A VIRTUAL ENGINEERING APPROACH FOR CHATTER PREDICTION

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Abstract

The prediction of machine behavior becomes more important for the development of machine tools because due to demand for shorter development phases. The concept of virtual engineering enfolds the holistic computational development of a product, which allows the evaluation of the machine design in early stages of development. The consequent application of this method for modeling and analyzing the behavior of a machine tool is described in the paper. As an example, a milling and drilling machine for workpieces with large dimensions is introduced and the most important parts for the analysis of machine dynamics were identified. Two models were built to represent the machine tool. The first one is a complex finite element model with volume elements, which needs a low number of assumptions related to the geometry and therefore it can be used to validate simplified models in the absence of a physical prototype in early design stages. The second one is a simplified beam model which is used to evaluate the dynamic behavior due to force excitation during machine operation and chatter prediction. The force functions for typical milling processes on the described machine are calculated and the conditions for the appearance of chatter are analyzed for the computation of stability lobes in time domain which help to identify the machining parameters yielding stable or unstable processes.

Keywords: Virtual Engineering, machine tool, milling, chatter, finite element analysis

Presenting Author's biography

Corinna Barthel studied mechanical engineering with emphasis on applied mechanics and graduated in August 2006. In her diploma thesis she investigated the methods and possibilities of the virtual development of a machine tool. Since September 2006 she is working as a researcher at the Institute of Mechanics in the area of computational mechanics.

1 Introduction

The prediction of the machine behavior during operation becomes more important in the design process, because of the demand for shorter development time with contemporaneous enhancement of design results. Due to the complex dependency of the properties, it is difficult to give universal design rules. Especially the properties of new machine tools can often be evaluated up to the assembly of a prototype. The analysis of the development process by the machine tools producers shows that decisions about the design are made without knowledge of the effects on the quality and the performance of the machine tool [1]. Solution approaches are simulation tools, which allow an early evaluation of the design and offer the possibility of improvements. The concept of virtual engineering enfolds the holistic computational development of a product. This process reaches from the 3D-based design via the simulation of the product behavior to realistic demonstration of the design and the behavior. Therefore the results of every single step pass over to next one. The result is a virtual product whose properties can be analyzed and evaluated. The gained cognitions can be used to derive an enhancement of the design. The tools required for a virtual engineering approach have been established since long time ago, but they have reached the required specification in combination with the appropriate interfaces recently, which enables the holistic computer-aided representation of a product. The whole process with the corresponding tools is illustrated in Figure 1.

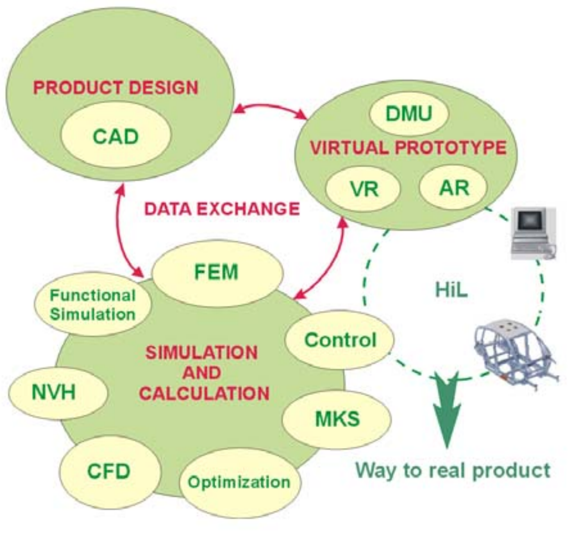

Figure 1 Process of virtual engineering [2]

