Available online at www.sciencedirect.com
ScienceDirect
Procedia CIRP 31 (2015) 70 – 75
www.elsevier.com/locate/procedia

15th CIRP Conference on Modelling of Machining Operations

High performance machining enabled by adaptive machine components

Burtscher, J. [a]*, Koch, S.-F. [a], Bauer, J. [a], Wagner, H. [a], Fleischer, J. [a]

[a]Karlsruhe Institute of Technology (KIT), wbk Institute for Production Science, Kaiserstrasse 12, 76131 Karlsruhe, Germany

* Corresponding author. Tel.: +49-721-608-42453; fax: +49-721-608-45005. E-mail address: jacques.burtscher@kit.edu

Abstract

A current drawback in high performance machining is the dynamic behavior of the machine tool. The dedicated Eigenfrequencies of the machine tool structure lead to not optimal cutting parameters due to the danger of chatter. Possibilities to overcome this drawback are on one hand the unwanted variation of the cutting parameters and on the other hand the variation of the dynamic properties of the machine tool components. The variation of the dynamic properties of these components requires high efforts and expenses.

An approach to easily vary the dynamic characteristic of the machine tool is a carriage made out of Carbon Fibre Reinforced Plastic (CFRP) with adequate mass shifting by filling the built in chambers with a fluid. This leads to shifted Eigenfrequencies and therefore to optimal cutting parameters without chatter. The mass ratio between the empty lightweight machine tool carriage made out of CFRP and the one filled enables a significant frequency shift.

Within this paper three issues will be addressed: first the frequency shift depending on the carriage's position will be examined, the frequency shift according to the filling strategy will be shown and finally the amplitude reduction related to the external excitation will be presented.

Peer-review under responsibility of the International Scientific Committee of the “15th Conference on Modelling of Machining Operations

Keywords: Machine tool, Chatter, Adaptive manufacturing

1. Introduction

Machine tools are on the one hand every year more performant in terms of accuracy, flexibility and productivity thanks to new concepts, devices or new materials [1]. On the other hand vibrations in machine-tool are the cause of numerous problems such as tool breakage, poor surface quality and an increased material wear, leading to higher energy consumption. Vibrations in machining operations are therefore the most limiting factor to increase the performance of the machine tools [1,2,3].

Vibrations can be classified in three categories depending on the source of the vibrations and their behavior. If the vibration is initiated by an external source and a periodic vibration still exists when the external source is removed then the vibration is called free vibration. In the case where the vibration only occurs during the presence of external forces, the vibration is called forced vibration. And if the vibration does not require any external force, the vibration is called self-excited vibration. In this case the vibration resource is inside the mechanism [2]. The self-excited vibrations that occur during the machining process are also defined as chatter [1,3], or self-excited chatter [2]. The chatter during the machining process is mainly caused by the regeneration of the waviness on the workpiece surface [1,2,3].

Chatter can be avoided by two general approaches [1]. Since the chatter is due to the interaction between the tool and the workpiece, the first approach consists of changing the machining operation parameters. Therefore the chatter can be represented - for a given machine tool - on a stability lobe diagram (SLD) depending on the axial depth of cut as a function of the spindle speed [4,5]. Thus the operator will be able to set adequate cutting parameters to get a chatter free surface quality for an optimal material removal rate by using at best the lobbing effect. The SLD of a machine tool can be generated by the use of stationary experimental frequency response function (FRF).

In this way it is possible to predict the chatter boundaries in order to establish the SLD under the consideration of varying material and machine parameters [6] or by modelling the

Peer-review under responsibility of the International Scientific Committee of the “15th Conference on Modelling of Machining Operations
doi:10.1016/j.procir.2015.03.039

machine behavior based on the FRF measurement [7,8]. In this approach, the machining operation and the definition of the machining parameters are defined based on the dynamics of the machine tool.

