INFLUENCE OF COMPOSITE SPINDLE RAM ON MACHINE TOOL DYNAMIC STIFFNESS

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DOI: 10.17973/MMSJ.2022_12_2022156

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Demands for improving machine tool productivity and accuracy can be addressed using alternative material structures with the potential to reduce the mass of moving bodies and decrease machine tool dynamic compliance. A useful option is to apply composite materials because they offer high damping and low density in comparison with steel or cast iron. The key question is how the stiffness and damping of a single composite or hybrid metal + composite component influences the behaviour of the machine tool. In this paper, simulation models for the prediction of machine tool dynamic compliance were prepared for a detailed analysis of a use case study using a hybrid and ductile iron spindle ram for a portal milling centre. A simplified model for the damping matrix formulation was assembled and the influence of the spindle ram damping on the dynamic compliance was tested along with stiffness and mass change, leading to conclusions about the effect of a single component redesign. The minor influence of the material damping of the hybrid structure is noted, and factors influencing the final dynamic compliance are discussed in detail.

KEYWORDS

machine tool, light-weight design, fibre composites, damping, simulation, stiffness

1 INTRODUCTION

The structural parts of a machine tool coupled with the feed drives and a control system are directly responsible for the performance of the machine tool in terms of the machining productivity and workpiece surface quality and accuracy. The behaviour of the frame structure can be defined by its mechanical properties; the most important properties are the static and dynamic stiffness of the assembly, mass of the motion axis components, and dimensional and shape stability. Improvements of the aforementioned mechanical properties can significantly help achieve the desired behaviour of the assembly.

Several approaches can be used to improve the mechanical properties of structural designs. An increase in the assembly dynamic stiffness improves machining productivity, since the dynamic stiffness influences the machining limits [Tlusty 1957], [Altintas 2004]. A decrease in dynamic compliance would require the improvement of either the static stiffness or the assembly damping. While both options are attractive in terms of results, it is very difficult to significantly improve the static stiffness of modern machine tool structural parts unless a significant increase in size or mass is allowed. Steel or cast-iron-based structural parts are usually designed with the help of finite element analysis and optimization methods. The standard approach is to effectively distribute the structural material for the required stiffness while trying to reduce the mass of the components. Mass reduction can even lead to optimized bioinspired structural components designs; [Zhao 2011] presented ones such example. Another approach to designing light and productive machines was presented by [Zulaika 2011]. In general, the potential for increasing the static stiffness of the structural components is limited.

Changing the assembly damping is another option to increase the dynamic stiffness. Common knowledge says that the damping of the machine tool frame is mostly influenced by the damping of connection interfaces, for example, by the type of guideway connection. A comparison of various linear guideway systems by [Astarloa 2022] showed that hydrostatic guides offer 3-4 times higher damping than roller-based guides. The damping parameters of linear guides were presented by [Oleaga 2022], showing different damping parameters in the interfaces between the spindle ram – frame, the frame – column, and the column – bed of a machine tool assembly. The modal damping ratios of the linear guides were obtained in the range of 2-6 % from experiments and 2-9% from FEA. Damping values of structural components are significantly lower (for example, steel 0.05-0.15 %, cast iron 0.10-0.20 %) and their contribution to the assembly damping is not as large as the contribution of the interfaces. A possible approach to increasing damping is to use active dampers and mechatronic approaches, as stated, for example, by Neugebauer [2007] or Novotny [2021]. In the aforementioned study by [Astarloa 2022], the damping of the rolling guideway system increased 30 times when the axis was equipped with an active damper system.

In general, it is difficult to significantly improve chatter-related machining productivity by steel or cast-iron design modification as these materials do not provide options for a significant improvement in stiffness or damping. Light-weight design approaches are an attractive option since they tend to reduce the mass of motion components while keeping the static stiffness of components and the assembly at acceptable levels. An example was given by [Triebe 2022], in a moving table mass reduction study, which stated that light-weight design approaches enable energy savings of 30 % or more. A study by [Kroll 2011] analysed several light-weight design approaches, including fibre composites, and showed that using fibre composites could lead to a possible increase in performance and decrease in energy consumption.

Fibre composites appear to be an attractive material option for the new design approaches due to their low density and great direction-oriented stiffness. They provide an option for mass reduction and damping increase and have the capability to reach similar bending stiffness as steel structures, as shown by [Lee 2004]. An extensive overview of the application of fibre composites in the machine tool industry was published in [Möhring 2015] and [Möhring 2017], which listed mechanical interfaces of composites, material costs, resistance to the machining environment and reproducible manufacturing processes as challenges for the wider spread of fibre composite application. The benefits of low specific weigh, good specific stiffness - mass properties, high damping, and low thermal expansion motivated the research and development of composite applications in rotational components such as spindles, shafts, and cutting tools, and also in large scale structural components such as spindle rams. A recent publication by [Birk 2019] showed the development of composite spindle rams for machine tools, including the effect of adhesive joints on spindle ram stiffness and damping. [Möhring 2021] published a fibre composite – hybrid solution for a spindle ram design, which was improved with integrated sensors for the monitoring of the design.

A general issue with large-scale applications of fibre composites in the machine tool industry is their material costs, which are 5-10-15 times higher than those of welded steel structures. The second complication is the orthotropic behaviour of fibre composites, which limits the potential to achieve low deformation during multiaxial loading. Local deformations at the connection interfaces are also an issue with structural components created from orthotropic composites. Therefore, it could be more effective to focus on the development of hybrid structures based on metals and fibre composites. [Kulisek 2021] presented an example of model spindle rams, which were designed from ductile iron reinforced by an inner composite tube with a focus on the damping improvement.