The advantage of such a holistic approach is the retrenchment of development time and costs, since necessary enhancements can be recognized and effected earlier in the development process, which ensure a high quality of the first physical prototype. Its application is especially of interest in the development of machine tools for the conditioning of parts with large dimensions, because they are only produced in low quantities. An example addressed in this paper is the Horimaster 5 of the SCHIESS GmbH, a horizontal milling and drilling machine. This machine was investigated by the consequent application of the methods of virtual engineering. The focus was especially put on the prediction of chatter, because this phenomenon has a high influence on the machining accuracy and productivity. An applicable model for this investigation has to have a low number of degrees of freedom since the time integration is required for the computation of the dynamic effects. In addition this model has to ensure a defined accuracy of the results. In early design stages, it is not possible to validate a created model using a physical prototype. The idea in this paper is therefore to build a complex model with a low number of assumptions, relating to geometry directly from CAD data in order to have a reference for the evaluation of a simpler model which can be used for the investigation of the dynamic effects at the appearance of chatter.

Because of the high complexity of a machine tool, not every part can be considered for the modeling. For this reason the machine was analyzed first, for identifying the parts which have a significant influence on the demanded calculations. In the next step the complex finite element model consisting of volume elements was built as a reference model in the absence of a physical prototype. Regarding the calculation time and memory capacity, a simplified finite element beam model with less degrees of freedom was developed for the extensive calculations for the chatter prediction. It was validated using the complex model.

Furthermore the process of milling and the conditions for the appearance of chatter were analyzed and represented by a mathematical model. The models of the machine and the process were implemented in the finite element software ANSYS® for the calculation of the machine behavior. As an example of the obtained results the stability lobe of the milling of GGG70 with a 6 teeth cutter is presented.

2 Model of the machine tool

For the calculation of the required results a physical model of the machine has to be established, which describes the most important properties in mathematical form. Because the Horimaster 5 is a large and complex system, a complete modeling of all the parts is not possible. The parts, which have an essential influence on the results, have to be identified first by an analysis of the system.

2.1 Real System

The Horimaster 5 of the Schiess GmbH is a horizontal CNC-milling and drilling machine. Parts with a size of $2 \times 2 \times 2$ m and a mass of 100t can be processed. Figure 2 shows the whole machine with the different moveable axes. More information about the machine can be found in [3].

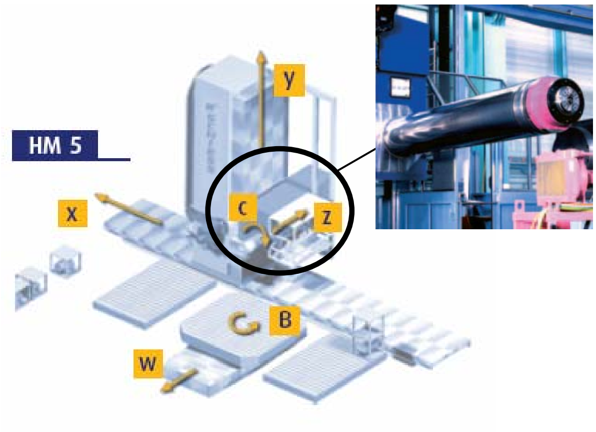

Figure 2 Machine Tool with ram [3]

The round and linear axes are equipped with wear-free hydrostatic guideways, which provide maximal efficiency for all desired machining tasks. The axes reach their defined positions by a CNC-control. It is possible to change the tools and heads automatically.

The assembly identified to be under highest load during machine operation, is the ram-assembly and the encompassing parts of its bearing (Figure 2). The ram is the non-rotating outer sleeve of the spindle. It contains all engines with their corresponding gears and driving shafts for the tool. The complete assembly is moveable along the z-axis by a ball screw. The ram is supported over its circumference with eight hydrostatic guideways and is protected against contortion with them.

2.2 Finite Element Model

For describing the elastic properties of the ram, which are very important in relation to its behavior caused by dynamic forces, the finite element method is used, since the influence of various geometries on the elastic properties can be considered.

First a finite element model with volume elements was developed, because it allows investigating the interaction of the ram-assembly considering the encompassing parts of the bearings and the whole complex geometry of the model. It can be described with using few assumptions related to geometry. The volume element model is applicable for the calculation of deformations and stresses caused by static loads and for the calculation of eigenfrequencies and eigenvectors. Due to the high number of degrees of freedom it is not convenient to compute the behavior of the tool under dynamic loads, but it can be used to validate a simplified model in the absence of a physical prototype in early development stages.

