Integrated System Simulation of Machine Tools

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Summary:

For the development of machine tools, dynamic behaviour of the complete system is of utmost importance for the efficiency and precision of the machines. The complete system consists of structural parts, drives in different axes, and control. During machining the interaction between work piece and tool generates cutting forces which may cause vibrations in the system. These vibrations have to be sufficiently damped by all system components. Finally, the machine tool has to provide high speed and high precision.

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

Zusammenfassung:

Bei der Entwicklung von Werkzeugmaschinen spielt das dynamische Verhalten des Gesamtsystems für die Effizienz und Präzision der Maschinen die entscheidende Rolle. Das Gesamtsystem besteht dabei aus den Strukturteilen, den Antrieben der verschiedenen Achsen und ihrer Regelung. Bei der Bearbeitung des Werkstücks durch das Werkzeug entstehen Kräfte, welche die Maschine zu Schwingungen anregen können, die durch die Systemkomponenten ausreichend gedämpft werden müssen. Am Ende soll die Maschine bei hoher Geschwindigkeit ein Höchstmaß an Genauigkeit erreichen.

Die FE (Finite-Elemente)-Methode hat sich bei der numerischen Untersuchung von Schwingungen in vielen Anwendungsbereichen etabliert. Daher lag es nahe, auch das Gesamtsystem Werkzeugmaschine mit diesem Verfahren zu berechnen. Dazu müssen die Antriebskomponenten und die Regelung in geeigneter Weise in das FE-Modell integriert werden. An Hand des Beispiels einer einfachen Drehmaschine wird die Modellierung des Gesamtsystems und die Berechnung von Eigenfrequenzen bis hin zur Stabilitätskarte dargestellt und erläutert.

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, the turning process of a turning machine tool is essential. So, the process forces between lathe tool and work piece have to be determined and taken into account during a time-history analysis.

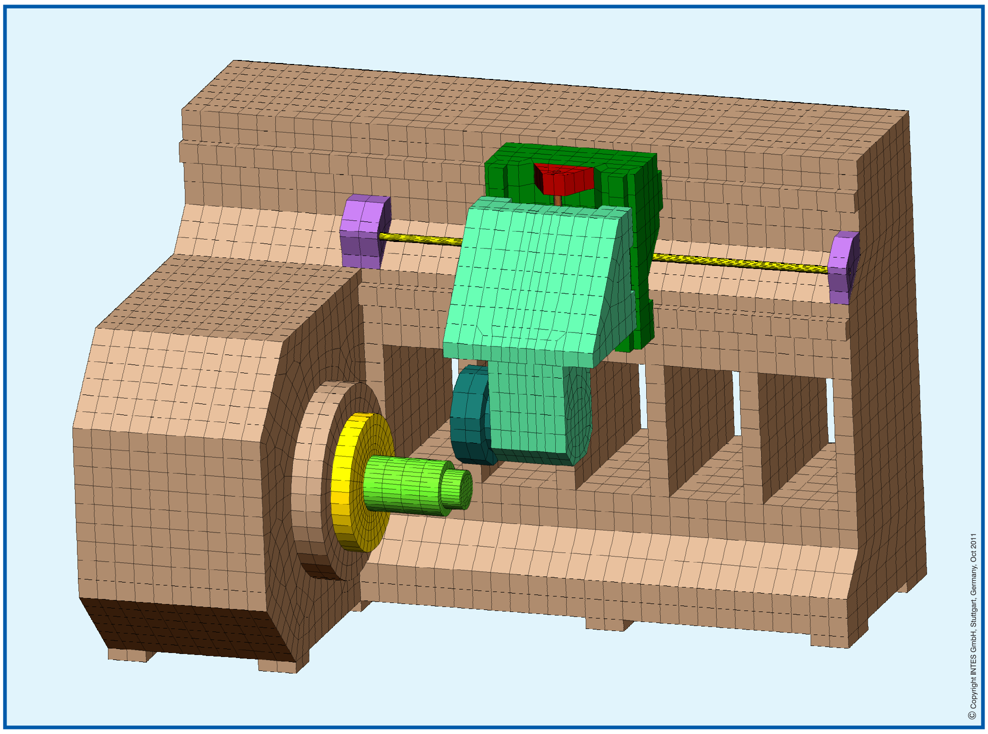

Fig. 1: Turning machine model (proposed by WZL)

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 indispensible 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.

In some parts, the paper presents results achieved in the research project VispaB which was funded by the German Federal Ministry of Education and Research under reference number 02PC1011.

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

The paper is based on a small model of a turning machine which has been provided by the Laboratory for Machine Tools and Production Engineering (WZL) at RWTH Aachen, Germany, during the VispaB project in order to provide a demonstration model for stability analysis. A proper parameter setting for the controller has been achieved in cooperation with INDEX-Werke in Esslingen, Germany.

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. The purpose of the example is to show instability effects. So, the model setup is not the best possible configuration, because this would result in a stable behavior in any case.

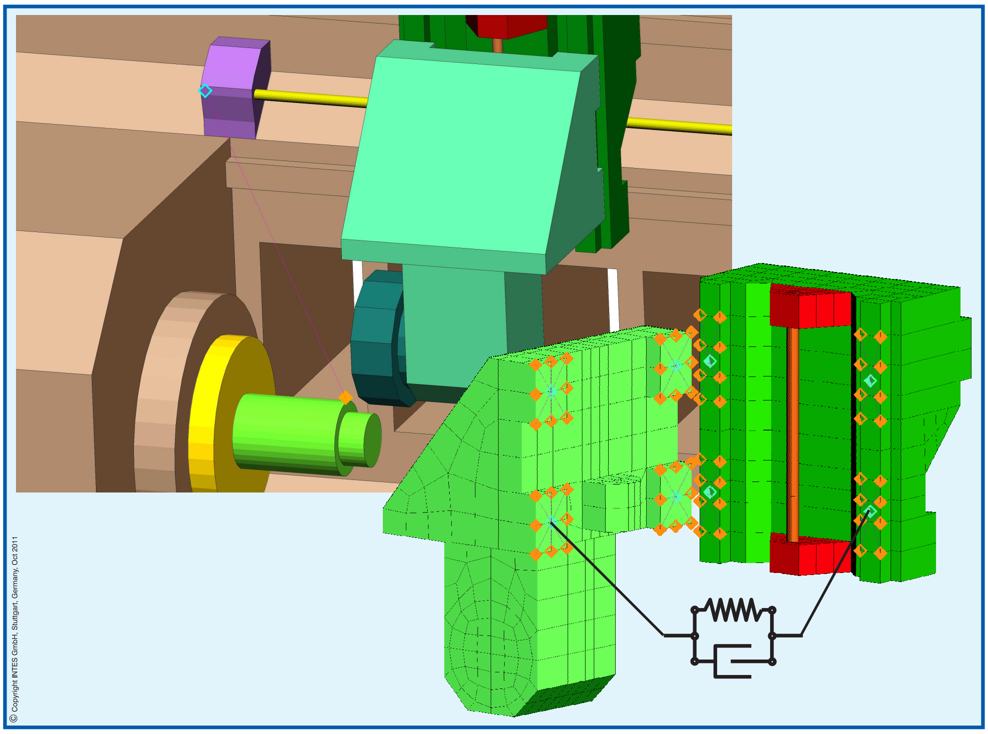

Fig. 2: Guide rail connection by spring-damper system

2 Example Model

The model has the following typical components:

- **Structural components**: Machine bed, carriages, and headstock 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 Fig. 2), where the spring and damper forces are connected to the solid structures by MPC conditions (i.e. weighted least square connections).

- **Ball screw drives**: They are modelled by beam elements (see Fig. 2). Their function is to transform a rotational motion of the drive to a translational motion of a carriage. This transformation is achieved by an MPC condition taking the diameter of the screw and the pitch of its thread properly into account.
- **Spindle with workpiece**: Spindle and workpiece are usually modelled with solid elements (see Fig. 3). The coupling to the headstock is done by a special element combined with suitable MPC conditions. The special element is used to provide rotation speed dependent stiffness and damping values for the spindle support.

