

# Integrated System Simulation of a Milling Machine

R. Helfrich, B. Kirchgäßner, G. Nötzel-Steidle,

INTES GmbH, Stuttgart, Germany

## Summary:

The dynamic behavior of the complete system is crucial for the development of machine tools. The complete system comprises structural parts, drives of the various axes, and their control. During the interaction of tool and workpiece, the dynamic forces are causing vibrations of the machine, which have to be damped out sufficiently by all system components. Finally, the machine tool should provide highest accuracy and cutting depth at high speed.

The FE (Finite Element) method is the standard method for numerical vibration analysis in many application fields. So, it is quite natural to use this method for complete machine tools. Of course, drive components and control have to be included in the FE model in an appropriate manner. Then, all known analysis methods like eigenvalue analysis, frequency response analysis, and time-history response analysis are available for the computation of the dynamic behavior of the machine tool. In particular, numerical stability analysis of machine tools is of utmost importance for the development of machine tools to predict stable process parameters and chatter frequencies.

A simple milling machine model is used to show the modelling and analysis of machine models from eigenfrequencies up to stability charts.

## Keywords:

System simulation, machine tool, dynamic analysis, stability chart, chatter frequencies, Finite Element Method

## 1 Introduction

During the long history of machine tool design, the main focus was always to produce high precision parts as fast as possible. During machining the interaction between work piece and tool generates cutting forces which may cause vibrations in the system. On the other hand, due to material and transportation costs, there is also a long-term trend to reduce the weight of the machines which also leads to more vibrations in critical frequency ranges.

Numerical analysis of vibrations is a standard application field of the FE (Finite Element) method. Beside real eigenmodes and frequencies, complex modes, frequency response analysis, and time-history response analysis are available to predict vibrational behavior. In addition, optimization methods can be used to propose model modifications which improve the characteristics of the machine tool like weight, static response, and dynamic response. In order to use these methods for machine tool analysis, the model has to be enhanced by a few additional modelling features:

- **Drive components:** For all active axes, guide rails, ball screw drives, electrical motors, and any transmission ratios between different components have to be added to a FE model of the structural components. Classical means of FE modelling like bars, springs, dampers, and MPC conditions can be used to model such features.
- **Controllers:** The combination of sensor, actuator, and control logic can be seen as a controller. Special controller elements are provided in FE analysis. In particular, linear controller elements allow for all dynamic analysis methods as mentioned above. Nonlinear controller elements are possible, but linear frequency response analysis is not supported for such elements.
- **Working process:** For stability analysis of a milling machine tool, the milling process is essential. So, the process forces between milling cutter and work piece have to be determined and taken into account during a time-history analysis.

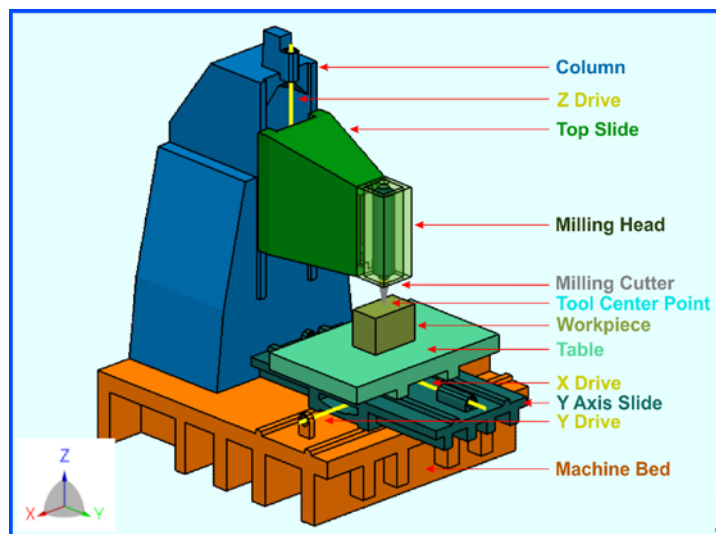


Fig. 1: Milling machine model

For the purpose of the FE model, there are two generally different views:

- **View on machine tool design:** For dynamic analysis of machine tools, the view is the optimum design of the machine tool and not the design of the control. So, it is indispensable for the correct view on machine tool dynamics to take control into account in order to include all relevant effects which influence the dynamic behavior.
- **View on the controller design:** Control design usually also needs some information on the elastic structure of the machine tool. This is provided by dynamic reduction methods through FE analysis. A special export of the reduced matrix system allows the subsequent design of controllers taking elastic information on the machine tool into account.