As described in the foregoing paragraph the ram is a very complex system with a series of functions. Therefore it is not possible to include every part of it in a simulation model. The calculation time would be too long and in addition to that many parts do not influence the result. The investigation of the static and dynamic properties of the assembly is mostly affected by the stiffness and the mass properties. Therefore, only those parts are taken into account, which have a significant contribution.

The creation of the finite element model took place with the 3D-geometry data, which was transferred to the preprocessor of the finite element software ANSYS®. The geometry was meshed with tetrahedral elements with quadratic shape-functions, because this element is applicable for the efficient meshing of the irregular geometry of the assembly and elements with quadratic shape-functions yield better results than linear elements, which generally make the assembly too stiff.

Some elements, like the two driving engines inside of the ram, are very stiff, so their elasticity can be neglected and they are modeled as a mass point, which is linked with the other parts by stiff beam elements.

The encompassing parts of the bearing are connected by linear springs, which represent the properties of the hydrostatic guideway.

The result of the meshing is a finite element model with 70598 elements and 119044 nodes (Figure 3).

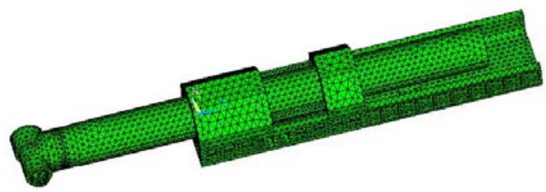

Figure 3 Finite element model with volume elements

For simulating the behavior of the ram-assembly during machine operation it is necessary to create a finite element model with a lower number of degrees of freedom. Therefore further assumptions have to be made.

The parts, which are identified to be significant for the ram-stiffness, are all large dimensioned in one direction in space, compared to the others. Therefore, it is possible to approximate them as a beam. The cross-section parameters are calculated using the CAD-software. The finite element model was created with beam elements based on the Timoshenko's theory, because the relationship between the dimension of the section and the dimension in the beam axis is very low, so the shear deformation cannot be neglected. The parts of the bearing cannot be considered because there is no possibility to reduce their number of degrees of freedom. The ram-assembly is supported against the inertial system. The

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simplified beam model of the ram consists of 183 elements and 191 nodes (Figure 4) and it was also implemented in ANSYS®. Due to this low number of degrees of freedom it is possible to realize time integration for investigating the dynamic behavior.

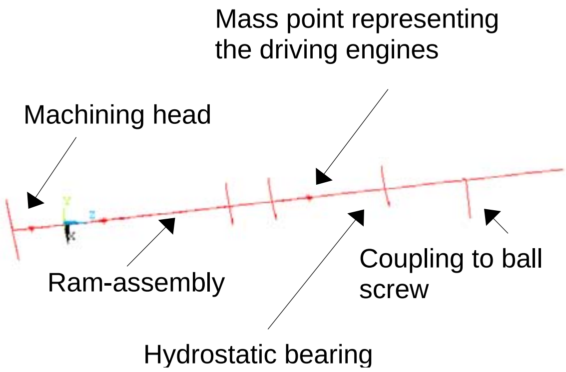

Figure 4 Beam element model

3 Model of the cutting process

Besides the modeling of the machine tool it is also necessary to derive a mathematical description of the cutting process. In this paper the process of slot milling which is a special type of face milling is considered exemplarily.

3.1 Milling

In Figure 5 the process of slot milling with the most important parameters is shown.