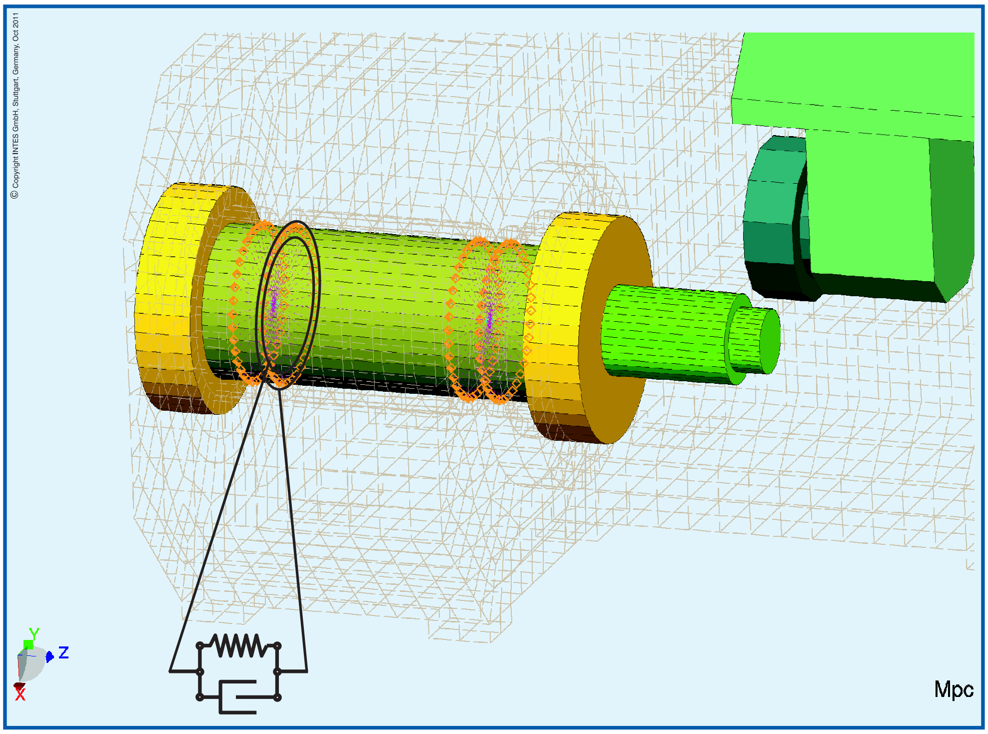

Fig. 3: Spindle supported in headstock with workpiece

A number of parts could also become a part of the model, but were omitted for simplification reasons, like tailstock and motors.

The machine bed is rigidly supported to the ground. Other conditions are of course possible, but were also omitted for simplification reasons.

The weight of the model is 9,455 kg.a

3 Control

In order to include a controller in an FE model, an element has been developed which represents a cascade controller as shown in the block diagram of Fig. 4. 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.

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. 4).

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.

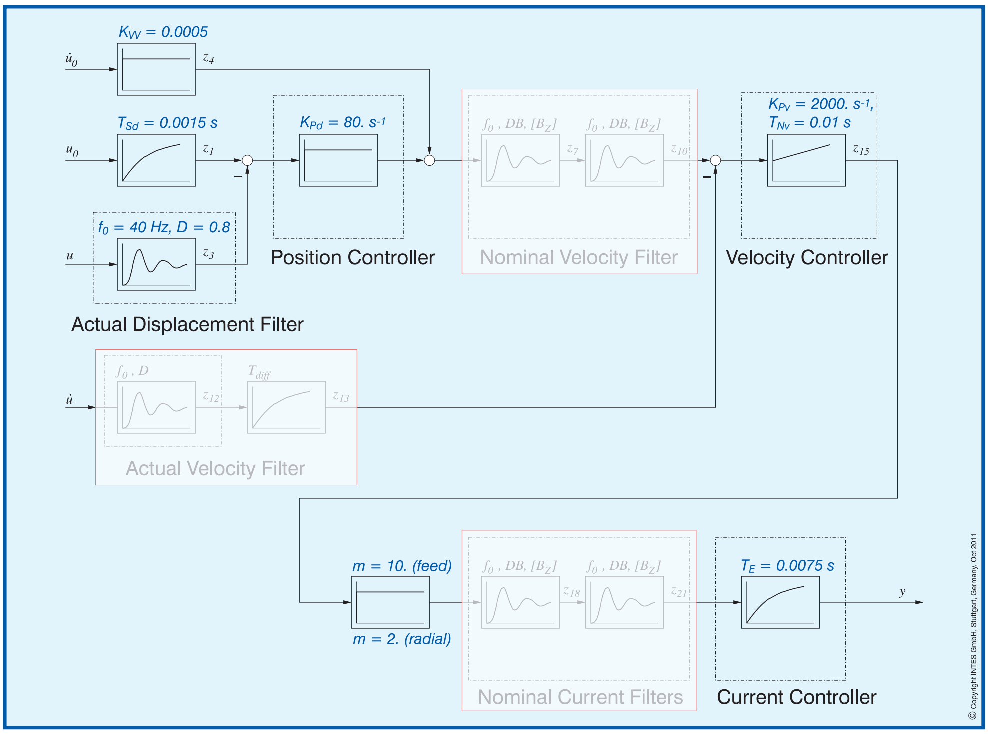

Fig. 4: Block diagram of the cascade controller CONTRL8

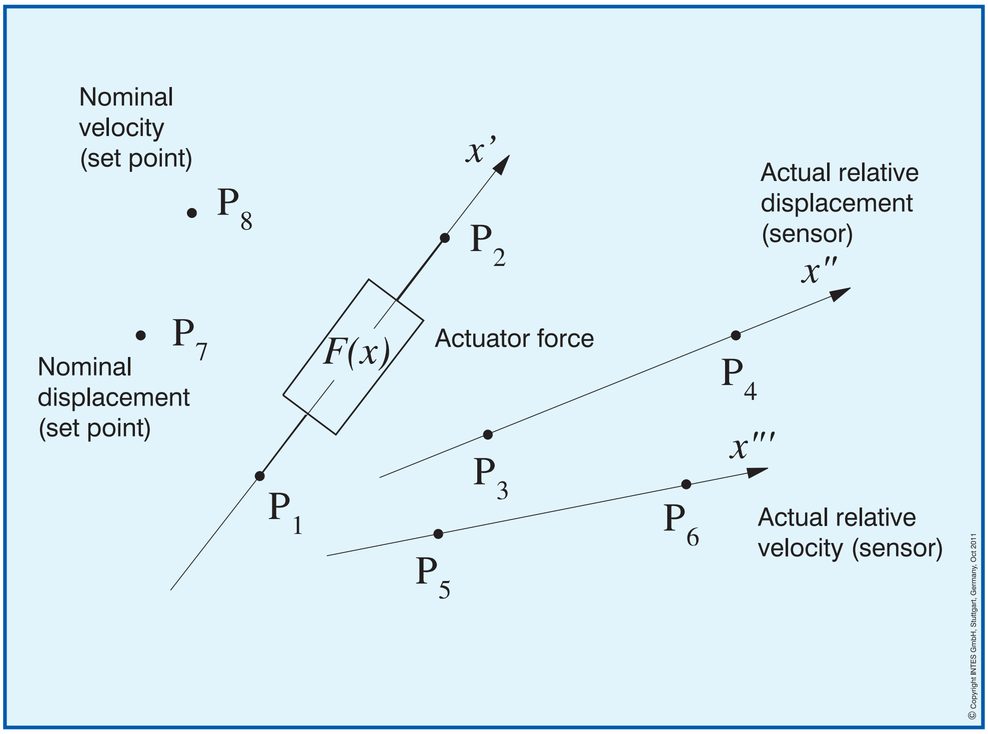

Fig. 5: FE representation of the controller element CONTRL8

The finite element representation of the control element is shown in Fig. 5, 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 Fig. 6).

- Node 5 and 6: Sensor to provide relative velocities (based on rotation speed sensors), which are located aat the ball screw drives.
- Node 7: Set point for nominal displacement (if any, see page 12).
- 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.

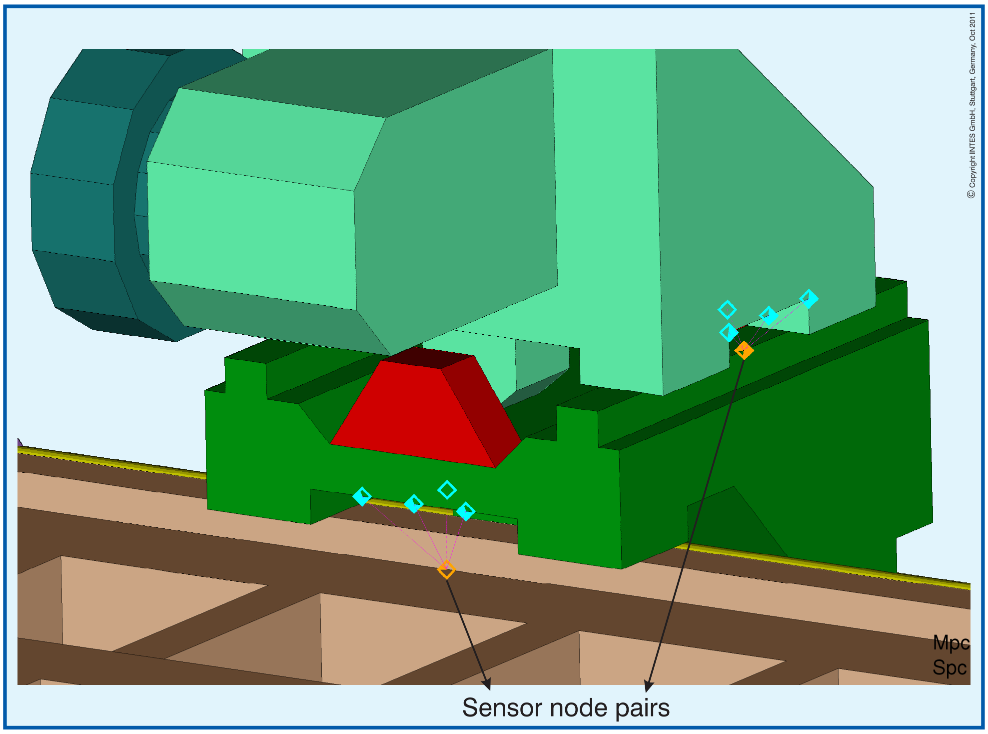

Fig. 6: Sensor locations to get relative displacements

In the example model, there are two 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 almost the same (see Fig. 4) except the parameter *m* which reflects the moved masses leading to higher value for the motion of the translational axis, because the carriage for the radial feed has to be moved in addition.

