APPLICATION OF A MULTI TUNED MASS DISC DAMPERS SYSTEM TO A PORTAL MILLING MACHINE

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The dynamic performances of the machine tools have decisive influence on the achievable accuracy and productivity of the machining processes. For reducing the unfavourable vibrations of machine tools, a system of multi tuned mass disc dampers (MMD system) is developed in this paper. The simulation model of a portal milling machine is built on its CAD model. Thenceforth, the simulation model is parameterized based on the measured natural frequencies, vibration modes and frequency response functions (FRF) of the portal milling machine. Utilizing the parameterized simulation model, the MMD system is designed, the influence of the tuning errors of the dampers is analysed and the robustness of the MMD system is evaluated. The effectiveness and the robustness of the MMD system is validated through FRF measurements and cutting tests on the portal milling machine.

KEYWORDS

simulation, tuned mass disc damper, multi-mass dampers, impact hammer test, cutting test, large scale machine tool

1 INTRODUCTION

The dynamic properties of the machine tools are amongst the deciding factors when it comes to the machining accuracy and the productivity [Schmidt 2019]. Many engineers and researchers are committed to find methods to improve the dynamic properties of the machine tools, such as optimization of the machine tool structure, process parameter tuning, spindle speed variation, active damping techniques and passive damping techniques [Munoa 2016]. The machine tool structure can be optimized in the design phase to increase the stiffness, however the structure change of existing machine tools means usually high cost. The process parameters and the spindle speed can be tuned for problematic machining processes. However, this can be done to a certain extent and requires individual and iterative tuning, which leads to a considerable overhead. In order to cost-efficiently improve the dynamic behaviors of the existing machines, the passive damping techniques are especially well suited [Schmidt 2019] [Munoa 2016]. The tuned mass damper (TMD) is one of them [Brecher 2017] [Munoa 2016]. Frahm has proposed the original idea of the TMD [Frahm 1911]. The TMD was first applied as a chatter suppression method by Hahn [Hahn 1951]. Traditionally, TMD consists of a single absorber mass connected to the main system using a specific attachment stiffness and optimal damping. One natural frequency of the TMD must be tuned accurately to match the critical natural frequency of the machining system. There are two major drawbacks of the TMD. At first, the effectiveness of the TMD is influenced strongly by the accuracy of the frequency tuning. Furthermore, the damping effect of the TMD is mostly realized by using the viscous materials, which are very sensitive to the external temperature and degrade over time [Li 2017] d[Friend 2000]. With the aim of improving the dynamic properties of a portal milling machine, a system of multi tuned mass disc dampers (MMD system) is presented in this paper. The MMD system overcomes the two major drawbacks of the traditional TMD. Multiple dampers are used to build the MMD system, hence its effectiveness doesn't strongly depend on the tuning accuracy of a single tuned mass disc damper (TMDD). Furthermore, the damping effect of the TMDD is mainly generated through friction and collision between its metal components [Brecher 2020] [Brecher 2021], thus the MMD system composed of the multiple TMDDs is not sensitive to the external temperature.

This paper is organized as follows: Following this introduction section, section 2 demonstrates the modeling of the portal milling machine. Subsequently, the simulation model of the machine is validated using the results of the vibration measurements. Based on the simulation model, section 3 presents the design and the simulations of the MMD system. The effectiveness and the robustness of the MMD system is validated by impact hammer and cutting tests in section 4. Conclusions and outlooks are provided in the last section.

2 SIMULATION MODEL OF THE PORTAL MILLING MACHINE

The simulation model of the machine is fundamental to develop the MDD system. In this section, the simulation model is built on

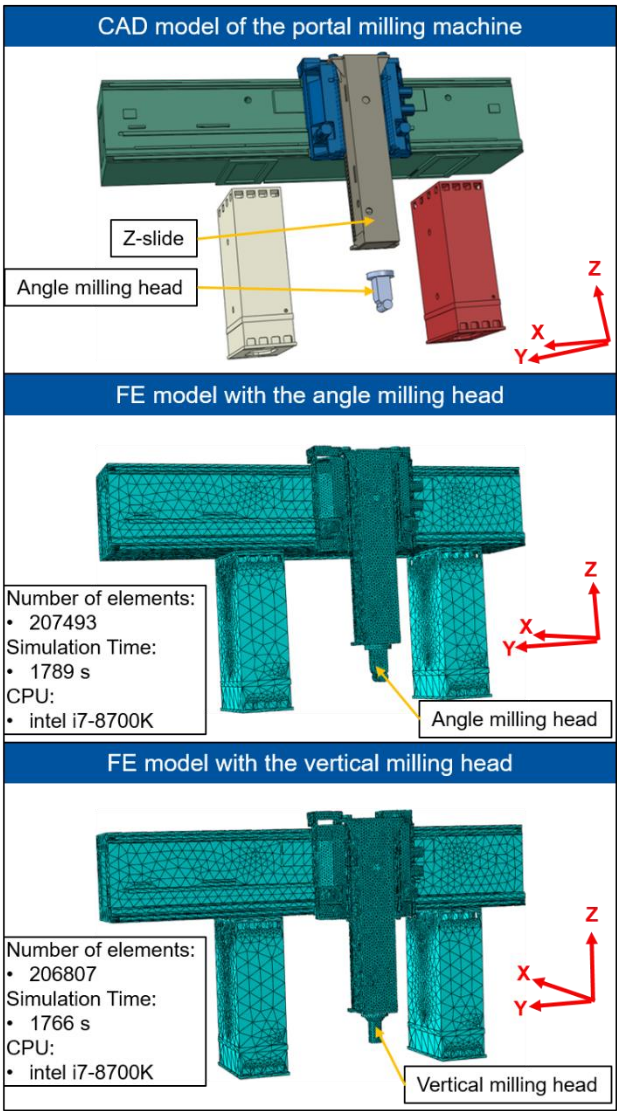

Figure 1. CAD model and simulation models of the portal milling machine

the CAD model of the Machine, thereafter the model is parametrized and validated using the results of the vibration measurements.

2.1 Modeling the portal milling machine

Fisrtly, the simulation model of the machine with the angle milling head is built on the CAD model (Fig. 1) of the machine and all the joints between the components of the machine are simulated as the bushing elements, which are parameterized after the vibration measurements of the machine.

2.2 Vibration measurements

Multiple frequency response functions (FRFs) are measured and an experimental modal analysis is conducted in this subsection. The smallest dynamic stiffness of the tool center point (TCP) occurs when the Z-slide reaches the lowest position. Accordingly the aforementioned measurements are carried out at the lowest Z-position. These measurements are necessary to parameterize and validate the simulation model of the portal milling machine.

