Optimizing the Dynamic System Behaviour of Machine Tools

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Abstract

The dynamic behaviour of a complete machine tool including workpieces is crucial for the development of such machines. The accuracy of the motion of tables and tools is absolutely essential for the accurate resulting surface shape of the workpieces. Fast positioning and fast machining are important, the first with very small to no vibrations and the latter with high cutting depths and no chatter.

The complete system comprises structural parts, drives of the various axes, their control, and rotating spindles. 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 real and complex eigenvalue analysis, frequency response analysis, and time-history response analysis are available for the computation of the dynamic behaviour 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.

For lightweight designs of machine tools, topology optimization provides the right features to save weight while keeping the controlled dynamic behaviour within the required limits. In addition, another important activity in developing machine tools is tuning, where the control parameters are modified to get the best possible machine behaviour. This is also an important application of optimization integrated in FE analysis, which allows a completely integrated tuning by simulation.

A simple milling machine model is used to show the modelling, analysis, and optimization of machine models from eigenfrequencies up to stability charts. Dynamic analyses and optimization of dynamic behaviour is a central point to be shown. All computations are carried out using one industrial FE software (PERMAS).

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 behaviour. 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.

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 behaviour.
- **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 published in [1] to [6], and documented in [7] to [9].

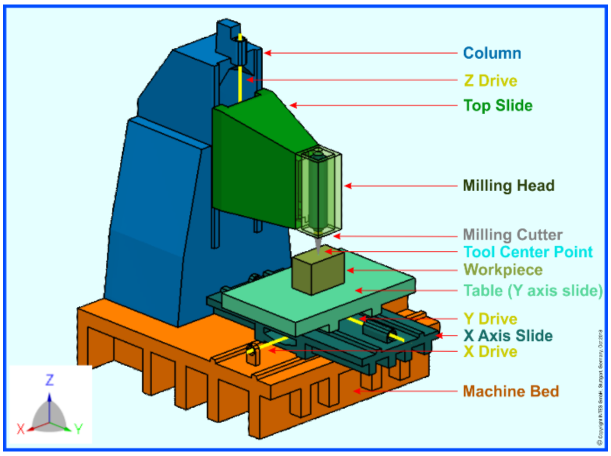

Figure 1: Milling machine model

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 behaviour in frequency and time domain. It follows a topology optimization of the table and the X axis slide and an optimization of the controller parameters. Finally, the combination of both methods is shown.

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 Fig. 2 for X, Y, Z drives), where the spring and damper forces are connected to the solid structures by surface coupling enabling incompatible meshes.
- **Ball screw drives**: They are modelled by beam elements (see Figs. 2). 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.

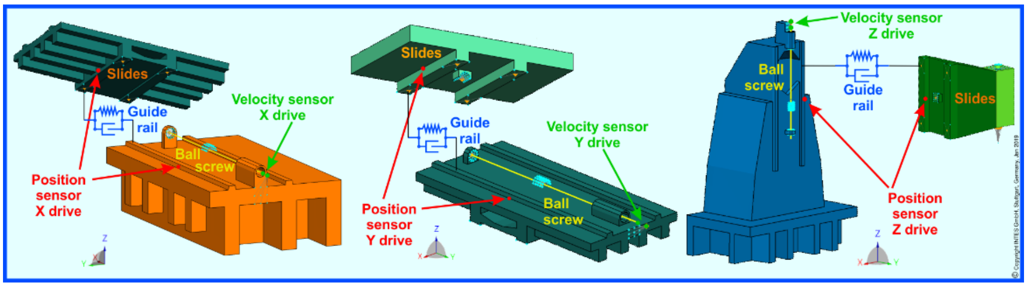

Figure 2: Configuration of X, Y, and Z drive with sensors and guide rail connections

A number of parts, like motors, could also become a part of the model, but were omitted for simplification reasons. 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 a FE model, an element is available which represents a cascade controller as shown in the block diagram of Fig. 3. 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.

