Design of two-axis controlled damper to suppress vibration of long slim rams

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Abstract

The contribution deals with the design of two-axis controlled damper to suppress vibration of long slim rams of larger cross-section. Proposed concept is based on a demand to damp down lowest (bending) eigenmodes of the ram in the X-Y plane. These dominant eigenmodes influence the power transferred to the transfer to the cutting process. Supposed position of the vibration damper body is close to the machining point. The proposed solution of the vibration damper is unique, as it is two-axis version with relatively high action force.

Keywords: damper, active damper, active vibration damper, vibrations

1. Introduction

The main goal of users of machine tools is to produce an accurate workpiece with quality surfaces in the shortest possible time that are contradictory requirements. These requirements put high demands on dynamics and accuracy of single axis' drives of the machine tool and at the same time on construction of a machine frame, mainly on its modal characteristics. One of issues related to this is ram vibrations. Rams have usually small cross-section and long overhang, mainly while machining holes, which may cause an instability while machining. Rams are generally compliant in X and Y axis direction and very stiff in Z (longitudinal) axis direction. Excited vibrations of the machine or its parts may influence accuracy and surface quality of the workpiece. Representative examples of machines with long slim rams are vertical lathes.

Aim of the work is to design an active vibration damper to reduce dynamic pliability of the ram in X and Y axis, to improve damping in these axis direction [1], that will enable to increase the power transferable to the cutting process.

2. Possibilities of vibration damping

2.1. Basic damper divide

Dampers could be divided according to the position:

- Damping on the workpiece side
- Damping on the tool side

This work is focused on the design of the damper on the tool side.

Another way to sort dampers is considering their manageability:

- Passive dampers
- Semi-active dampers
- Active dampers

1.1.1. Passive dampers

Passive dampers are the most common and the best-known method of vibration damping. Principle of passive damping is based on the external additional structure connected to the primary damped system to make the system unresponsive to supposed external excitation. Passive damper consists of additional mass, connected to the primary system via spring element and damping element. (Fig. 1)

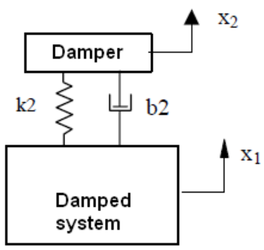

Fig. 1. *Basic scheme of passive damper [2].*

Natural frequency is possible to be adjusted by the appropriate combination of mechanical parameters: mass, rigidity and damping of the additional element. If the natural frequency of the damper corresponds to the natural frequency of the system, the system is damped, as the mass of the damper takes over a part of the vibration energy of the system. Main disadvantages of the passive damper are narrow efficient frequency range and big additional mass [2]. Advantages comparing to other damper applications are easy construction and reliability. Passive dampers are suitable mainly for systems with consistent excitation frequency. Passive dynamic damper was patented in 1911 by H. Frahnem. [3, 4, 5, 6]

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1.1.2. Semi-active dampers

Semi-active damper is a combination of passive and active damper. Passive element is replaced by a controlled element. Unlike an active damper application, energy is got out of the primary system and only a capacity of a damping power is controlled. (Fig. 2)

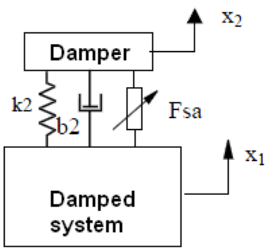

Fig. 2. *Basic scheme of semi-active damper [2].*

Unlike the passive damping, semi-active dampers enable damping constant to be changed fluently. Comparing to the passive dampers, they provide far better damping effect but more complicated construction. Comparing to the active damper, advantages are stability of the system and lower energy demands. Despite they do not achieve same efficiency as active damper applications, they are widely applied due to favourable efficiency to energy consumption ratio [2]. Construction of semi-active dampers is based on various principles, for instance hydraulic damper containing liquid reacting to the magnetic field, hydraulic damper controlled by valves, or damper with linear electric motors. [1, 5, 7]

1.1.3. Active dampers

Active dampers eliminate main disadvantage of passive and semi-active dampers, which is their sensitivity to accurate adjustment of mechanical parameters of the system. Passive element between the system and the damper is replaced by an active one. (Fig. 3)

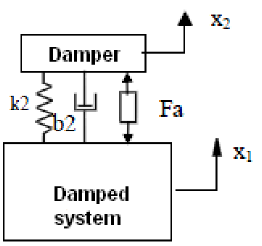

Fig. 3. *Basic scheme of active dynamic damper [2]*

Active element of the damper enables external energy to be brought into the system. Source of the external energy could be e.g. electric, magnetic, hydraulic or pneumatic actuator. Due to this, active damper is able to work in a wide frequency range. Efficiency of the active damper is limited by power, mechanical and energetic limits of the active element. The main disadvantage of active damper application is possible destabilisation of the system in case of inappropriate control. Such a system has worse dynamic characteristics than an undamped system. [1, 2, 5, 8]

3. Design of active damper

There are significant frequency differences in the ram caused by changing overhang while machining. That is why a controlled external force source – active vibration damper – needs to be used.