The positions of the sensor points can be seen from Fig. 6. 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 cases without and with active controller. For an excitation in z-direction, the response in frequency domain is shown in Fig. 9 and Fig. 10.

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 49 modes has been calculated up to 1500 Hz. Some important eigenfrequencies are:

Mode no.	Frequency [Hz]	Fig.
1	70	11
2	80	12
3	126	
4	174	
5	194	
10	244	

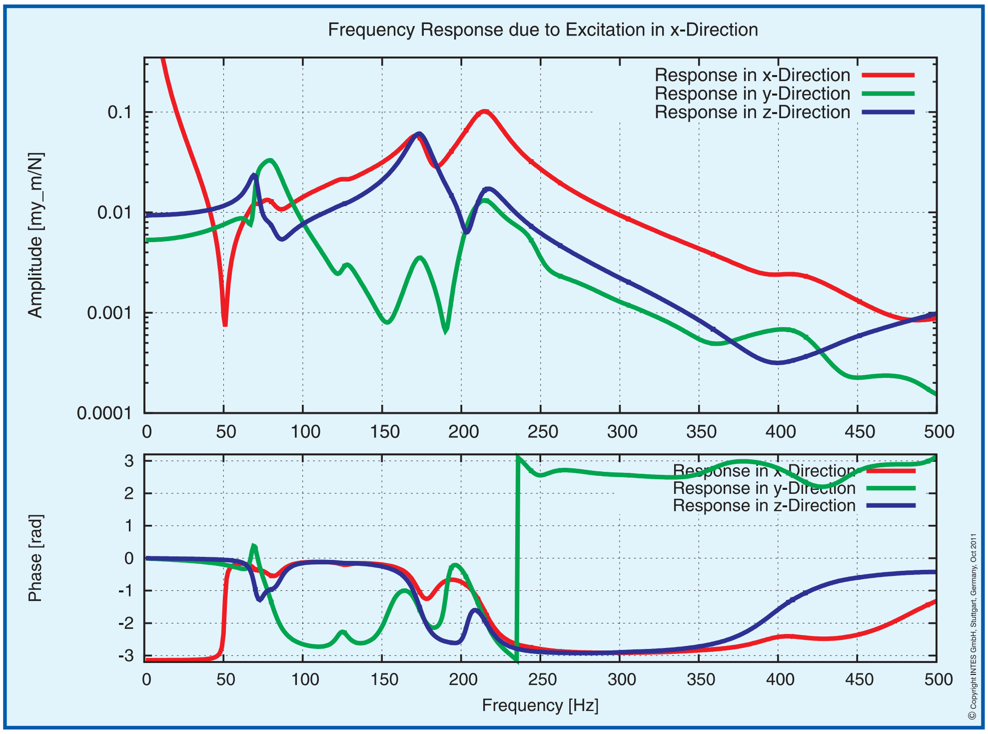

Fig. 7: Frequency response for excitation in x-direction without active controller

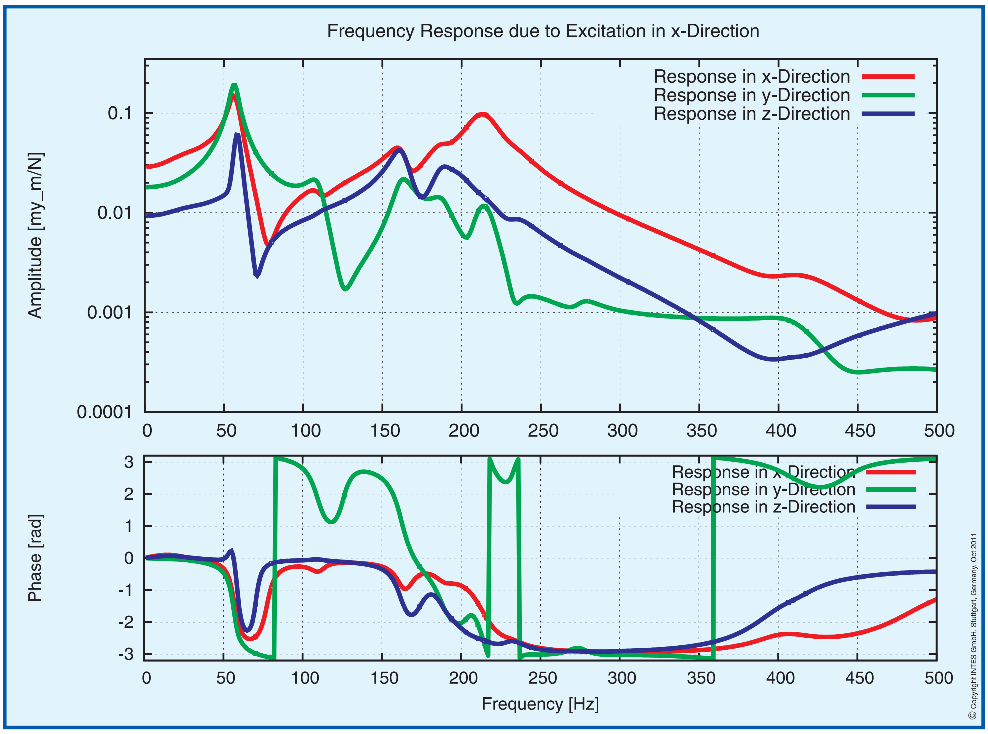

Fig. 8: Frequency response for excitation in x-direction with active controller

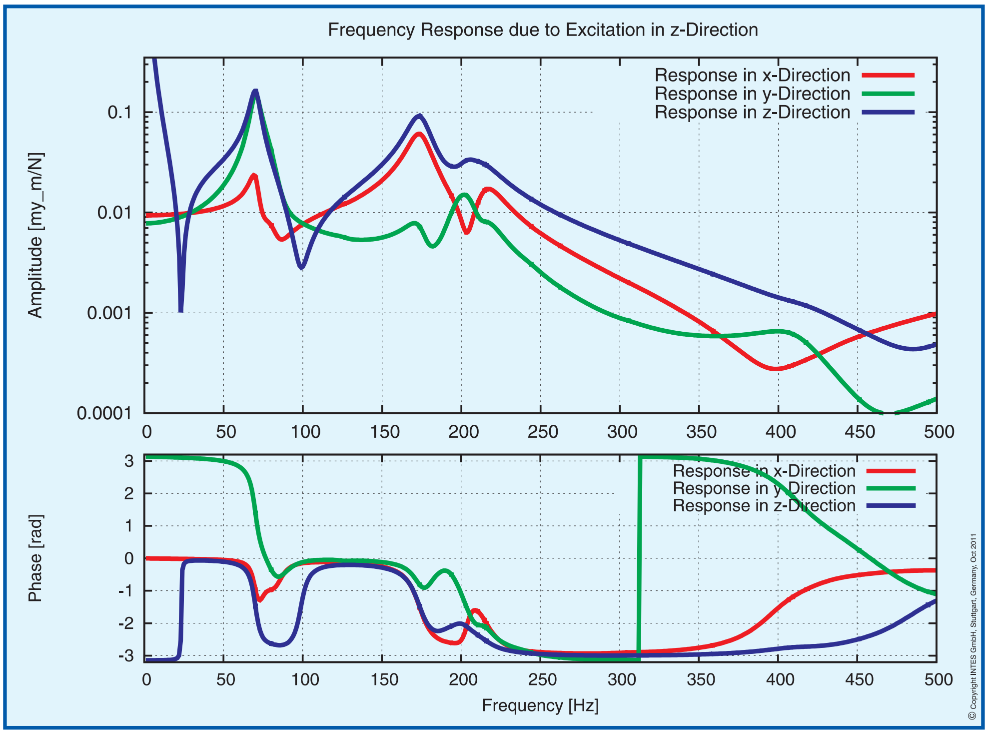

Fig. 9: Frequency response for excitation in z-direction without active controller