In the second approach, instead of setting adequate cutting parameters in function of the machine tool, the machine tool dynamic behavior is changed according to the machining process. [2] distinguishes between passive and active strategies. Passive strategies are when the machine is adapted to the machining by replacing some components by others. These strategies are implemented off-line. Active strategies are based on machine tool components that can adapt their dynamic behavior in order to shift or change the lobes frontiers of the SLD. In this approach, the machining operation and the definition of the machining parameters remain unchanged and the machine dynamic is adapted on-line based on the workpiece and the machining process.

This paper will present an approach to easily vary the dynamic characteristics of a machine tool. A new carriage made out of Carbon Fibre Reinforced Plastic (CFRP) with adaptive dynamic characteristics has been developed based on first experimentations of [8]. The variation of the component dynamics is based on the mass variation of the carriage leading to shifted Eigenfrequencies. In this way chatter free machining is possible without changing the cutting parameters. The mass ratio between the empty lightweight machine tool carriage made out of CFRP and the one filled enables a significant frequency shift.

Within this paper three issues will be addressed: first the frequency shift depending on the carriage's position will be examined, the frequency shift according to the filling strategy will be shown and finally the amplitude reduction related to the external excitation will be presented.

2. State of the Art

As introduced before the chatter reduction can be obtained by changing the cutting parameters in accordance with the LSD to select a stable combination of spindle speed and axial depth or by changing the system behavior. This paper will present an approach to change the machine dynamics, thus the state of the art will focus on the actual techniques that are implemented to reduce passively or actively the chatter of a machine tool by changing the system behavior. [1,10]

Off-line strategies (passive techniques) are implemented once the machine tool and the cutting tool are selected, whereas on-line solutions (active techniques) are implemented on the machine to act during the machining operation to reduce the vibrations [10]. [11] compares the use of an active and a passive damper in the case of grinding machines. Fig 1 a shows how the resonance frequency of a system can be dodged or reduced by the use of a damper. In this case a damper is fixed to the wheel head of the cylindrical machine. The machining at 121 Hz is obviously improved however the machining from 100 Hz to 110Hz could be subject to higher vibrations. In contrast the active damping showed on fig 1 b is more flexible. Vibration can in this way be immediately suppressed by a changing of the machine behavior through a closed loop control of the vibrations at the workpiece.

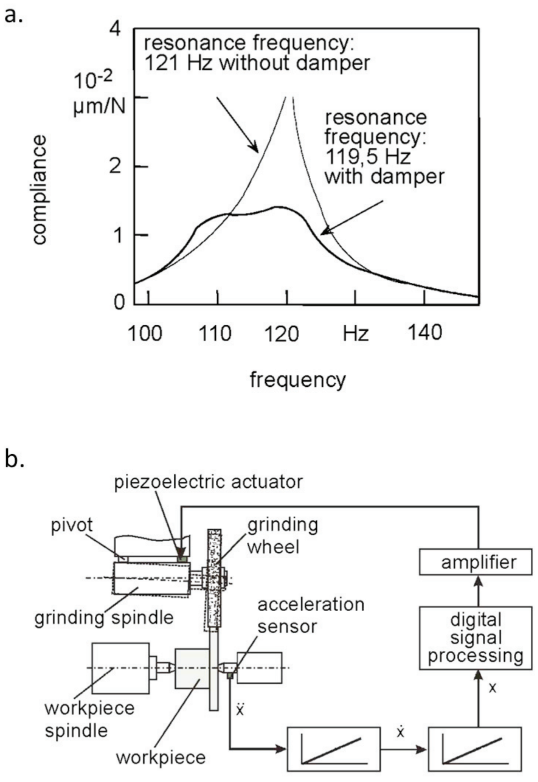

Fig. 1. Reduction of chatter by the use of damper. [11]

The use of a variable pitch cutter to suppress the chatter vibrations is presented in [12]. An analytical model has been developed to determine the optimal design of the pitch angles. In combination with an adaptive force control model the cycle time has been reduced significantly [12].

