Article

Vibration Damping Analysis of Lightweight Structures in Machine Tools

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Abstract: The dynamic behaviour of a machine tool (MT) directly influences the machining performances. The adoption of lightweight structures may reduce the effects of undesired vibrations and increasing the workpiece quality. This paper aims to present and compare a set of hybrid materials that may be excellent candidate to fabricate the MT moving parts. The selected materials have high dynamic characteristics and capacity to damp mechanical vibrations. In this way, starting by the kinematic model of a milling machine that highlights the main critical factors, this study paper evaluates a number of prototypes made of Al Foam sandwiches (AFS), Al Corrugated sandwiches (ACS) and materials reinforced by carbon fibres (CFRP). A set of prototypes has been fabricated, represented the Z-axis ram of a commercial milling machine. The static and dynamical properties have been evaluated by using both FE simulations and experimental tests. The obtained results show that the proposed structures may be a valid alternative to the conventional materials of MT moving parts, increasing machining performances. In particular, the AFS prototype highlighted a damping coefficient that is 20 times greater than a conventional ram (e.g. steel). The CFRP structure is able to satisfy the machining requirements with a reduced weight of 48.5%, while the ACS prototype showed a good trade-off between stiffness and damping.

Keywords: hybrid materials; machine tool structures; modal analysis; machine tool kinematics; aluminium metal foams; aluminium corrugated sandwiches; CFRP materials; FE simulations; damping

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1. Introduction

The dynamical behaviour of a Machine Tool (MT) plays a crucial role in satisfying the main machining requirements, like high-speed operations, precision in axes positioning and capability to quickly remove high quantity of workpiece material [1]. These performances are directly related to the materials used in MT construction. In this way, materials of MT foundations and moving parts need to be selected with high dynamic characteristics and capacity to damp mechanical vibrations.

The MT structures guarantee to withstand the forces generated by the process and the machine motions [2]. Forces acting on the MT structure during motion represent a serious limitation for the precision and productivity and, therefore, they are evaluated during machine design and material selection [3,4]. In particular, structural deformations may lead to unwanted displacements of the tool tip point (TTP), causing undesired deviations and degradations of surface quality during finishing. In order to avoid these undesirable effects, the adopted materials must satisfy a set of requirements such as high static stiffness for bending and torsion, high value of elastic modulus, yield strength, good dynamic characteristics and dimensional stability [5].

These reasons have motivates researchers to study and design lightweight mobile parts of MTs, increasing stiffness and capacity in vibration damping guaranteeing the required precision.

Specially, the challenge is to identify the right and cost effective structural materials that are able to meet the requirements in terms of productivity, accuracy and eco-efficiency [6].

This paper aims to evaluate the static and dynamic performances of a set of materials used in MT structure fabrication. In particular, this work focuses on the capacity of a mobile part to guarantee high-speed operations reducing undesired effects. In fact, high cutting and transfer speeds represent essential requisites to satisfy productivity and sustain competition. Nevertheless, while recent cutting tool materials have enabled an increment of the cutting speed, the productivity remains still limited due to a low transmission speed of moving parts, generally made of cast iron. In contrast with machine's conventional steel moving frames, which work with speed in a range of 0.3–0.9 m/s and speedup of 0.3–4.2 m/s^2, the modern machines need to manage accelerations of 14 m/ s^2 and speeds from 3 m/s [7]. The utilization of massive steel may generate undesired vibrations in high transfer speed conditions, compromising seriously the quality of the workpiece [8–9]. In this way, hybrid materials that are able to damp mechanical vibrations may be preferred in manufacturing machine tool structural parts.

Vibrations are undesired effects that directly influence the quality of a workpiece. Their undesirable effects are usually considered in the design phase of a MT, analyzing the structure conception in terms of mass reduction, stiffness increase and material selection. In MTs, three distinctive types of vibrations are noted and classified: forced, self-generative and free vibrations [10,11]. The first type of vibrations is generated by periodically forces and arising within the machine. These vibrations may be originated by multiple events as intermittent cutting, unbalanced rotating components, spindle or gears wear-out issues or unstable effects. State of art shows a number of solutions that permits to limit this effect (e.g. active vibration control - AVC systems, mechatronic model-based on compensation). The second class of vibrations - self-generative - may occur as a result of some process in the machine. In particular, one of the main undesirable dynamic process phenomenon is the machine tool chatter. It is generated by the interaction between the machine structure and metal cutting process. These vibrations have frequencies in a range from 200 to 1,000 Hz. This undesirable effect may be mitigated by optimizing spindle speed or depth of cut. Alternative solutions are the use and integration of active structural control systems to alter dynamics by installing actuators or sensors in closed-loop. The last group of vibrations is related to free vibrations. These vibrations may be originated by material imperfections, inertia forces or shocks from MT basements. They may occur during strong accelerations, when inertial forces generate oscillations impacting on geometrical errors of the workpiece. In this case, the MT structure design based on stiff and light-weight materials may improve the machining performance, limiting undesirable vibration issues.

