Effect of Hydrostatic Guideway on Reduction of Machine Tool Ram Vibration

Lazák Tomáš, Stach Eduard, Sulitka Matěj

CTU in Prague, Faculty of Mechanical Engineering, Department of Production Machines and Equipment, Technická 4, 166 07 Prague 6, Czech Republic

Abstract

The guideways provide translational movement of machine parts and have a major impact on the resulting utility properties of the machine tools, such as machining accuracy, surface quality and productivity. It is generally understood that hydrostatic guideways have better damping properties than linear guideways with rolling elements. However, quantitative expressions of better damping appear in the literature very sporadically. Therefore, this paper aims to compare hydrostatic and linear guideways and to assess the impact of higher damping on a ram vibrations of a large machine tool. The forced oscillations amplitude of the ram 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 reduce the forced oscillation amplitude of the first eigenfrequency 15 times in case of the modeled machine tool.

Keywords: hydrostatic guideway; vibrations; damping; 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 [1]. Damping can be increased by manufacturing structural parts from cast iron or composite material [2]. Damping improvement is also achieved by various part fillings, e.g. aluminum foam and glass balls [3]. Another source of damping are guideways that moveably connect machine tool parts. Linear guideways (guideways containing rolling elements) exhibit lower damping in comparison with hydrostatic (HS) guideways [4]. 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 induce dynamic forces that lead to machine tool structure vibrations. Damping dissipate energy of vibrations and reduce vibrations amplitude. The higher is damping the smaller is vibration amplitude. This paper assesses whether hydrostatic guideways significantly reduce vibration amplitude. The paper also propose a methodology to compare different kinds of guideways with respect to damping.

2. Model description

This chapter propose a methodology to compare HS and linear guideways with respect to damping. Furthermore, the chapter describe damping model of HS guideways and FE model of studied machine tool ram.

2.1. Guideways comparison approach

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 [5]. Rolling elements are elastic bodies with corresponding stiffness but very low capability of damping.

The HS guideway comprise a rail (prism) and HS pocket. The pocket 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 thro narrow gap between the land and the rail. 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 opposing surface of the rail are referred to as a HS cell and narrow gap is also referred to as a throttling gap.

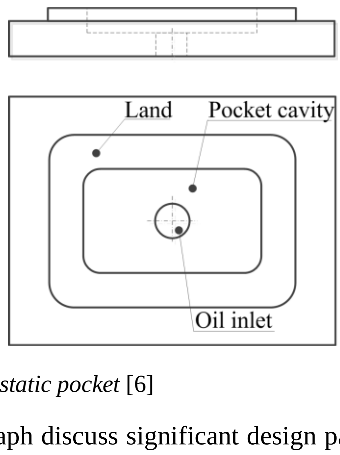

Fig. 1. *Hydrostatic pocket* [6]

Next paragraph discuss significant design parameter that enable 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 contrary operating life of HS

* Corresponding author: tomas.lazak@fs.cvut.cz

guideways is almost not limited since the surfaces of rail and pocket are not in mechanical contact. Therefore, service life is not suitable parameter for guideways comparison. Installation dimensions are not 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, design requirement are very different and not suitable for comparing. Operation of HS guideways require energy whereas linear guideways are passive components. 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 with respect to 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 beneficial parameter for evaluation of mathematical model results. For two guideways with the same stiffness are their eigenfrequencies equal. 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) [7], 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.

$$b_{\mathrm{HS}} = \frac{\eta A_L l^2}{h^3} = \frac{\eta d l^3}{h^3} \tag{1}$$

Computed damping of one HS pocket is $5{,}6 \cdot 10^5\ Nsm^{-1}$.

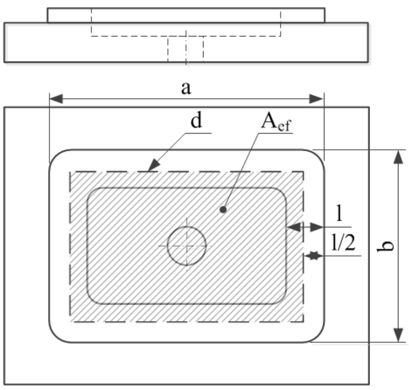

Fig. 2. *Dimensions of HS pocket* [6]

In order 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 Y axis are examined. An excitation force is applied at the tool in the direction of Y axis. The ram is modeled of beams in 2D space and describes bending and axial displacement. The ram is made of steel and thus its structural damping is assumed to be $0{,}5\ \%$ [8]. In analysis, the damping is modeled as Rayleigh damping. 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).