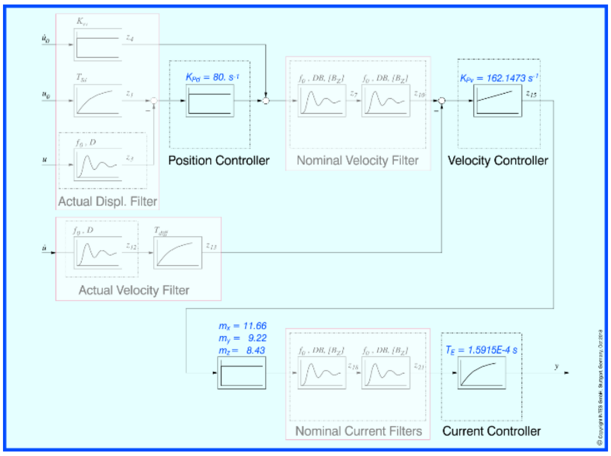

Figure 3: Block diagram of the cascade controller CONTRL8

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. 3).

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.

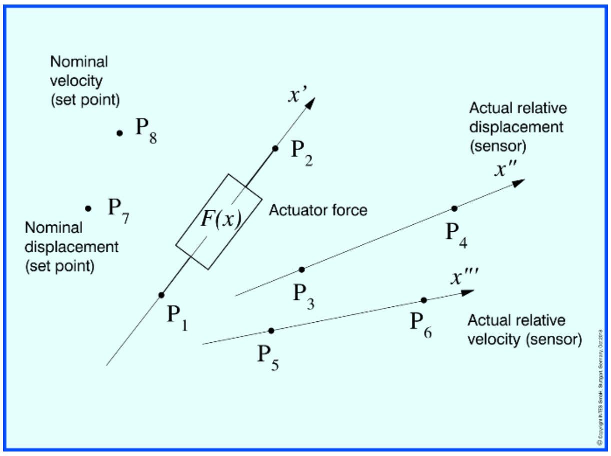

Figure 4: FE representation of the controller element CONTRL8

The finite element representation of the control element is shown in Fig. 4, 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. 2).
- Node 5 and 6: Sensor to provide relative velocities (based on rotation speed sensors), which are located at the ball screw drives (see Fig. 2).
- 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 two for both the motion in X and Y direction. The parameters of the controllers are the same (see Fig. 3) except the parameter m which reflects the moved masses.

The positions of the sensor points can be seen from Fig. 2. 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. 5 and Fig. 6 show the frequency response graphs for an excitation in X and Y direction with active controller.

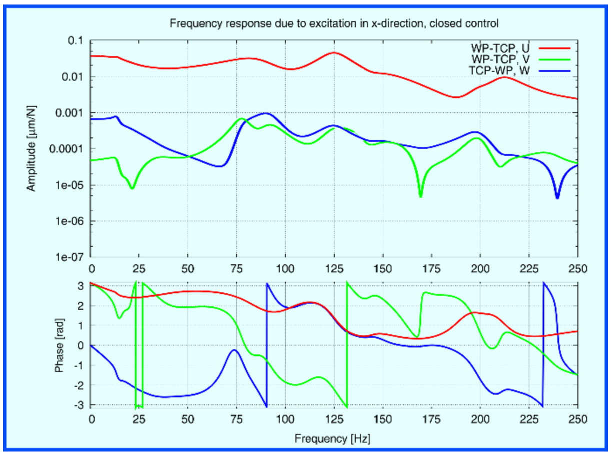

Figure 5: Frequency response for excitation in X direction with active controller

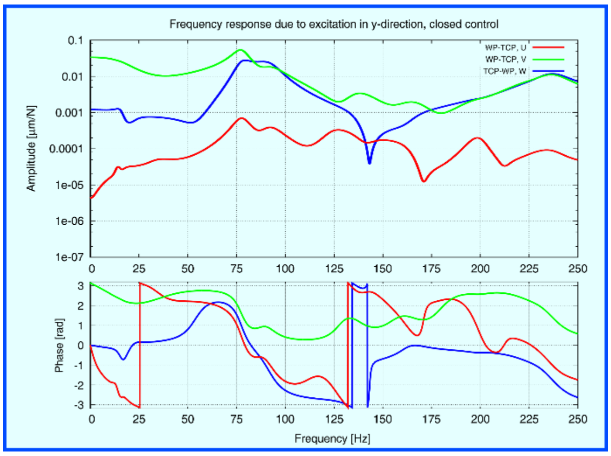

Figure 6: Frequency response for excitation in Y direction with active controller

A number of 268 modes has been calculated up to 2500 Hz. The first ten eigenfrequencies are shown in Table 1.

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.