In general, new material structures offer significant mass reduction along with an improvement in the damping of structural components. However, the effect of the composite-based light-weight approach on the machine tool assembly is usually not evaluated or described. There is a limited amount of data on how the component damping improvement results in the assembly damping change.

Although the stiffness and mass effect can be simulated with standard approaches, for example by harmonic analysis using FEA, the influence of single-component damping on the entire assembly behaviour presents a more difficult task. There has been a high demand for research and development of finite element (FE) based methods for machine tool dynamic behaviour prediction, especially with the spread of machine tool digital twins and virtual machining. Publications by [Brecher 2013], [Rebelein 2017], [Semm 2018], and [Zaeh 2019] describe approaches focused on modelling of machine tool behaviour using linear damping and nonlinear friction effects, using the local damping effects of machine tool structural parts, connection interfaces, and motion mechanism like ball-screw drives. These methods usually use experimental approaches to identify damping ratios of the interfaces and develop particular approaches on how to implement the values into the global damping matrix of the machine tools. The successful application of these methods would help to evaluate composite or hybrid application during the design phase. However, these detailed models require a large quantity of identified data for connection interface behaviour. A simplified approach is still needed during comparison of design options and decision - making about a new machine tool.

As the overview showed, composite materials are finding their way into machine tool structural components. They offer low density, good damping, and good directionally - oriented stiffness behaviour, as published by the cited papers. The influence of composite-based design on machine tool behaviour is still not very clear when applied to large structural components such as spindle rams or transverse beams, which require high static stiffness parameters under multiaxial loading. Two important questions are what are the benefits of composite-based structural components to machine tool frame properties, especially if only a single component of a machine tool is redesigned, and how can changes to the design mass, stiffness and damping improve the properties of the machine tool assembly.

To answer those questions, a case study of a portal milling centre with a hybrid spindle ram was analysed, focusing on the mass, stiffness and damping influence of the spindle ram on the dynamic properties of the machine tool, mainly dynamic compliance. The focus is on using simulation tools along with experimental measurements and drawing conclusions about the effect of hybrid spindle rams on machine tool behaviour. The paper is organized as follows: in Chapter 2, a case study of a hybrid spindle ram for a portal milling centre is presented, including an analysis of the stand-alone spindle ram and machine tool assembly behaviour based on experiments and finite element simulation. In Chapter 3, a damping matrix is assembled using the modal damping ratio of the machine tool model components, and the influence of the hybrid spindle ram on the dynamic stiffness amplitude is evaluated. In Chapter 4, the effects of the spindle ram mass, stiffness, and damping change on the machine tool assembly behaviour are discussed.

2 A CASE STUDY OF A HYBRID SPINDLE RAM DESIGN FOR PORTAL MILLING CENTRE

2.1 *Hybrid spindle ram design and material properties*

A hybrid spindle ram for a 5-axis vertical milling centre was designed, using a ductile iron body which was reinforced by an inner composite tube. The design (external dimensions 420 mm x 400 mm x 2530 mm, see Fig. 1), uses the benefits of the metal body, which enable standard designing of connection interfaces for linear guideways, ball-screw drives and milling head attachments. The inner composite reinforcement provides a further increase in the stiffness of the design. Using the benefits of low-density composites, the added mass of the composite body was 110 kg compared to more than 1000 kg of the reference spindle ram mass. The interface between the composite and the metal body was created by adhesive joints using an epoxy-based adhesive.

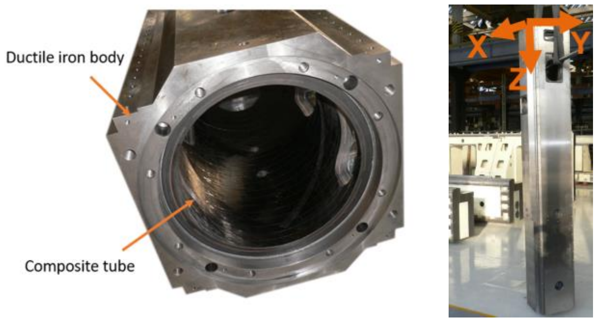

Figure 1. Hybrid spindle ram based on ductile cast iron and inner composite reinforcement (left) and the boundary conditions during experimental modal analysis (right)

For the structural component of the machine tool motion axis, only fibres that lead to composite layers with Young's modulus comparable to or greater than steel, ductile or cast iron should be considered. Therefore, the composite reinforcement was designed using high-strength composite fibres (HSC) with Young's modulus in the fibre direction around 235 GPa and ultra-high modulus fibres (UHM) with Young's modulus in the fibre direction around 780 GPa. The composite body consisting of these fibres was produced by a filament winding technology, using epoxy-based resin to create the composite layer (HSC-E or UHM-E).

The composite reinforcement; see Fig. 1 and Fig. 2, was designed as an internal tube with a conical ending at the side of the spindle ram – milling head interfaces. The cylindrical side had a 240 mm internal diameter and 31 mm wall thickness, while the extended ending had a 290 mm internal diameter and 21 mm wall thickness. The lay-up was designed with a focus on balancing the stiffness in bending, transverse shearing, and torsion, which generally requires a combination of layers with a 0, ±45 degree orientation. The lay-up consisted approximately of 15 % HCS-E layers with a ±45 degree orientation, 45 % UHM-E layers with a ±45 degree orientation, 37 % UHM – E layers with a 0 degree orientation, and 3 % HSC – E layers with a 88 degree orientation. Due to the winding technology, the actual winding angle of layers with a nominal winding angle of ±45 degrees was in the range of 41 degrees to 53 degrees. The 0 degree orientation indicates the direction of the tube axis.