From a practical point of view, all these vibrations have equal importance on workpiece quality, machining performance and accuracy. The state of art presents a set of strategies that may be adopted to mitigate the vibration effects. These techniques aims to compensate vibrations using passive or active control strategies. Table 1 summarizes a set of studies that suggest different approaches in MT vibration control and mitigation.

This study deals with the importance of the material selection in MT structure building in order to increase damping performance without sacrificing the stiffness. A performance comparison of a set of hybrid materials is presented. The selected materials are Aluminium Foam Sandwiches (AFS), in closed cells configuration, Corrugated-Core Sandwich panels and Fiber reinforced (FR) materials. These alternatives are also evaluated considering the conventional materials (e.g. cast iron or steel), usually adopted in MT structure fabrication.

Metal Foams represent an innovative category of materials. They are characterized by a low density with good shear and fracture strength. Metal Foams guarantee both lightweight and stiff structures with low costs [12,13]. These performances are reached through the novel mechanical, thermal and physical proprieties.

Corrugated-Core Sandwiches are another valid alternative to guarantee bending stiffness and strength when minimal mass is required. These steel sandwich panels are lightweight structural offering 30-40 % of weight savings compared with the conventional structures. The lightness of the

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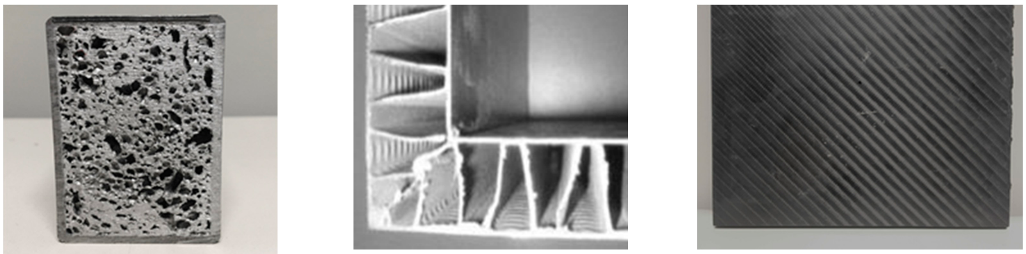
panels is achieved through the layer build-up, that permits to increment the wall thicknesses of the structure. They have a broad range of applications, in particular they are suitable to improve the stiffness and reliability of constructions absorbing energy during blasts and impacts [14-16].

Table 1. Material comparison: structural index versus damping coefficient.

Vibration Damping	Characteristics	References
Active methods	• Use of additional devices such as actuators • Use of advanced and complex control algorithms • Knowledge of vibrations' eigen-frequencies • Model-based strategy	[1], [19], [20], [21], [22], [23], [24], [25]
Passive methods	• Based on viscoelastic materials, viscous fluids, magnetic or passive piezoelectric, lightweight materials • Vibration energy dissipation or redirection • Cost effective • Dampers are usually in small size and ease to be installed	[26], [27], [28], [29], [30], [31], [32], [33], [34]

The last considered category is represented by composite materials reinforced by carbon fibres (CFRP). These materials are particularly interesting for applications in machine tool structures due to their ratio of mechanical strength to density [35,36]. CFRP materials consist of a binding matrix system and a reinforcement of fibers or particles.



(a) (b) (c)

Figure 1. Lightweight MT structure materials: Al Foams Sandwiches (AFS) (**a**), Al Corrugated-Core Sandwiches (**b**) and composite materials reinforced by carbon fibres - CFRP (**c**).

The selected CFRP materials have fibers with diameter of 4–10 μm. The matrix is made of either of polymer resin, epoxy and polyimide. The characteristics of fibers (strength, orientation, length) and matrix (volumetric content, layer) define the mechanical properties of the CFRP materials. In particular, the composites reinforced by unidirectional fibers have the highest mechanical performance in fiber direction. This point needs to be considered in the structural design.

2. Materials and Methods

In order to guarantee the machining performance requirements, a strong mass reduction and a high increment of stiffness of MT moving parts are becoming essential properties to satisfy the damping and maintain the thermal stability of the MT structures. Typical examples of machine structures are the mobile parts of milling machines (e.g. ram), grinding machines or coordinates measuring machines where the position accuracy and machining precision are strictly required. Usually, the conventional materials applied for the MT construction are welded steel, cast iron and, sometimes, Al alloys. Although these materials represent a very consolidated technology, the need to achieve high performances pushes to investigate new class of materials able to assure of the geometric configuration of the machine elements even under static, dynamic and thermal loads.

The strategy to select the most suitable materials is based on the analysis of a set of parameters. These parameters focus on structural and damping indicators [37]. The main structural index is described by equation (1) as follows:

$$C = E^{1/3} / \rho, \tag{1}$$

where E is the Young's module and ρ is the material density. This indicator links the material mass and stiffness. In this way, the weight is reduced and the stiffness is increased by selecting materials with high values of structural indexes. A further indicator is the loss factor (damping coefficient) that represents the attitude of material to damp vibrations [38]. Table 2 lists the main values of these indicators a comparison between the main materials, conventional and novel, adopted for MT moving part fabrication.

Table 2. Material comparison: structural index versus damping coefficient.