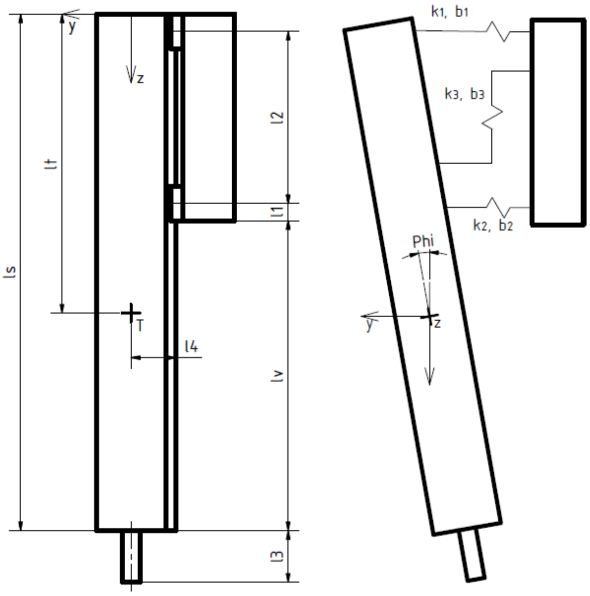

Fig. 3. *Model of machine tool ram*

For purpose of analysis the linear guideway is designed for machine toll ram with service life of five years in five-day two-shift operation. Suitable linear guideway is designated BMA 30 with ball elements and preload V3 supplied by Schneeberger. Load-deformation graph of one carriage is shown in Fig. 4. Derived linearized stiffness equals $640\ N/\mu m$. Damping of linear guideway is very small and therefore it is modeled as structural damping 1 % [8].

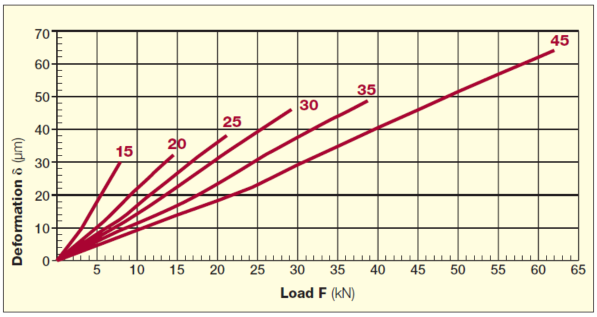

Fig. 4: Load-deflection relation of guideway carriage [9]

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.

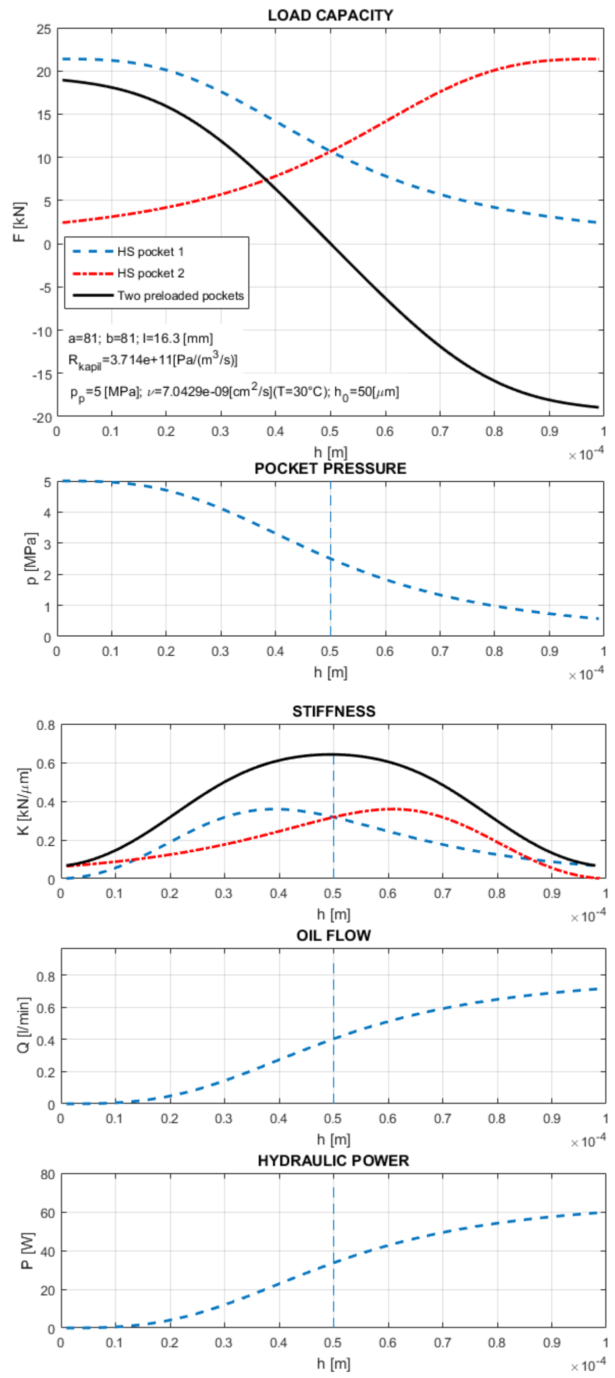

Fig. 5. *HS pocket parameters*

Designed throttling gap height equals 50 μm. Regulation of throttling gap height is performed by capillary regulator.

