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Damping in ram based vertical lathes and portal machines

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ABSTRACT

Chatter vibrations originated by the machine structure are a major limitation for the productivity of ram based machines performing heavy duty operations. Consequently, the damping of the machine structure has a capital importance. It is known that interfaces and guideways are the main origin of damping. Recently, the use of active dampers has been introduced in industry. In this work, the damping of hydrostatic and rolling guideways with and without active damping has been experimentally identified and compared using receptance coupling. The results show that hydrostatic guidance can introduce 3–4 times more damping than a roller based system. However, the introduction of active damping is a game changer enhancing damping more than 30 times.

1. Introduction

Ram based vertical lathes and portal machines are widely used in the production of large parts for energy and aeronautics sectors [1]. The use of rams provides accessibility for internal and external operations at expense of reducing stiffness. Therefore, heavy duty operations are limited by chatter vibrations, and damping becomes a key factor in machine's performance [2].

The major part of damping occurs on machine interfaces. More specifically, damping of assembled structures can be about 30 times higher than the damping of individual components [3]. Concerning ram based machines, vibrations are usually related to low frequency (<200 Hz) ram bending modes [4], where the guideways play an important role. Traditionally, sliding contact and hydrostatic guideways with high damping were used. The recent search for higher precision excluded the use of sliding contact guides, and rolling element contact guides have gained popularity due to their high performance and modular integration [5]. Brecher et al. [6] experimentally modeled linear rollers and concluded that the increase of preload reduces damping. Semm et al. [7] included dynamic characteristics of the roller guides in FEM models for dynamic simulation.

However, the low damping properties offered by roller guides make some machine tool builders prone to keep hydrostatic guides in large-scale machines, despite their higher production, assembly and maintenance cost [4]. The damping of hydrostatic guideways highly depends on the viscosity of the fluid, the temperature, the chamber geometry and the gap of the interface [8]. Lazak et al. [9] concluded, thanks to a theoretical study, that the hydrostatic guidance can increase the dynamic stiffness more than 15 times. Other authors have limited the magnitude of this improvement measuring experimentally the receptance of ram based moving column machines with different guidance systems and obtaining theoretical stability lobes [10].

Alternatively, damping can be increased integrating specific damping devices [2]. A traditional method is to locate a tuned mass damper in the structure [11]. In this case, passive absorbers are generally too bulky to use and not always robust enough to handle dynamic variations of ram based machines [12]. Active dampers applied in machining centres [14] and ram based moving column machines [12] were proposed to overcome these limitations [13].The potential of active solutions has been experimentally proven, doubling the material removal rate in many applications [12]. This concept is already on the market, increasing the damping offered by rolling guides [10].

However, a strict and fair comparison on the damping provided by different configurations is missing in the literature. Therefore, the objective of the present paper is to dynamically compare linear roller guides, hydrostatic guidance and active damping solutions. For that purpose, two twin test benches with different guiding systems and with the possibility to add active damping have been developed. Later on, the receptance coupling substructure analysis (RCSA) has been adapted to extract the stiffness and damping values of the different guidance systems. Finally, damping and cutting capabilities have been experimentally evaluated.

2. Design of demonstrators

In ram based portal machines and vertical lathes, in the most flexible positions with large overhang, the critical modes are directly

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related to the bending and rotation of the ram, where the ram guideway's dynamic properties have direct influence. This is not the case in ram type traveling column machines where other machine elements are also involved in critical modes [12].

Therefore, two demonstrators have been designed to perform a trustable quantitative evaluation of the current design solutions. The demonstrators consist in a ram embraced by a saddle and driven by a ball screw. It can be located in the bed of any machine tool to perform turning operations by rotating the part using the main spindle of the machine. Following this procedure, two guiding technologies have been analyzed: recirculating roller guides and hydrostatic guides. Moreover, active damping has been added to the roller guiding solution as the third configuration design for portal machine rams (see Fig. 1c).

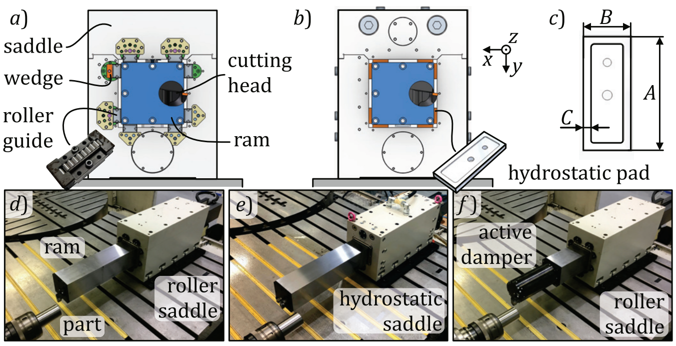

Fig. 1. Roller guides based ram (a), hydrostatic guides based ram (b), and geometry of hydrostatic pads (c). Demonstrators mounted in machine bed: roller ram (d), hydrostatic ram (e), roller ram + active damper at the ram tip (f).