Material	$E^{1/3}/\rho$ [GPa$^{1/3}$/(Mg/m^3)]	η
Cast iron	0.63	$1.2\times10^{-3}\div1.7\times10^{-3}$
Steel	0.77	$6.0\times10^{-4}\div1.0\times10^{-3}$
Al alloys	1.50	$2.0\times10^{-4}\div4.0\times10^{-4}$
Mg alloys	1.90	$1.0\times10^{-3}\div1.0\times10^{-2}$
Al Corrug. Sandw.	2.52	$4.0\times10^{-3}\div1.0\times10^{-2}$
Al Foams	2.67	$1.0\times10^{-4}\div1.0\times10^{-3}$
CFRP (unidir)	4.00	$1.5\times10^{-3}\div3.0\times10^{-3}$

This preliminary investigation shows that the CFRP materials have the highest structural index. In the same way, Aluminium Foams highlight good structural and damping proprieties and they may be a further suitable solution to satisfy the structure requirements. Table 2 underlines also that Mg alloys have the highest value of loss factor, nevertheless these materials are expensive due to the complexity of their fabrication processes.

By starting from these considerations, this study aims at evaluating the static and dynamic properties of a number of materials, that may be selected in MT moving part fabrications. The material properties have been analyzed focusing both on a theoretical model and on experimental testes performed on a set of prototypes. These prototypes reproduced the ram (Z axis) of a commercial milling machine. All the presented solutions are a great challenge and they may open interesting perspectives for MT applications.

2.1. Analysis of vibration damping in machine tools

It is useful to study the static and dynamic behaviour of the machine structures using a parametric model. This model is evaluated in a simplified form to provide a better understanding of experimental tests [1]. In particular, it is suitable to represent the kinematics of a number of machines, such as gantry or travelling column machines (e.g. milling or boring machines). As defined by Apprich et al. [39], this model depends on movement, position and type of machining. For this reason, the main parameters need to adjusted in-real time in order to compensate the vibrations effects. Nevertheless, the kinematic representation of a general machine permits to understand the main critical variables to be evaluated in designing an effective lightweight structure.

The presented kinematic model is developed on the machine configuration described by Figure 2. It is assumed that the machine may be represented as two rigid bodies: body A, that is the MT moveable frame, and body B, that represents the Z-axis ram.

Body A, with the mass equal to m_1, has 3 rotational degrees of freedom (DOFs) constrained to the basement. These DOFs permit to evaluate its natural frequencies. In this way, a simple finite segment method (FSM) may model the stiffness using one rigid mass and 3 springs for each degree

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of freedom. This approach is based on modelling a flexible body as a finite number of rigid elements that are linked by dampers and springs. It needs of the use of the rigid multibody formulations. The spring s_1 defines the bending in Z-axis, s_2 describes the torsion around the Y-axis while s_3 shows the inflection in X-axis. The dampers d_i (i = 1, 2, 3) regulate the resonance, without influence the natural frequencies of the body. It is assumed that the translational movement of body A is not permitted.

Body B, with mass equal to m_2, is the ram of the machine tool. It has a transaction DOF in Z-axis that represents the rigid movement of the slide. It has also a DOF defined by s_3 and d_3 in Y-axis.

Assuming that a set of external forces (F_x, F_y, F_z) is applied to the machine ram (body B), the dynamical behaviour of the system may be evaluated applying the Newton-Euler equations for a general coordinate z = [α, β, γ, L_Y, L_Z], where L_Y and L_Z are the distance between the body's barycenters (in Y-Z plane).

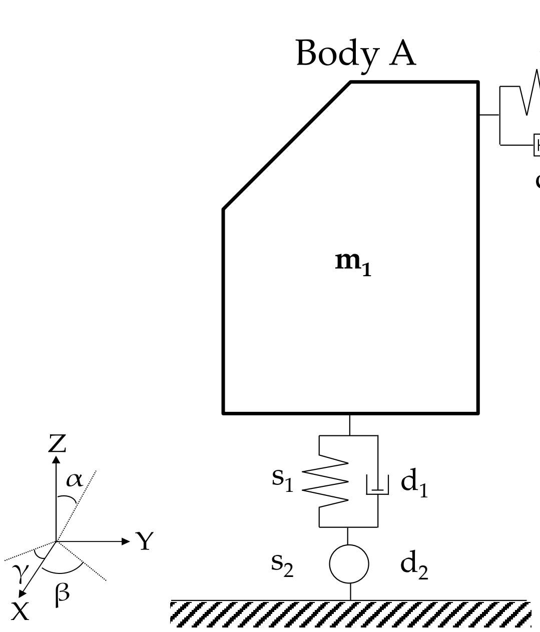

Figure 2. General machine kinematic model.