The vibrations that can occur during the machining are due to the structural modes of the machine tool. An approach is presented in [13] to avoid chatter by shaping the tool trajectory to avoid the excitation of the structural modes. Another possibility to avoid vibrations is to superimpose contrary vibrations originating from both machining operations at both spindles. This can be realized through a double sided milling. [14]

[15] presents an approach to improve the damping capability of boring tools. The damping - realized with a magneto rheological fluid - is then adjusted to the machining operation to suppress the chatter occurrence.

[16] presents a measure to reduce the ability of a system to chatter. An improved resilience behavior is characterized by its Nyquist representation where the maximal negative real part is reduced and the Eigenfrequency is higher. The figure 2 illustrates this measure in the case of a mass-spring-damping system. The factors impacting the dynamic behavior of the system are its mass, its damping and its stiffness. The reduction of the maximal negative real part can be obtained

through the increase of the systems damping and stiffness as well as through the decrease of the systems mass.

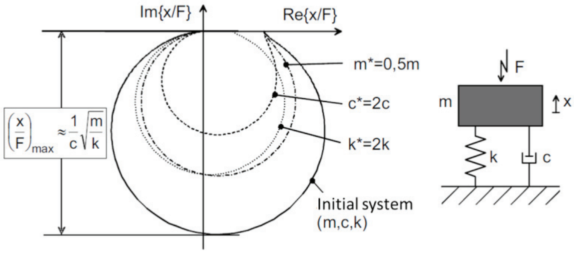

Fig. 2. Measure to reduce the ability of a system to chatter. [16]

3. Approach

As introduced before, the active reduction of chatter is obtained with components that adapt themselves to the machining parameters and the machine dynamics during the machining operation. Since the Eigenfrequencies of a vibrating system is mainly defined by its stiffness, damping and its mass. Since the mass of a system is easily changeable, the approach presented in this paper is based on an adaptive mass shifting. For this purpose water will be pumped into the new component in order to change its mass in a controllable way.

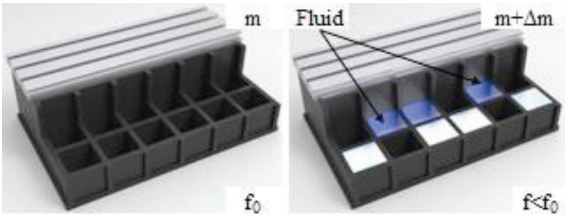

Fig. 3. Approach [9]

When filled the carriage mass is higher and therefore its Eigenfrequency is lower, see fig. 3. A structure made of CFRP is developed, see fig 5. The effect of Eigenfrequency shifting is increased because the mass ratio of the filled carriage in comparison to the empty one is higher with a lightweight structure made of CFRP. By changing the carriage dynamics, all the machine tool is affected and a new SLD is valid. The filling of the chamber is therefore chosen with respect to the cutting parameters imposed by the machining operation to achieve chatter free machining

4. Results

4.1. The test bench

The frequency shift is measured on a test bench, see fig. 4. The test bench is composed of a carriage that can move along a feed axis itself fixed on machine bed. The machine bed is made out of a steel and aluminium structure that, among other things, contains the control system, a water tank and a water pump. The bottom of the carriage is made out of a CFRP structure that is divided into 7 chambers. The top of the carriage is closed by an aluminum plate which represents the working area of the supposed machine tool. On the carriage an unbalanced motor is fixed in order to recreate a milling operation.

Fig. 4. Test bench.

The feed axis, composed of a guide rail and screw ball permits the carriage to move along the X direction over a range of more than 500 mm. The position of the carriage is measured by an incremental glass scale. The control of the test bench is assured by an I/O module from a dSpace system permitting Hardware in the Loop (HiL) simulation and controlling. This module is connected to a PC where a Matlab-Simulink model of the control has been created and then compiled on the module. In this way it is possible to move the carriage under control in position or to fill / empty the chamber of the carriage. To this end the chambers are equipped with water level sensors connected to the dSpace module.