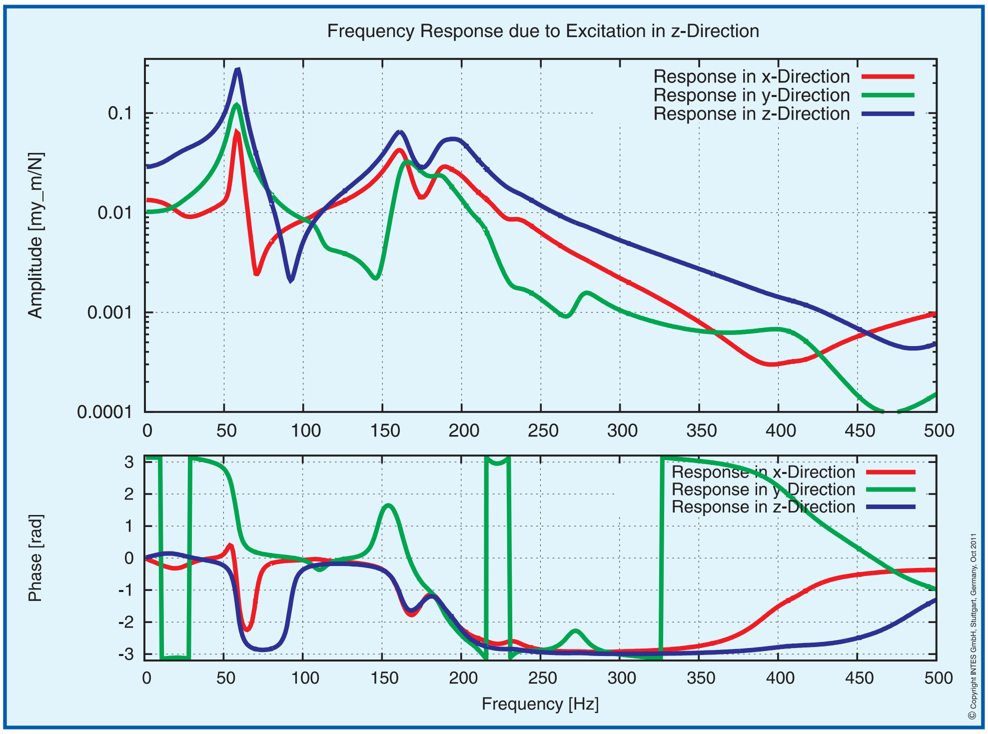

Fig. 10: Frequency response for excitation in z-direction with active controller

Modal damping is applied to all vibration modes by a value of 0.04.

Modal response analysis works with a limited number of modes. The higher 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 quasistatic 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.

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.

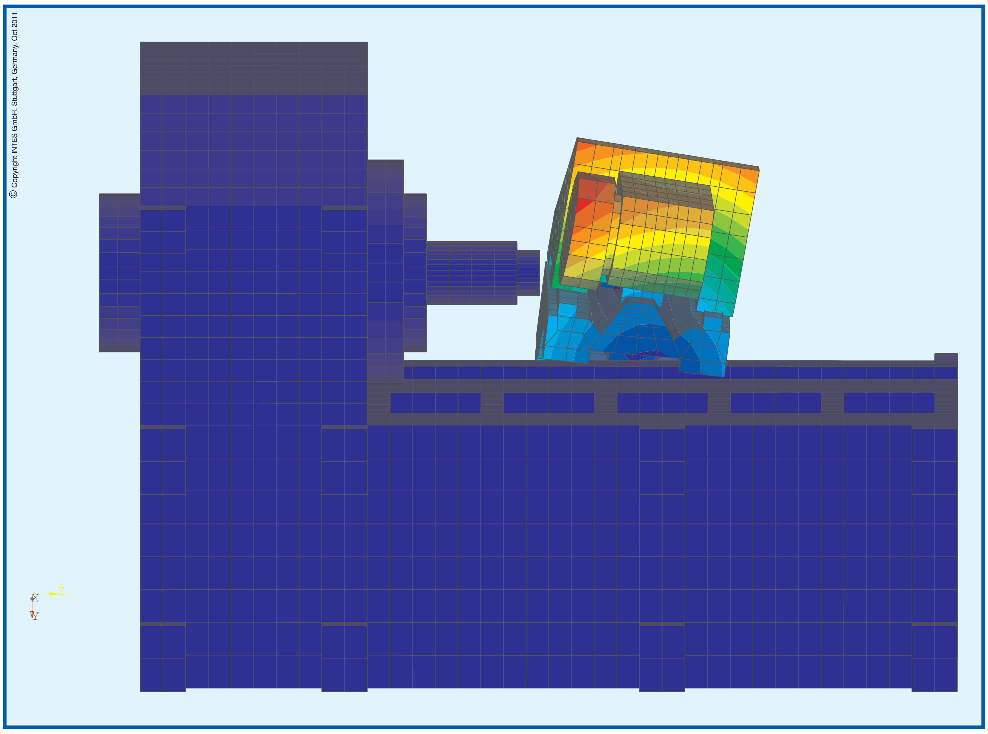

Fig. 11: First real eigenmode at 70 Hz

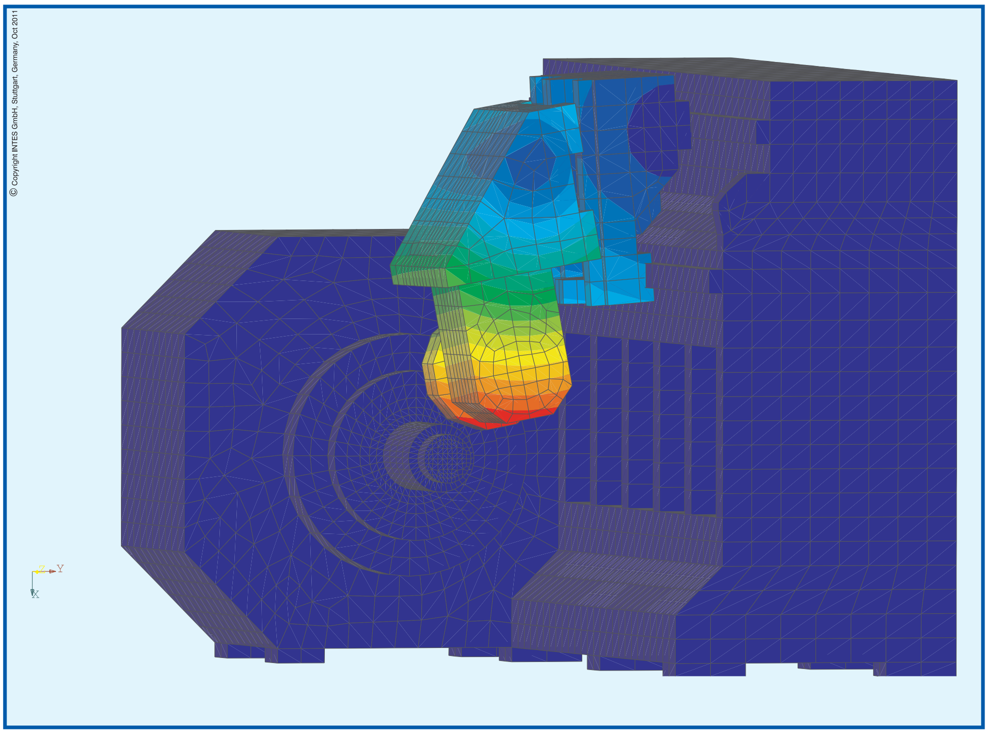

Fig. 12: Second real eigenmode at 80 Hz

Comparing the response without and with active control, we see a shift of the first two 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. Fig. 13 and Fig. 14 show the complex mode shapes at 57 and 59 Hz which correspond to real modes 2 and 1 in Fig. 12 and Fig. 11.