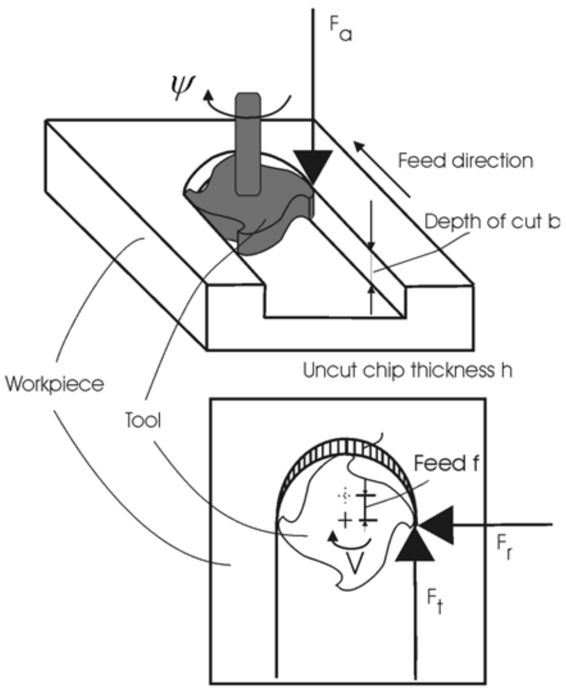

Figure 5 Slot milling

In milling processes, material is removed from the workpiece by a rotating cutter. It has often multiple cutting edges. The cutting edges on the milling cutter enter and leave the cut at each rotation, and they are actually cutting for less than half of total machining time [5].

The resulting milling forces depend on the workpiece material and the machining variables of the process cutting depth b and uncut chip thickness h. The cutting depth is constant for the considered process of face milling, but the uncut chip thickness depends on the feed rate f and varies with the instantaneous angle of immersion φ. The uncut chip thickness of a single cut z of the tool h_z is calculated from Eq. (1):

$$h_z = f_z \sin\varphi_z \cdot \sin\kappa \tag{1}$$

The angle κ is constant for the considered milling operation. It is about $\kappa = 90°$.

The angle φ of one cut z varies with the spindle angle of rotation $\psi(t)$ as shown in Eq. 2:

$$\varphi_z(t) = \psi(t) + \varphi_{z0} \tag{2}$$

The angle φ_{z0} is the instantaneous angle of the single cut z at time $t = 0$.

The resulting force from the cutting process is subdivided into three parts: The tangential force F_t, the radial force F_r and the axial force F_a (Figure 5). For calculation of the forces a number of approaches exist. In this paper the approach of [4] is followed. The cutting forces at a single cut z are described by two terms, which represent the most important effects of chip removal (Eq. 3)

$$\begin{aligned} F_{tz}(t) &= K_{tc}\, b\, h(\varphi) + K_{te}\, b \\ F_{rz}(t) &= K_{rc}\, b\, h(\varphi) + K_{re}\, b \\ F_{az}(t) &= K_{ac}\, b\, h(\varphi) + K_{ae}\, b \end{aligned} \tag{3}$$

The cutting edge coefficients K_{te}, K_{re}, K_{ae} describe the influence of rubbing and plowing on the cutting force and the cutting coefficients K_{tc}, K_{rc}, K_{ac} describe the influence of shear. The identification of the coefficients results from cutting experiments.

The entire cutting force results from the sum of the cutting forces calculated in Eq. (3) on each cut of the tool. It has to be noticed, that a cut is not in contact with workpiece over the whole revolution of the tool. For example, for a full immersion cutting process the cut is only in contact from $\varphi = 0°$ until $\varphi = 180°$. This dependence is taken into account by a function $\delta_z(\varphi)$ for every cut z, which is equal to 1 if the cut is in contact and equal to 0 if the cut is not in contact [5]. The whole cutting force is calculated from Eq. (4).

$$\begin{aligned} F_{t\,sum}(t) &= \delta_z(\varphi) \sum_z F_{tz}(t) \\ F_{r\,sum}(t) &= \delta_z(\varphi) \sum_z F_{rz}(t) \\ F_{a\,sum}(t) &= \delta_z(\varphi) \sum_z F_{az}(t) \end{aligned} \tag{4}$$

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The resulting force function for all the three parts is a periodic but non-harmonic function. Figure 6 shows the tangential force for an ideal rigid machine tool. The forces in the other directions look similarly.