Measurements of the FRFs

The setup for the measurements of the FRFs is shown in Fig. 2. Three FRFs (Gxx, Gyy and Gzz) of the tool center point (TCP) are measured. The dynamic exciting force is generated by the piezo exciter and the vibration signals are recorded by the accelerometer (acceleration signal) and the position sensor (position signal). The measured FRFs are presented in Fig. 3. The possible tendency of the machine to chatter can be recognized from the FRFs Gxx and Gyy. In addition to a necessary criterion of a phase drop below -90 degree, correspondingly weakly damped resonance peaks are necessary so that the manufacturing process can chatter. Furthermore, the differences between the FRFs measured by the accelerometer (on the dummy tool) and the position sensor (between the dummy tool and the support structure on the machine table) are very small, which means that the machine table has only a negligible influence on the dynamic properties of the machine. Thenceforth the machine table is not considered in the simulation model, and the FRFs can also be measured properly with the impulse hammer.

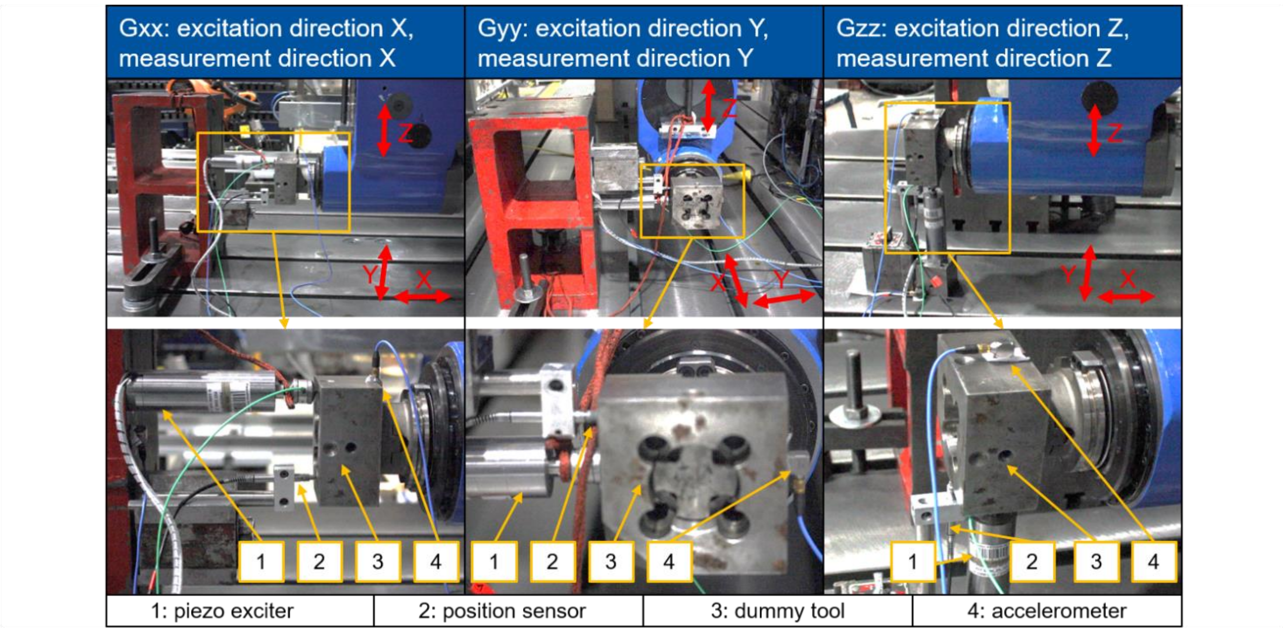

Figure 2. Setup for the measurements of the FRFs

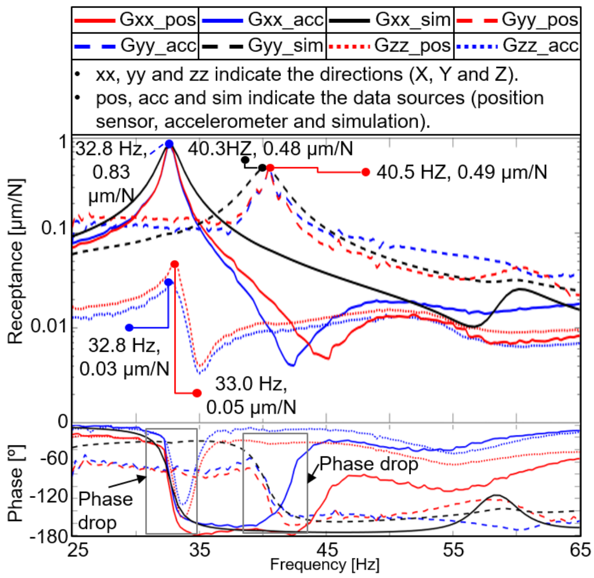

Figure 3. Measured and simulated FRFs of the portal milling machine with the angle milling head

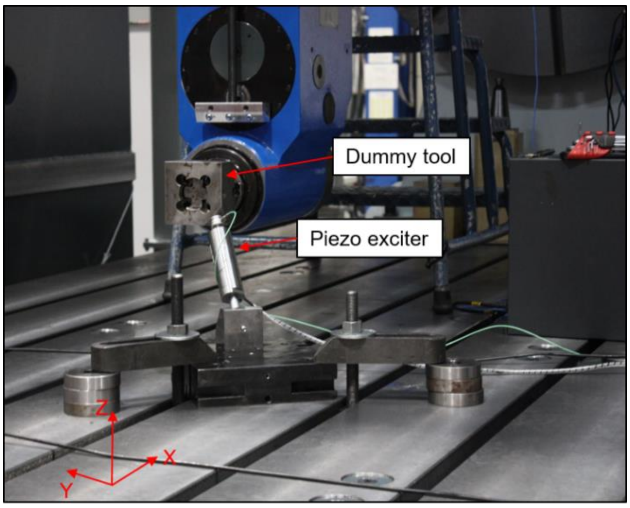

Figure 4. Setup for the EMA

Experimental modal analysis

Fig. 4 shows the setup for the experimental modal analysis (EMA). The machine coordinates are kept constant during the EMA. The portal milling machine is excited by the piezo exciter through the dummy tool, while the acceleration signals are measured at the 100 grid points (vertex points of the geometry in Fig. 5 and Fig. 6). Fig. 5 and 6 demonstrate the bending modes of the Z-slide in X and Y directions. The resonance points identified from the measured FRFs (Gxx: 32.8 Hz, Gyy: 40.5 Hz) can be clearly assigned to these two bending modes.