The chambers design is shown in fig 5. The chambers are symmetrically positioned with respect to the x and y axis to allow a filling of the chambers without misbalancing the carriage. The structure also allows the carriage to be rigid, because of ribs that cross the carriage in the diagonals. The chamber C is separated in the middle by a wall made of CFRP too with an opening on the bottom to let the water fill both sides of the chamber. Even though each chamber can be filled seperatly, the groups of compartments labeled either A or B will be filled at the same time. The chambers are filled up to the top to prevent the water to slosh back and forth so that the carriage isn’t subject to any additional oscillation.

4.2. Eigenfrequency shift

The Eigenfrequency shift has been measured on the test bench with modal analysis equipment. A Single Input, Multiple Output (SIMO) test has been done on the test bench. In this way acceleration sensors have been placed on the carriage exterior at some adequate positions in order to measure the response to a dirac impulse of the hammer. See fig 5. The sensor positions do not coincide with the corner or the symmetry axis of the carriage, these positions could be a node of the Eigenform.

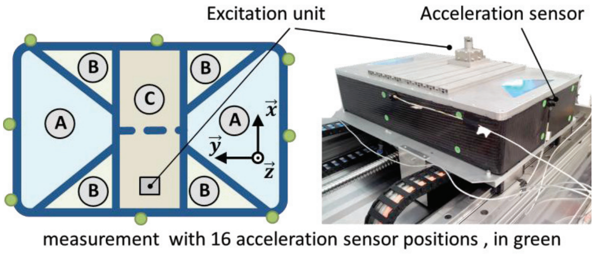

Fig. 5. Chamber design, and sensor positioning.

The analysis of the signals is done by a Fourier transformation in order to determine the transfer function of the machine tool. This function will bring to light the resonance modes of the test bench. To verify the approach, the ball screw has been clamped, and the carriage has been hit by the hammer. The measurements took place for 3 different carriage positions on the x direction and for different filling stages of the chambers. The design of experiment is shown on table 1.

Table 1. Eigenfrequency shift - Design of experiment.

Test	Chambers A	Chambers B	Chamber C	X position (mm)
1	Empty	Empty	Empty	0
2	Full	Empty	Empty	0
3	Full	Full	Empty	0
4	Full	Full	Full	0
5	Empty	Empty	Empty	250
6	Full	Empty	Empty	250
7	Full	Full	Empty	250
8	Full	Full	Full	250
9	Empty	Empty	Empty	500
10	Full	Empty	Empty	500
11	Full	Full	Empty	500
12	Full	Full	Full	500

The results of the measurements are displayed in fig. 6. The resilience is calculated as the norm of the displacement in x, y, and z direction for a hammer impulse in respectively x, y and z direction. The resilience in x direction also has the biggest influence on the fequency shift for an excitation in the x direction since the ball screw is in a simplified model comparable to a spring and the y and z direction are blocked by the guide rail in this case. This effect can be observed on fig. 6, indeed, the Eigenfrequency of the carriage at the position x = 0 mm is higher than the one for the carriage at the position x = 500 mm.

The frequency shift is related to the carriage filling. When the carriage is fully filled a larger frequency shift is noticeable. In the case x=250mm, a higher frequency shift of a not fully filled carriage in comparison to the frequency shift of the fully filled carriage is achieved, see fig. 6. This effect could be due to the complexity of the structure of the carriage where some Eigenfrequency of the carriage itself superimpose with the Eigenfrequency of the machine structure.