The elastic properties and density of unidirectional layers for both types of carbon fibres and effective properties of the homogenized tube are given in Tab. 1. For improved damping, two layers of cork material (Young's modulus is approximately 40 MPa) were integrated into the composite design; see the brown layers in Fig. 1.

	HSC-E	UHM-E	Homogenized tube
E_1 [GPa]	127.7	383.1	130
E_2 [GPa]	5.07	3.57	33
E_3 [GPa]	5.07	3.57	4.5
G_{12} [GPa]	3.42	2.91	48
G_{13} [GPa]	3.42	2.91	3
G_{23}[GPa]	3.42	2.91	3
ρ [kg.m^{-3}]	1458	1627	1600

Table 1. Material properties of unidirectional composite layers (fibres and epoxy resin) and effective properties of homogenized composite tube

The effective properties show that the reinforcement material has significantly lower density, and has important stiffness parameters (E_1, G_{12}) that are similar with parameters of cast iron. These properties could be further improved by removing compliant cork layers and optimizing the composite reinforcement.

2.2 *Verification of finite element models of hybrid ram*

The composite reinforcement was designed as a thick-walled structure with a complicated internal structure, which was composed of layers of high-strength or high-modulus carbon fibres and integrated cork layers to increase damping. A finite element model with a fully defined composite lay-up was made; see Fig. 2. The composite reinforcement was simulated using solid-shell elements (continuum shells) with multiple elements per wall thickness. To define the properties through the thickness, the composite lay-up was divided into several subsections. The interface between the ductile iron body and the composite tube was modelled by bonded contacts, assuming a minor influence of the epoxy-based adhesive on the ram stiffness behaviour and on the total mass of the ram.

As the study focused on the prediction of the machine tool assembly with the spindle ram, it was necessary to answer the question about the accuracy of the hybrid spindle ram when predicting stiffness or modal properties. Therefore, the modal analysis was performed by FEA and its results were compared with the experimental modal analysis measurements. During the experimental measurements, both rams were placed on a flexible rope to approximate the free - free boundary conditions, and to minimize the effect of boundary conditions on the modal analysis results; see Fig. 1. For simulation, free-free boundary conditions were used neglecting the rigid mode shape results. The comparison is given in Tab. 2. The comparison demonstrated that the finite element model of the hybrid spindle ram provides sufficient precision in predicting modal behaviour.

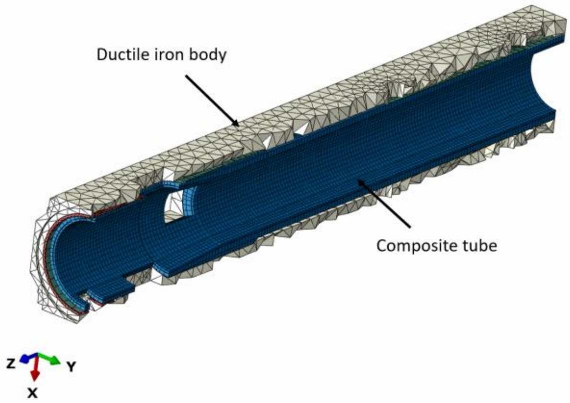

Figure 2. Detailed FEA model of hybrid spindle ram and composite reinforcement

	EMA	FEA	Mode shape
f_1 [Hz]	326	327	Bending
f_2 [Hz]	328	336	Bending
f_3 [Hz]	554	572	Torque
f_4 [Hz]	700	701	Bending
f_5 [Hz]	701	704	Bending

Table 2. Comparison of computational and experimental modal analysis results performed on FE model and real spindle ram

2.3 *Stand-alone hybrid spindle ram experimental behaviour*

The first approach to hybrid spindle ram evaluation was performed using experimental modal analysis (EMA). The hybrid spindle ram behaviour was compared with that of a reference ductile iron ram (without a composite tube reinforcement) as a stand-alone body. The aim of the experimental work was to compare the natural frequencies, damping ratio and dynamic compliance amplitudes on the most important mode shapes.

Both the reference and hybrid spindle rams had the same ductile iron body (same dimensions, connection interfaces, etc.). The hybrid spindle ram was created using the reference spindle ram design, which was reinforced by bonding of the internal composite tube. The aim of the work was to compare the frequencies and amplitudes of dynamic compliance on important mode shapes, along with damping of the reference and hybrid ram.

The configuration of the experiment is shown in Fig. 1. The stand-alone rams were measured using fixation by flexible ropes, which were attached to the front of the spindle ram. A B&K PULSE analyser was used for data collection in the experiments. A modal hammer was used to excite the measured structures. A one-axial or three-axial accelerometer was used to measure the vibration response. The modal parameters (natural frequency, damping ratio and mode shapes) were estimated from the measured data after the experiments had been performed. The results are shown in Fig. 3 (excitation in X) and Fig. 4 (excitation in Y).