Mode No.	Eigenfrequency [Hz] of starting model
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

Table 1: Eigenfrequencies of the starting model

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

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, one typical imposed position change is an abrupt jump in X or Y direction as shown in Figs. 7 and 8 under a spindle speed of 3000 rpm.

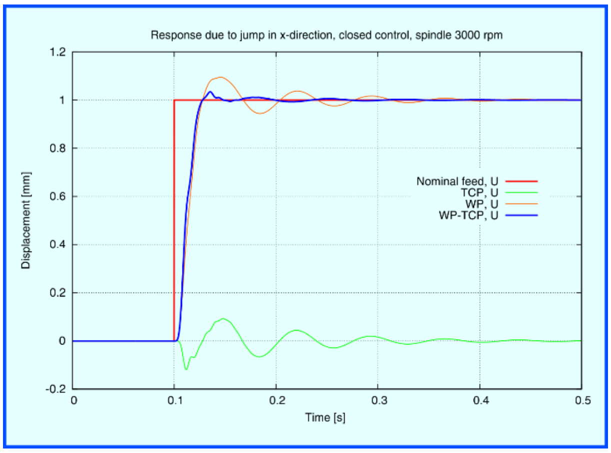

Figure 7: Sudden jump of X axis slide in X direction

The workpiece values are taken from the sensor. The TCP values are taken from the tool centre 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.

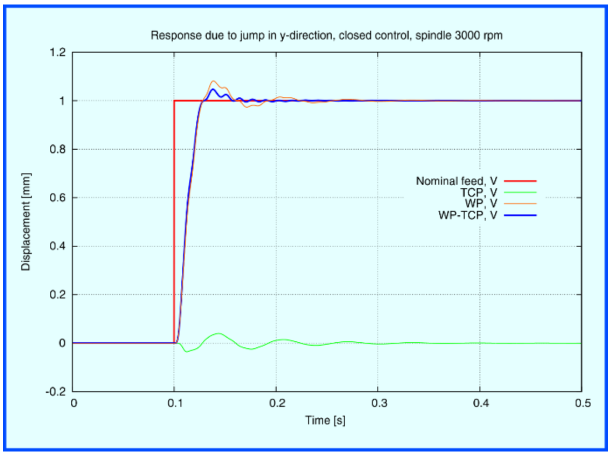

Figure 8: Sudden jump of table in Y direction

6. Optimization Strategy

In this paper, two directions of optimization are presented:

- The right adjustment of controllers is not so easy. So, it might be helpful to use optimization techniques to adjust the parameters of all three controllers (see Fig. 3).
- Facing lightweight design requirements, reducing weight using topology optimization can be quite helpful. Here, the topology optimization is applied to the table and the X axis slide.

In addition, the load cases have to be selected for the optimization:

- The sudden jump as described in Figs. 7 and 8. This leads to a parameter optimization for a modal time-history response analysis.
- The frequency response analysis as used for the Fig. 5 and 6. Here, the topology optimization is applied in a modal frequency response analysis.

The main question is, which loading case is the ruling case. Following scenarios were used:

- A combined parameter and topology optimization in frequency domain. It turned out that the resulting structure and controller parameters do not result in a desirable response to a sudden jump.
- The optimized controller parameters from a sudden jump analysis showed an improvement also in the frequency response analysis.

Finally, the selected procedure is as follows:

- To optimize the controller parameters with the sudden jump analysis.
- Then, to optimize the topology in a frequency response analysis.

- Subsequently, the modified topology is used in a sudden jump analysis to optimize the controller parameters again.
- For the resulting parameter set, another frequency response analysis is performed to compare the results.

7. First Optimization of Controller Parameters

The sudden jump analysis is performed by a modal time-history response analysis between 0. and 0.2 s (see Figs. 7 and 8). The objective function is the difference between the sudden jump and the resulting response where the jump occurs at 0.1 s.

