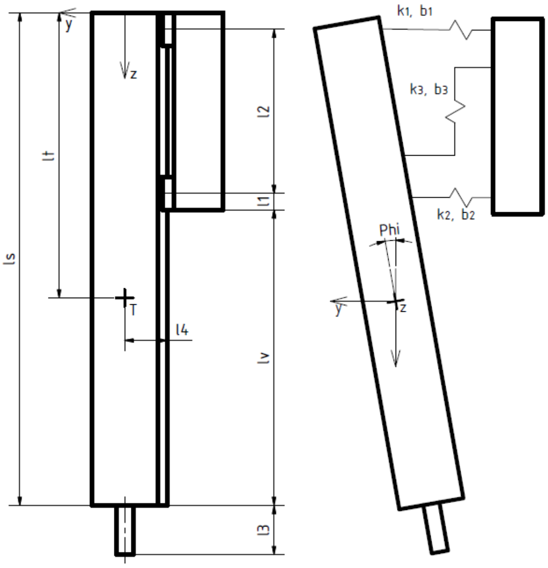

Figure 3. Model of machine tool ram

For analysis, the linear guideway is designed for machine toll ram with the service life of five years in five-day two-shift operation. The suitable linear guideway is designated BMA 30 with ball elements and preload V3 supplied by Schneeberger. Load-deformation graph of one carriage is shown inFig. 4. Derived linearized stiffness equals$640\ N/\mu m$. Damping of the linear guideway is very small and therefore the damping of the machine tool with linear guideways is assumed to be structural damping1 % [Altintas 2001]. The ram is made of steel. In the analysis, the damping is modeled as Rayleigh damping.

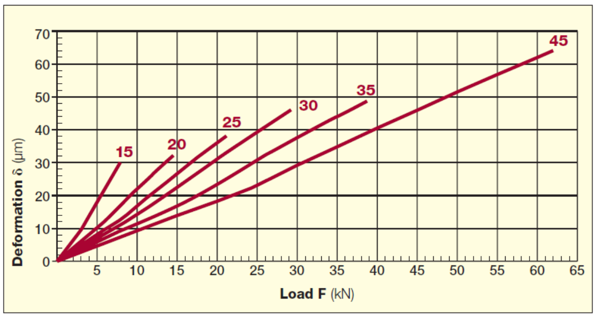

Figure 4. Load-deflection relation of guideway carriage (the red numbers indicate the size of the guideway) [Schneeberger 2017]

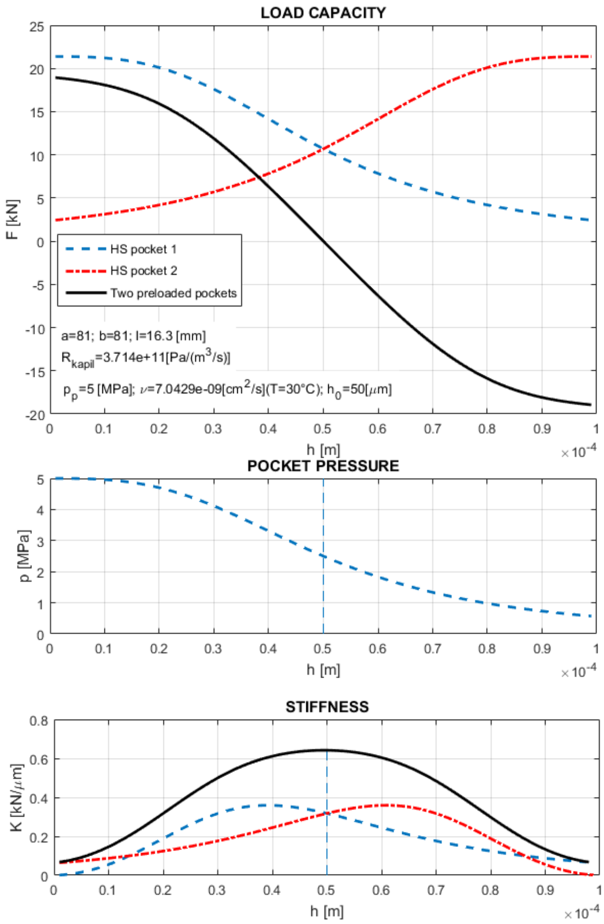

Figure 5. HS pocket parameters

The HS guideway is designed with equal stiffness and load carrying capacity as linear guideway. Thus two hydrostatic pockets stiffness equal $640\ N/\mu m$. Load-carrying capacity, pocket pressure, stiffness, oil flow and required power are shown in Fig 5.