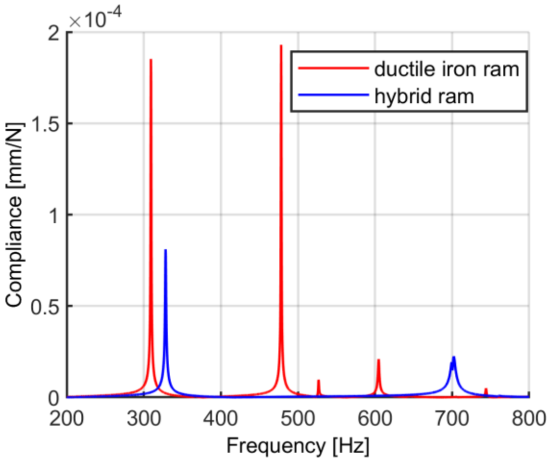

Figure 3. Comparison of stand-alone ram behaviour: dynamic compliance amplitude for excitation and response in X direction from experimental modal analysis

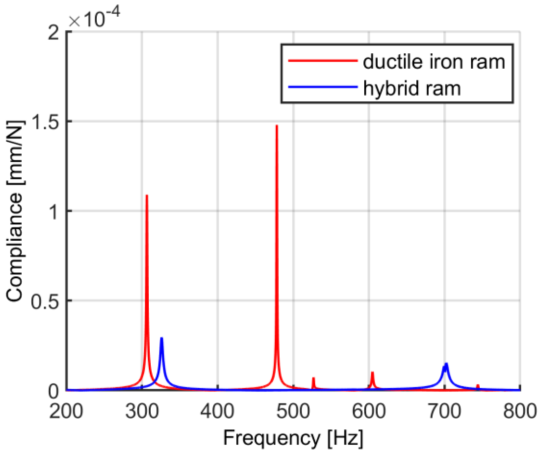

Figure 4. Comparison of stand-alone ram behaviour: dynamic compliance amplitude for excitation and response in Y direction from experimental modal analysis

The experimental comparison of dynamic compliance demonstrates the ability of composite reinforcement to improve the design, as the hybrid spindle ram reached significantly higher dynamic stiffness on most of the comparable mode shapes. Although the mass of the hybrid body was higher, the increase in stiffness was sufficient to improve the natural frequencies, which occurred for the first bending mode shape and torque related mode shapes. The frequency and damping values of the reference and hybrid rams for the selected important mode shapes are given in Tab. 3. The important mode shapes were selected as those with the highest dynamic compliance or important due to expected structural behaviour. The values were evaluated for the first bending mode shape (1a for excitation in X, 1b for excitation in Y) and the second (2a, 2b) bending mode shape along with the torque mode shape (T).

For excitation in the Y direction, the hybrid design outperformed the reference body in both frequency and damping values, and the damping of the hybrid ram was multiple times higher. However, for excitation in the X direction, the damping parameters were similar to the damping values of the reference design.

Mode	Reference Ram		Hybrid Ram	
	f [Hz]	ζ [%]	f [Hz]	ζ [%]
1a	306	0.16	328	0.19
1b	309	0.10	326	0.44
T	478	0.06	554	0.35
2a	706	0.12	700	0.12
2b	650	0.11	701	0.34

Table 3. Stand-alone spindle rams - comparison of frequency and damping ratio s

2.4 *Simulation of machine tool assembly with hybrid spindle ram*

The second approach for hybrid spindle ram evaluation was performed using simulation of the machine tool assembly, which was used to evaluate the effect of the spindle ram mass and stiffness change on the assembly behaviour. The analysis was performed using a finite element model of the existing machine tool, which is operated together with the reference ductile iron ram. The assembly model is shown in Fig. 5. The machine tool operation space is 1.5 m in Z, 3.2 m in Y, 7 m in X, and the static stiffness of the tool tip is in the range of 40-60 N/um when loaded in the X or Y direction. The finite element model of assembly consists of solid bodies, which were used for modelling of the columns, a transverse beam, cross-slides, a spindle ram model, and simplified rails and gear-racks. Connection interfaces – housings of guideways, ball-screw drives and bed to ground clamping were modelled using spring-like bushings.

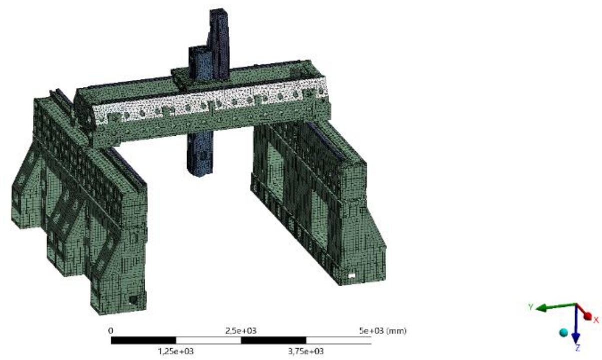

Figure 5. FE model of the portal vertical milling centre

To evaluate the quality of the model, a verification of the predicted dynamic compliance frequency response function with the experimentally obtained behaviour was performed for the machine tool assembly with the reference ductile iron spindle ram. Dynamic behaviour simulation was performed using modal analysis to simulate 100 mode shapes in the simplified tool control point (TCP), which was placed at the front of the spindle ram. The parameters were then exported from the finite element solver to a state-space model using modal reduction and the frequency response function of dynamic compliance was calculated. The simulated responses use a proportional damping model, where a different modal damping value can be set for the whole assembly. The analysis of machine tool behaviour was performed for the configuration when the TCP was loaded in the X direction. The comparison between the measured and simulated responses is shown in Fig. 6.