2.1. Requirements

Five main requirements have been defined to allow a fair comparison of the guiding systems: Firstly, the *geometries of the ram and saddle must be equal* in the demonstrators, including the connection points between them. Both demonstrators include a 100 × 100 mm cross section prismatic ram, held in a saddle in 2 planes, with 8 pads in each plane, with a maximum overhang of 700 mm (Fig. 1a,b). Standard recirculating roller guides have been used, with their rolling race on the ram surface itself.

Damping only affects resonances and therefore it is important to *assure demonstrators with similar natural frequencies*. FEM models have been used to predict a natural frequency of 235.8 Hz for a 500 mm overhang. For the hydrostatic testbench, pockets have been designed to assure equal stiffness as roller guides. Thus, the damping is the only difference between them.

The bending mode of the ram *should not be dynamically coupled* with any mode coming from the supporting bed. This effect is positive for machining, because it increases the overall damping, but it can distort the comparison. A milling machine bed has been selected as support structure after an experimental study where only modes below 200 Hz were detected. The possibility to avoid coupling with overhangs below 500 mm was predicted. The low influence of the milling head/part side was confirmed too.

During the cutting tests, the *absence of process damping* is mandatory, as its effect may masque the damping coming from the guiding systems. Therefore, the machining tests must be carried out at process damping free lobe numbers (<10) [2]. This requirement is fulfilled by the proper selection of the relation of part's material and diameter, the spindle speeds and the natural frequency of the main mode. Cylindrical workpieces (C45K) have been clamped in a 50 mm HSK100 mechanical power chuck, using the machine's milling head as a lathe spindle.

Additionally, the insert geometry and position have been selected to minimize the effect of the Y axis in the regenerative effect. The chip thickness direction lies in the XZ plane where the *X direction is predominant*.

2.2. Roller guided ram

IKO RWB14 recirculating rolling element linear guides have been used, which offer a theoretical stiffness around 450 N/µm. A screw driven wedge mechanism is used to control the preload of the rollers, which lay over a cylindrical part housed in the saddle to avoid misalignments between the ram and the rollers. Once the preload is applied in the assembly, the motion of beforementioned parts is restricted by bolted joints.

2.3. Hydrostatic guided ram

Hydrostatic pads have been designed to offer the same stiffness as the recirculating rollers. This design criteria has given as a result A = 60 mm, B = 25 mm and C = 4 mm pockets (see Fig. 1b), which have been manufactured in Biplast-V® material and glued to the saddle through an adhesive. Once manufactured and measured the actual dimensions of the ram, the pads have been ground to ensure a 0.02 mm gap in each side of the ram. ISO VG68 oil was used at 100 bar line pressure, and an oil cooling unit maintained the oil at 25 °C. The inlet and outlet oil temperature were measured to ensure constant temperature during the tests.

2.4. Roller guided ram enhanced by active damping

A special head with an integrated electromagnetic inertial active damper (⌀100 mm × 176 mm) with an inertial mass of 4.5 kg has been attached to the free end of the roller ram, acting in parallel to the cutting force path. With this configuration, the actuator does not exceed the external dimensions of the ram, allowing full operability. In this way, for the same total overhang L_{OH}, the damper substitutes part of the ram. As the mass of the active damper's casing is similar to the end of the ram that is substituted, a small variation of the natural frequency is observed. This actuator can introduce forces up to 30 N in a 50–400 Hz frequency bandwidth, hence maintaining the scalability of the demonstrator with respect to current industrial solutions. Concerning active control strategy, direct velocity feedback (DVF) algorithm has been applied with high and low pass filters of 100 Hz and 650 Hz, respectively, and a control gain of 740 Vs/m.

3. Damping characterisation model (RCSA)

In order to characterize the damping provided by the different ram configurations, a simple mathematical model has been developed (Fig. 2) to extract the dynamic parameters of the guideways based on tool point experimental receptances.

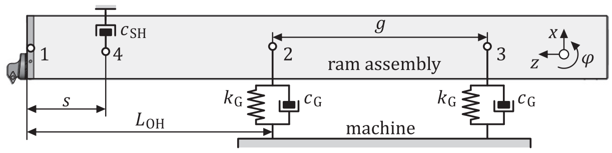

Fig. 2. The parametric model of the ram.