Results of the vibration measurements

These two bending modes of the Z-slide at the lowest position can be critical for the portal milling machine in some cases. Because of the lower dynamic stiffness and the stronger phase drop, the bending mode in X direction is possibly more critical than the bending mode in Y direction. This is the usual case for a machine with a long Z-slide. In order to research the effectiveness and the robustness of the MMD system, the MMD system is designed to damp the vibration in X direction at the lowest Z-position in this paper.

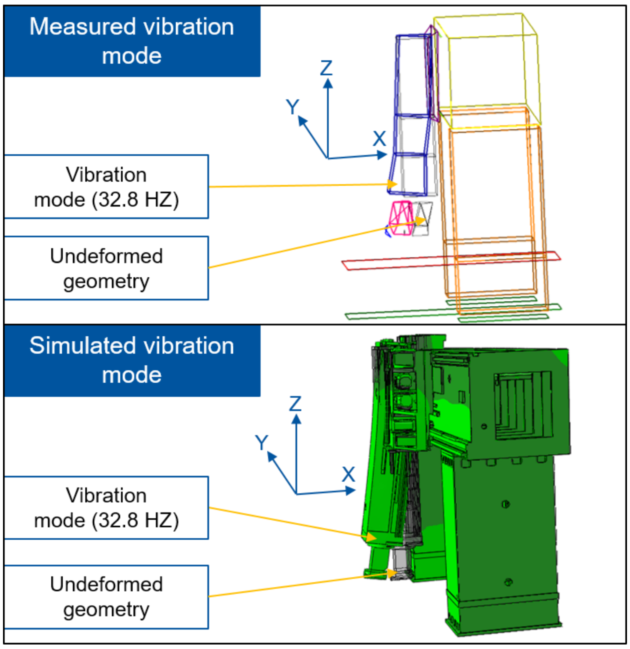

Figure 5. Bending mode of the Z-slide in X direction

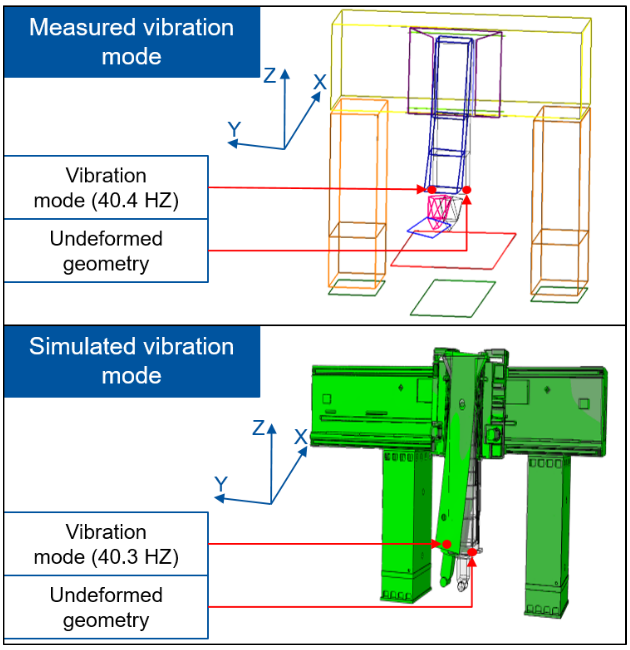

Figure 6. Bending mode of the Z-slide in Y direction

2.3 Validation of the simulation model

The bushing elements (joints) in the simulation model are tuned based on the measured results. After the tuning, the resonance frequency and the corresponding receptance identified from the simulated Gxx are identical with the measured results (Fig. 3). Furthermore, the simulated bending modes can clearly match the measured bending modes in Fig. 5 and 6. On account of these comparisons, the simulation model represents the dynamic properties of the machine with the angle milling head correctly. The machine is equipped with another vertical milling head, it can perhaps be more critical than the angle milling head considering the vibrations during the milling process due to the larger power. Because the vertical milling head was in maintenance during the measurements in section 2.2, the dynamic properties of the machine with this milling head can only be predicted by the corresponding simulation model (Fig. 1).

Utilizing the simulation model of the machine with the vertical milling head, a MMD system is designed to damp the vibration of the machine in the next section.

3 DESIGN AND SIMULATIONS OF THE MMD SYSTEM

In this section, a MMD system is designed utilizing the in last section validated simulation model. Furthermore, the effectiveness and the robustness of the MMD system are evaluated through simulations.

3.1 Design of the MMD system

The aim of this subsection is to design the MMD system to reduce the vibration of the portal milling machine in X direction. As shown in Fig. 7, the basic concept is to use multiple groups of dampers to damp the critical vibration of the machine in a wide frequency range, and each group consists of multiple dampers [Brecher 2015] [Brecher 2016] [Schmidt 2019].

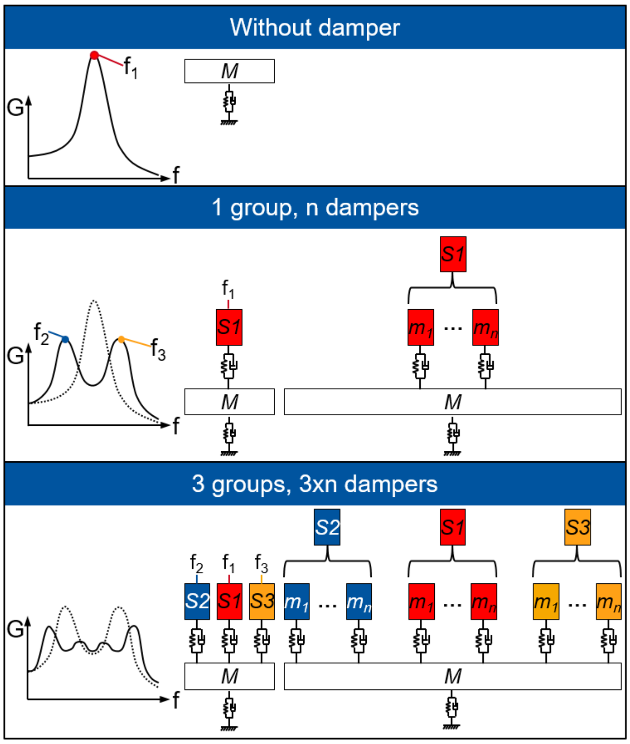

Figure 7. Basic concept of the MMD system [Schmidt 2015] [Schmidt 2019]

The TMDDs (Fig. 8) presented in [Brecher 2020] [Brecher 2021] are used as the single dampers to compose the MMD system. Each of the TMDDs is attached to the primary structure through a magnetic foot with a M6 threaded hole, a M6 screw connects to the magnetic foot and works as a cantilever beam. The identical metal discs are installed on the M6 screw, the axial positions of the metal discs along the M6 screw are adjusted by changing the number of the front and back washers. The washers are not pulled together by force, thus the clearances between the screw, the washers and the metal discs are maintained after assembling the TMDD. These clearances are important for the damping effect of the TMDD, which is generated by collision and friction between the metal discs and the M6 screw. The natural frequency of the TMDD depends on the total mass and the axial positions of the metal discs. [Brecher 2020]