Using modal damping $\zeta = 6\,\%$ for the first amplitude (25 Hz) and modal damping $\zeta = 3.7\,\%$ for the second dominant amplitude (38-45 Hz), the predicted response reached amplitudes of dynamic compliance to the response, which was captured by experimental modal analysis in the real assembly. A notable frequency shift (38 Hz vs 45 Hz) between the predicted and measure response was obtained on the mode shape with the dominant amplitude. The 18 % difference was taken as acceptable for the model, which will be used further to assess and evaluate the benefit of the hybrid design. Although the basic assembly model did not cover the experimentally obtained assembly behaviour more precisely, an assumption was made that the single component change (in this case, spindle ram) will be correctly predicted in the assembly behaviour change.

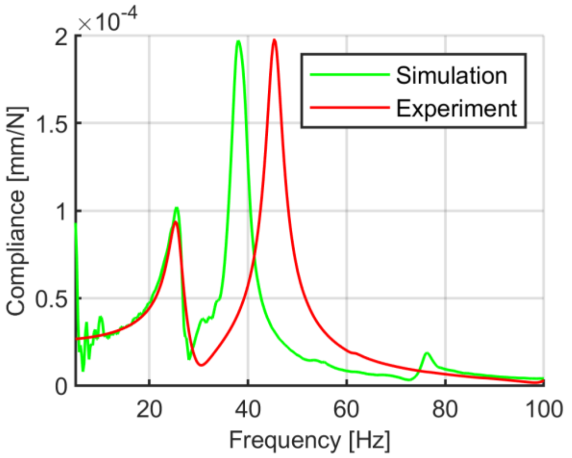

Figure 6. Configuration with ductile iron ram: measured and simulated dynamic compliance frequency response function – X direction

The verified machine tool assembly model was used with both the reference and the hybrid spindle ram and the same assembly proportional damping value settings with the assumption that the damping of the assembly would not be influenced by the change in the spindle ram material-based damping. Comparison of the assembly dynamic compliance with the reference and with the hybrid spindle ram is shown in Fig. 7.

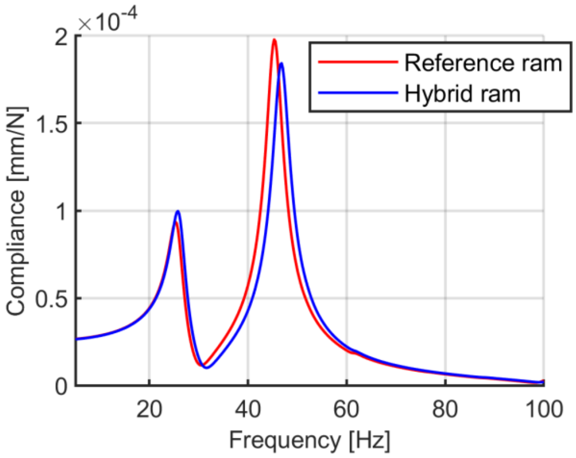

Figure 7. Simulation of assembly behaviour - dynamic compliance comparison – effect of composite reinforcements – excitation in X

The comparison demonstrated the effect of the hybrid spindle ram design on the assembly behaviour. The most important results are:

- Chatter-related productivity is influenced by the reduction in the dominant amplitude of dynamic compliance. The reinforcement of the hybrid design stiffness reduced the dynamic compliance by 7 % when the damping effect of the spindle ram was not evaluated.
- The hybrid design led to a 3 % increase in the eigenfrequency of the dominant mode shape compared to the reference spindle ram.

The simulation was performed for a loading case in which the simplified tool tip was loaded in the X direction. This loading case was selected as the major contribution to assembly compliance should be for this loading caused by bending and transverse shearing of the spindle ram, bending and torsional deformation of the transverse beam, and by deformation of connection interfaces between the motion axes.

The results demonstrate the limits of a single structural component change in machine tool assembly behaviour. The design change, i.e. the change in mass and stiffness through composite reinforcement, would not cause a significant increase in the chatter-related machining productivity. This analysis did not investigate or determine the possible effect of increasing the spindle ram damping on the assembly behaviour, a question, which needs to be answered using various models or approaches.

3 INFLUENCE OF SPINDLE RAM DAMPING ON ASSEMBLY DYNAMIC COMPLIANCE

3.1 Damping model description

The influence of the hybrid spindle ram's stiffness and mass on the machine tool behaviour was analysed using proportional damping values, which were given to the whole assembly. An additional option would be to perform a harmonic analysis directly in a finite element solver, for example, by creating a global damping matrix using the Rayleigh damping model or the constant modal damping ratio of the whole. Neither approach is suitable for analysing how single structural component damping influences the assembly behaviour or answering the key question of whether a component with higher damping can improve the dynamic stiffness of the machine assembly, as most of the assembly damping happens in connection interfaces.