This paper concentrates on the first aspect: machine tool design using system simulation by suitable FE analysis.

Previous work has been documented in [1], [2], and [3] and published in [4], [5], and [6].

The paper is based on a simple model of a milling machine in order to provide a demonstration model for stability analysis of milling machines. The model is introduced first to show the various components of the machine tool model (see Fig. 1). Then, modelling and effect of control is described, followed by the dynamic behavior in frequency and time domain. Three controllers are applied for the control of the axes of the workpiece and another one for the control of the tool axis. Finally, the generation of stability charts and the detection of chatter frequencies is described.

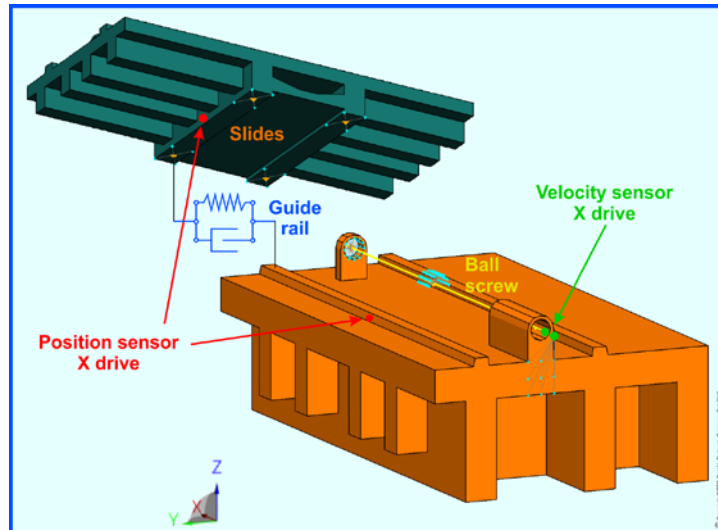


Fig. 2: Configuration of X drive with sensors and guide rail connections

## 2 Example Model

The model has the following typical components:

- **Structural components:** Machine bed, X, Y, and Z drive, milling head, and column are usually modelled by solid elements (see Fig. 1).
- **Guide rails:** They are part of structural components, but their proper connection is modelled by spring-damper combinations (see Figs. 2, 3, 4 for X, Y, Z drives), where the spring and damper forces are connected to the solid structures by surface coupling enabling incompatible meshes.

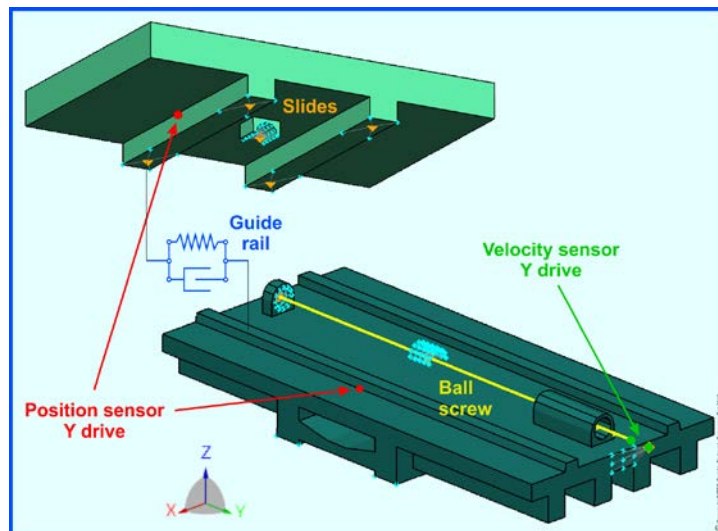


Fig. 3: Configuration of Y drive with sensors and guide rail connections

- **Ball screw drives:** They are modelled by beam elements (see Figs. 2, 3, 4). Their function is to transform a rotational motion of the drive to a translational motion of a carriage. This

transformation is achieved by a functional connection taking the diameter of the screw and the pitch of its thread properly into account.

- **Main spindle and milling head:** The spindle is usually modelled with solid elements. The bearings are modelled with special elements, which provide speed dependent stiffness and damping.