By using the D'Alembert's principle for dynamic equilibrium condition in the Lagrange's form, the multibody system is described by equations 2 and 3, as follows [39,40]:

$$M(u,t)\cdot\ddot{z}+k(u,\dot{u},t)=q(u,\dot{u},t), \tag{2}$$

$$\sum_{i=1}^{p}\left[J_{T_i}^{T}\cdot m_i\cdot J_{T_i}+J_{R_i}^{T}\cdot I_i\cdot J_{R_i}\right]\cdot\ddot{z}+\sum_{i=1}^{p}\left[J_{T_i}^{T}\cdot m_i\cdot\bar{a}_i+J_{R_i}^{T}\cdot I_i\cdot\bar{a}_{R_i}+J_{R_i}^{T}\cdot\tilde{\omega}_i\cdot I_i\cdot\omega_i\right]=\sum_{i=1}^{p}\left[J_{T_i}^{T}\cdot f_i+J_{R_i}^{T}\cdot h_i\right], \tag{3}$$

where:

- J_{R_i}, J_{T_i} : Jacobian matrixes of rotation and translation of the i-th body,
- I_i : Inertia matrix of the i-th body,
- $\bar{a}_i$: Acceleration of the i-th body,
- ω_i : Angular rate of the i-th body,
- $\tilde{\omega}_i$: Rotation matrix of the i-th body,
- f_i , h_i : External forces and moments of the i-th body,
- m: Mass matrix

In this way, a general coordinate *z* of the multibody system is expressed by a number of *f* differential equations, where the movement of the machine ram is represented by Inertia matrix and moments l_i, acting on the body. In particular, the model parameters (e.g. Jacobian matrixes, speeds,

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accelerations) depend on the position and they need to be calculated from theoretical considerations. However, it is also interesting to highlight that the active forces are not pose-dependent [36,41]. This parameter model may be useful both for the numerical studies (e.g. finite element analysis - FEA) and for the experimental campaigns (e.g. static and dynamic tests). In fact, it underlines those variables need to be evaluated to describe the machine dynamical behaviour and, as consequence, how to compensate the undesired effects of vibrations.

2.2. The lighweight structure prototypes

In order to compare the lightweight structures, a set of prototypes has been fabricated. The prototypes represented the Z-axis ram (300 x 1,500 mm) of a commercial milling machine and they were made of AFS, in close cell configuration, Corrugated-Core Sandwiches and CFRP material, respectively. Figure 3 shows an overview of the three prepared samples. The prototype dimensions have been designed satisfying the MT requirements, guaranteeing the first bending modes between 0 and 500 Hz, normally the excited range in cutting machining. The vertical axis prototypes have been equipped by commercial guideways reproducing the same characteristics of the final version of a machine tool element.

(a) **(b)** **(c)**

Figure 3. Lightweight MT structure prototypes: Al Metal Foams (**a**), Al Corrugated-Core Sandwiches (**b**) and composite materials reinforced by carbon fibres (**c**).

3. The stiffness analysis of the lightweight structures

3.1. Numerical Analysis of the structure stiffness

To investigate and compare the performance of the selected materials, the first phase was to study the structure behaviour when a static load was applied, using a set of simulations. A model was built by a finite element analysis (FEA) that is able to represent the mechanical properties of the structures. The study was executed on a ram of a commercial milling machine. In particular, the simulation wanted to compare the conventional material (e.g. steel) with the selected materials, simulating a set of external loads that may usually occur during the working operations. The analysis aimed to evaluate the static performance of the lightweight structures and to provide a preliminary assessment of the design of the prototypes. In fact, the simulations needed to consider the material mechanical properties, dimensions and weight. The kinematic behaviour was modeled by a commercial FE software.

Figure 4 illustrates an example of FE analysis performed on the Aluminium Metal Foam sandwich structure. A set of loads was applied at the TTP of the ram and the stiffness was calculated in X, Y and Z directions. The effect of gluing was not considered both for AFS sample and aluminum corrugated core sandwich sample. Table 3 lists the sample weight comparison while Table 4 highlights the results obtained from the FEA simulations.

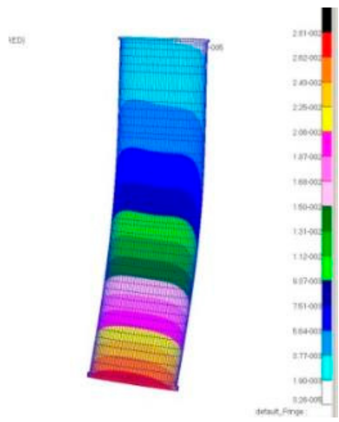
Table 3. The material mass comparison.

Configuration	Mass [kg]	Conventional Mass [kg]	Saving [%]
AFS ram	116.0	97.0	+19.6%
Al Corrug. ram	78.0	97.0	-19.5%
CFRP ram	50.0	97.0	-48.5%

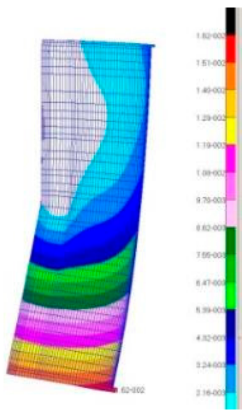
It is noted that all materials show a better performance than the conventional ram (steel) both in Y and Z direction. In particular, the AFS ram material shows the highest stiffness in Z direction, its performance is greater than the convention material (steel) by 152%. Nevertheless it is the heaviest structure, as shown in Table 3. This fact, that has been considered in the simulation, is due to the oversizing of sample ribs and flanges required for the prototype installation. The CFRP material underlines also significant properties, it has the best stiffness in Y direction with a significant saving of weight. Instead, Al Corrugated Sandwiches may have some issue in X direction, this weakness may be due to its structural configuration, however it guarantees high stiffness both in Z and Y axis.