In this tailored measurement setup, the guiding systems are assumed to be completely linear and identical. The eight guide pads (4 in tensile/compressive loads and 4 in shear loads) in each side plane are modeled as a single spring and damper, with stiffness k_G and damping c_G (Fig. 2). These support elements are holding the ram in place through m: = {2, 3} points, while the response is measured and predicted in n: = {1} point.

Assuming an ideally stiff ground (machine bed) the response of the assembled ram structure can be determined by using the multipoint RCSA methodology [15].

$$\mathbf{H}_{n,n} = \mathbf{R}_{n,n} - \mathbf{R}_{n,m}\left(\mathbf{K}_{m,m}\ \mathbf{R}_{m,m} + \mathbf{I}\right)^{-1}\mathbf{K}_{m,m}\ \mathbf{R}_{m,n}. \tag{1}$$

The matrix valued $\mathbf{H}(\omega)$ and $\mathbf{R}(\omega)$ receptance FRFs describe the assembled and the free (constrain-less) dynamics of the ram, respectively, with the following block definitions

$$\mathbf{R}_{\alpha,\beta}(\omega) := \left[R_{ia,jb}(\omega)\right]_{i,j\in\alpha,\beta;\ a,b\in X}, \mathbf{H}_{n,n}(\omega) := \left[H_{na,nb}(\omega)\right]_{a,b\in X}. \tag{2}$$

for all combination of receptances of transversal x and angular displacements φ, where X:={x, φ}.

Therefore, the translational response $H_{1x,1x}$ at the end of the ram is one of the elements of the 2×2 $\mathbf{H}_{n,n}$, while the support is established by the 2×4 $\mathbf{R}_{n,m}$ and 4×4 $\mathbf{R}_{m,m}$ through the contact stiffness function defined uniformly as 4×4 $\mathbf{K}_{m,m}$. $\mathbf{R}_{m,m}$ includes the 2×2 free receptances of contact points 2 and 3 ($\mathbf{R}_{2,2}$, $\mathbf{R}_{3,3}$) and the crossed submatrix ($\mathbf{R}_{2,3} = \mathbf{R}_{3,2}$). The dynamic stiffness function is a diagonal matrix $\mathbf{K}_{m,m}(\omega) := \boldsymbol{\kappa}(\omega) = \mathrm{blkdiag}_{i \in m}[\mathrm{diag}[K_G(\omega)\ \ 0]]$ where the first and third element of the diagonal are a complex function $K_G(\omega) := k_G + c_G\,\mathrm{i}\omega$.

The transversal free (unconstraint) dynamics ($\mathbf{R}_{\alpha,\beta}$ in (2)) of the ram assembly are described by using the Timoshenko-beam model for the monolithic ram and a rigid mass for the head inertia also connected by RCSA to have an agile enough mechanical representation for parameter optimization.

Instead of using inverse receptance coupling [15], the present paper proposes to define an error between measured $\Phi_{1x,1x}(\omega)$ and the modeled FRFs $H_{1x,1x}(\omega; k_G, c_G)$ for different overhang (OH) values L_{OH} (see Fig. 2) as

$$\epsilon(k_G, c_G) := \int_{\omega_{\min}}^{\omega_{\max}} \overline{\Delta H}(\omega; k_G, c_G)\Delta H(\omega; k_G, c_G)\mathrm{d}\omega \tag{3}$$

where $[\omega_{\min}, \omega_{\max}]$ is the bandwidth of interest and $\Delta H(\omega; k_G, c_G) := \Phi_{1x,1x}(\omega) - H_{1x,1x}(\omega; k_G, c_G)$. In this way, a simple two-parameter but nonlinear optimisation can be performed on the real valued error $\epsilon(k_G, c_G)$ for each L_{OH}, extracting k_G and especially c_G values for different guiding system configurations.

In order to mimic the dynamic effect of an active damper located inside the ram, a skyhook damper is defined at point 4, since DVF control strategy introduces the force as viscous damping [12,14]. Point 4 is selected as the centre of mass of the active damper, which is parametrized by a constant distance s. The skyhook damper is connected to the ram according to (1) with m:={2, 3, 4}. In this case, a 6×6 extended stiffness function is considered as $\mathbf{K}_{m,m}(\omega) := \mathrm{blkdiag}\ [\boldsymbol{\kappa}(\omega)\ \ \mathrm{diag}[c_{SH}\mathrm{i}\omega\ \ 0]]$. Since the roller pad properties (k_G, c_G) can be previously determined when the actuator is deactivated, a single parameter optimization for the additional skyhook damping c_{SH} is needed similarly to (3).