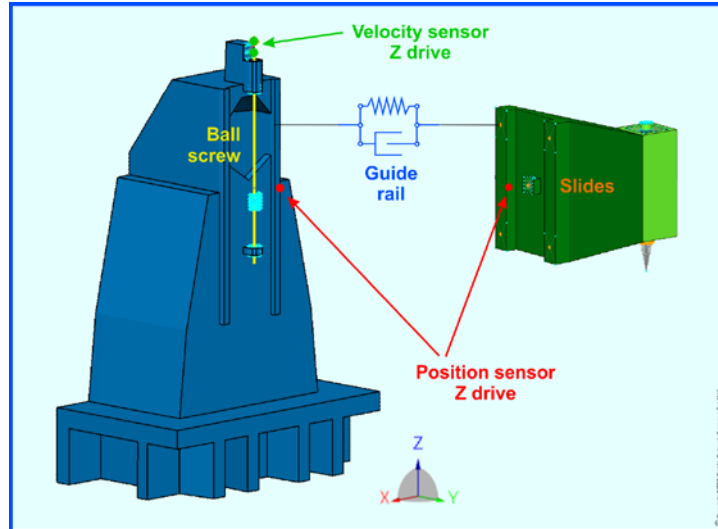


Fig. 4: Configuration of Z drive with sensors and guide rail connections

A number of parts could also become a part of the model, but were omitted for simplification reasons, like motors. The machine bed is flexibly supported to the ground by springs and dampers. The weight of the model is 8,345 kg.

### 3 Control

In order to include a controller in an FE model, an element is available which represents a cascade controller as shown in the block diagram of Fig. 5. This cascade controller is able to represent position and velocity controllers beside some filters and a current controller. This control element comprises the typical control steps used in machine tools and has been developed based on the experiences of machine tool manufacturers.

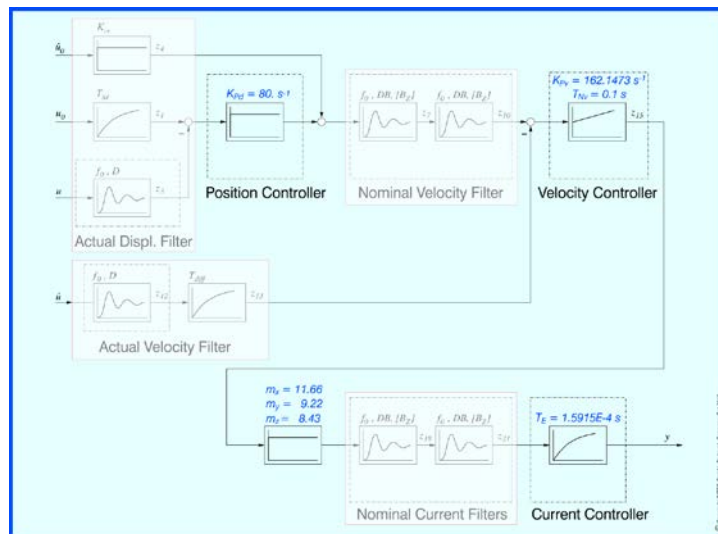


Fig. 5: Block diagram of the cascade controller CONTRL8

Such controllers are leading to a system of linear differential equations which can be exactly incorporated in the linear vibration analysis by the FE method. The resulting finite element CONTRL8 allows the selection of the actual components of a controller and the setting of the parameters of these components (see Fig. 5).

The parameter settings have to reflect the actual controller which will be used for the machine tool. So, the parameters have to be provided to the analyst by the controller development. In this way, the analyst will be able to support the machine design by stress and durability analysis.

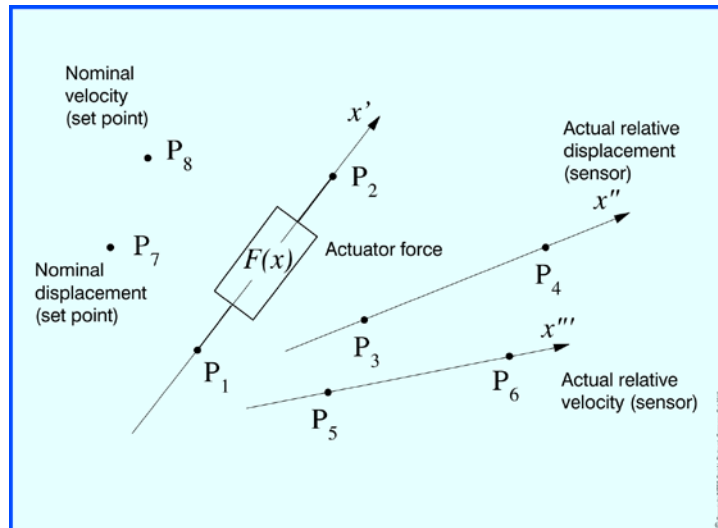


Fig. 6: FE representation of the controller element CTRL8

The finite element representation of the control element is shown in Fig. 6, where the eight nodes have following functions:

- Node 1 and 2: The actuator force is applied here.
- Node 3 and 4: Sensor to provide relative displacements (see Figs. 2, 3, 4).
- Node 5 and 6: Sensor to provide relative velocities (based on rotation speed sensors), which are located at the ball screw drives (see Figs. 2, 3, 4).
- Node 7: Set point for nominal displacement (see below).
- Node 8: Set point for nominal velocity (if any).