Table 4. FE results: the material stiffness comparison.

Configuration	Kx [kg/µm]	Ky [kg/µm]	Kz [kg/µm]
Conventional (steel) RAM	2.2	3.4	22.0
AFS RAM	2.5	3.9	55.6
Al Corrug. RAM	1.9	4.4	42.8
CFRP RAM	2.7	5.1	39.8



(a)



(b)

Figure 4. FE model: the deformed shape in X-orientation (**a**) and in Y-orientation (**b**).

In any case, the FEA results needed to be validated and confirmed by a set of experimental tests. In fact, the characterization of samples has been a complex process since these materials have isotropic and stochastic features. Their characteristics may change in base on manufacturing processes or chemical composition.

3.2. Experimental test capaign: the stiffness analysis of the lightweight structures

An experimental campaign has been performed to validate the results of the simulations. As shown, the structure stiffness plays a crucial role to satisfy the MT requirements.

The material stiffness has been evaluated applying a set of external force in the X and Y direction, that are the most critical directions to guarantee the machining quality. The static dynamometer force was controlled by acting on a screw and avoiding peak or step stress. The corresponding deflections were measured at the tool tip point by a number of dial gauge sensors. Finally, the stiffness was quantified calculating the ratio between the force and the corresponding

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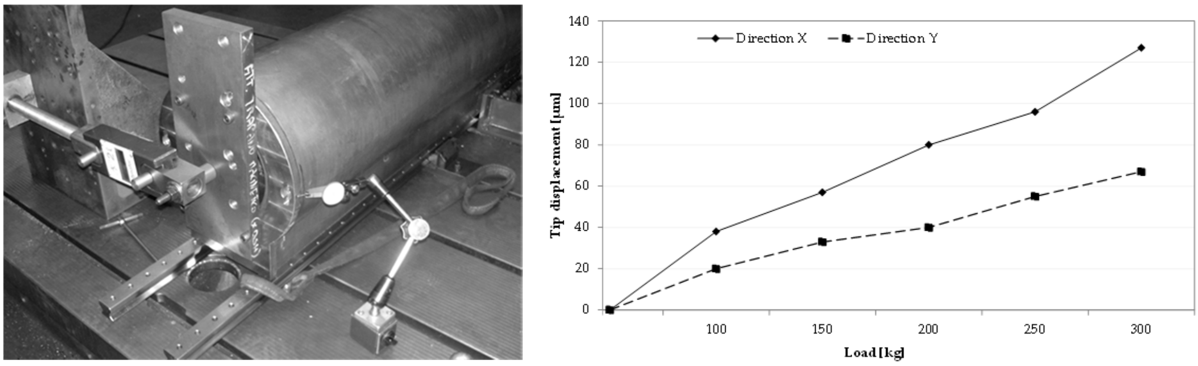
displacement at the application point. Figure 5(a) illustrates an overview of the test bench used to execute the static tests. The applied forces varied a specific range from 10 kg to 300 kg, reproducing the potential loads that occur during the machining.

In order to confirm the numerical analysis, a comparison between the FEA simulations and the experimental test results was evaluated. The FEA model validation is fundamental to develop further activities (e.g. modal analysis and dynamical structure behavior), as shown in the next sessions. Figure 5(b) highlights an example of the results obtained from the AFS ram. The stiffness value has been calculated and compared with the simulated results in X and Y directions. This comparison showed the substantial matching between the FEA model and the experiment tests, with an error of 6% in X direction and 2% in Y direction, as listed in Table 5. The AFS structure is the most critical ram to be characterized, in fact it has a complex internal configuration that depends of the prototype fabrication process.



(a) (b)

Figure 5. Test bench setup overview (**a**) and the AFS ram experimental results (**b**).

Table 5 summarizes the comparison between the lightweight structures and the conventional ram (steel). The results confirm the considerations highlighted from the FEA simulations. CFRP material shows the best performance in both directions (X-Y), while the Al Corrugated sandwich structure underlines some issues in X direction.

Table 5. Static test results: the material stiffness comparison.

Configuration	Direction	K [N/µm]	Steel RAM [N/µm]	Comparison
AFS ram	X	2.5	2.2	+13.6%
	Y	4.6	3.4	+35.3%
Al Corrug. ram	X	1.75	2.2	-20.5%
	Y	4.4	3.4	+29.4%
CFRP ram	X	2.3	2.2	+4.5%
	Y	5.5	3.4	+61.8%

This preliminary investigation shows that the selected lightweight materials are good candidates to be used in MT moving part fabrication. They have suitable mechanical properties to guarantee the machining performance in terms of stiffness. In particular, a significant saving of weight is noted both for CFRP and Al Corrugated sandwiches structures. The AFS material underlines the best tradeoff of stiffness in X-Y directions, nevertheless some limitations in terms of weight are observed due the oversizing of sample ribs and flanges required for the prototype installation.