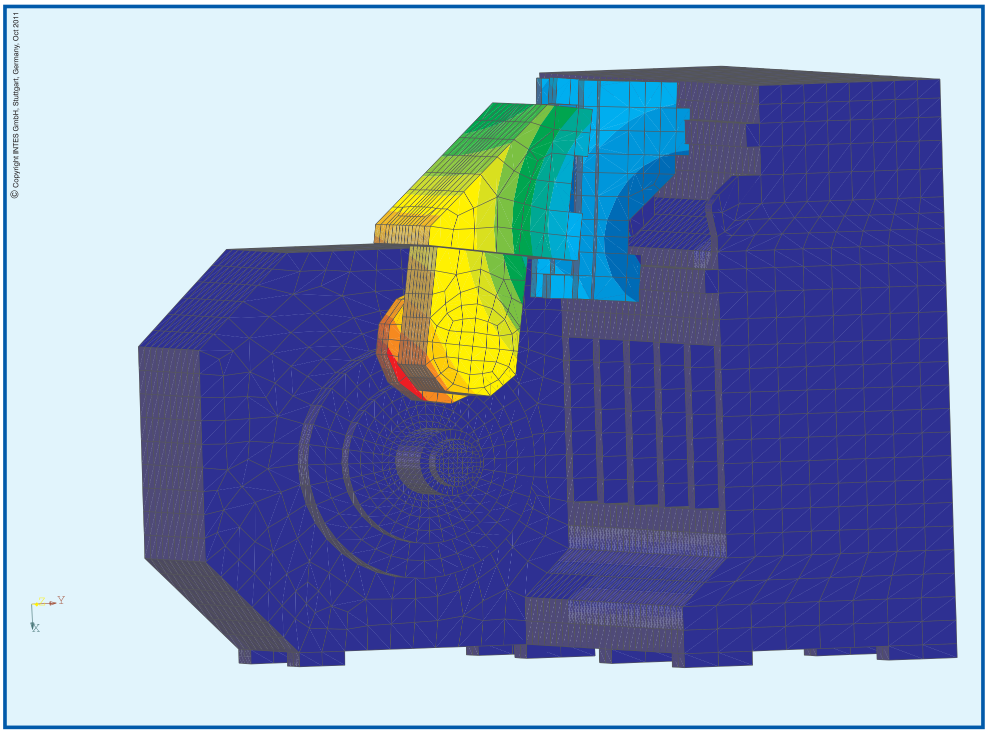

Fig. 13: Complex eigenmode at 57 Hz

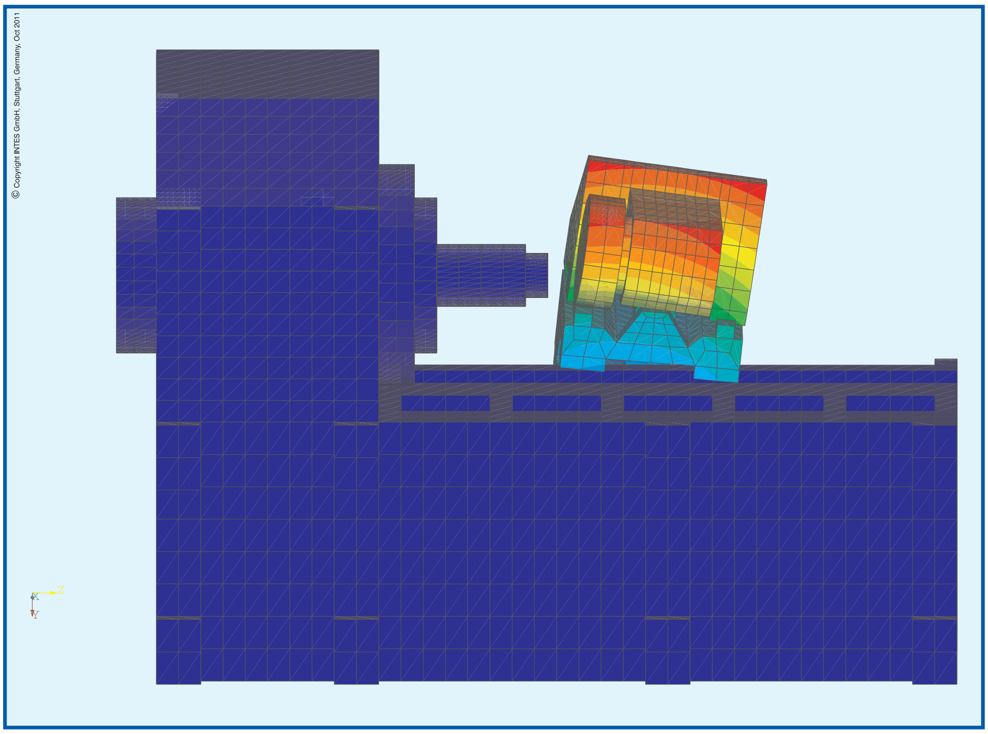

Fig. 14: Complex eigenmode at 59 Hz

Because the spindle is rotating, the influence of the rotor effect on the response should be investigated. Fig. 15 and Fig. 16 show the response curves for excitations in x- and z-direction, respectively.

The influence of the rotor effect is not significant for this model, as one can see from the campbell diagram in Fig. 17.

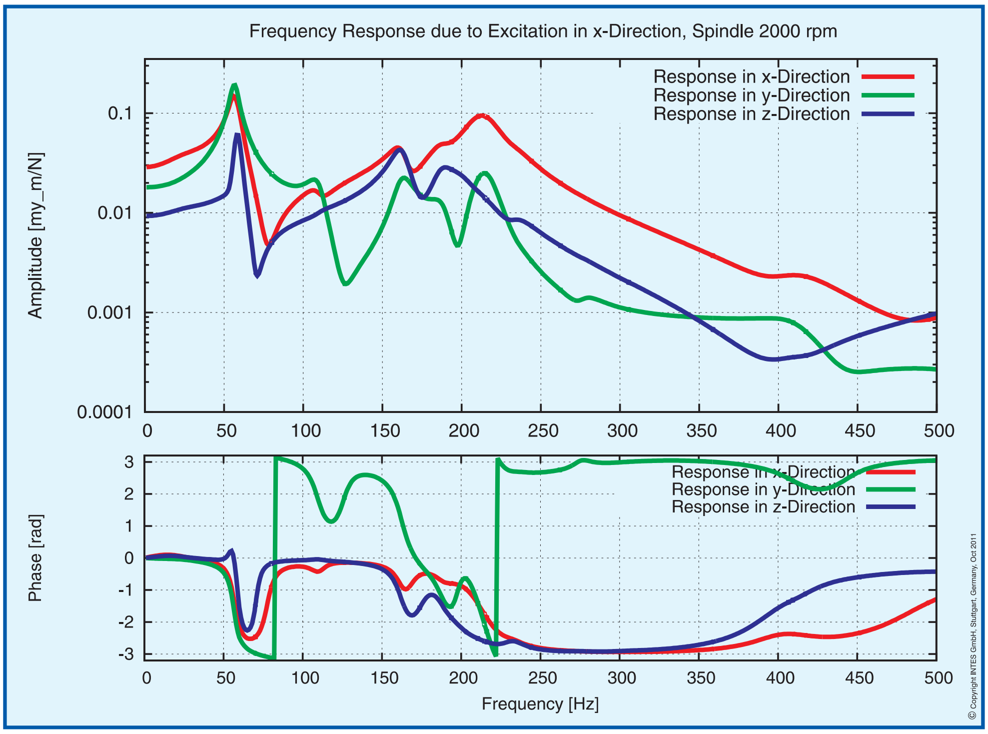

Fig. 15: Frequency response for excitation in x-direction with active controller and a rotational speed of 2000 rpm for the spindle

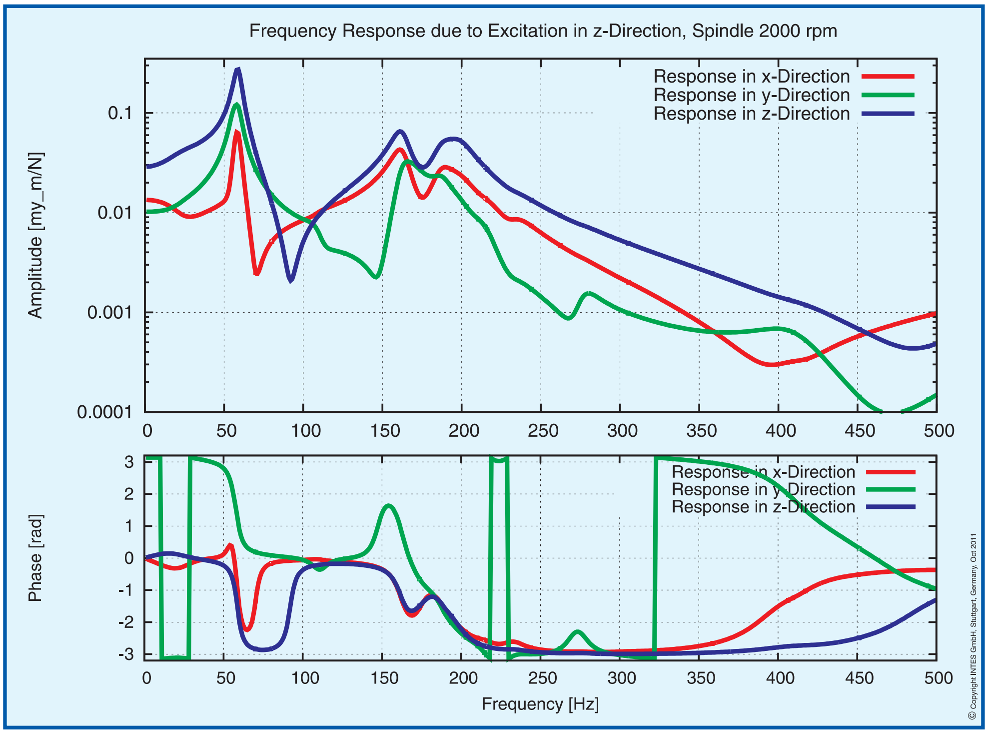

Fig. 16: Frequency response for excitation in z-direction with active controller and a rotational speed of 2000 rpm for the spindle