It is worth mentioning that due to the availability of such control elements, a fully coupled and integrated FE analysis is provided.

In the example model, there are three controllers, one for the translational motion parallel to the spindle axis (Z axis) and one for the motion in radial feed direction (X axis). The parameters of the controllers are the same (see Fig. 5) except the parameter  $m$  which reflects the moved masses.

The positions of the sensor points can be seen from Figs. 2, 3, 4. Two sensor nodes are needed to get the relative displacement or relative velocity. One of the two nodes is located on the bed and the other node is located on the carriage.

#### 4 Real Modes and Frequency Response Analysis

One important advantage of using the FE approach for linear controllers is the applicability of modal methods for frequency response analysis. Fig. 7 and Fig. 8 show the frequency response graphs for an excitation in X direction for both without and with active controller. Fig. 9 and Fig. 10 show the frequency response graphs for an excitation in Y direction for both without and with active controller.

The modal basis is not changed by the controller, i.e. real eigenmodes are identical for both the controlled and uncontrolled model. A number of 268 modes has been calculated up to 2500 Hz. The first eigenfrequencies are:

Mode no.	Eigenfrequency [Hz]
1	14.12
2	16.03
3	23.09
4	32.36
5	34.59
6	35.85
7	78.25
8	88.30
9	93.80
10	124.66

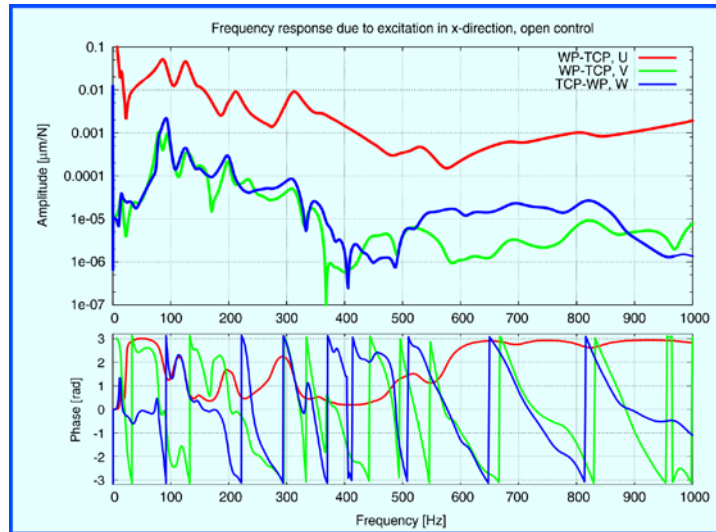


Fig. 7: Frequency response for excitation in X direction without active controller

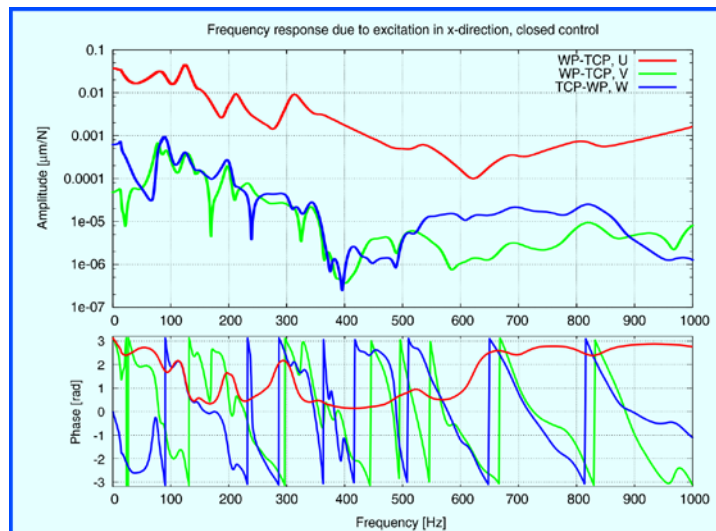


Fig. 8: Frequency response for excitation in X direction with active controller

The first six modes are vibration modes of the ground stiffnesses with the mass of the machine. From mode number 7 elastic machine modes are calculated. Modal damping is applied to all vibration modes by a damping ratio of 0.03.