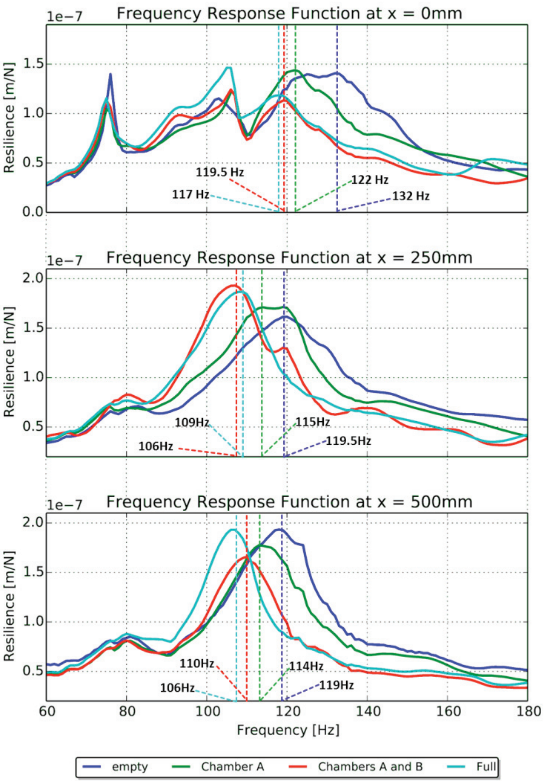

Fig. 6. FRF of the carriage at different positions and for different fillings.

The highest achievable Eigenfrequency shift is of about 11.3%, 11.3% and 10.9% at respectively x = 0 mm, x = 250 mm and x = 500 mm.

4.3. Vibration reduction

The impact of the presented approach for chatter reduction cannot be quantified with a FRF. A second experiment has been set up to measure the vibration reduction as a function of the filling strategies. For this purpose a forced vibration is created by an unbalanced motor and an acceleration sensor is placed on the carriage in order to measure the reduction of the resulting vibrations for different filling combinations of the chamber.

The unbalanced motor voltage is directly related to the motor rotation speed and thus to the vibration frequency. For this experiment, the voltage has been set to get the largest amplitude on the acceleration sensor; the carriage is excited at one of its Eigenfrequencies. The vibration reduction in the case where all the chambers are filled at the same time is showed in fig. 7 for the carriage position x = 250 mm. The reduction is mainly to be seen in the x direction. This has also been noticed by establishing of the FRFs since the ball screw acts like a spring, see preceding paragraph.

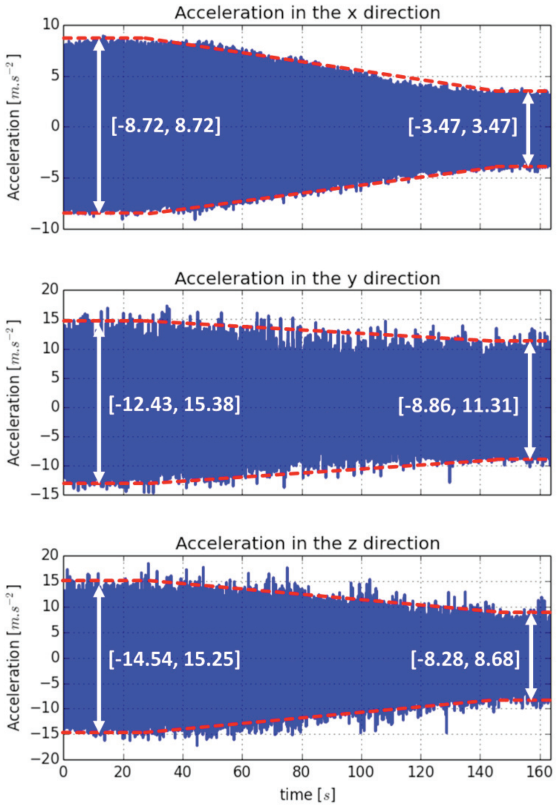

Fig. 7. Amplitude reduction of the vibrations by filling the carriage

The table 2 gives the amplitude reduction for different filling strategies for the carriage position x = 250 mm. It is interesting to notice that the fully filled carriage doesn't reduce the total vibration the most, see test 6 and 7 of the table 2. In fig. 6 for the case x = 250 mm the frequency shift is also larger where chambers B and C are filled. This shows that the position of the added mass has an impact on the reduction of the vibration amplitude. A possible superimposition of the Eigenfrequency of the carriage and the one of the supporting structure can explain a lower frequency shift by a fully filled carriage in comparison to a partially filled one. The current measurements do not permit a sophisticated explanation.

Table 2. Vibration reduction - Results.