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4. The modal analysis of the lightweight structures

4.1. Numerical Analysis of the structure dynamical behaviour

The materials of MT structures need to be selected with high dynamic characteristics and capacity to damp mechanical vibrations. In this way, a modal analysis is needed to study the dynamical characteristics of a structure under vibrational excitation.

The first step was to develop a numerical simulation in order to identify the main structural modes. By using experimental tests it was also interesting to calculate the structural loss factor (η) through the half-power bandwidth technique [42-45]. The damping characteristics of a viscoelastic material may be defined by the adjustable indicators T*(ω) [46-48], as shown in equation (3):

$$T''(\omega) = T'(\omega) + i \cdot T''(\omega) = T(\omega) \cdot (1 + i \cdot \eta(\omega)), \tag{3}$$

where T(ω) = T′(ω) is the storage value, T′′(ω) is the loss parameter and η(ω) is the loss quantity that is well-defined by the equation (4):

$$\eta(\omega) = \frac{T''(\omega)}{T'(\omega)}, \tag{4}$$

The storage value T(ω) and the loss quantity η(ω) are determined by the excitation range discretizing the viscoelastic to the other damping phenomena [49-51]. There are a couple of experimental methods to extrapolate the loss quantity η(ω). Usually, the half-power bandwidth procedure is a suitable approach to quantify damping. This method accepts the frequency response of a system with a single DOF. The loss quantity is calculate as the ratio between the frequency and the reaction that is reduced by 3.01 dB with respect to the max level at resonance rate. In case of multiple DOFs, the calculation is more complex and the use of further techniques in required [44].

The dynamical simulation has been performed on all lightweight structures and the obtained results have been validated through an experimental campaign. In order to understand the adopted approach, the complete study is only described for the AFS structure. The simulation results of the free-free modal analysis are presented in Table 6 and Figure 6. The main AFS ram modes have been calculated and compared with the experimental tests.

Table 6. Numerical modes and frequencies of the AFS ram.

N. Mode	FE Model Freq. [Hz]	Description
1	650	Breathing
2	677	Breathing and Bending Z-X
3	709	Breathing
4	782	Bending mode Z-Y
5	816	Torsion Z axis

4.2. Exerimental test campaign of the structure dynamical behaviour

To confirm the numerical analysis, a set of experimental modal analysis was executed using the "hammer test". This experimental campaign was executed on all structures (conventional and lightweight rams) in order to compare their dynamical properties. A test bench was equipped with a 4-channel FFT (Fast Fourier Transformation) instrumentation, an amplifier, an impulse hammer, an accelerometer and a force transducer. The prototypes were excited by a hammer in two different directions and positions in order to have a complete modal response.

Figure 7 illustrates the Frequency Response Function (FRF) of the AFS structure. The results show a substantial mismatching of the first modes between the numerical analysis and the experimental tests. In fact, the first two modes of the tests occur at 441 Hz and 483 Hz, while the FE simulation highlighted these modes at 650 Hz and 677 Hz. This fact is due to the constrains'

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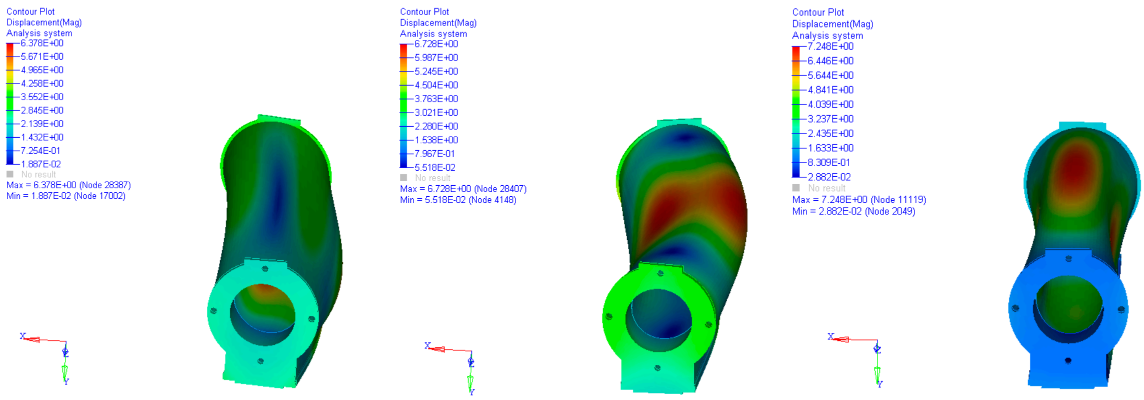
interaction with the support structure and they couldn't be considered in the simulation. In particular, the FRF diagram (Figure 7) describes the relationship between the displacement and the point of the applied load (center of ram). This analysis is useful to compare the vibration performance with the force contribution. It confirms that the first two modes of the experimental tests correspond to the breathing modes.



(a) **(b)** **(c)**

Figure 6. The FE modes 1–3 at 650 Hz (**a**), 677 Hz (**b**) and 709 Hz (**c**) performed on the AFS ram.