Modal response analysis works with a limited number of modes. The neglected modes could have some influence on the response results. Therefore, it is good practice to amend the modal space by so-called static displacement modes which improve the response results (e.g. in the quasi-static frequency range). Static load cases to derive the static mode shapes are e.g. the element forces of spring and damper elements in the model. In this way, the quasi-static deformation caused by spring

and damper forces are represented in the modal space, although the corresponding eigenfrequencies are not contained in the limited frequency range. The modal damping factor for the additional static mode shapes is set to 1.0.

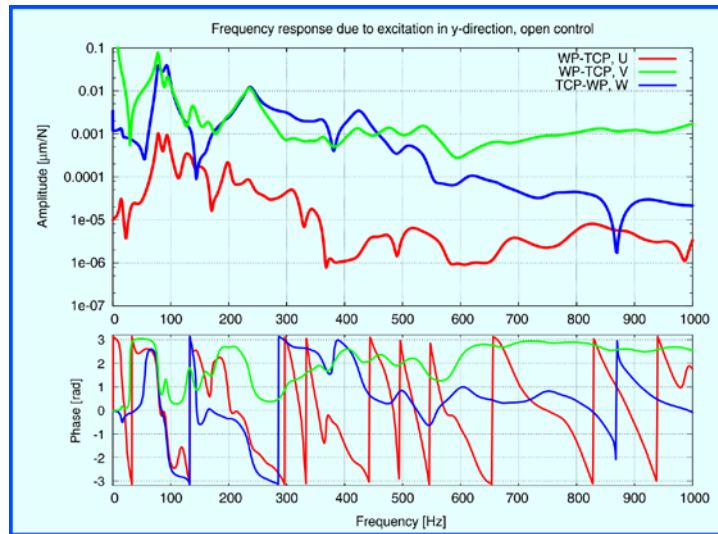


Fig. 9: Frequency response for excitation in Y direction without active controller

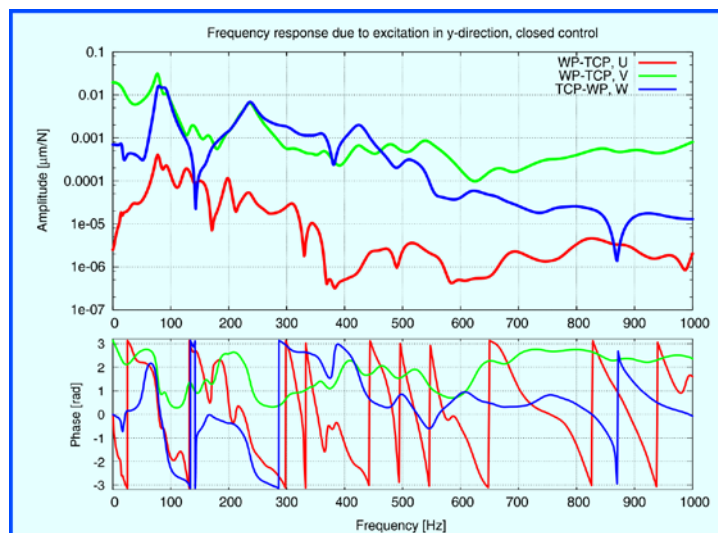


Fig. 10: Frequency response for excitation in Y direction with active controller

The frequency response results show the transfer function between TCP (Tool Center Point) and workpiece. Both nodes have the same coordinates, but they are not coupled.

Comparing the response without and with active control, we see a shift of the first peaks to lower frequency due to the effect of the control elements on the stiffness. Complex eigenvalue analysis can be used to calculate shapes and frequencies of the complex modes. The complex mode pairs at a spindle speed of 3000 rpm for the real modes 7 to 10 with the equivalent modal damping ratios are as follows:

Complex mode no.	Frequency [Hz]	Equivalent damping ratio
83	77.43	0.054
85	84.77	0.152
87	91.77	0.071
89	124.86	0.059

Fig. 11 shows the complex mode shape at 84.77 Hz compared to real mode shape at 88.30 Hz.