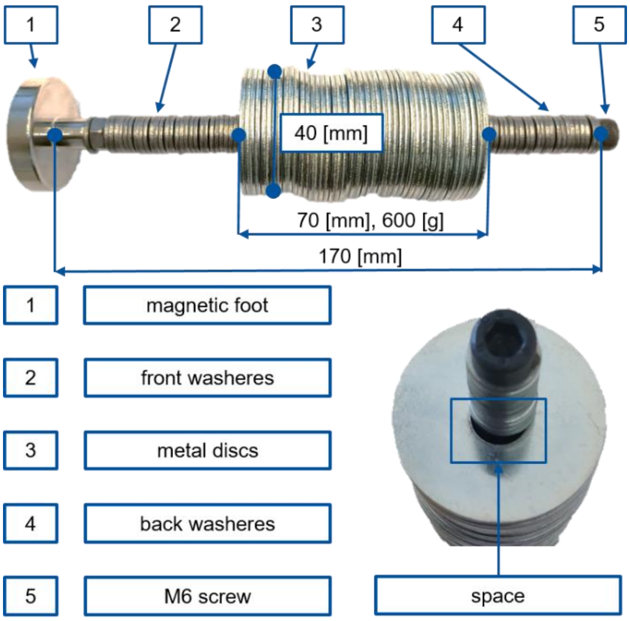

Figure 8. Structure of the TMDD [Brecher 2020] [Brecher 2021]

Firstly, the total mass m_{total_MMD} and the total number of the damper groups n_{total_groups} of the MMD system are defined. The mass ratio

$$\mu = \frac{m_{total\_MMD}}{m_{eff}} \quad (1)$$

is used to determine the m_{total_MMD}, where the m_{eff} is the mass of the single mass system reduced from the simulation model of the machine. The m_{eff} can be calculated accurately only after defining the m_{total_MMD} and the n_{total_groups}, hence the total mass of the Z-slide and the vertical milling head 2841 kg (extracted from the simulation model) is used as an approximation of the m_{eff}. According to the measured data in [Brecher 2016] [Schmidt 2019], a properly designed MMD system with $\mu = 6.5\ \%$ and $n_{total\_groups} = 7$ is able to damp the targeted primary structure in a wide frequency band [Brecher 2016] [Schmidt 2019]. Therefore these two parameters are selected for the MMD design, the derived m_{total_MMD} is 184.7 kg (eq. 1). The vibration mass of a TMDD is 0.6 kg (Fig. 8), thus the total number of the TMDDs is $\frac{184.7\ \text{kg}}{0.6\ \text{kg}} \approx 308$, each damper group has $\frac{308}{7} = 44$ TMDDs. With the aim of reducing the computation cost of the upcoming simulations, two TMDDs are reduced to a single mass damper with the effective direction X as shown in Fig. 9 (V3). With this reduction, each damper group has 22 single mass dampers. The mass of each single mass damper is 1.2 kg. The damping value of the bushing element in X direction is 0.02 Ns/m, because the damping value of the single TMDD in X direction is 0.01 Ns/m, which is determined by the half power method [Thomson 1993]. The stiffness of the bushing element in X direction depends the corresponding tuning frequency. The damping values and stiffness in other directions are neglected. The diameter of the metal discs used to compose the TMDDs is 40 mm (Fig. 8), which means the minimum distance between the centre of two mounted TMDDs is 40 mm. On account of that the minimum distance between the mounting points of two single mass dampers is considered as 80 mm.

The second step is to determine the potential mounting surfaces for the MMD system and to calculate the number of the mountable TMDDs on the surfaces. The vibrations of the machine are critical in X and Y directions. The most critical direction depends on the real cutting conditions. However the Z-slide has smaller dynamic stiffness in X direction than in Y direction. For this reason, the TMDDs are designed to be installed closing to the TCP, and their effective directions are required to be in X direction of the machine. Considering the aforementioned requirements, the surfaces F1 to F4 are the optimal surfaces to mount the MMD system (Fig. 9). The 7 groups of the attachment points are schematically demonstrated in Fig. 9, each group of the first 6 groups (P1 to P6) includes 22 attachment points, they are equally divided in the left side (V1) and right side (V2) of the Z-slide. As a result of the space limitation, the seventh group P7 has only 17 attachment points. Hence the total

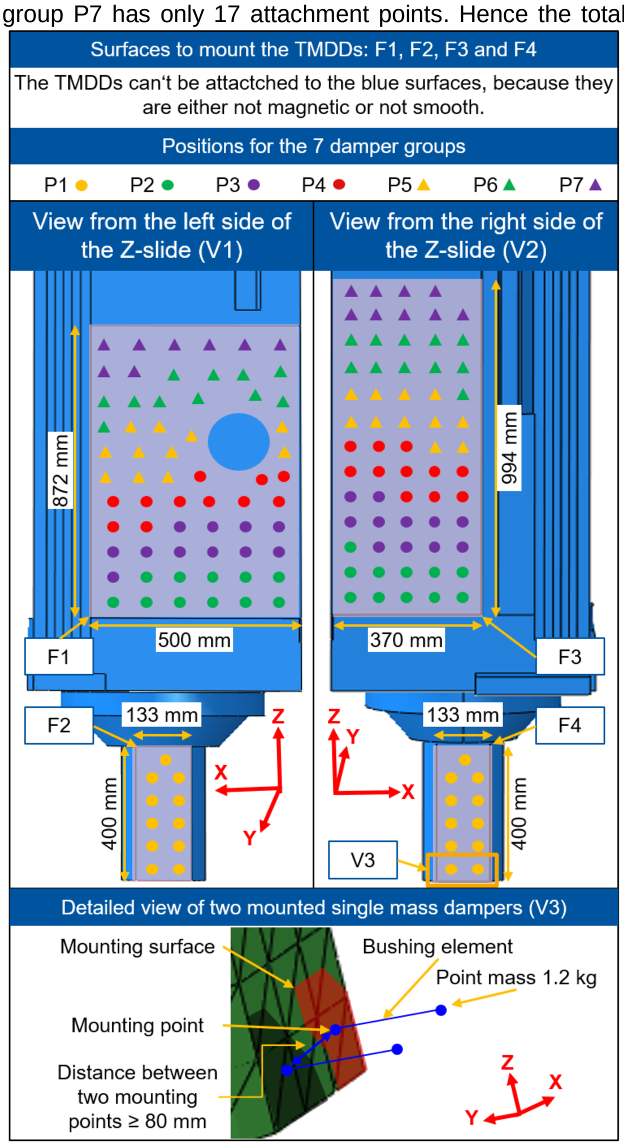

Figure 9. Detailed specifications of surfaces for MMD system mounting

number of the attachment points is 149. The simulation model is reduced to 149 single mass systems for the 149 attachment points of the TMDDs applying the pseudo kinetic energy method [Brecher 2016] [Schmidt 2019]:

$$E_{PK,s} = \frac{1}{2}\sum_{e=1}^{E} m_e^{el}\left(\vec{\dot{u}}_{e,s}^{el}\right)^2 \tag{2}$$

$$M_{eff,k,s}^{R} = E_{PK,s}/\left(\frac{1}{2}\left(\dot{u}_{R,k}\right)^2\right) \tag{3}$$

with

s: vibration mode number

$E_{PK,s}$: pseudo kinetic energy of the vibration mode s

e: index of the finite element

E: total number of the finite elements

m_e^{el}: mass of the finite element e

$M_{eff,k,s}^{R}$: effective mass of the mode s

k: index of the damper mounting point

$\dot{u}_{R,k}$: velocity of the damper mounting point k

The critical vibration of the machine is mainly caused by the bending vibration mode in X direction (32.2 Hz), hence this vibration mode is selected to determine the effective mass for the 149 mounting points. The 149 effective mass are not identical, because the damper mounting points have different velocities. In order to simplify the calculations of the required tuning frequencies for the single mass dampers, the effective mass of the mounting points in each group (P1 to P7) are assumed to be the average effective mass of this group (third column of Tab. 1). The tuning frequency of the first group is selected to be the natural frequency of the bending mode in X direction (32.2 Hz). The natural frequencies of the machine with the MMD system are predicted using the method presented in [Den Hartog 1985], [Brecher 2016] and [Schmidt 2019]:

$$f_{i\_L} = f_{sys}\left(\left(1+\frac{\mu_{group}}{2}\right) - \sqrt{\mu_{group}+\frac{\mu_{group}^{2}}{4}}\right)^{\frac{i\_L}{2}} \tag{4}$$

$$f_{i\_U} = f_{sys}\left(\left(1+\frac{\mu_{group}}{2}\right) + \sqrt{\mu_{group}+\frac{\mu_{group}^{2}}{4}}\right)^{\frac{i\_U}{2}} \tag{5}$$

with

f_{sys}: natural frequency of the machine without MMD system

μ_{group}: ratio of the mass of one damper group and the corresponding effective mass of the machine

i_L: number of the mounted damper groups with the tuning frequencies lower than f_{sys}

i_U: number of the mounted damper groups with the tuning frequencies higher than f_{sys}

f_{i_L}: new resonance frequency lower than f_{sys}

f_{i_U}: new resonance frequency higher than f_{sys}

The optimal tuning frequencies of the TMDDs are derived using the H_∞ method [Den Hartog 1985]:

$$f_{opt} = \frac{f}{1+\mu_{group}} \tag{6}$$

with

f_{opt}: optimal tuning frequency of the TMDD

f: targeted resonance frequency of the machine (f_{i_L} and f_{i_U})

The fourth column of Tab. 1 lists the optimal tuning frequencies of the TMDDs. The corresponding nominal effective frequency band f_{db} is $38.3\ \text{Hz} - 25.8\ \text{Hz} = 12.5\ \text{Hz}$. Based on these calculated tuning frequencies, the suitable TMDDs are chosen and their nominal tuning frequencies are listed in the fifth column of Tab. 1. The stiffness of the bushing elements in X direction of the single mass dampers are derived from the tuning frequencies [Den Hartog 1985]:

$$\mathrm{k} = \mathrm{m}(2\pi f)^2 \tag{7}$$

3.2 Simulations of the MMD system

Before installing the MMD system on the machine, its effectiveness and robustness are investigated in this subsection by simulations.

Simulations considering the effectiveness of the MMD system

Because the vibration in Y direction is not considered for the design of the MMD system, only the simulated Gxx of the vertical milling head without the MMD system is shown in Fig. 10 (legend: without MMDs). It's slightly different from the simulated results of the machine with the angle milling head (natural frequency is 0.6 Hz lower and the receptance at the resonance frequency is 0.02 µm/N lower).

The simulated FRF in X direction with the selected TMDDs (column 5 in Tab. 1) is shown in Fig. 10. The maximum receptance is reduced from 0.805 µm/N by 79 % to 0.171 µm/N. This result gives the impression, that the designed MMD system is promising to reduce the vibrations. However, the tuning errors Δ of the TMDDs can't be avoided, hence the natural frequencies of the TMDDs in each damper group distributing around its nominal natural frequencies. In this paper, the maximum tuning error 5.7 Hz is considered [Brecher 2021], the natural frequencies of the TMDDs in each group are assumed to distribute equidistant from 5.7 Hz lower than its nominal natural frequency to 5.7 Hz higher than its nominal natural frequency. As a result of that, the actual natural frequencies of all the TMDDs are in a wider frequency range than the designed nominal frequency band f_{db} 12.5 Hz. Consequently, it is possible, that the vibration of the machine around its natural frequency is only weakly damped. In order to avoid this problem, the designed nominal effective frequency band is scaled to two smaller bands (10.75 Hz and 9 Hz). Accordingly, the tuning frequencies of the 7 damper groups are modified and the suitable TMDDs are selected as listed in Tab. 1 (columns 6 and 7). The simulated FRFs with different nominal effective frequency bands f_{db} considering the tuning errors Δ are compared in Fig. 10. The reduction of the maximum receptance keeps constant (-79 %) after considering the tuning errors and scaling the nominal effective frequency band of the MMD system. However, the distance between the new critical frequency and the original natural frequency of the machine (32.2 Hz) decreases from