Controller	Parameter	Initial value	Optimized value
X	K_{Pd}	80 s^{-1}	76.70 s^{-1}
	K_{Pv}	162.1473 s^{-1}	166.8293 s^{-1}
	m_x	11.66	12.47
	T_E	1.5914E-4 s	6.2778E-4 s
Y	K_{Pd}	80 s^{-1}	73.97 s^{-1}
	K_{Pv}	162.1473 s^{-1}	166.5899 s^{-1}
	m_y	9.22	9.80
	T_E	1.5914E-4 s	7.5000E-4 s
Z	K_{Pd}	80 s^{-1}	78.47 s^{-1}
	K_{Pv}	162.1473 s^{-1}	165.3772 s^{-1}
	m_z	8.43	8.82
	T_E	1.5914E-4 s	7.5000E-4 s

Table 2: Initial and optimized controller parameters

From the sudden jump response, we see that two properties of the response are important. An ideal solution is to reduce the delay of the response after the sudden jump and to reduce the overshooting amplitudes. But there is a conflict between both objectives. If the response becomes steeper to reduce the delay, higher overshooting amplitudes can occur. In case of longer delays, the amplitudes will be reduced more likely. So, the decision was to keep the slope and reduce the amplitudes by optimization of controller parameters.

The controller parameters before and after the optimization are shown in Table 2 and Figs. 9 and 10 compare the response of the initial controller settings with the optimized ones in X and Y direction.

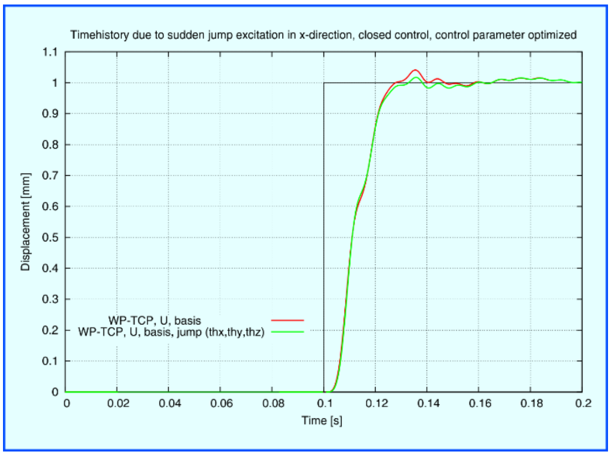

Figure 9: Initial response to sudden jump in X direction (red) and after optimization of controller parameters (green)

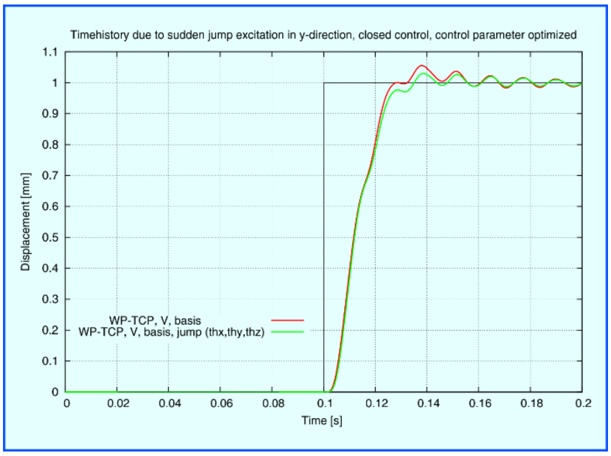

Figure 10: Initial response to sudden jump in Y direction (red) and after optimization of controller parameters (green)

8. Topology Optimization in the Frequency Domain

The design space for topology optimization is defined as part of the table and part of the X axis slide (see Fig. 11).

Before starting the frequency response optimization, the controller parameters were updated following the result of the sudden jump optimization (see Table 2). Then, the result of the topology optimization is shown in Fig. 12.

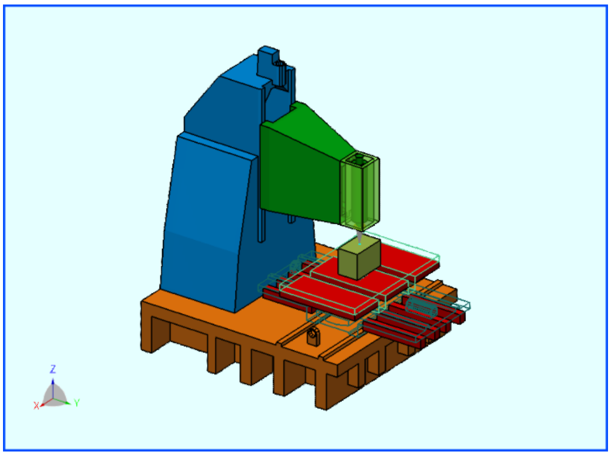

Figure 11: The design space for topology optimization (red)