Test	Chambers A	Chambers B	Chambes C	Reduction of the vibration amplitude	
				x direction	total
1	Empty	Empty	Full	26,7%	27,1%
2	Empty	Full	Empty	38,1%	27,9%
3	Full	Empty	Empty	34,1%	23,9%
4	Empty	Full	Full	52,3%	35,7%
5	Full	Empty	Full	45,5%	29,2%
6	Full	Full	Empty	48,9%	45,2%
7	Full	Full	Full	60,2%	38,7%

The vibration reduction obtained by the filling of the chambers cannot be compared directly with the amplitude of the FRF shown in fig 6. For instance the filling of the chamber A increases the amplitude of the FRF in fig 6 whereas the vibration will be reduced by 23.9% as listed in table 2. Nevertheless the maximum vibration reduction in the x direction is obtained by filling the carriage at the maximum.

4.4. Decrease of the negative real part

In order to improve the resilience behavior of a mass-spring-damping system, one has to reduce the maximal negative real part. This is obtained as explained in section 2 through the increase of the systems damping and stiffness as well as through the decrease of the systems mass, see fig 2. In this way [16] shows that lowering the mass permits the stabilization of a previous unstable process. The Eigenfrequency is then shifted in a higher region.

In the presented approach the mass shifting only allows to increase the mass in comparison to the initial system. Fig 6 shows that the Eigenfrequencies are shifted to lower regions. In order to see if the resilience behavior is improved by the mass increase the Nyquist representation of the FRF of the system is shown in fig 8. The Eigenfrequency not being at a phase of -90° (even for the initial system) shows that it is not possible to compare this approach with a mass-spring-damping system.

It may be obvious that the biggest frequency shift is achieved by fully filling the carriage, because this leads to the biggest variation in mass. Yet out of fig 8, the fully filled carriage (cyan in fig 8) in comparison to the carriage filled with chambers A and B (red in fig 8) does not have a better resilience behavior. The maximal negative real part is bigger for the fully filled carriage in all three carriage positions.

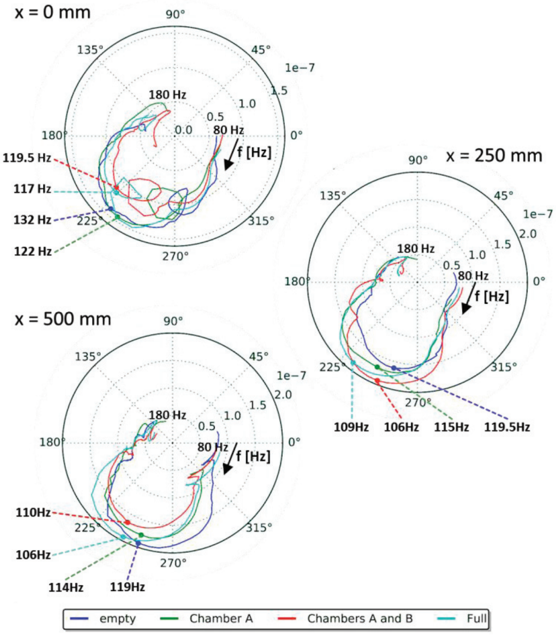

Fig. 8. Decrease of the negative real part in a Nyquist representation.

A mass increase indicates a degradation of the dynamic behavior of a mass-damping-system. The present approach permits the system to be configured to the best state regarding its position, the machining operations and its susceptibility to chatter. For instance if the machining doesn't excite the system at $f=119.5\ Hz$ in the case $x = 0\ mm$ the carriage filled with chambers A and B seems to be the best filling configuration to reduce the susceptibility of the system to chatter whereas an empty carriage at $x = 250\ mm$ presents the best filling configuration.

5. Conclusion and outlook

Future trends of production processes are pointing towards a flexible production combined with a high productivity. In order to fulfill the demand with a reduced amount of different machine tools it is necessary to have machine tools that adapt to the product and its machining and not to adapt the machining operation to the machine in order to reduce the production cost.