In order to evaluate the AFS structure damping, the most interesting modes are the bending in X-Z plane at 667 Hz, the bending in Y-Z plane at 758 Hz and the torsion at 816 Hz, as shown in the simulation. They appear with a limited pick, highlighting the high capacity of the material to damp the vibrations. The loss factor η has been calculated for each mode, underlining an important grow of the damping performance coinciding to inflection and torsional mode of the Z-axis. The AFS damping coefficient of the first modal frequency (bending) was equal to 1.7%.

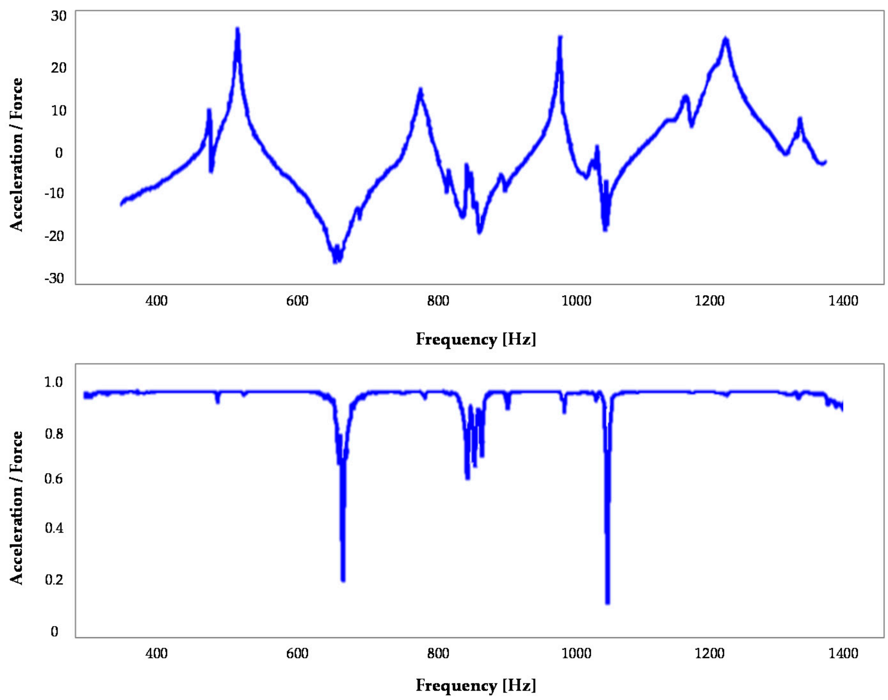


Figure 7. The Frequency Response Function (FRF) of the excited point (center of ram) in Y-direction.

In the same way, the study has been replicated for all prototypes. Table 7 lists the damping coefficients of the lightweight structures and the conventional ram. It is noted that the hybrid

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materials have significant dynamical properties to reduce the vibration effect. The AFS material has a damping capacity that is 20-30 times greater than a conventional structure. The CFRP structure has good damping properties and it achieves the first modal frequency close to 1200 Hz.

Table 7. The damping coefficients of the selected materials.

Configuration	1st Freq. [Hz]	Modal Damping [%]
Conventional (steel) RAM	670	0.08
AFS RAM	667	1.70
Al Corrug. RAM	745	0.17
CFRP RAM	1286	0.23

5. Discussion

This study aims to present and evaluate the static and dynamical properties of a set of hybrid materials that may be used to fabricate MT moving parts. As shown in Table 8, these materials are excellence candidates to substitute the conventional structures (e.g. cast iron or steel). In fact, they guarantee high mechanical properties and high dynamic characteristics to damp vibrations.

Table 8. The lightweight structural properties compared to a conventional structure.

Configuration	ΔKx	ΔKy	ΔKz	ΔMass	ΔDamping
Conventional (steel) RAM	0	0	0	0	0
AFS RAM	+ 11.4 %	+14.7 %	152.7%	+19.5%	20 times
Al Corrug. RAM	-13.6%	29.4%	94.5%	-19.6%	2 times
CFRP RAM	+22.7%	50.0%	80.9%	-48.5%	3 times

The experimental tests show that the AFS structure has a stiffness greater than the conventional ram stiffness of 152.7% in Z-axis, 11.4% in X-axis and 14.7% in Y-axis. This point guarantees to withstand the forces generated by hard machining and machine motions. This material has also a high capacity to damp the vibrations, in fact, the loss factor is 20 times higher than the factor of the other evaluated materials. The main limit is represented by the weight. In this study, the AFS prototype was the heaviest structure due to the particular configuration to be installed in the milling machine. Nevertheless, a design review may be evaluated to obtain a weight saving. The CFRP ram highlights also good mechanical characteristics in terms of stiffness and damping. However, the main advantage is the saving of weight that is lower than steel of 48.5%. The Al Corrugate-core sandwiches are a further alternative to fabricate the MT moving parts. This structure underlines a good trade-off between the mechanical properties and the cost of fabrication. Nevertheless, some issues of stiffness have been noted in X direction due to the sandwich configuration. A new study of the design may solve this problem. Figure 8 summaries the comparison between the lightweight structures and the conventional ram (steel).