4. Experimental damping measurement

The objective of this section is to characterize the damping provided by each ram configuration from experimental measurements. For this characterization, the demonstrators have been placed on the selected milling machine and 3 overhangs have been chosen (450, 475 and 500 mm). Receptances at these L_{OH} are not influenced by other modes coming from the supporting structure (Fig. 3). However, the effect of mode coupling is clearly seen in most of the other overhangs. The compliance of those modes does not increase with the overhang, showing higher damping values when it occurs (Fig. 3).

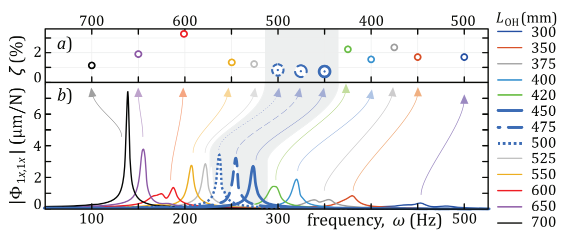

Fig. 3. Experimental FRFs obtained for roller guides at all possible overhangs (b); fitted damping values vs overhangs (a).

4.1. Receptance measurements (FRFs)

The easiest way to extract damping information of each ram configuration is by means of receptance measurements. For that purpose, direct FRFs at the end of the ram have been measured in the three uncoupled overhangs by means of a dynamometric hammer and an accelerometer. Fig. 4 shows the receptance comparison between different ram configurations.

While natural frequencies ω_n remain similar, receptance amplitudes clearly reflect damping differences between different ram configurations. The relative damping ζ values extracted from these responses by means of a curve fitting method are shown in Table 1. The results show that although relative dampings of the critical modes on the cutting point are increased more than 3 times when hydrostatic guides are used, the introduction of active damping can almost flatten the receptance curve by multiplying relative dampings up to 30 times.

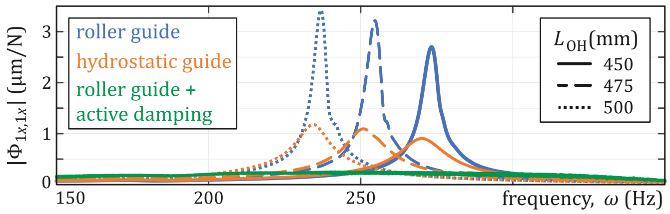

Fig. 4. Experimental FRFs obtained for different overhangs and ram configurations.

Table 1
Experimental relative damping ζ and estimated k and c values for different ram configurations and overhangs.

Ram configuration	L_{OH} (mm)	General		Individual elements		
		ω_n (Hz)	ζ (%)	k_G (N/μm)	c_G (Ns/mm)	c_{SH} (Ns/mm)
Roller guides based ram	450	273.2	0.7	489.6	14.8	–
	475	254.3	0.7	514.5	18.4	–
	500	236.8	0.7	530.0	23.9	–
Hydrostatic guides based ram	450	270.6	2.4	447.3	38.8	–
	475	251.8	2.4	455.5	45.6	–
	500	234.5	2.5	472.7	59.3	–
Roller based ram + active damping	450	267.5	22.8	421.8	11.9	4.8
	475	252.7	20.5	466.7	23.3	4.6
	500	236.5	17.7	463.1	32.8	4.8

4.2. Damping estimation by RCSA

The relative damping values obtained from receptances are referred to general dynamic parameters of the complete system at the tool tip. The stiffness and damping provided by each element (guides, active damper) are calculated by using the method described in Section 3.

Considering distances g:=450 mm and s:=107 mm, the stiffness k_G and damping c_G values of the guide pads, as well as the damping introduced by the active damper c_{SH} have been estimated (Table 1). Concerning stiffness values, a similar k_G (a difference lower than 10%) has been estimated for the three different configurations, which was one of the requirements for the demonstrators. If the guide pad dampings are compared, it can be clearly observed that the c_G value for each plane of the hydrostatic guides is around 2.5 times bigger than that provided by roller guides.

When these c_G values are compared to the skyhook damping supplied by the active damping system, the relatively low values of c_{SH} stand out. However, it must be considered that the damping application point is different for the guides and the damper. Indeed, active damping is introduced in a high displacement point whose modal vector is around 10 times larger than the one at the guiding pads location. Therefore, in absolute terms, it is more efficient to introduce damping devices in locations with high modal displacements. In most cases, this is only possible including an additional inertial mass.