Based on a demand to position the active damper at the end of machine tool ram, following requirements results:

- Compactness – to generate relatively big damping force with minimum installation dimensions
- Wide frequency bandwith of the actuator - there are significant frequency differences in the ram caused by changing overhang
- Mechanical robustness and reliability – requirement for maintenance-free solution (no significant wear of components)
- Stability of actuator control
- Damping provided by the mass of the actuator itself – typical concept of the damper, where dumping is realized between the two objects

Given requirements are fulfilled only by electromagnetic active dampers. Active dampers based on other principles does not meet at least one of the requirements above. [5]

Disadvantage of contemporary actuators is that they are not able to work concurrently in two axis directions.

3.1 Required vibration damper

Required vibration damper was designed for rams of a cross-section of 350x350 mm and action force 400 N, to damp down vibrations within the frequency range 40-100 Hz for the first eigenmode of ram vibrations. The damper was to be designed as an independent connectable unit.

3.2. Damper active mass calculations

Considering maximal required dimensions 350x350 mm and damper action force 440 N, a damper to damp down vibrations on a general frequency is preferred. If the X axis is introduced to the direction of active mass movement and the active mass is presume to vibrate with an amplitude A, on a frequency f and the active mass is presume to oscillate harmoniously around a value of ΔL/2 in a stable condition, it can be written:

$x = A \cdot \sin(2\pi f \cdot t)$ mass position (1)

$\dot{x} = 2\pi f \cdot A \cdot \cos(2\pi f \cdot t)$ mass speed (2)

$\ddot{x} = -4\pi^2 f^2 \cdot A \cdot \sin(2\pi f \cdot t)$ mass acceleration(3)

Applying Newton's second law to the equation (3), relation for a dynamic force value is:

$$F_{dyn} = \pm 4\pi^2 \cdot m_{eff}\left(\frac{\Delta L}{2}\right) f^2 \quad (4)$$

To damp low frequency vibrations larger inertial mass of the damper is required, as well as a longer stroke, see Chart 1. [8, 9]

Chart 1. *Calculations of deflection amplitude*

Fdyn	440	N			
deflection amplitude [mm]	movable mass [kg]				
frequency [Hz]	8	10	12	14	16
10	13,93	11,15	9,29	7,96	6,97
20	3,48	2,79	2,32	1,99	1,74
30	1,55	1,24	1,03	0,88	0,77
40	0,87	0,70	0,58	0,50	0,44
50	0,56	0,45	0,37	0,32	0,28
60	0,39	0,31	0,26	0,22	0,19
70	0,28	0,23	0,19	0,16	0,14
80	0,22	0,17	0,15	0,12	0,11
90	0,17	0,14	0,11	0,10	0,09
100	0,14	0,11	0,09	0,08	0,07
110	0,12	0,09	0,08	0,07	0,06

Based on calculations in Chart 1 and considering maximal dimensions of the damper, required action force, technical and technological options, maximal active mass size and its deflection was selected.

3.3. Damper conception

The proposed solution of the controlled damper is unique, as it is two-axis version with relatively high action force. Damper motors have constant force of 440 N, or 750 N respectively, using water cooling. Basic arrangement of the damper is presented on the Fig. 4. The damper is connected with a ram via the external (blue) frame, where necessary vibration sensors (X and Y axis directions) are placed. Two movable platforms are connected in a series to the blue frame using steel stripes (light blue). Steel stripes have a function of spring loading and linear guide (parasitic feed in perpendicular axis direction could be ignored for small strokes). First moving platform (light green) is able to oscillate in the Y axis direction connected with steel stripes. Second moving platform (grey) is connected to the green one via steel stripes and is able to oscillate in X axis direction towards the green one. Overall movement of the second one could be considered planar. Between the second (gray) platform and the frame, there are linear motors installed in X and Y axis direction. The motors are able to apply accelerating forces Fx an Fy to the platform, as indicated in Fig. 4. Position sensors, measuring position of the primary motor elements to secondary motor elements, need to be installed because of the necessity of linear motors commutation. (Fig. 4)