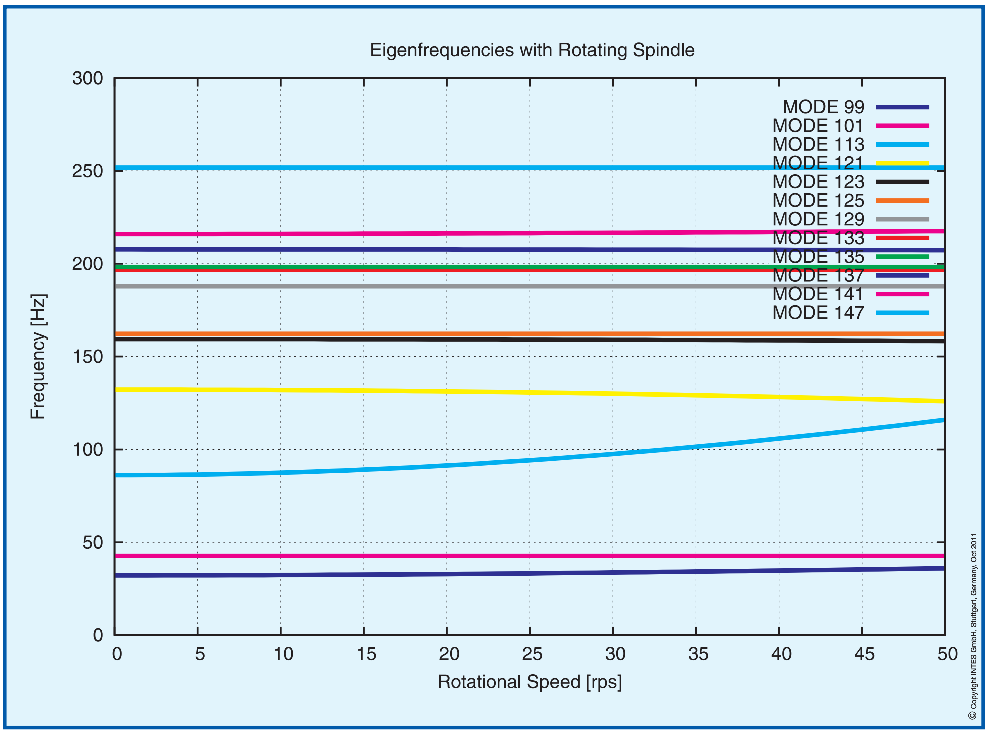

Fig. 17: Campbell diagram for the machine tool model with rotating spindle

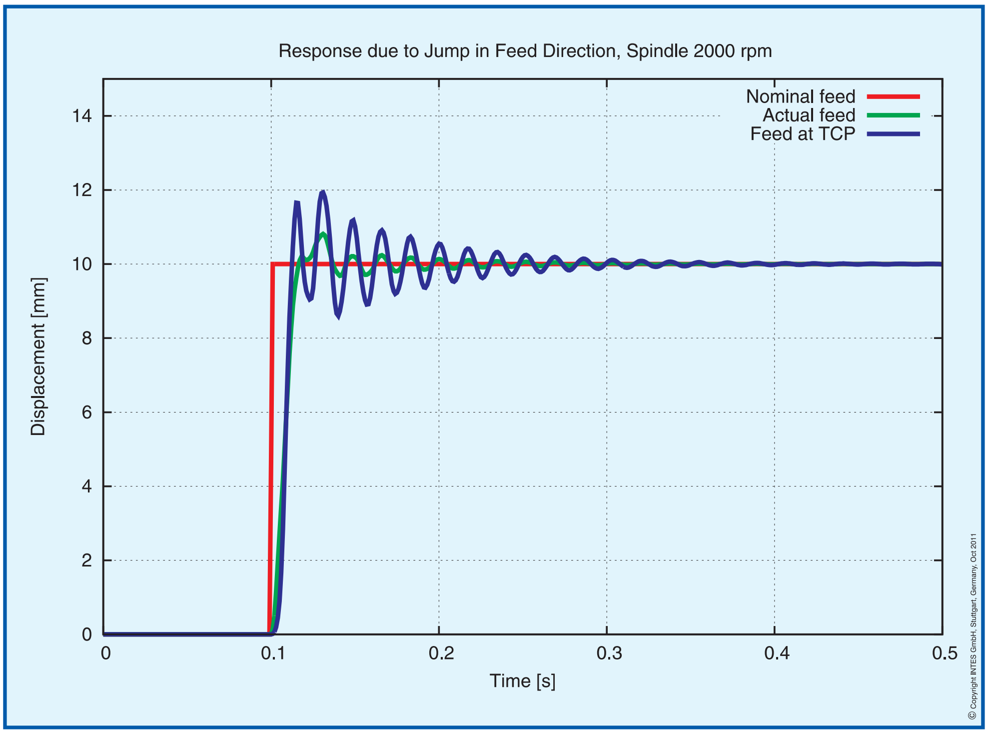

Fig. 18: Time-history response for a jump in x-direction (translational feed)

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 as shown in Fig. 18 and Fig. 19 both under a spindle speed of 2000 rpm.
- A smooth jump using a sine wave as shown in Fig. 20 and Fig. 21 also under a spindle speed of 2000 rpm.

The actual values are taken from the sensor. The TCP values are taken from the tool center point. The difference stems from the additional elasticity between sensor and TCP.

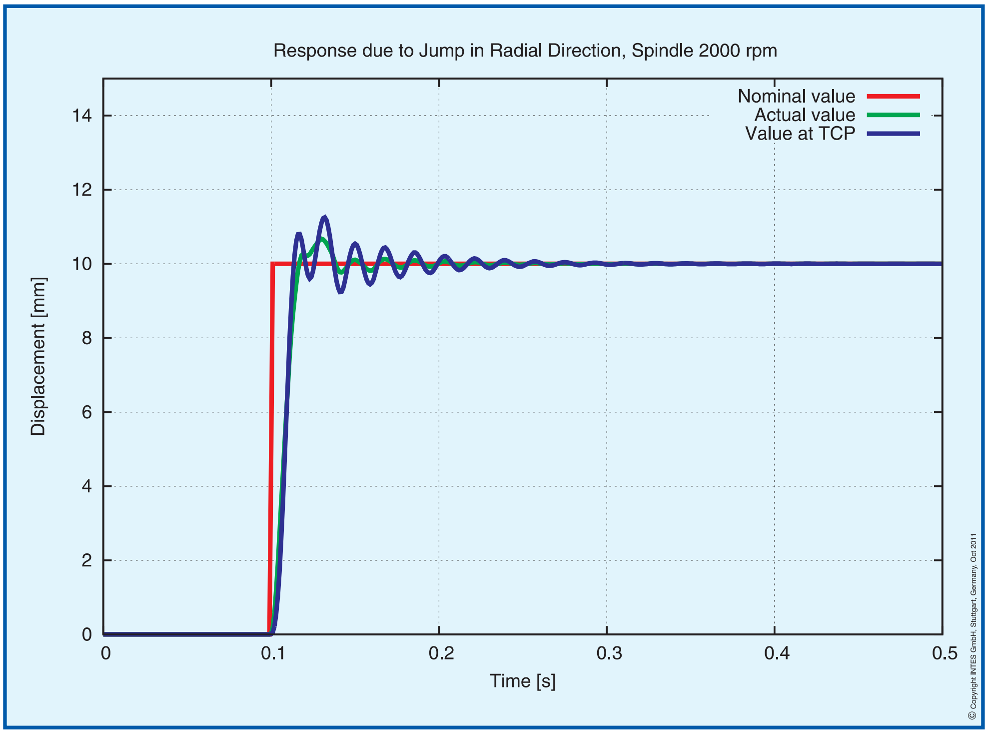

Fig. 19: Time-history response for a jump in z-direction (radial feed)

