Numerical and experimental analysis of vibration damping performance of polyurethane adhesive in machine operations

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Machining operations inherently cause structural vibrations of the work piece and machine involved, especially during roughening operations. This clearly adversely affects the surface roughness and tooling lifetime. These effects may become extremely pronounced in case the stiffness of the work piece to be machined is low and machining vibration excitation frequencies fall in the range of structural resonance frequencies of the work piece. Additional fixation is costly, time-consuming and moreover it may induce structural loading and deformation which, after finishing, causes an out of tolerance work piece.

This work presents an alternative approach in which Polyurethane foam adhesive bonding is applied as a temporary measure to reduce structural vibration during milling operations on a large welded sheet metal construction. The first phases analysis the actual structural vibrations that occur during milling operations. Experimental and numerical modal analysis is carried out in order to determine the excitation and resonant vibration behaviour of the sheet metal structure. A dedicated FE – model is set up which enables the necessary insights in where and how critical vibration levels may be reduced. The second part of the work discusses the development of suitable vibration dampers applying Polyurethane foam adhesive. For this purpose, experiments are carried out to determine the vibration dampening performance of different foam layer thicknesses. These involve hammer excitation of dedicated small-scale adhesively bonded samples. FE modelling is applied to optimize the mounting of bonded Polyurethane dampeners and to predict the effect on structural vibrations during milling operations. This part extensively

outlines the specific experiments involved and the strategy for modelling the viscoelastic foam dampeners in the structural numerical model in a robust but effective way. The third part describes the validation measurement campaign and extensively compares structural vibration levels before and after providing the foam dampening measures. It concludes by showing that critical vibration levels may be reduced by over 58% with the method applied.

I. INTRODUCTION

There is an increasing demand of lightweight structures in all branches of the industry. These lightweight structures typically involve the use of thin-walled constructions with a high overall, but low local stiffness. Hence more vibrations are induced when machining these parts, causing out of tolerance work pieces. Considering total time and effort of machining and tooling lifetime, vibration reduction is very important as it directly leads to faster productions speeds and extend the tooling lifetime.

An early and significant breakthrough in chatter and vibration prediction is presented by Weck M. et al [1]. In this paper, some discrete machining parameters are calculated with which the machining operation runs stable. These discrete parameters are then implemented in a CAD/CAM system. To calculate these parameters, the stiffness characteristics of the work piece and the transfer function of the machining system are assumed to be known. Conclusions from this work are that the vibration levels for each frequency strongly depend on the cutting depth.

As an extension of this work, E. Budak and Y. Altinas [2] define a new general formulation to predict Chatter Stability in milling. By using time-domain simulations and analytical-single frequency solutions, they are able to define stability diagrams. These diagrams clearly show an area with stable cutting conditions and an area in which chatter occurs. Conclusions are that both spindle rotation speed and axial depth of cut significantly affect the position and shape of the estimated stability lobes. To calculate these stability lobes,

the work piece stiffness and the transfer functions of the machining system are also assumed to be known. A deterministic approach is thus applied here.

As it is important to increase production speeds, it is important to understand the cause of the chatter that occurs in the unstable region of the stability diagrams. Chatter can be classified in primary chatter and secondary chatter [3]. Primary chatter consists of vibrations on frequency's that are much lower than the spindle revolution frequency. This scatter originates from friction between tool and work piece [4][5], stress distribution on the normal rake face [6], thermoplastic behaviour of the chip material [7] and mode coupling [8]. Secondary scatter consists of unstable chaotic vibrations with a higher frequency than that of the machine spindle rotation. These vibrations, caused by regenerative effects (amplified oscillation loop between spindle and work piece), are due to unstable behaviour of the uncut chip thickness [4][6].

Several methods have been investigated to cope with machining chatter and vibrations. One method is to intervene on the machine system by predicting and measuring chatter. According to Delio T. et al. [9] microphone measurements are very useful to detect scatter. Also, the use of laser displacement sensors [10], accelerometers [11] , eddy current [12], probes and plate dynamometers [13] has been analysed to measure machine vibration scatter. Both passive and active damping systems have been implemented into machining systems to reduce vibrations. One example is the use of special hydrostatic spindle bearings. These bearings are able to absorb and dissipate vibration energy.

Another approach is to avoid chatter by using variable pitch cutters [14] or varying the spindle speed by using Spindle Speed Modulation [15] [16].

The methods described above are focussing on the dynamical behaviour of the machine system. These dynamical properties depends on both the dynamical behaviour elements of the system and their interactions. Therefore it is very important to calculate their modal parameters and transfer function and estimate the

relative vibrations between tool and workpiece [17]. These calculations are quite complex and the proposed solution have some practical implications thus few industrial applications are known.