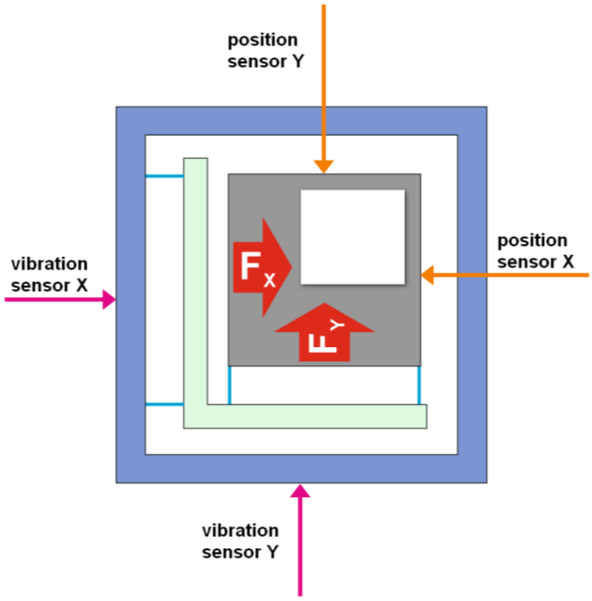

Fig. 4. *Vibration damper scheme [10]*

Damper module contains interior spaces that are possible to be used for media connection between a ram and a tool holder.

3.4. Damper design

Basis of the damper comprise of an active mass (Fig. 5), which is fitted with secondary components of linear electro-motors. Secondary components are turned 90° to each other, to ensure motor effect in both X and Y axis direction. Active mass of the damper is mounted with a pair of steel stripes. These stripes enable the single direction movement. The steel stripes are fixed to the C profile, which is connected to the damper frame via a pair of steel stripes. (Fig. 6) This type of connection of the active mass enables it to move in two perpendicular directions of X and Y axis. Total height of the damper is chosen to keep a required gap between primary and secondary part, when the primary parts are screwed on the part. Manufacturing and mounting inaccuracy are eliminated thanks to the precise underlay between primary component of the linear electro-motor and the damper cover. A pair of optical distance sensors, placed on the damper cover (Fig. 7), and a pair of reflectors placed on the active mass, are installed to measure position of the active mass to the secondary motor parts. The two-axis damper frame is closed with the covers to complete the construction. (Fig. 8)

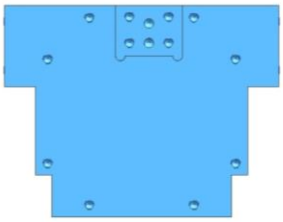

Fig. 5. *Active mass*

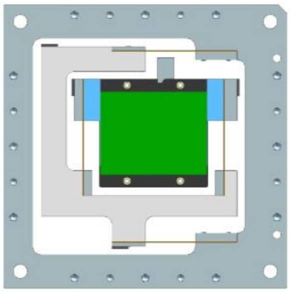

Fig. 6. *C profile connected to the frame*

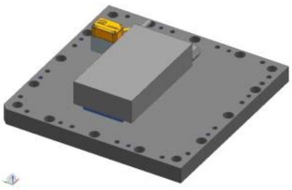

Fig. 7. *Active damper cover*

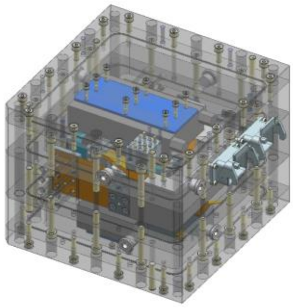

Fig. 8. *Active damper*

Main component of the active damper is the linear electromotor with constant force of 440 N, or 750 N respectively, using water cooling and short-term force of 1000 N. Second main component is an optical distance sensor of a range of 50 mm and a repeatable accuracy of 5 µm. For power supply, signal cables entries industrial connectors are used. Last but not least purchased component of the active damper is a connection material, specified in the manufacturing drawings.

3.5. Control design

Linear motors control in this construction arrangement causes a problem of commutation. Position of the moving mass needs to be measured using contactless measuring, because of construction, operation and technological reasons. An optical sensor, having either analogue or digital communication interface outputs, was chosen to be suitable for this application. However, the sensor is not possible to be connected to the drive amplifier as a commutation or feedback sensor. Sensor with a suitable range or a signal is not commercially available. The converter needed to be designed to convert analogue signal of the position sensor to simulated TTL incremental encoder, which is possible to be connected to the drive amplifier. The sensor signal converter will be realized in the controlling computer National Instrument cRIO-9064 (Fig. 9), fitted with analogue input and digital output modules. Control programme will be designed in the LabVIEW environment.

Fig. 9. *Control computer cRIO-9064 [12]*

3.6 Testing stand conception

New vibration damper development requires comprehensive function testing focused on both damper hardware testing and control algorithms design and testing. Considering the time demands, the development and testing is not possible to be realized directly in the machine, because of an economic pressure to utilize full machining capacity. That is why a testing stand with a ram was designed and constructed (Fig. 10), to enable the development and long-term testing of the dampers. This ram substitutes the existent vertical lathe ram.