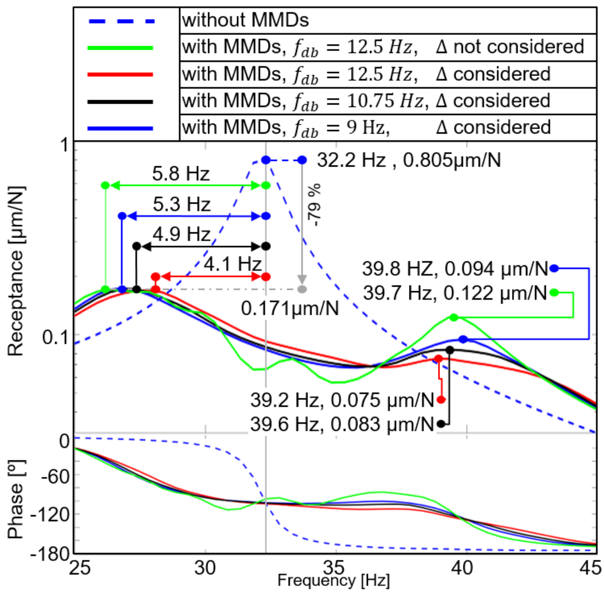

Figure 10. Influence of the tuning errors of the TMDDs

Position group	Damper group	Effective mass [kg]	Calculated tunning frequency [Hz] (frequency band 12.5 [Hz])	Nominal tunning frequency of the TMDD [Hz] (frequency band 12.5 [Hz])	Nominal tunning frequency of the TMDD [Hz] (frequency band 10.75 [Hz])	Nominal tunning frequency of the TMDD [Hz] (frequency band 9 [Hz])
P1	S1	865.3	32.2	31.9	31.9	31.9
P2	S2	1554.2	34.5	34.3	33.7	33.1
P3	S3	1900.1	29.1	29.2	30.2	30.8
P4	S4	2213.6	36.4	36.3	35.6	34.3
P5	S5	2890.0	27.2	27.3	28.2	29.2
P6	S6	3462.8	38.3	38.3	37.6	36.9
P7	S7	5058.0	25.8	25.6	26.8	27.8

Table 1. Tuning frequencies of the TMDDs

5.8 Hz to 4.1 Hz after considering the tuning errors Δ using the MMD system with the 12.5 Hz nominal effective frequency band. This distance increases by decreasing the nominal effective frequency band of the MMD system from 12.5 Hz to 9 Hz. Therefore, the tuning errors have negative influence on the damping effect of the MMD system, and this influence can be mitigated by scaling the original nominal effective frequency band f_{db} to a smaller band. Thus the TMDDs listed in column 7 (f_{db} 9 Hz) of Tab. 1 are used to compose the MMD system.

Simulations considering the robustness of the MMD system

The real natural frequency of the machine can be different from the simulated value 32.2 Hz because of several reasons, such as simulation errors, change of components and varying vertical position of the Z-slide. Thus it's necessary to examine the robustness of the designed MMD system before assembling the TMDDs and installing the system on the machine. Because the nominal tuning frequencies of the TMDDs of the MMD system vary from 27.8 Hz to 36.9 Hz (column 7 of Tab. 1), it's intuitive to expect, that the machine with the natural frequency in this band can be damped adequately. This assumption is proved by the following simulations. Firstly, the simulation model of the machine is adjusted to shift the natural frequency f_{sys} from 32.2 Hz to 27.8 Hz and 36.9 Hz, then the MMD system is added to the simulation model and the simulations are conducted. The simulated FRFs are shown

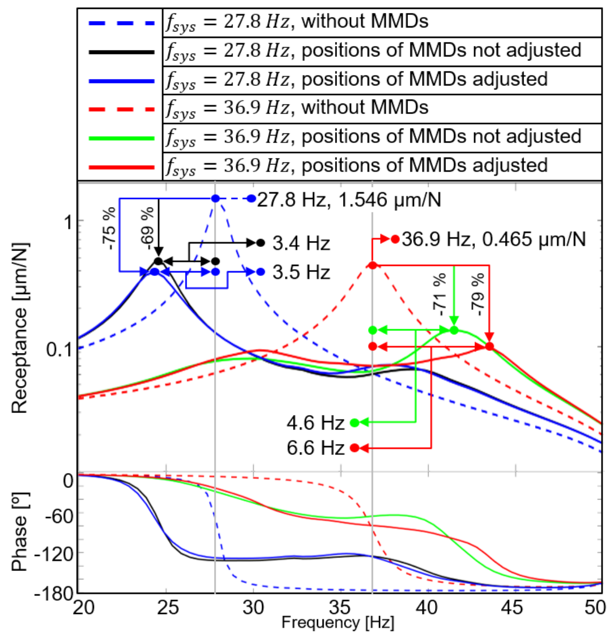

Figure 11. Robustness of the MMD system

in Fig. 11. For the simulation model with the natural frequency 27.8 Hz, the maximum receptance is reduced by 69 % when the MMD system is positioned as listed in column 1 and 2 in Tab. 1. Through optimising the positioning of the TMDDs (P1:S7, P2:S5, P3:S3, P4:S1, P5:S2, P6:S4, P7:S6), the reduction reaches 75%. For the simulation model with the natural frequency 36.9 Hz, the maximum receptance is reduced by 71 % when the MMD system is positioned as listed in column 1 and 2 in Tab. 1. Through optimising the positioning of the TMDDs (P1:S6, P2:S4, P3:S2, P4:S1, P5:S3, P6:S5, P7:S7), the reduction reaches 79 %. It's clearly to be identified, that the designed MMD system is robust again the variation of the natural frequency between 27.8 Hz and 36.9 Hz.

4 EXPERIMENTAL VALIDATION

After the evaluation of the designed MMD system using simulations in the last section, the effectiveness and robustness of the MMD system is experimentally proved by the measurements of the FRFs and the cutting tests in this section.

4.1 Measurements of the FRFs

The FRFs are measured by the impact hammer tests (Fig. 12, front view). The resonance frequency of the machine without the MMD system is 36.3 Hz (Fig. 13), it's near the upper frequency limit of the MMD system 36.9 Hz (4.1 Hz higher than the simulated value 32.2 Hz because of the higher Z-slide position). In order to generate a better damping effect, the MMD system is mounted as demonstrated in Fig. 12, which is identical to the optimized positioning of the MMD system for the simulation model with the natural frequency 36.9 Hz in the last section.

Three measured FRFs with the TMDDs are demonstrated in Fig. 13. The dynamic stiffness of X direction increases through using the TMDDs. If the entire MMD system (S1 to S7) is used, the maximum receptance is reduced by 71 %. The phase drop is flattened by using the MMD system. Without the MMD system, the phase angle is greater than -90 ° up to 36.4 Hz. If the entire MMD system (S1 to S7) is installed, the phase angle is not significantly smaller than -90 ° up to 40.5 Hz. The frequency range for absolutely stable milling is increased. The results show the potential of the designed MMD system to improve the dynamic performances of the machine.

4.2 Cutting Tests

Fig. 14 presents the setup for the cutting tests. The machining processes are usually stable. In order to generate chatter in X direction at the lowest Z-slide position, the aggressive process parameters are used for the 8 cutting tests C1 to C8 listed in Fig. 15 (with other process parameters, the machine can perhaps chatter both in X and Y directions at other Z-slide position). They are divided into two groups (4 cutting tests for each). The Z machine coordinate is kept constant for all the cutting tests. The identical process parameters are used in each of the two groups (Fig. 15). Within a group, only the configuration of the MMD system is varied. Chatter occurs at the chatter frequency 39.1 Hz during the test C1 (without damper), the maximum dynamic displacement of the tool side Umax is 550.1 µm. During the test C2 (damper group S6), the machining process is already significantly more stable. Umax is reduced by 96.5 % to 19.1 µm. During the test C3, Umax drops further by 44.0 % from 19.1 µm to 10.7 µm because the damper group S4 is also used. C4 shows a reduction of Umax from 10.7 µm by 33.6 % to 7.1 µm through the use of the entire MMD system (S1 to S7). The frequency corresponding the Umax from C2 to C4 is the tool passing frequency 35.9 Hz, which means the externally excited vibrations dominate the vibration pattern.