The approach presented here permitted to shift the Eigenfrequency of about 11% and to reduce the vibration by up to 60% in the x direction and up to 45% for the overall vibration. The FRF and the vibration reduction in the x direction have the biggest impact on the frequency shift and the total vibration reduction. This approach could permit manufacturers to increase the productivity of a their machines by reducing the machining operation time

It is also possible to imagine for the future, in the case where this approach is implemented on a machine tool, a machining strategy to reduce vibrations. In this way the chambers would be filled according to the machine dynamics and to the machining operation based on the NC program. This strategy could reduce the time between the submitting of the order and the beginning of the manufacturing and would permit a machining with a good material removal rate.

Upcoming research topics will concern the implementation of this approach into a real machine tool to actively reduce the vibrations or the risks of chatter. This project will be founded by the Central Innovation Program SME (Zentrales Innovationsprogramm Mittelstand – ZIM).

References

[1] Quintana G, Ciurana J. Chatter in machining processes: A review, International Journal of Machine Tools & Manufacture, 51:2011, pp. 363-376

[2] Cheng K. Machining Dynamics: Fundamentals, Applications and Practices. Springer Science & Business Media, 2008 ISBN 184628368X, 9781846283680

[3] Siddhpura M, Paurobally R. A review of chatter vibration research in turning, International Journal of Machine Tools & Manufacture, 61:2012, pp. 27-47

[4] Altintas Y, Budak E. Analytical Prediction of Stability Lobes in Milling, Annals of the CIRP, 44/1/1995, pp. 357-362

[5] Zhongqun L, Qiang L. Solution and Analysis of Chatter Stability for End Milling in the Time-domain, Chinese Journal of Aeronautics, Volume 21, Issue 2, April 2008, Pages 169–178

[6] Faassena RPH, van de Wouwa N; Oosterlingb J.A.J, Nijmeijera H. Prediction of regenerative chatter by modelling and analysis of high-speed milling

[7] Zaghbani I, Songmene V. Estimation of machine-tool dynamic parameters during machining operation through operational modal analysis, International Journal of Machine Tools & Manufacture, 49 (12-13) (2009), pp. 947–957

[8] Hongrui C, Bing L, Zhengjia H. Chatter stability of milling with speed-varying dynamics of spindles, International Journal of Machine Tools and Manufacture, Volume 52, Issue 1, January 2012, Pages 50–58

[9] Koch SF, Bauer J, Wagner H, Horsch J, Brecht S, Fleischer J, Characterization of an eigenfrequency adaptable machine tool carriage, 6th CIRP International Conference on High Performance Cutting, HPC2014

[10] Altintas Y, Weck M. Chatter stability of metal cutting and grinding. CIRP Annals-Manufacturing Technology, 2004, vol. 53, no 2, p. 619-642.

[11] Inasaki I, Karpuschewski B, Lee HS. Grinding chatter–origin and suppression. CIRP Annals-Manufacturing Technology, 2001, vol. 50, no 2, p. 515-534.

[12] Budak E, Kops L. Improving productivity and part quality in milling of titanium based impellers by chatter suppression and force control. CIRP Annals-Manufacturing Technology, 2000, vol. 49, no 1, p. 31-36.

[13] Ramesh R, Mannan MA, Poo AN. Tracking and contour error control in CNC servo systems. International Journal of Machine Tools and Manufacture, 2005, vol. 45, no 3, p. 301-326.

[14] Shamoto E, Mori T, Nishimura K, Hiramatsu T, Kurata Y. (2010). Suppression of regenerative chatter vibration in simultaneous double-sided milling of flexible plates by speed difference. CIRP Annals-Manufacturing Technology, 59(1), 387-390.

[15] Sathianarayanan D, Karunamoorthy L, Srinivasan J, Kandasami GS, Palanikumar K. (2008). Chatter suppression in boring operation using magnetorheological fluid damper. Materials and Manufacturing Processes, 23(4), 329-335.

[16] Weck, M.: Werkzeugmaschinen 5. Springer-Verlag Berlin Heidelberg 2006, 352-354. ISBN: 9783540329510