Other methods are focussing on the work piece. This is very interesting for batch productions of thin-walled products. Yang Y. estimates the displacement response function of a thin-walled structure using simplification methods based on Euler-Bernoulli beam assumption. The transfer function is used to determine external damping properties

[18].

Also tuned masses can be attached to the work piece surface to avoid scatter during milling. Tests based on this principle are conducted by Kolluru K. [19] and Bolsunovsky S. et al. [20]. Other methods implying tuned masses are given by Altinas Y. and Weck M. [4]. These methods offer satisfying solutions for smaller thin-walled work pieces where the masses are simply applied by use of adhesive tape.

For larger thin-walled structures, which are the most flexible elements in the machining process, it is even more convenient to intervene on level of the work piece. Using external damping methods are known to be very effective as a measure to counteract vibrations. Ma J. et al. consider magnetorheological fluids flexible structures to attach the work piece on strategic positions to tackle vibrations [21]. Though the tests looked promising adding external fixtures is time consuming moreover, they can induce loading and deformation of the structure.

II. MATERIALS AND METHODS

A.Determining actual structural vibrations

The work presented in this paper aims to reduce vibrations on a specific work piece by use of Polyurethane adhesive fixtures. An experimental approach is used to define reference values to be used as input for numerical vibration simulations. The complete milling process, as shown in figure 1 and figure 2, is divided into five different milling sequences based on the milling parameters. Structural vibrations at 10 Points are

measured by a laser vibrometer and 4 accelerometers which are placed on strategic locations to capture relevant vibration levels for each milling sequence. A total of 75 measurements are executed during the measurement campaign. At each sensor location, vibration levels are monitored for a frequency range of 0,5 Hz to 200 Hz and the frequency response functions between the sensors are extracted. As the milling parameters are known for each measurement, the main excitation frequencies are determined by the spindle rotation speed and the number of tooth. The excitation force can be extracted from the machining power given by expression (2.1) [22].

$$P_c = K_P C Q W \quad (2.1)$$

Where P_c is the machining power, K_P represents the power constant and is related to the material hardness of the work piece, C is the Feed Factor which is a function of the feed rate and the power constant, Q equals the Metal Removal Rate and W , called the Tool Wear Factor, which is a term defined by the type of operation, e.g. slab milling.

FIG. 1. Milling process of the structure.

FIG. 2. Milling process with Polyurethane adhesively bonded fixtures.

B. Finite element modelling

This research is based on a mixed experimental-numerical approach to analyse the damping performance of Polyurethane foam adhesive and to estimate its performance to reduce vibration levels in machining operations. The steel sheet structure is modelled using CQUAD8 shell mesh showed in figure 3, applying linear material definitions for the steel. This model set-up allows quick calculation times. Before the milling operation, the thin shell structure is welded, causing local deviations in material properties. These welded zones are integrated in the simulation using adapted material properties. Small stiffeners are simplified using beam elements. Fixtures of the work piece are simulated identically using constraints with restricted DOF's and CBUSH-elements as seen in figure 4.

FIG. 3. Meshed Finite Element Model of the structure.

The FE model is solved for modal structural behaviour. The natural frequencies and mode shapes of the structure are calculated and the simulation model is extended to calculate the structural vibration responses due to the enforced vibrations by machining. The implementation of these spectral local input forces represents the real milling forces which are applied during machining. Milling force values are retrieved from real milling parameters (type of mill, rotation speed, feed). This enables the simulation of non-stable cutting parameters (4). The simulated structural vibrations are analysed by placing virtual sensors on the simulation

model corresponding with the true sensor locations during the measurements. The milling process for which very high vibration levels are observed in reality is simulated. The matching modal deflections and frequencies are then used to optimize the Polyurethane foam fixture location and the corresponding damping properties. Of course practical implementations must be considered here as the foam dampening system are only temporarily. For all simulation work, Simcenter 12 interface is used with the NX Nastran solver environment.

FIG. 4. Example of the applied constraints and milling force.

C. Development of polyurethane dampers

As outlined above, Polyurethane adhesive fixtures are used as a damping method. The damping ratio (2.2) D of a single degree of freedom (DOF) spring damper system, under assumption of viscous damping, can be calculated from the logarithmic decrement as given by expression (2.3). Here n represents the number of periodic amplitudes and X the corresponding amplitude level.

$$D = \frac{J}{\sqrt{4\pi^2 + J^2}} \quad (2.2)$$

$$J = \frac{1}{n} \ln \frac{X_n}{X_{n+1}} \quad (2.3)$$

Performing a hammer excitation test on a single DOF spring-damper system is an effective way to determine the damping ratio of a damper. Therefore samples, as shown in figure 5, are made representing a spring damper system. Two masses are adhesively bonded with Polyurethane adhesive on a stiff structure. One

sample uses a mass of 1028g adhesively bonded with Polyurethane with a bond thickness of 16mm. The second sample has a mass of 1500g with a bond thickness of 34mm. The difference in masses and bond thickness's results in different natural frequencies of the sample when excited. This enables to define the damping ratio of Polyurethane for two different frequencies. Each sample is subjected to three tests with each seven excitations. This ensures stable values for a reliable prediction of the damping properties of Polyurethane.