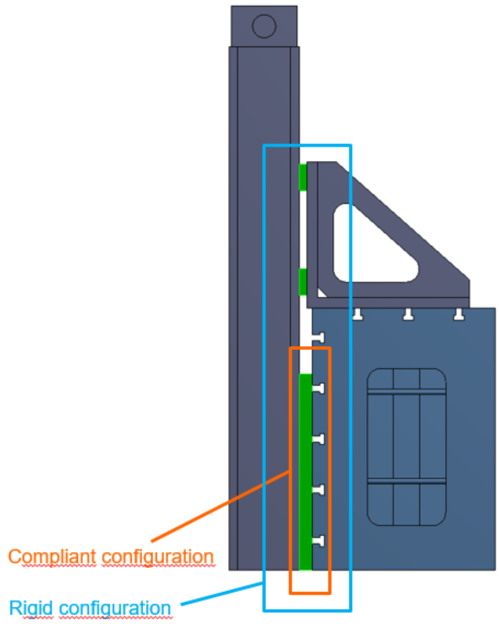

Fig. 10. Conception of the testing stand with a ram to test controlled vibration damper [10]

The testing stand is designed for minimum two configurations to adjust natural frequency of the ram substitute. These two first natural frequencies differ to each other. One of them represents rigid configuration and the other compliant configuration of the testing stand.

The testing stand design was based on the requirement to minimise parasitical torsional deformation impact on the vibration of the ram end, the lowest natural frequency values and the biggest frequency difference between compliant and rigid variant.

The rigid configuration of the testing stand is ensured by connection between the ram substitute and a basis compliant between the ram substitute and a cantilever. It is proposed to test vibration damping of higher frequencies in range of 70-80 Hz.

The compliant configuration of the testing stand is ensured by connection between the ram substitute and a basis. It is proposed to test vibration damping of lower frequencies in range of 50-60 Hz.

To design the testing stand and to estimate its natural frequencies, a FEM model was created. It was corrected through iterations until a construction arrangement achieved first natural frequencies in the required range of rigid and compliant testing stand. Results of the final version are shown on the Fig. 11. [10]

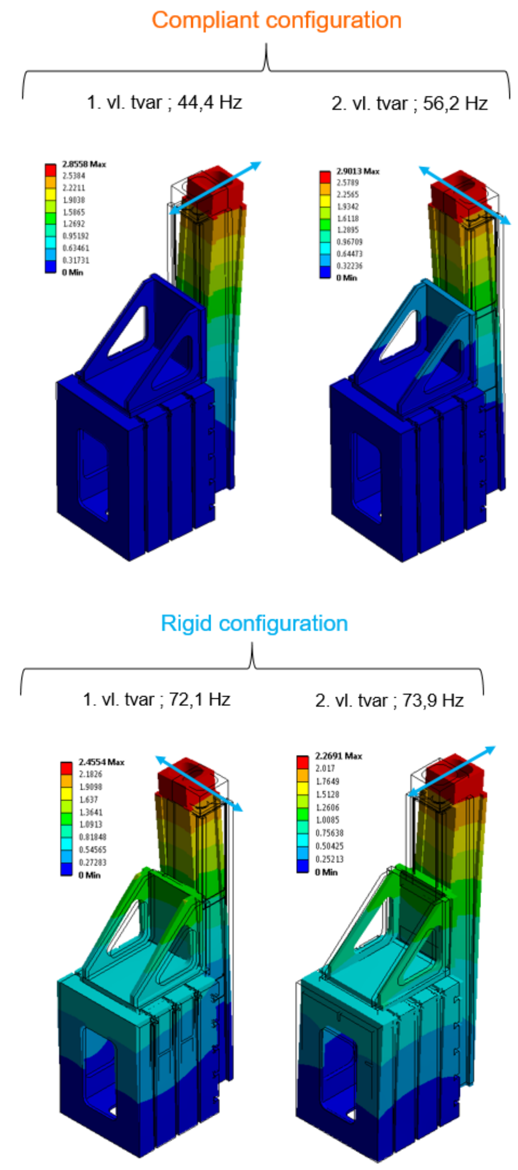

Fig. 11. *Result parameters of the testing stand construction arrangement [10]*

4. Conclusions

New damper was designed for vibration damping vibrations of long slim rams of a cross-section of 350x350 mm. Proposed concept is based on a demand for vibration damping lowest eigenmodes of the ram in the X-Y plane. It was designed as a two-axis damper. Supposed position of the vibration damper body is close to the cutting process. The damper body is designed to be placed between the ram and a turning or a milling head. The proposed solution of the vibration damper is unique, as it is two-axis version with relatively high action force. Damper motors provide constant force of 440 N, or 750 N respectively, using water cooling and short-term force of 1000 N. Nowadays, the testing stand and the vibration damper is mounted.

Terminology

f	frequency (Hz)
F_{dyn}	amplitude of dynamic force (N)
ΔL	stroke length (m)
m_{eff}	effective accelerated mass (kg)

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