MMD position	P1 ●	P2 ●	P3 ●	P4 ●	P5 ▲	P6 ▲	P7 ▲
MMD group	S6	S4	S2	S1	S3	S5	S7
Nominal tuning frequency [Hz]	36.9	34.3	33.1	31.9	30.8	29.2	27.8
Length of the front washers [mm] (Fig. 8)	45	51	54	57	60	64.5	69

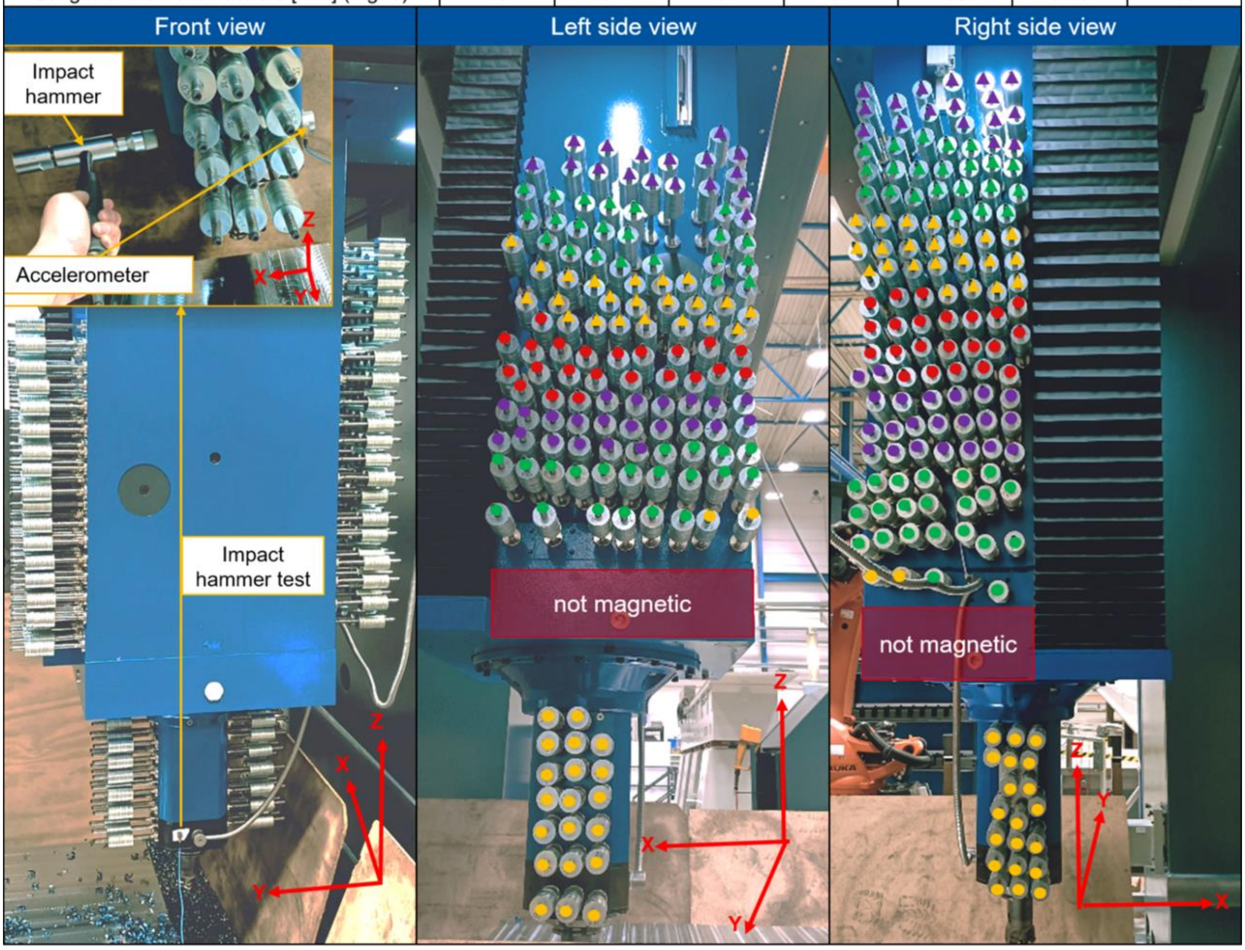

Figure 12. Visualization of MMD system installation on the real machine

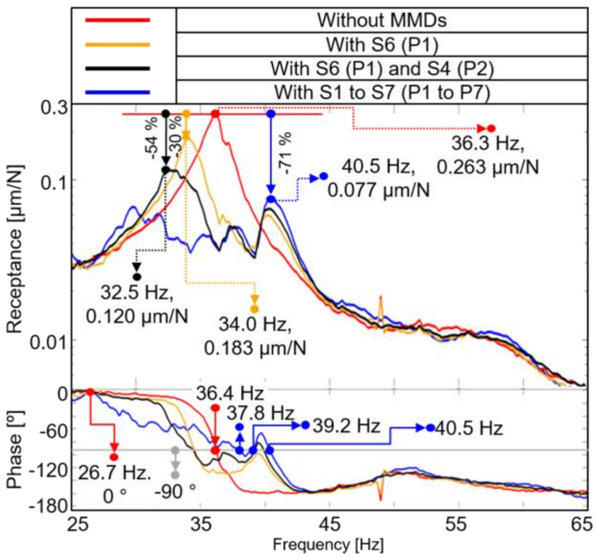

Figure 13. Results of the impact hammer tests

1	Encapsulated, triaxial acceleration sensor on the tool side
2	Cutter head (Cemented carbide): • 80 mm diameter • 6 cutting edges
3	Feed direction
4	Encapsulated, triaxial acceleration sensor on the workpiece side
5	Workpiece: s235 steel