5. Cutting tests

Damping is a key parameter in the search for chatter stability. The objective of this section is to validate the damping comparisons obtained by dynamic response measurements with cutting tests. As explained in Section 2, a milling machine head has been used as a lathe spindle by clamping the workpiece in a mechanical power chuck. In the performed cutting tests, the spindle head is also responsible for the cutting feed while the ram overhang L_{OH} remains constant. This way, the dynamic response of the ram is not changed during the cuts and the cutting stability results can be compared with the damping values obtained from the experimental FRFs (Section 4). A cutting tool with high radial load has been selected to excite the horizontal bending mode. Cutting conditions are summarized in Table 2.

Table 2
Cutting tool details and cutting conditions.

Head reference	Sandvik 570-SDXCR-40–11
Insert reference	Sandvik DCMT 11 T3 04-PF 4425
Nose radius	0.4 mm
Lead angle	62.5°
Workpiece diameter and material	35 mm (C45k)
Feed per revolution	0.15 mm/rev
Spindle speeds (N)	2700, 2850 and 3000 rpm

Three different spindle speeds have been selected in order to cover the stability lobe effect, so at least one of them is close to a critical spindle speed for each overhang and ram configuration. In this way, the minimum depth of cut a_p has been measured, as shown in Fig. 5a. It can be observed that the minimum a_p of roller guided ram is below 0.2 mm, since stable cutting was not possible at certain spindle speeds. Meanwhile, the minimum a_p for hydrostatically guided ram was measured at 0.4–0.6 mm, which proves the improvement provided by the hydrostatic guides. However, this improvement is substantially overcome when active damping system is added to the roller guided ram, which permits chatter free cutting for all analyzed conditions (up to 4 mm a_p). Since damping equally affects all the stability lobes, the results can be extrapolated to the other stability lobes as far as process damping is not involved. Fig. 5b,c compares the vibration level in time and frequency domains at certain cutting conditions (L_{OH} = 500 mm, N = 3000 rpm, a_p=0.4 mm). It shows that huge chatter appears for both guiding systems, although the vibration regeneration is much slower in case of hydrostatically guided ram. Chatter vibration is completely removed when active damping is used.

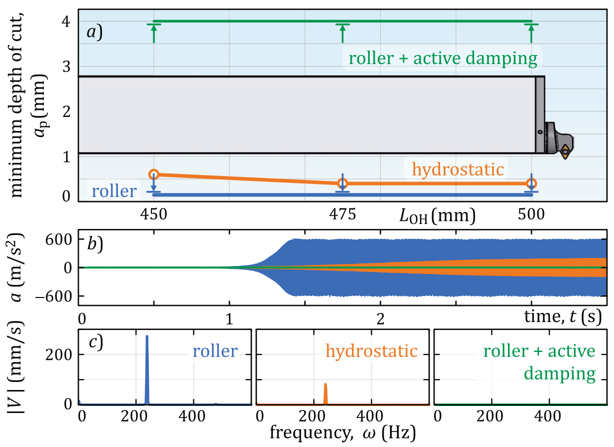

Fig. 5. Minimum depth of cut for each ram configuration and overhang (a); Acceleration time signal (b) and vibration spectrum (c) during cutting operation (L_{OH}=500 mm, N = 3000 rpm, a_p=0.4 mm).

It is noteworthy that, since the selected spindle speeds are located in lobe number 6, the lobe effect clearly appears and the stability can be improved by tuning the spindle speed.

6. Conclusion

Chatter vibrations are one of the main limitations of the performance of heavy duty operations in ram based vertical lathes and portal machines. In this kind of machines, the critical modes are completely characterized by ram bending modes, wherein guiding systems play an important role. This paper presents a fair quantitative comparison of the damping provided by different solutions proposed for ram based portal machines by means of 3 demonstrators: a roller guided ram, a hydrostatic ram and a roller guided ram enhanced by an active damping system.

The experimental dynamic response measurements show that the damping at the cutting point is 3–4 times larger when hydrostatic guides are used instead of roller guides. A similar conclusion is obtained when the damping coefficient value is estimated by means of RCSA method. However, the guiding system influence is no longer significant when the active damping system is introduced, since the damping is applied in parallel into a point with a high vibration displacement. It leads to an increase of around 30 times the damping on the cutting point. These results are confirmed by cutting tests where the stability margin is increased accordingly to the damping provided by each solution. In this way, the hydrostatic guided ram shows a quite higher stability than roller guided ram. Meanwhile, active damping solution removes chatter in all studied conditions.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Acknowledgements

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