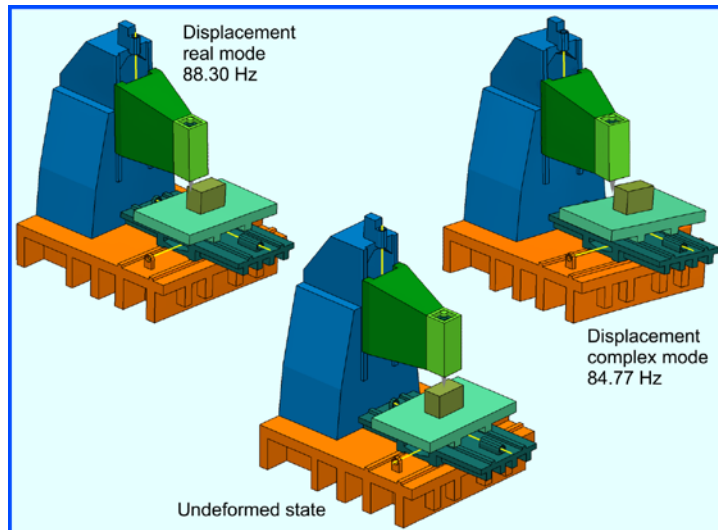


Fig. 11: Comparison of real mode 88.30 Hz and complex mode 84.77 Hz

Because the spindle is rotating, the influence of the rotor effect on the response should be investigated. The influence of the rotor effect is not significant for this model as shown in the Campbell diagram of Fig. 12.

## 5 Time-History Response Analysis

For the quality of control, the response in time domain is significant after applying a set point for a position change, for example. Two typical imposed position changes are:

- An abrupt jump in X direction as shown in Fig. 13 under a spindle speed of 3000 rpm.
- A smooth jump using a sine wave as shown in Fig. 14 also under a spindle speed of 3000 rpm.

The workpiece values are taken from the sensor. The TCP values are taken from the tool center point. The difference is small and stems from the additional elasticity between sensor and TCP. The other axes show a smaller overshooting than the X direction.