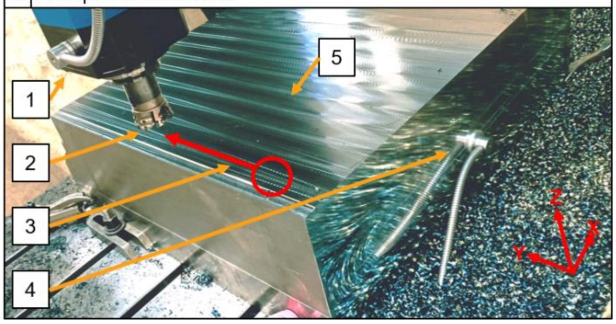

Figure 14. Setup for the cutting tests

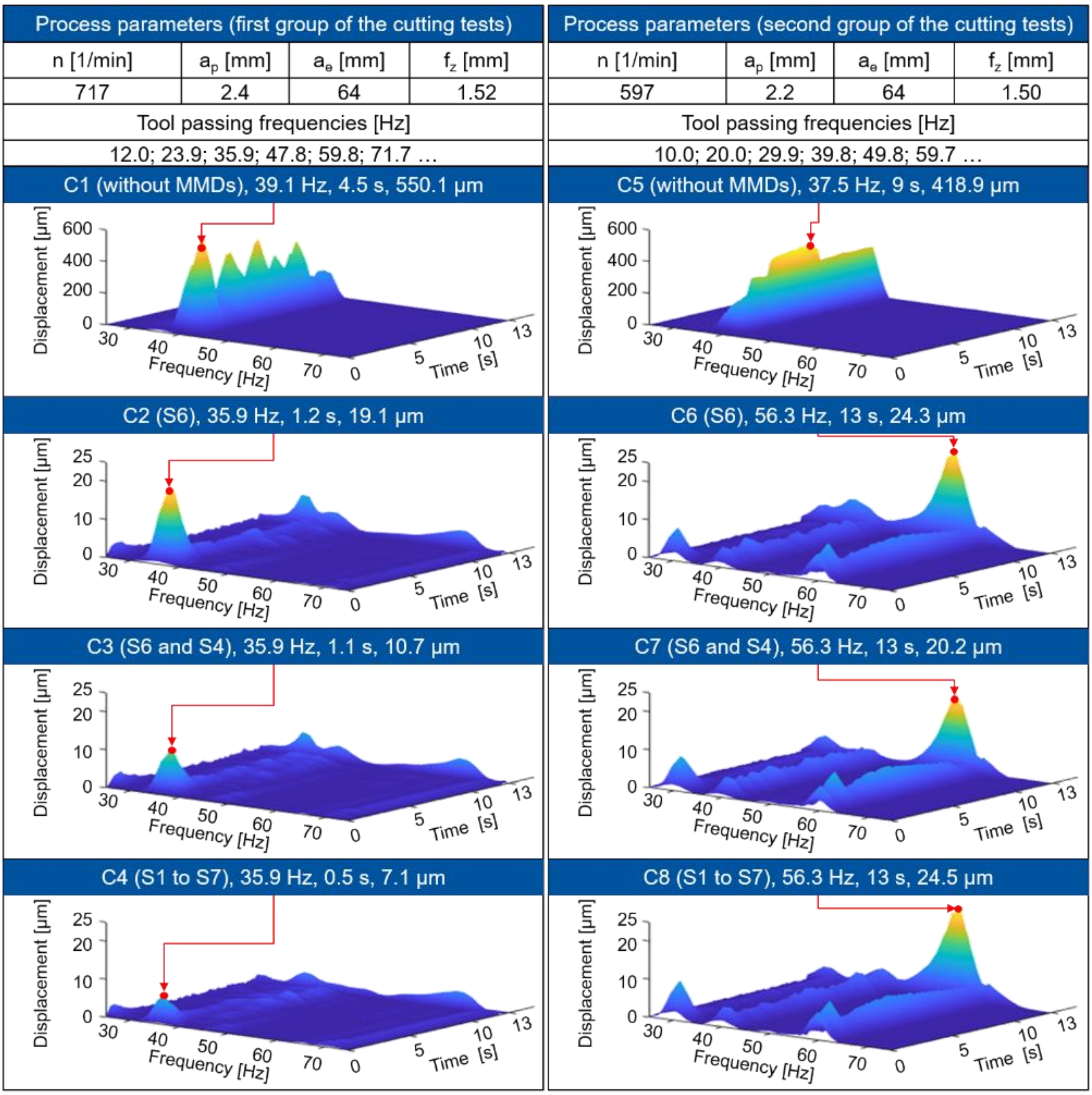

Figure 15. Results of the cutting tests

At C5 (without damper), chatter occurs at the chatter frequency 37.5 Hz, with the maximum dynamic displacement of the tool side Umax being 418.9 µm. At V6 (damper group S6), the machining process is already significantly more stable. Umax is reduced by 94.2 % to 24.3 µm. At C7, Umax drops further by 16.9 % from 24.3 µm to 20.2 µm because the damper group S4 is also used. C8 shows no further improvement, because the corresponding frequency of the Umax with 56.3 Hz is outside the scope of the MMD system.The results of the cutting tests illustrate the effectiveness and the robustness of the MMD system.

5 CONCLUSION

This paper applies a method to design the MMD system consisting of multiple groups of dampers to reduce the vibration of machining systems. The first step of this method is to build and validate the simulation model of a machining system. Utilizing the validated simulation model of the machining system, the mounting positions for all groups of dampers are determined and the simulation model is reduced to a single mass vibration system at each of these mounting positions. The average mass of the single mass vibration systems in each group is used to find the nominal tuning frequency of the dampers in this group using the Den Hartog method. The designed MMD system is evaluated by simulations before installing it on the machine, which gives the impression, that the designed MMD system is promising to reduce the possible vibration of the machining system.

In order to validate the designed MMD system, several impact hammer and cutting tests are conducted. The real natural frequency of the machine without the MMD system 36.3 Hz is 4.1 Hz higher than the simulated value 32.2 Hz (because of the higher Z-slide position). However, the maximum receptance of the machine is reduced by 71 % and the stimulated chatter under the aggressive process parameters is avoided with the designed MMD system. The effectiveness and the robustness of the designed groups of the TMDDs (MMD system) is thereby validated. Furthermore, only with the S6 dampers, the tendency to chatter can be significantly reduced. The total oscillating mass of the group S6 is 26.4 kg, which is less than 1.4 % of the total mass of the Z-slide and the milling head (over 2000 kg). In addition, the material of the group S6 costs 315 €. This shows the high efficiency of the designed TMDDs. The MMD system is a quick solution for the unexpected vibration problems during the machining processes. However the installation of the MMD system affects the outer dimensions of the Z-slide significantly, which limits the application of the MMD system on the cavity machining processes.

The presented design method of the MMD system aims for vibrations caused by the tool side of the machining system. During the machining of thin wall workpieces, the vibrations are manly caused by the workpiece side. Thus it's meaningful to expand the presented method to consider the workpiece vibrations. Furthermore, the possibility of combining the MMD system with other vibration reduction methods is to be explored.

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