The amount of relative damping depends on the mass and stiffness of the vibrating system considered. To tune the damping properties of the Polyurethane fixation, the viscous damping constant has to be determined. A modified formulation for the viscous damping C_A (2.4) is found.

$$C_A = 2D\sqrt{\frac{km}{A}} \quad (2.4)$$

In case the spring constant k of the Polyurethane and the mass m of the fixation is known, the viscous damping of the Polyurethane C_A can be tuned to approach the critical damping C_{cr} by modifying the adhesive bond area A and the spring constant k which is related to the Polyurethane adhesive bond thickness.

FIG. 5. Samples used for hammer excitation.

D. finite element model updating

Determining and dimensioning of the fixtures of the structure is performed by virtual testing. An accurate simulation model is required to conduct a proper optimization of the milling process. A first milling process during which significant vibration output is observed in reality, is simulated. An optimization routine minimizes the difference between the experimental vibration measurements and the numerical simulated counterpart. The optimized FE model is achieved by adapting the material properties and the material damping of the steel sheet structure. The two other milling processes are then used as a validation tool for the FE-model. Figure 6 shows the mode shape of the structure with a frequency of 43 Hz on the updated model.

FIG. 6. Example of a structural vibration mode shape of 43Hz.

III. RESULTS

A. Virtual testing

As mentioned in section IIA a total of 75 measurements are conducted. Only those cases are simulated for which significant vibration levels are captured. One measurement is discussed as an illustration of the process. The simulated vibration spectrum of all four virtual accelerometers is shown in figure 7 and 8. Both plots are representing the vibration levels while the work piece is exposed to the same external excitation, to be specific, the same milling parameters and excitation point. Figure 7 represents the simulated amplitude levels of the test without the Polyurethane adhesively bonded fixations, figure 8 displays the same situation after

implementation of the damped Polyurethane fixtures. The spectrum plots are showing the captured amplitudes of each virtual sensor in three directions. The virtual sensors are places corresponding to the real sensor location. The frequency spectrum plots clearly show a significant reduction of the amplitudes of most accelerometer location after adding adhesively bonded Polyurethane foam fixtures. A slight frequency increase of the peak amplitudes is a result of the increased stiffness of the work piece caused by the introduced Polyurethane fixtures. An overview of some corresponding frequencies with significant amplitude reductions is presented in table I.

Freq. without PU fixation	Freq. with PU fixation	Amp. without PU fixation	Amp. with PU fixation	Amp. reduction
29 Hz	37 Hz	0,28 mm	0,02 mm	-93%
75 Hz	83 Hz	0,08 mm	0,005 mm	-94%
89 Hz	101 Hz	0,08 mm	0,012 mm	-85%

TABLE I. Example of amplitude reduction for the excitation illustrated in figure 7 and figure 8 by use of Polyurethane adhesively bonded fixtures.

The mode deflection of the large amplitude peaks, corresponding with a frequency of 29 Hz shown in figure 7 and with a frequency of 37 Hz shown in figure 8, are presented in 9 and 10 respectively. Analysing the different mode shapes on frequencies with high amplitude output allows to adapt the fixture locations and tune the Polyurethane damping properties. After practical considerations, the adhesively bonded polyurethane fixtures are placed as shown in 11. In this figure, the green marked locations suggest an extra internal Polyurethane fixation e.g. connecting the vertical struts with each other, the yellow marked locations suggest an additional Polyurethane fixture and red marked locations indicate points where a stiff fixation is desired.

FIG. 7. Amplitude spectrum of all accelerometers without Polyurethane foam fixtures.

FIG. 8. Amplitude spectrum of all accelerometers with Polyurethane foam fixtures.

FIG. 9. Mode shape of 29 Hz of the workpiece without Polyurethane adhesive fixtures.

FIG. 10. Mode shape of 37 Hz of the work piece with Polyurethane adhesive fixtures.

Applied excitations for the vibration simulation shown in 7 and 8 are determined analytical. These induced vibrations correspond with the real milling parameters during the measurement campaign. A summarize of the used milling parameters in the discussed simulation are presented in table II. Figure 12 shows a picture of the mill used during the real vibration measurements for which the milling parameters are presented in table II. After initial simulations, the cutting parameters are tuned in accordance to the measurements.