6. Conclusions

This paper deals with the dynamical behaviour of the machine tool structures, focusing on the evaluation of the main properties of a set of lightweight materials. These materials may represent a valid alternative to conventional structures that are often unsuitable to satisfy the machining requirements. In fact, the modern machines need to satisfy high and strict requirements in terms of operation speed, precision, quality and capability to quickly remove high quantity of workpiece material. These performances are directly related to the materials used in MT construction.

Starting by the kinematic model of a milling machine that highlights the main factors impacting on the MT dynamical behaviour, the paper evaluates a number of prototypes made of Al Foam sandwiches, Al Corrugated sandwiches and CFRP material. The state of art shows that these materials are good candidates to be used in moving part fabrication. In fact, they have significant structural indexes and loss factors, as shown in Table 2. In order to compare the lightweight structures, a set of prototypes has been fabricated. The prototypes represented the Z-axis ram (300 x 1,500 mm) of a commercial milling machine. The static and dynamical properties of the selected materials have been evaluated using simulations and experimental tests. The first part of the study illustrates that the lightweight materials have good structural index. A set of loads was applied at the tool tip point of the prototypes and the stiffness was calculated in X, Y and Z directions. The obtained results confirm the simulations highlighting that the AFS structure has the highest stiffness, nevertheless some limits on the weight is noted due to the installation constraints. A design review may overpass this problem. The CFRP structure underlines a good stiffness and an important saving of weight.

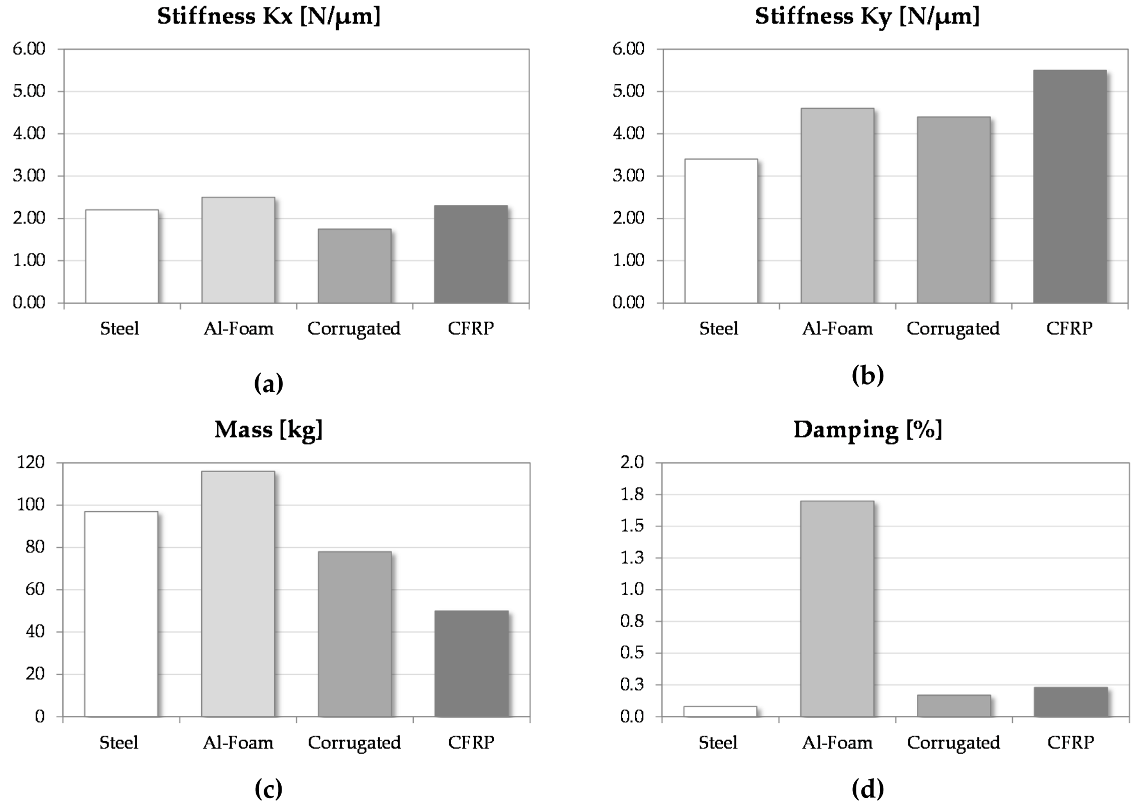

Figure 8. The result comparison: stiffness X-axis (a), stiffness Y-axis (b), Mass (c) and Damping (d).

The second part of the study focuses on the modal analysis. A set of simulations were executed to understand the main vibration modes of the lightweight structures. Then, an experimental campaign was executed through the "hammer test" in order to study the dynamical behaviour. The AFS ram shows a damping that is 20 times greater than the conventional ram. In the same way, the other hybrid materials highlight good capacity to damp vibrations. The obtained results illustrate that the selected materials may be an excellent alternative to conventional materials. Nevertheless, some limits are noted in terms of fabrication complexity and costs. For this reason, a design review of the prototypes is required to optimize the weight and, as consequence, the costs. The next steps of the research will be based on the selection of further hybrid materials to be used for fixed part application.

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