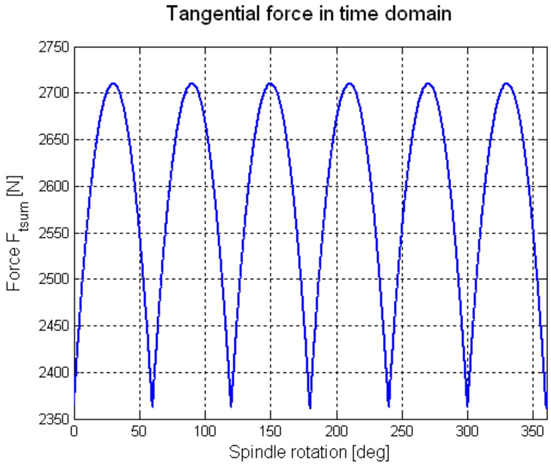

Figure 6 Tangential Force for an ideal rigid machine tool in time domain (6 teeth cutter, v=240rev/min, b=2mm, f=0.3mm, material GGG70)

In frequency domain the tooth passing frequency f_c can harmonics ($2f_c$, $3f_c$ etc.) of it can be recognized. The amplitude of the tooth passing frequency has the highest amplitude and the amplitude of the harmonics become smaller with rising frequency.

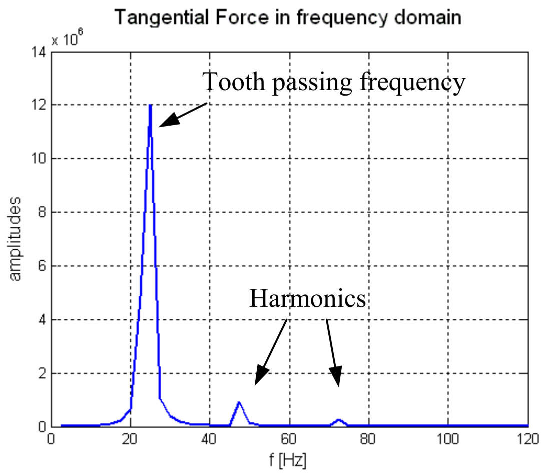

Figure 7 Tangential Force for an ideal rigid machine tool in frequency domain (6 teeth cutter, v=240rev/min, b=2mm, f=0.3mm, material GGG70)

3.2 Chatter

Chatter vibrations result from a self-excitation mechanism in the generation of chip thickness during machine operation [1]. The reaction of the cutting process on the elastic machine structure can induce an instable process. The relative movement between machine and tool rises until a stationary value. The energy for perpetuation of the vibration is taken periodically from the engine. In comparison with forced vibrations due to the forces resulting from the cutting process, the amplitudes are considerable higher, which results in a bad surface quality of the workpiece. There are a number of mechanisms, which induce chatter. This paper is dealing with the most important one: the regenerative chatter of chip thickness. At this mechanism the surface roughness due to the foregoing cut influences the current cut. If the machine vibrates, the vibrations are mapped on the workpiece. Due to this waviness on the surface the uncut chip thickness h is influenced. It follows the surface roughness. Eq. (1) has to be extended by two parts. The first one represents the current deflection of the tool, which has a significant influence due to the high amplitudes during chatter. The second one represents the waviness of the surface due to the vibration of tool at time $t = T$, when the foregoing cut was at this position.

The extended calculation of the uncut chip thickness h_z is shown in Eq. 5, where $x(t)$ is the deflection of the tool in x-direction and $y(t)$ is the deflection in y-direction. The notation is also illustrated in Figure 8.

$$\begin{aligned} h_z(t) = f_z \cdot \sin\varphi + [y(t) - y(t-T)] \cdot \cos\varphi \; + \\ + [x(t) - x(t-T)] \sin\varphi \end{aligned} \tag{5}$$