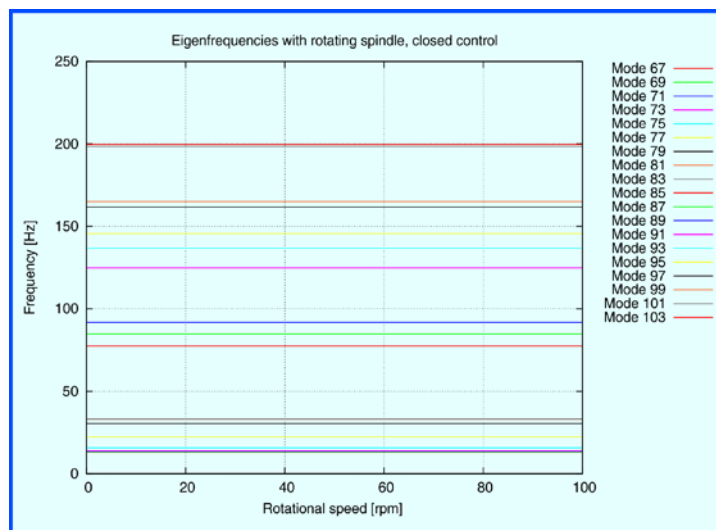


Fig. 12: Campbell diagram



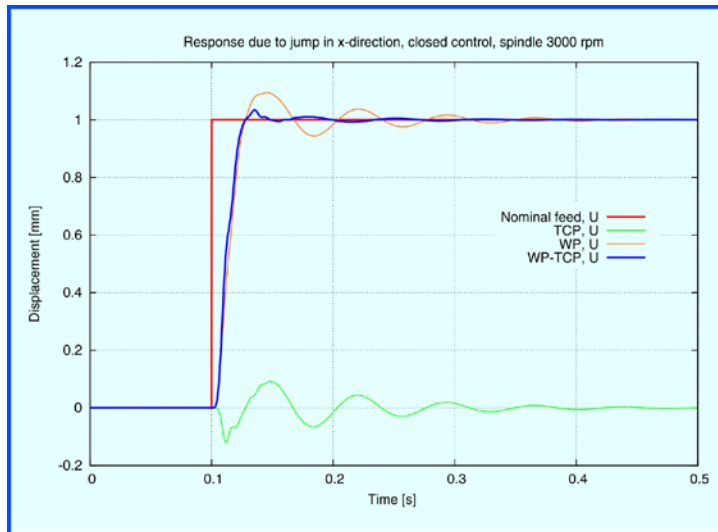


Fig. 13: Abrupt jump of table in X direction

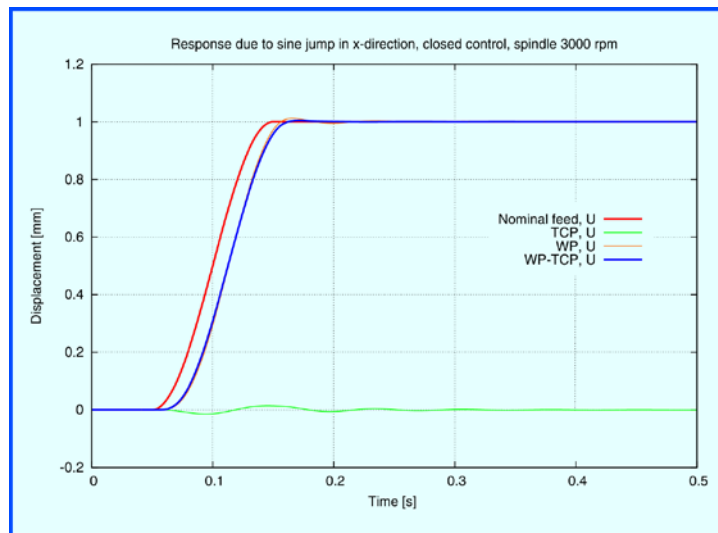


Fig. 14: Sine jump of table in X direction

In addition, a circularity test is performed where a circle path has to be followed by the TCP (see Fig. 15). The motion starts with zero velocity and limited jerk until final velocity of 10 m/s is achieved.

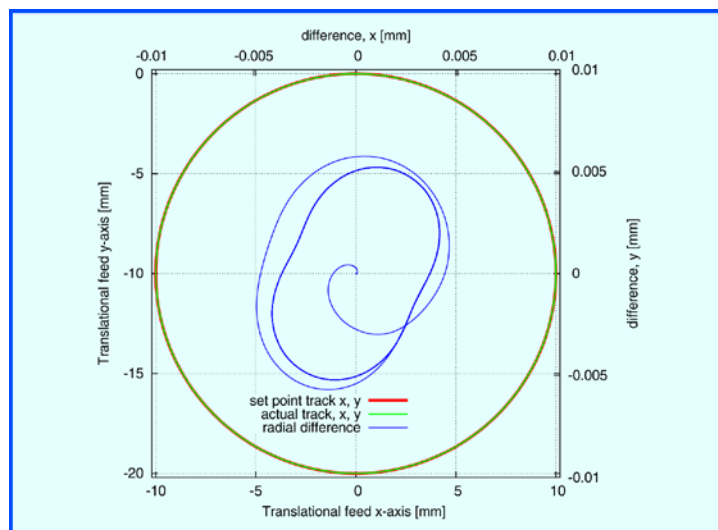


Fig. 15: Circulatory test with radius 10 mm

## 6 Stability in Operation

For the design of a machine tool, the prediction of instabilities during operation is of utmost importance to the manufacturer. The goal is to design a machine which works stable with high accuracy at high speed. The source of instability is the interaction between tool and workpiece, where the energy for self-exciting vibrations is coming from the rotating spindle and the table motion.

In order to make the nonlinear process predictable by FE analysis a cutting force model has been developed which is implemented by a nonlinear control element. In this way, time-history analysis in modal or direct approach is available to evaluate the response in time domain. The cutting force model depends on the cutting depth and the feed per revolution. The cutting force model requires a number of coefficients which reflect the interaction of a specific milling cutter with the workpiece. The cutting force model of Kienzle & Victor [7], [8] is used here.