Cutting parameter	value
Spindle RPM	600 min^{-1}
Feed rate	350 mm/min
Cutter diameter	100 mm
Number of teeth	5
Cutting depth	2 mm
Milling operation	slab milling
Main excitation frequency	10 Hz
Second excitation frequency	50 Hz
Radial milling force	511 N

TABLE II. Example of milling parameters used as an excitation input for the vibration simulation.

FIG. 11. Optimized fixation points of the work piece using Polyurethane foam adhesive dampers.

FIG. 12. Picture of the mill used to determine the excitation parameters (table II) in accordance to the discussed vibration simulation.

The Polyurethane adhesive is implemented in the vibration simulations by defining spring-damper elements. As mentioned in section IIC the damping properties are determined using hammer excitation tests. The damping properties extracted from the hammer excitation tests are presented in table III and IV for sample 1 and sample 2 respectively.

Sample 1	
Mass	1028 <i>g</i>
Adhesive thickness	16 <i>mm</i>
Viscous damping	0,72 $\frac{Ns}{m^2}$

TABLE III. Viscous damping of sample 1.

Sample 2	
Mass	1500 <i>g</i>
Adhesive thickness	34 <i>mm</i>
Viscous damping	0,65 $\frac{Ns}{m^2}$

TABLE IV. Viscous damping of sample 2.

B. Experimental data

During the test campaign a total of 1650 data sets can be extracted from all measurements. Only few are analysed and simulated for which significant vibration levels are captured. Figure 13 shows the frequency response function of accelerometer C in relation to accelerometer B during the first measurement corresponding to the previous vibration simulations. Accelerometer C is mounted close to a fixture point on top of the structure and Accelerometer B is placed on the back of a vertical strut of the structure as illustrated in figure 14.

As an extend acceleration levels of the sensors are compared to confirm the vibration reduction. Figure 15 and 16 represents actual acceleration levels measured by Accelerometer A. Figure 15 is created during the initial measurement campaign, Figure 16 represents the same measurement after implementation of adhesively bonded Polyurethane fixations. Amplitude reductions of 58% is noticed by analysing the graphs. These values deviate from the ones obtained by the vibration simulations. An explanation is found by analysing the induced scatter of the milling tool. Real milling excitation are not only due to spindle rotation and cutting parameters, they consist of a broad frequency spectrum with complex amplitude variations. For simulation purposes, these excitations are simplified using a limited frequency spectrum with discrete amplitude values. Though a satisfying reduction can be achieved using the fixations described in this paper.

The goal of the presented work is to increase production speeds by reducing the milling vibrations. A summary of the used milling parameters before and after the implementation of the adhesively bonded Polyurethane fixations is given in V. The spindle rotation speed is increased by 12% and the feed rate is almost doubled keeping the same cutting depth. This increase in milling parameters allows to save 45 minutes production time for each part which correspondence with almost one third in production speed.

FIG. 13. Frequency response function between accelerometer C and accelerometer B measured during initial test campaign.

FIG. 14. Location of accelerometers C and B.

Cutting parameter	Initial value	Value with Polyurethane damped fixations	Difference
Spindle RPM	600 min^{-1}	670 min^{-1}	+12%
Feed rate	350 mm/min	670 mm/min	+91%
Cutting depth	2 mm	2 mm	0%

TABLE V. Increase in cutting parameters by using adhesively bonded Polyurethane damping.

FIG. 15. Actual acceleration levels measured by accelerometer A during initial test campaign.

FIG. 16. Actual acceleration levels measured by accelerometer A after implementation of adhesively bonded polyurethane fixtures.

IV. CONCLUSIONS

A stepwise approach is presented in which experimental data is used to interact with vibration simulations as a way to reduce critical vibration frequencies of a work piece induced by milling operations. First an initial test campaign will provide the necessary data to determine the excitation parameters for which significant vibrations are captured. The sensor data is used as reference data to update the virtual vibration simulations. Therefore virtual sensors are placed on the simulation model corresponding to the real sensor locations. Improvement of the vibration behaviour of the simulation model is obtained by applying virtual Polyurethane dampers on strategic locations. Damping properties of the polyurethane adhesive is obtained by using hammer excitation tests. Variation of the Polyurethane adhesive bond thickness and the bond area are considered. The procedure followed enables an adequate estimation of the vibration behaviour of the milled structure using finite element software. As a process validation, the Polyurethane fixations are placed as suggested by the simulation model and a second milling process is executed.

The main conclusion of this work is that a vibration reduction of 58% is achieved by adding Polyurethane adhesively damped fixtures on strategic locations. This allows to increase the feed rate by a factor of two and the spindle rotation by 12%. However, deviations are found between the simulated improvements and the parameters observed during the milling process since the simulations considers simplified excitations with a limited frequency spectrum and discrete amplitude values. Overall, very positive results are obtained and the proposed methodology offers a quick and robust solution for industrial applications.

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