The time T is calculated from Eq. (6), where n_z is the number of the cuts and v is the number of revolution per time of the spindle:

$$T = \frac{1}{n_z \cdot v} \tag{6}$$

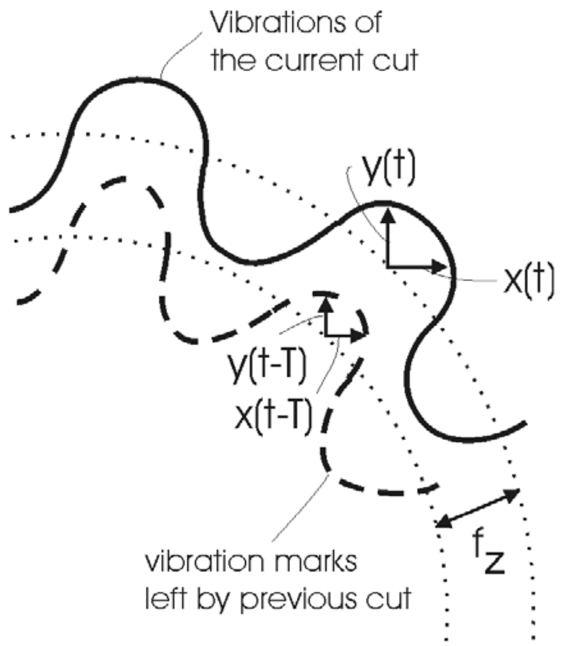

Figure 8 Mechanism of regenerative chatter

The conditions, which involve the self-induced vibrations, depend on the parameters of the cutting process. The variable parameters with given workpiece and tool are depth of cut b and spindle speed v. The stability is only being given for certain combinations of these two parameters. This dependency can be plotted as a stability lobe. The calculation of such a stability lobe can be realized analytically with investigations in frequency domain or numerically with investigation in time domain. In this paper the second one is used because all nonlinearities remain. In [6] a criterion for detection of chatter vibrations is established. It allows the distinction between the permanently occurring stationary vibrations during milling and vibrations due to chatter. A non-dimensional coefficient is calculated, see Eq. (7), where $h_{s,max}$ is the maximum uncut chip thickness in a simulation with a rigid machine tool and $h_{d,max}$ is the maximum uncut chip thickness in a time domain simulation with an elastic machine tool.

$$\eta = \frac{h_{d,\max}}{h_{s,\max}} \tag{7}$$

A cutting process is unstable if the coefficient reaches:

$$\eta > 1.25 \tag{8}$$

Now, it is possible to perform a number of FEM-simulations automatically to find the stability limits.

4 Results and Discussion

First, the beam element model was validated by a comparison with the volume element model. It reaches a good agreement up to 120 Hz. The eigenfrequencies are 5% higher than those from the volume element model. The eigenmodes higher than 120 Hz are not representable with the beam element model because they are determined by vibrations of the parts of the ram bearing, which cannot be represented by the beam model. To summarize, the beam model can be used for calculation of the behavior of the ram due to dynamic force excitation with frequencies up to 120 Hz.

The analysis of the force function in milling shows a periodic but non-harmonic function in time domain. In frequency domain the tooth passing frequency and harmonics of it can be recognized. The amplitudes of the harmonics become considerably smaller with higher frequencies, so the model can be used for milling force functions with a tooth passing frequency up to 120 Hz.

As an example for this paper, the behavior of the machine is calculated due to a slot milling process with the following parameters: number of cuts n_z = 6, material GGG70. The coefficients are extracted from the literature, see [7].

The calculation of the stability lobe is performed for a spindle speed from Ψ = 1 rev/min to Ψ = 1200 rev/min. For higher speeds the tooth passing frequency is higher than 120 Hz and the model would not be validated. The stability lobe was calculated with an accuracy of 3 rev/min for the spindle speed and 0.05mm for the depth of cut. The implemented search algorithm starts with a large search interval and after recognizing chatter an interval bisection algorithm starts to find the chatter bound at the stated accuracy. The calculation of the stability lobe required many hours. The results are shown in Figure 9.