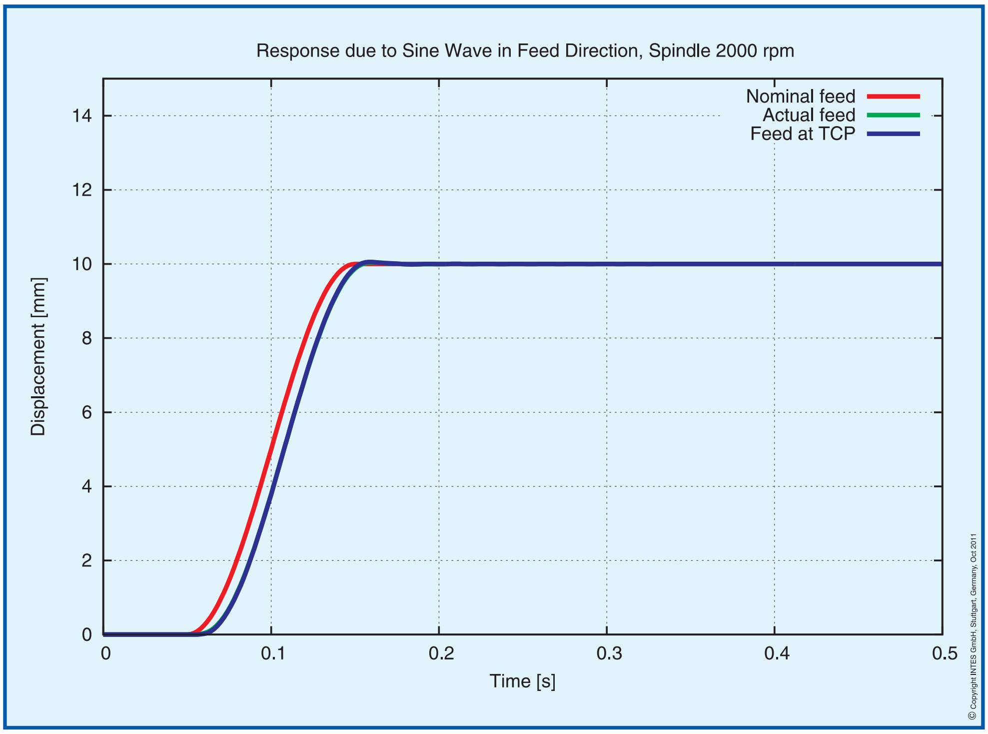

Fig. 20: Time-history response for a jump in x-direction (translational feed)

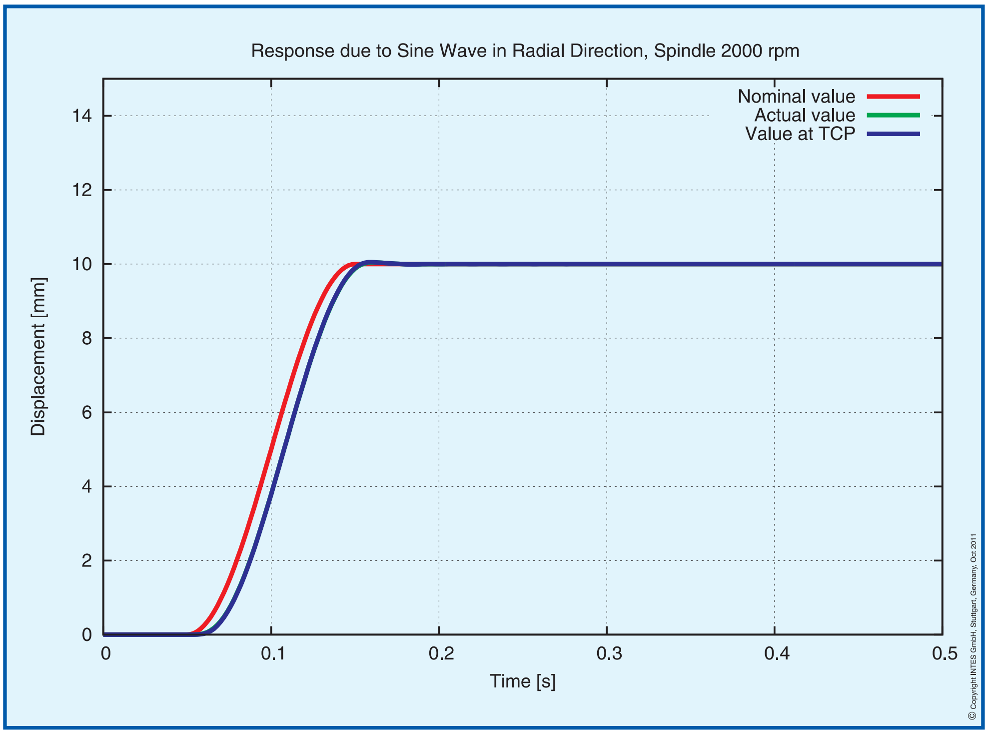

Fig. 21: Time-history response for a jump in z-direction (radial feed)

In addition, a certain geometrical contour can be given which has to be followed by the TCP. As contour shape a semicircle is used here as shown in Fig. 22. The nominal shape is starting at point 1 with a pure translational feed in z-direction which abruptly changes the motion in point 2 to a radial feed in x-direction. Then, it follows a semicircle with a constant angular velocity. The duration of the motion is 1 second. The actual shape is following the nominal shape with some time delay as we can see from point 3, where the nominal motion stops. Hence, the actual position in point 2 is not yet on the circle and the controller needs a short time to damp the vibrations. Finally, the radius of actual shape differs from the set point by a constant value of about 0.2 μm (see point 4).

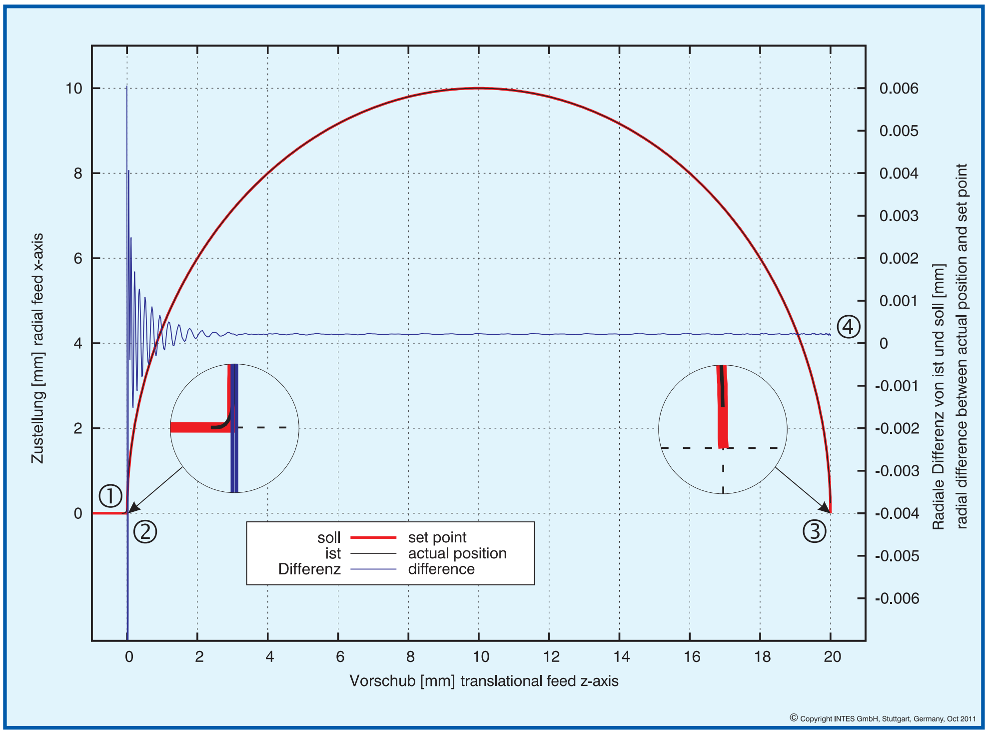

Fig. 22: Contouring the workpiece with a semicircle

6 Considering Stability

For the design of a machine tool, the prediction of instabilities during operation is of utmost importance for 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 selfexciting vibrations is coming from the rotating spindle.

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 speed, the cutting depth, and the feed per revolution. The cutting force model requires a number of coefficients which reflect the combination of a specific turning tool with the material of the workpiece. These coefficients have to be calibrated based on a few experiments with the same combination.

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, different cutting speeds, and different rotational spindle speeds, one gets a stability chart as shown in Fig. 23. The cutting process is assumed to be a so-called longitudinal turning, where the tool moves parallel to the rotational axis. While the chart represents varying cutting forces, cutting speeds, and rotational spindle speeds, the feed per revolution in longitudinal turning direction is kept constant (at 0.3 mm/rotation). Because the stability depends on the combination of turning tool and workpiece material, this information is an essential part of the stability chart (i.e. the tool is a SANDVIK CoroKey CNMM 16 06 08-MR 2025 and the workpiece material is of carbon steel C45N).