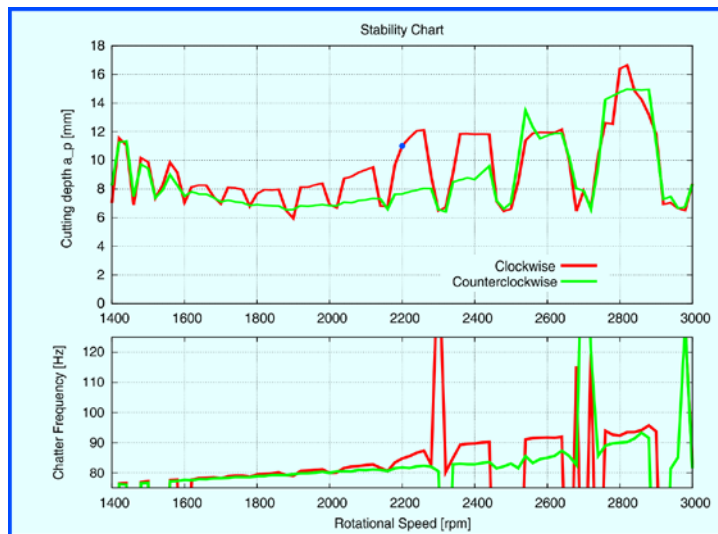


Fig. 16: Stability chart and chatter frequencies

If excited vibrations are not damped sufficiently in time domain, an instability is found for a specific rotational speed. After scanning the parameter field for different cutting depths and different rotational spindle speeds, one gets a stability chart as shown in Fig. 16. The cutting process is assumed to be a so-called face milling. While the chart represents varying cutting forces and rotational spindle speeds, the feed per revolution is kept constant (at 0.72 mm/rotation).

One important effect of spindle rotation is the dependency of the stability on the rotating direction of the spindle. Here, clockwise rotation allows for higher cutting depths than counterclockwise rotation of the spindle.

Because beginning of chattering can be heard by the operator, a prediction of chatter frequencies shows that the typical frequencies are mainly between 70 and 100 Hz, which in fact are in the audible frequency range of human ears (see Fig. 16). A few outliers produce even higher chatter frequencies.

In order to show an example for stable and instable behavior, the following operating point is used which is between clockwise and counterclockwise stability curve (see blue dot in Fig. 16):

- Cutting depth 11.025 mm
- Rotational speed 2200 rpm

Fig. 17 shows the forces at TCP for the stable clockwise case and Fig. 18 shows the forces at TCP for the instable counterclockwise operation.

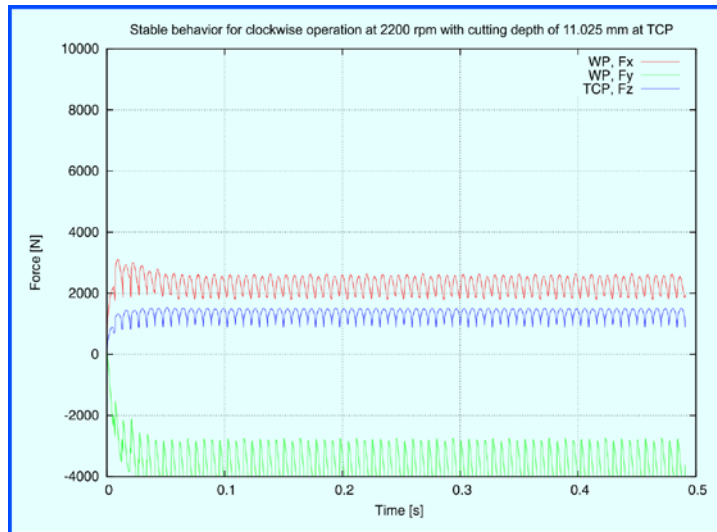


Fig. 17: Stable behavior for clockwise operation at 2200 rpm

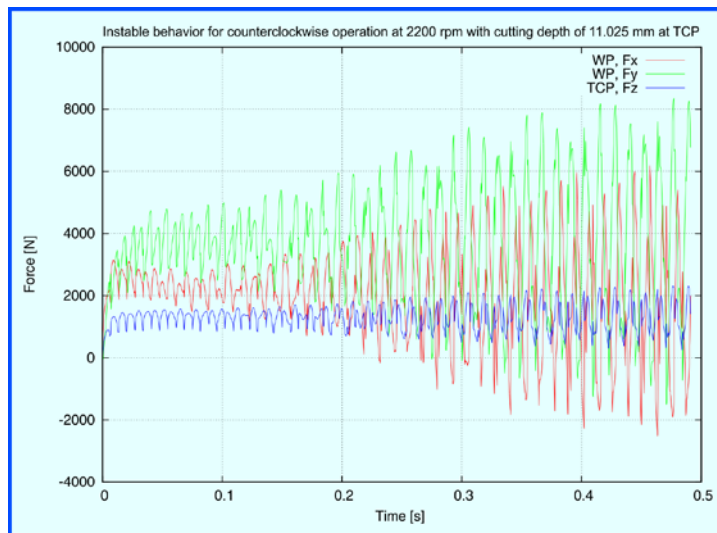


Fig. 18: Instable behavior for counterclockwise operation at 2200 rpm

## 7 References

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