Therefore, a simulation model was assembled with the objective of evaluating the effect of the structural parts' stiffness and damping on the dynamic behaviour of the machine assembly. The model used the description of the basic motion equation in modal coordinates y, see (1). The basic approaches used by FEA are given in (2), where the first two members on the right side correspond to the Rayleigh damping and the last member corresponds to the modal damping ratio set to the whole assembly. As can be seen, no member of the single component damping is included in equation (2). Therefore, a simplified model was used to build up the global damping matrix in modal transformation, which is given in (3), to analyse the component's damping influence.

$$[I]\{\ddot{y}\} + [\Phi]^T[C][\Phi]\{\dot{y}\} + [\lambda^2]\{y\} = [\Phi]^T\{F\} \qquad (1)$$

In equation (1), I represents an identity matrix, y represents modal coordinates, $\{\phi\}$ presents mode shapes, C represents a global damping matrix, λ represents a diagonal matrix containing eigenvalues $\lambda = \mathrm{diag}(\Omega)$ and F represents a nodal force vector.

$$[\Phi]^T[C][\Phi] = \alpha I + \beta[\Lambda^2] + 2\zeta[\lambda] \qquad (2)$$

In equation (2), α represents a global mass matrix multiplier, β represents a global stiffness matrix multiplier and ζ represents a global constant damping ratio. In equation (3), j represents an index of a material used in the model. Therefore, ζ_j represents a constant damping ratio of material j, Ω represents an excitation frequency and K_j represents a part of the global stiffness matrix that is given by material j.

$$[\Phi]^T[C][\Phi] = [\Phi]^T \sum_{j=1}^{m} \left(\frac{2\zeta_j}{\Omega}\right)[K_j]\,[\Phi] \tag{3}$$

The presented approach uses many assumptions and simplifications to formulate the damping matrix. The most important are as follows:

- The damping ratio of the assembly components is given by a constant value ζ_j for the whole component behaviour.

- The damping of the connection interfaces (for example, guideways, simplified ball-screw drives) is derived from their stiffness definition in the assembly and the set damping parameter ζ_j.

- No mass-related damping is included in the model, and only linear behaviour is assumed.

- Overall, this model presents an approximation of the assembly behaviour with limited validity in terms of the real behaviour but provides an option for evaluation of the influence of the assembly components' damping on the assembly dynamic behaviour.

A diagram of the model application is given in Fig. 8. The most difficult task was identifying the components *Kj*, parts of the global stiffness matrix *K*, which are created by each component (structural part or connection interface) of the model. For the connection interfaces, which were defined by a bushing or a spring, the matrix *Kj* was obtained using *Kj=K*-K*, where *K** was the global stiffness matrix of the modified assembly with the connection interfaces of double the stiffness. It was not possible to use a similar approach for structural bodies (by multiplying the Young's modulus twice), as the finite element solver can apply internal model renumbering. Therefore, a more complicated approach was used to identify the contribution of the components to the global stiffness matrix for structural bodies. A damped modal analysis of the assembly was performed, using only the local stiffness multiplier β_j of a single structural body to set the global damping matrix C. The matrix K_j was then defined as C/β_j.

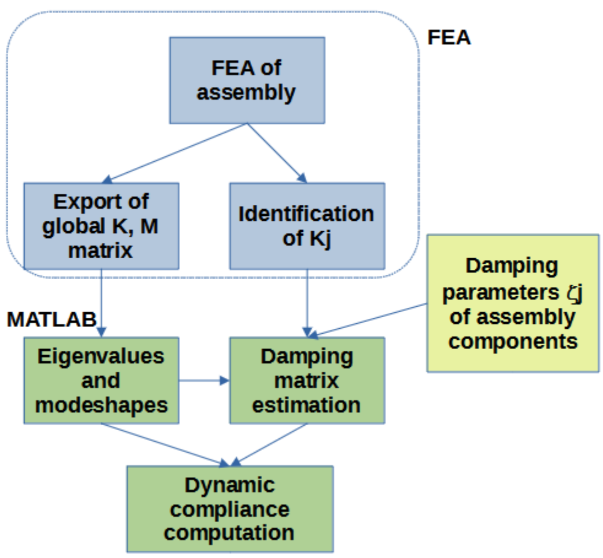

Figure 8. Diagram of damping matrix assembly and the dynamic behaviour simulation

The idea for the formulation of the damping matrix was to apply average damping ratios ζ_j of structural parts made of steel, ductile/cast iron or fibre composites, which were obtained from experiments, as, for example, given in [Kulisek 2021]. The next step was to apply the parameters of connection interfaces damping either from the literature, experimental identification or as identified values, which lead to a required assembly behaviour. For the connection interface parameter settings, the goal was to obtain dynamic compliance on the dominant mode shape corresponding to the dynamic compliance of the mode shape with the required damping ratio of the whole assembly (for example, using proportional damping given to the whole model).

3.2 Application on the machine tool assembly

The described approach was applied to the machine tool model that was used for the hybrid ram stiffness and mass evaluation in Chapter 2. To reduce the size of the numerical problem, the subassembly as shown in Fig. 9 was used to analyse the dynamic behaviour. For loading in the X direction this column simplification did not influence the assembly behaviour.

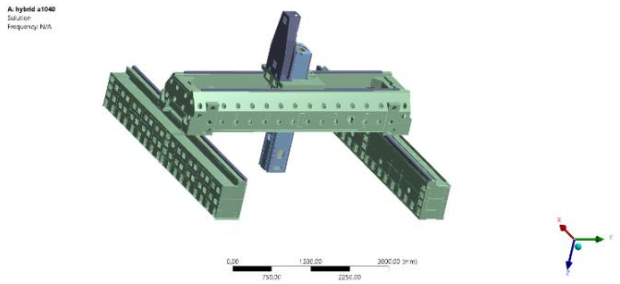

Figure 9. Machine tool sub-assembly for evaluation of damping influence

As reference behaviour, a model with a uniform damping parameter ζ = 3.3 % was used for the whole assembly. The model corresponds to a finite element harmonic analysis of the assembly with the same constant damping ratio. Using the formulation of Equation (3), model 1 was created using the damping ratios for components and connection interfaces as given in Tab. 4.