One important effect of spindle rotation is the dependency of the stability on the rotating direction of the spindle. Here, counterclockwise rotation allows for higher cutting depths than clockwise 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 between 40 and 70 Hz which in fact are in the audible frequency range of human ears (see Fig. 23).

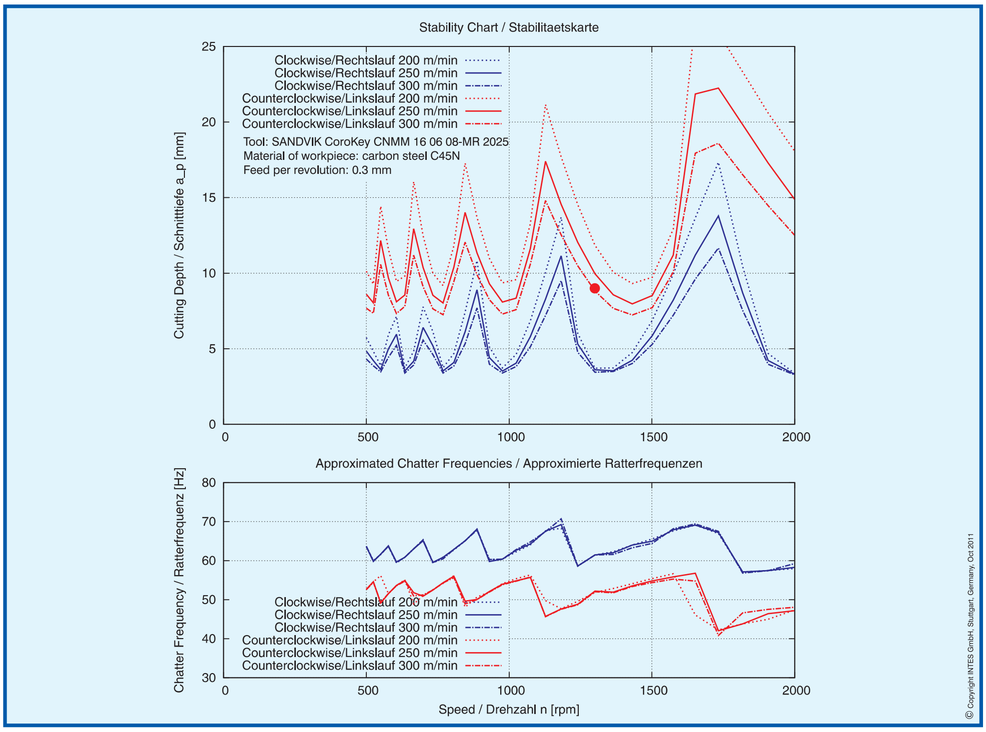

Fig. 23: Stability chart and chatter frequencies for the turning machine model

In order to show an example for stable and instable behaviour The following operating point is used which is between clockwise and counterclockwise stability curve (see dot in Fig. 23):

- Cutting depth 9 mm

- Rotational speed 1,300 rpm
- Cutting speed 250 m/min

Fig. 24 shows the forces at TCP for the stable counterclockwise case and Fig. 25 shows the forces at TCP for the instable clockwise solution.

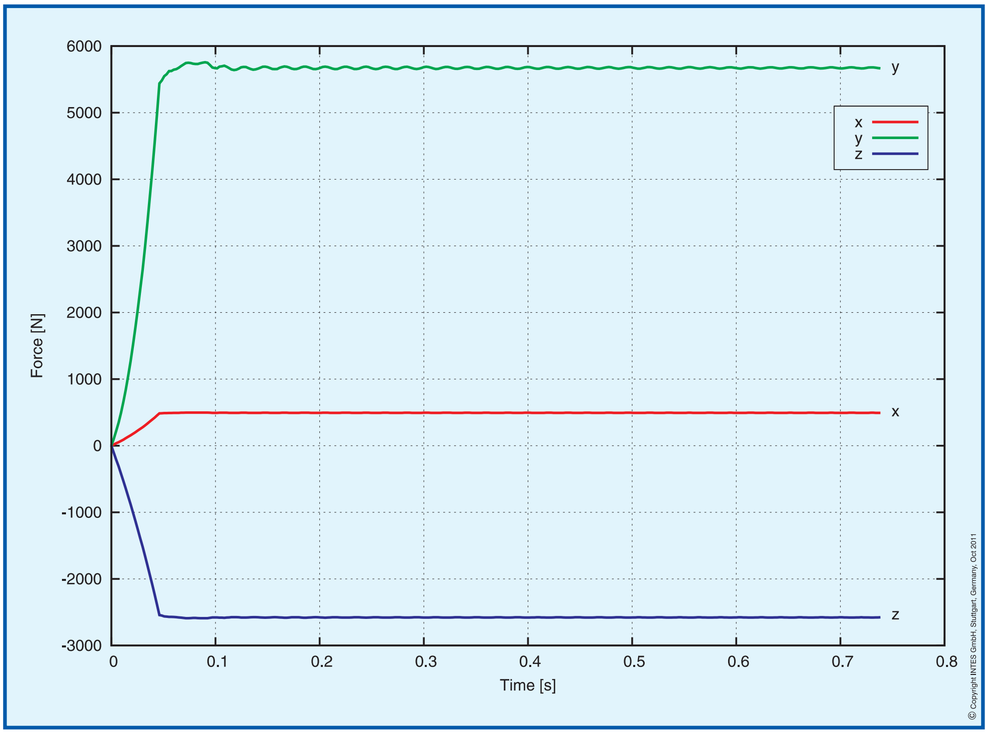

Fig. 24: Stable behaviour counterclockwise

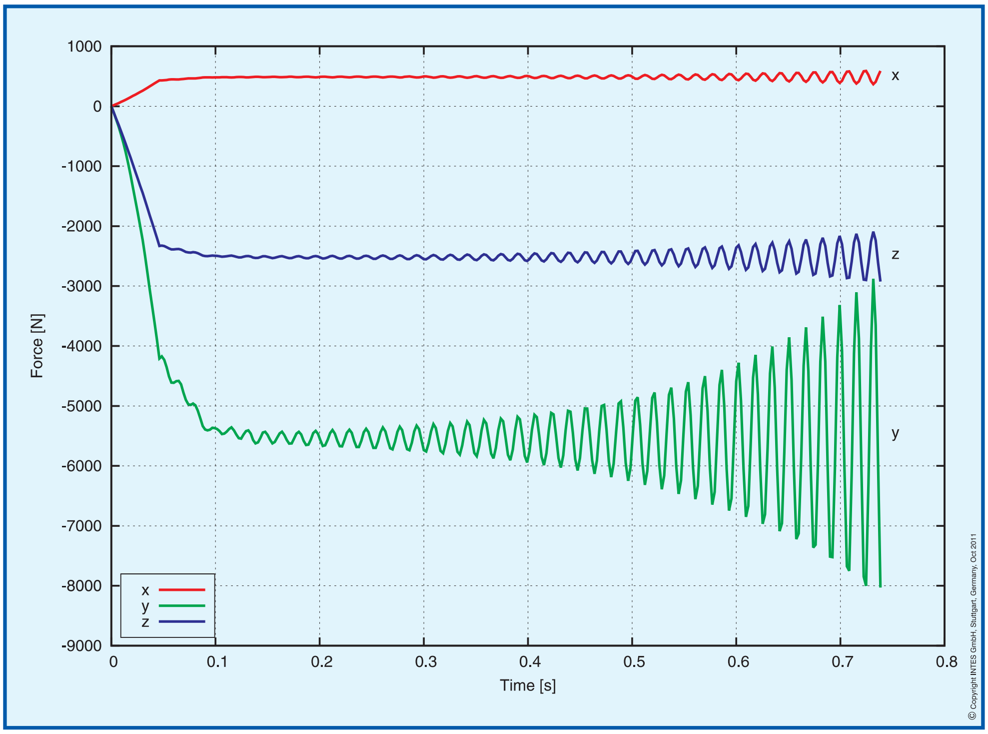

Fig. 25: Instable behaviour clockwise

7 Conclusions

The paper shows an example of a turning machine, where the FE model includes control and a cutting force model to study stability and chatter frequencies. The fully coupled and integrated system analysis is performed by FE analysis with focus on system behaviour in frequency and time domain and special emphasis on creating a stability chart and on computing chatter frequencies.

Possible extensions of the presented material include:

- More detailed modelling (including tailstock, motor, etc.).
- Topology optimization of structural parts taking control and dynamic conditions (like eigenfrequencies, frequency response) into account.
- Shape optimization for getting optimized positions of supports to ground.
- Sizing optimization of controller parameters to move eigenfrequencies and to improve frequency response.

Due to a linear controller for position and velocity control, valuable frequency response results can be computed. Using a nonlinear controller for the cutting force model is used to generate stability charts. Above all, both approaches can be used in the modal space drastically reducing the computing time in case of larger models.

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