3 CALCULATED RESULTS

Calculated transfer curve is shown in Fig 6. The curve values are divided by a value of static compliance $9{,}510^{-8}\ m/N$. Therefore, all values greater than zero indicate that dynamic deformation is greater than static deformation and vice versa.

The harmonic force is applied in the horizontal direction at the tool center point and deflection of tool center point is also calculated in horizontal direction. The deflection amplification of the first eigenfrequency is greater in the case of the linear guideway. It is assumed that phase is not essential for machining accuracy and surface quality and therefore it is not plotted.

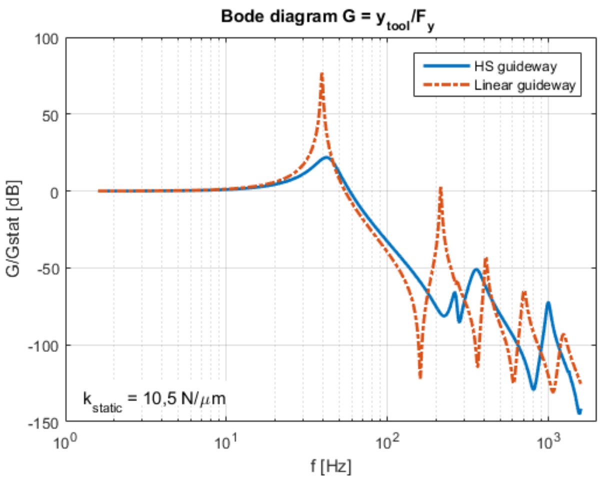

Figure 6. Transfer curve of tool center point deflection with respect to horizontal force

An amplitude of tool center point forced oscillations is depicted in Fig 7. Driving force equals $100\ N$. The amplitude of the first resonant frequency for MT with linear guideway is $442\ \mu m$whereas the amplitude of MT with HS guideway equals $28\ \mu m$. The amplitude of MT with linear guideway is 15 times higher.

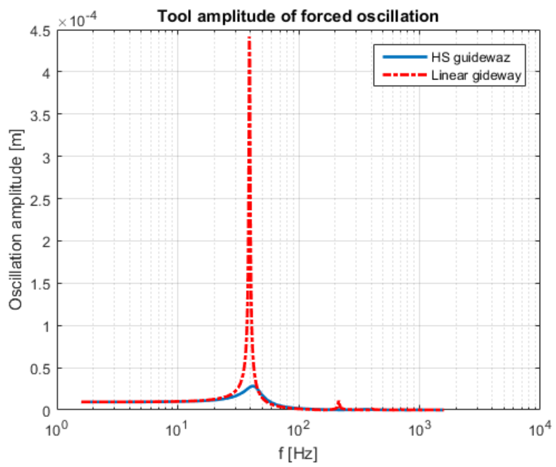

Figure 7. Amplitude of tool center point forced oscillations deflection

	HS guideway	Linear guideway	Difference
Amplitude [μm]	28	442	15,6×

Table 1. Forced oscillation amplitude of first eigenfrequency $40\ Hz$ induced by force $100\ N$

Calculated vibration amplitude of the first eigenfrequency is written in the Tab 1.

4 CONCLUSIONS

This paper compared hydrostatic and linear guideways concerning dynamic properties on the example of the large machine tool vibrations. The paper assessed the impact of higher damping of hydrostatic guideways on forced oscillation amplitude of the tool center point. The amplitude of tool center point was calculated by the FEM model of the deformable ram and stiffness and damping model of guideways. Results indicate that hydrostatic guideway reduced the forced oscillation amplitude of the first eigenfrequency times.

For future work, calculated transfer functions can be used for estimating limit chip thickness. Then in general for assessing whether it is beneficial to use hydrostatic guideway instead of the linear guideway. It is also planned to verify the dynamic model of hydrostatic guideways experimentally.

ACKNOWLEDGMENTS

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