Model 1 corresponds to the basic idea of using the average damping values of the structural components and the identified values of the connection interfaces and other parts of the model. For the metal parts (spindle ram, cross-slides, transverse beam and columns), values were taken from the higher end of the known range (steel parts ζ = 0.02-0.15%, cast iron or ductile iron parts ζ = 0.05-0.25%). In the analysis, no differences between the damping of steel or ductile/cast iron components were applied. To match the dynamic compliance amplitude in the dominant mode shape (45-46Hz), the connection interfaces had to be set with the damping ratio ζ = 25%, which is significantly higher than values found in the literature.

The second set of settings was used in model 2, where the damping of the guideways was defined more closely to the commonly used parameters: ζ = 8 %. To obtain the required dynamic compliance at the damping of the dominant mode shape, the structural components had to be increased by an order of magnitude in comparison with the expected values.

A comparison between the reference model with uniform damping and model 1 and model 2 with different damping values of components is shown in Fig. 10.

	Model 1: ζ_i [%]	Model 2: ζ_i [%]
Metal structural bodies	0.2	2.4
Spindle ram – CFRP	0.4	4.8
Ball-screw drives	1.5	2.4
Gear-rack	2.0	2.4
Guideways	25	8.5
Column - ground	35	12

Table 4. Constant damping ratios of assembly components for the damping influence study

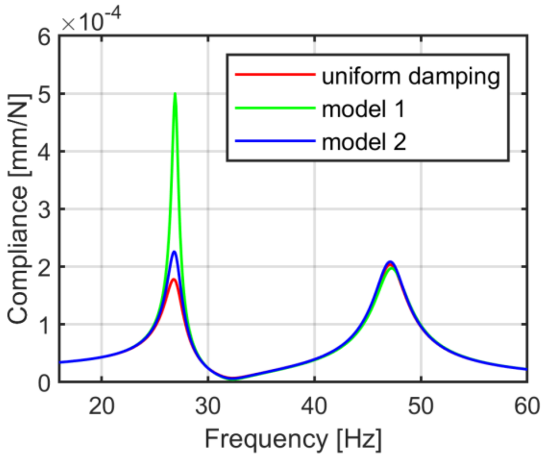

Figure 10. Assembly behaviour – comparison of dynamic compliance amplitude for different component damping settings.

The comparison of the dynamic behaviour demonstrates the limits of the presented model. The realistic average damping values of structural parts and high connection interface damping values matched the dynamic compliance on the mode of interest, the dominant mode shape. On a different mode shape, model 1 predicted significantly higher dynamic compliance, which would correspond to a system with very low damping. Model 2, with artificial structural component damping values, performed with a better match to the uniform damping model. But this should be expected, as the damping values of each component were closer to the global damping ratio of the uniform model. As the focus was on the mode shape with dominant compliance, the assumption is that the model with more realistic damping settings would reflect the behaviour in a more conservative or realistic way when modifying the damping values of a single component.

3.3 Influence of the spindle ram damping on assembly behaviour

Using the model 1 settings, the damping influence on the assembly compliance was evaluated. The first analysis focused on the spindle ram damping effect; see Fig. 11. It was performed by comparing the basic model 1 with the models where the damping of the composite reinforcement and the whole hybrid spindle ram were increased. The second analysis, see Fig. 12, was focused on the guideway damping influence. It used model 1 settings along with a 50 % increase in guideway damping.

A similar analysis of spindle ram damping was performed using the model 2 settings with artificially high damping structural component values. Again, the composite tube damping influence was non-existent. If the whole spindle ram had three times higher damping than the original body, the dynamic compliance was reduced by 14.8 %.

The comparison demonstrates that increasing the spindle ram damping (composite body or whole spindle ram) did not influence the dynamic compliance in a recognizable way. The amplitude was reduced by 0.1 % when the composite tube damping was increased 3 times, or by 1.3 % when the whole spindle ram damping was increased 3 times. On the other hand, a change in the connection interfaces damping did lead to a major reduction in dynamic compliance.

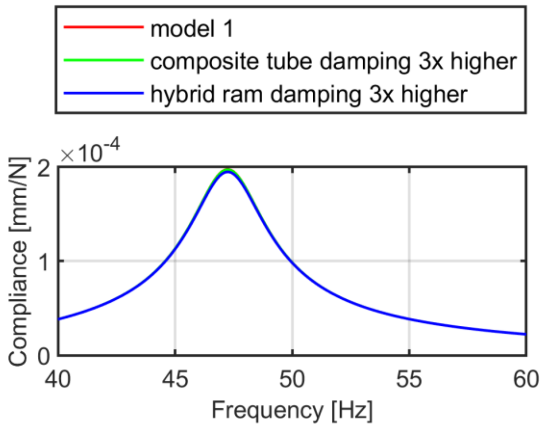

Figure 11. Effect of the damping increase (3x) of composite reinforcement or the whole spindle ram on the assembly dynamic compliance for model 1 settings