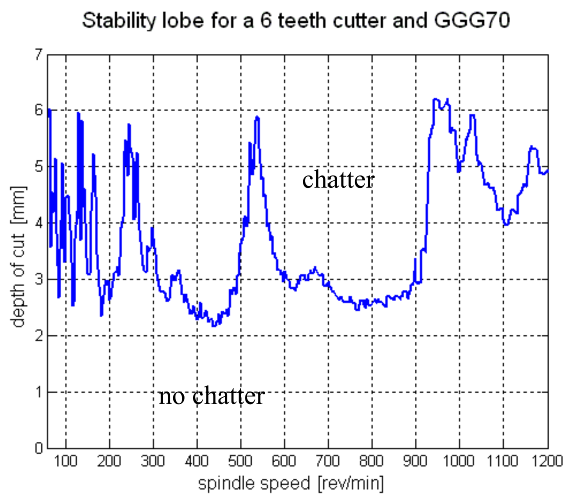

Figure 9 Stability lobe

The figure shows which value of the operation parameters, depth of cut b and spindle speed v, have to be chosen to ensure a stable process. In this example the milling process is always stable for a depth of cut smaller than 2 mm.

In Figure 10 the vibrations of the cutter for a stable process are shown. The amplitudes have a constant and low value which leads to a good surface quality.

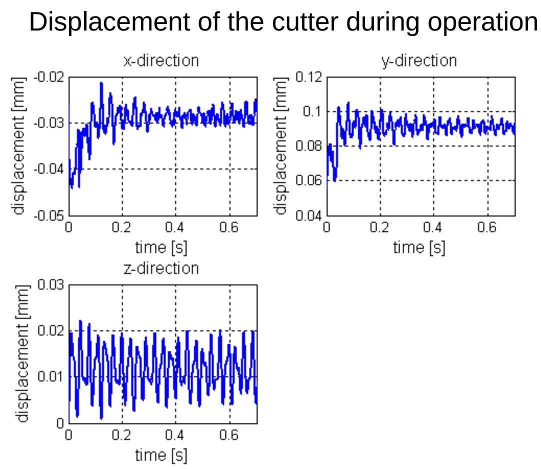

Figure 10 Vibrations for a stable cutting process with a 6 teeth cutter and GGG70 (v=240rev/min, b=2mm, f=0.3mm)

In comparison to Figure 10 the vibrations of the cutter for an unstable process are shown in Figure 11. The amplitudes are growing during the cutting process and at t=0.7s they are 10 times higher than those of the stable process. This induces chatter marks on the workpiece and results in a bad surface quality.

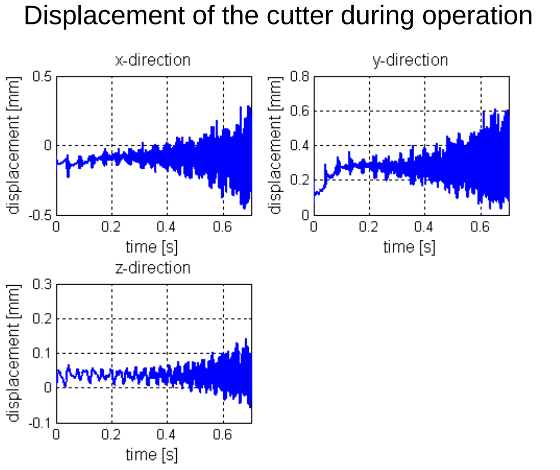

Figure 11 Chatter vibrations with a 6 teeth cutter and GGG70 (v=240rev/min, b=6mm, f=0.3mm)

5 Conclusion

In the paper was shown, that a holistic computer-aided development of a machine tool is possible and can make the development process more efficient. Due to the consequent use of the method of virtual engineering many different investigations of a machine tool can be performed before assembling a physical prototype. With the aid of a very precise model with many degrees of freedom it is possible to validate a simpler model, which allows to investigate the dynamic behavior during chatter appearance. The very important stability lobes can be calculated for a various number of workpiece-tool combinations and with the results improvements of the design can be initiated in an early stage of development. Furthermore, an instruction for the user of the machine can be published, which includes for example recommended spindle speed for a given tool and workpiece material.

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