3. Calculated results

Calculated transfer curve is shown in Fig. 6. The curve values are divided by a value of static compliance $9{,}5 \cdot 10^{-8}\ m/N$. 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 calculated in horizontal direction. The deflection amplification of first eigenfrequency is greater in case of linear guideway. It is assumed that phase is not important for machining accuracy and surface quality and therefore it is not plotted.

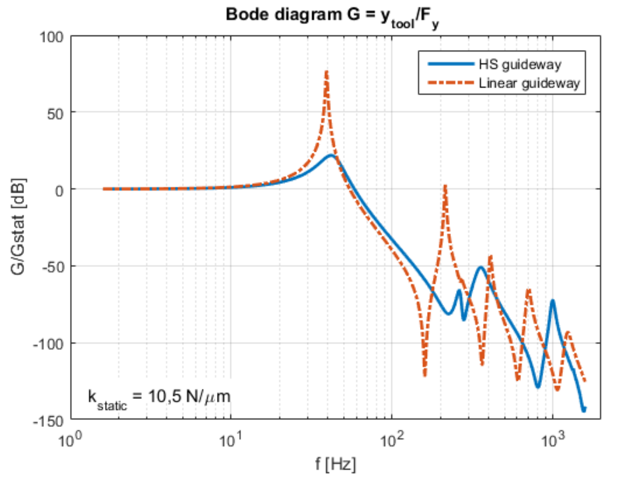

Fig. 6. *Transfer curve of tool center point deflection with respect to horizontal force*

Amplitude of tool center point forced oscillations is depicted in Fig. 7. Driving force equals 1000 N. The amplitude of the first resonant frequency for MT with linear guideway is 4417 μm whereas the amplitude of MT with HS guideway equals 282 μm. The amplitude of MT with linear guideway is 15 times higher.

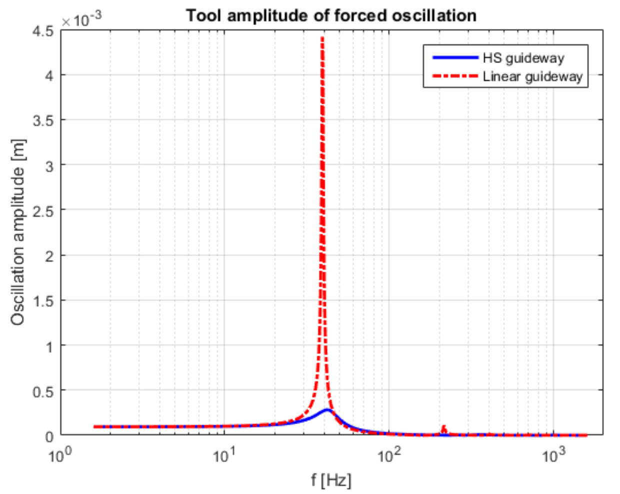

Fig. 7. *Amplitude of tool center point forced oscillations*

Calculated results are written in the Table 1.

Table 1. *Forced oscillation amplitude of first eigenfrequency 40 Hz induced by force 1000 N*

	HS guideway	**Linear guideway**	**Difference**
Amplitude [μm]	282	4417	15.6 x

4. Conclusion

This paper compared hydrostatic and linear guideways with respect to 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 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 15 times.

For a 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 linear guideway. It is also planned to experimentally verify dynamic model of hydrostatic guideways.

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Symbols

a	horizontal dimension of HS pocket (m)
A_L	land area (m^2)
b	vertical dimension of HS pocket (m)
b_i	damping of i-th carriage or HS pocket $(N \cdot s \cdot m^{-1})$
b_{HS}	damping of HS pocket $(N \cdot s \cdot m^{-1})$
d	circumference of effective area (m)
f	frequency (Hz)
G	transfer function of dynamic compliance
F	force (N)
h	throttling gap height (m)
k_i	stiffness (N/m)
k_{static}	static stiffness (N/m)
l	land (sill) (m)
l_i	dimensions of machine tool ram (m)
p	pressure (Pa)
p_p	pump pressure (Pa)
P	Power (W)
Q	oil flow $(m^3 \cdot s^{-1})$
y_{Tool}	Tool center point deflection
δ	deformation (m)
η	dynamic viscosity (Pa·s)

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