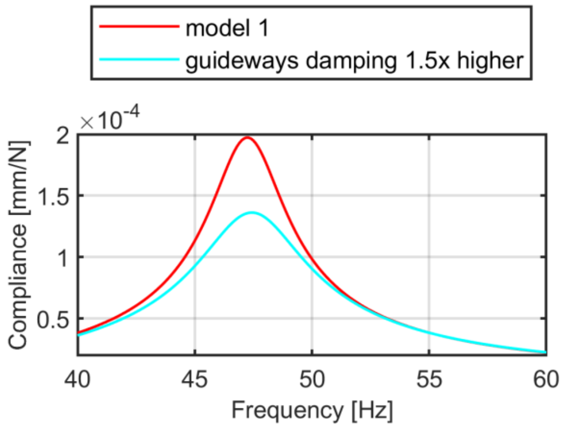

Figure 12. Effect of damping in the connection interfaces - housings

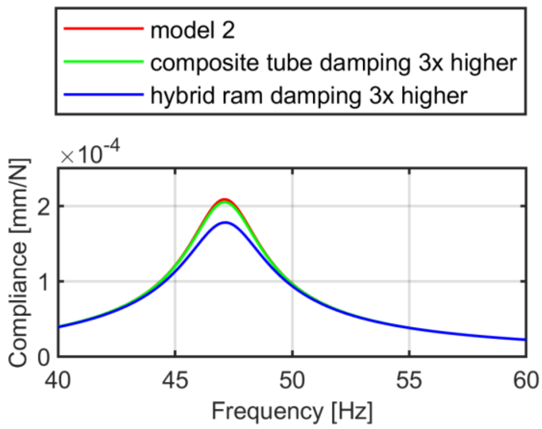

Figure 13. Effect of the damping increase (3x) of composite reinforcement or the whole spindle ram on the assembly dynamic compliance for model 2 settings

4 DISCUSSION

A case study of a hybrid ductile iron spindle ram with composite reinforcement was presented, using the properties of a real high-stiffness portal milling centre for general machining. A new hybrid spindle ram was designed using a thick-walled composite tube bonded to the ductile iron body. This design change led to a significant dynamic stiffness improvement of a stand-alone component due to the composite-related stiffness and damping improvement. FEA-based simulation models predicted results with good accuracy for the stand-alone component.

A simulation-based approach to evaluate the effect of a single component damping change on the dynamic stiffness was assembled and tested on a model of the machine tool. The model was assembled with many simplifications, using damping ratios of components and connection interfaces either from experiments or the literature, or artificially set to fit the damping ratio of the assembly to the required value. It was performed with acceptable results on the selected mode shape and used connection interface damping values, which were significantly increased compared to the usually published parameters.

As a result of the simulation sensitivity analysis, the effect of composite reinforcement damping on the dynamic stiffness of the assembly was negligible. The results are consistent across the different settings of the components and damping values. This leads to the conclusion that when new composite-based structures are introduced to machine tool structural components, the increase in damping should not be considered an important property for machine tool improvement. The focus in damping improvement should be on the connection interfaces or active dampers, and other mechatronic systems.

For the composite tube or the entire ram damping, a three-fold damping increase was estimated as an optimistic scenario when applying the composites of different lay-ups. As can be seen from the experimental results for the hybrid spindle ram prototype (see Tab. 3), it was not possible to consistently achieve such an increase on the real prototype through the most important mode shapes. In the direction X, the damping of the hybrid ram was comparable to that of the ductile iron body. Therefore, even the best improvement using the unrealistic settings of the structural component damping (see Fig. 13) would be minimized when the real hybrid spindle ram damping improvement is applied.

Although the potential of composites is promising, the effect of a single structural part such as a spindle ram on the assembly behaviour is limited. Although the hybrid spindle ram was significantly improved as a stand-alone system compared to the reference ductile iron body, the effect of the improvement on the machine tool assembly was not significant. In the presented case of a high stiffness portal milling centre, the frequency in the dominant mode shape increased only by 3 % while the dynamic compliance amplitude was reduced by only 7 %. Improvement of this magnitude would be hardly acceptable if material and manufacturing prices were taken into account. The composite design can have many input variables (fibre, thickness and orientation of each layer). Therefore, the results of this case study with a hybrid spindle ram prototype could be further improved by optimizing composite reinforcements.

From the results, the following generalized design recommendation can be derived:

- Composite parts of machine tools should be designed with a focus on achieving the highest static stiffness possible for the given load while trying to reduce the mass of the components.
- For machine tools and other production machines with similar connection interfaces, there is no need to integrate additional damping layers into the composite bodies, as the damping improvement would not be reflected in the assembly behaviour, and the reduction of static stiffness due to the presence of compliant damping layers could limit the assembly stiffness.
- Assuming the machine tool is a complex dynamic system, light-weight material structures or reinforcements should be applied only when the whole assembly can be optimized. The approach of replacing a single structural component with an improved one, using the same dimensions and connection interfaces, does not usually lead to a major improvement in the design of a modern machine tool.

5 CONCLUSION

A hybrid material structure consisting of fibre composites and cast iron was analysed for application in machine tool structural components with high stiffness demands. A hybrid spindle ram based on a ductile iron body and inner composite reinforcement was manufactured and experimentally tested along with the reference metal component. Using a simulation model, the effect of changes in mass, stiffness and damping due to the hybrid structure design on the machine tool assembly behaviour were analysed. Using an assembled damping model, the benefits of the hybrid design damping increase were evaluated as non-existent. For cases similar to the tested machine tool and its structural parts, the focus during machine tool related composite – hybrid structure design should be on static stiffness – mass behaviour.

ACKNOWLEDGMENTS

The authors would like to acknowledge the funding support from the Czech Ministry of Education, Youth and Sports under the project CZ.02.1.01/0.0/0.0/16_026/0008404 'Manufacturing Technology and Precision Engineering'. Support from the TAJMAC-ZPS company with design and experiments and from the Compo Tech Plus with the composite reinforcement design is also gratefully acknowledged.

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