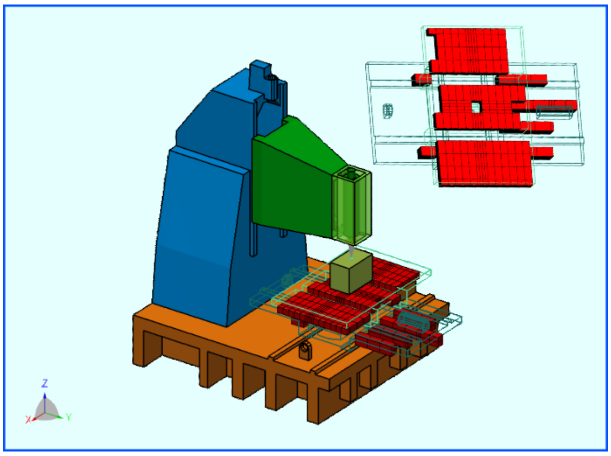

Figure 12: Remaining elements after topology optimization in frequency domain using the updated controller parameters

The weight history during topology optimization is shown in Fig. 13. From that, weight savings of 115 kg were achieved for X axis slide and table together.

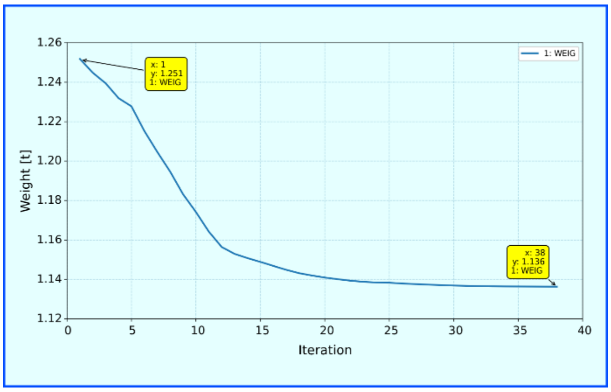

Figure 13: Weight history of X axis slide and table in topology optimization

9. Second Optimization of Controller Parameters

The controller parameters before and after the first and second optimization are shown in Table 3 and Figs. 14 and 15 compare the response of the initial controller settings with the optimized ones after first and second sudden jump optimization.

Controller	Parameter	Initial value	First optimized value	Second optimized value
X	K_{Pd}	80 s^{-1}	76.70 s^{-1}	77.85 s^{-1}
	K_{Pv}	162.1473 s^{-1}	166.8293 s^{-1}	166.3304 s^{-1}
	m_x	11.66	12.47	12.40
	T_E	1.5914E-4 s	6.2778E-4 s	7.0070E-4 s
Y	K_{Pd}	80 s^{-1}	73.97 s^{-1}	60.79 s^{-1}
	K_{Pv}	162.1473 s^{-1}	166.5899 s^{-1}	158.3734 s^{-1}
	m_y	9.22	9.80	8.56
	T_E	1.5914E-4 s	7.5000E-4 s	7.5000E-4 s
Z	K_{Pd}	80 s^{-1}	78.47 s^{-1}	78.67 s^{-1}
	K_{Pv}	162.1473 s^{-1}	165.3772 s^{-1}	165.774 s^{-1}
	m_z	8.43	8.82	8.86
	T_E	1.5914E-4 s	7.5000E-4 s	7.5000E-4 s

Table 3: Initial and optimized controller parameters

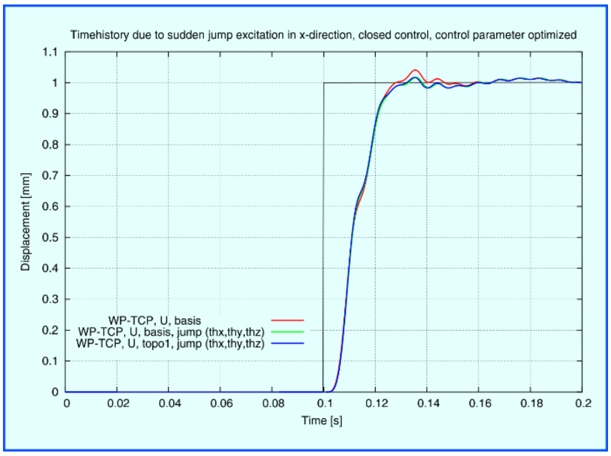

Figure 14: Initial response to sudden jump in X direction (red) and after first (green) and second optimization of controller parameters (blue)

In addition, the comparison of the frequency response results is shown in Figs. 16 and 17, where the amplitude of the distance between workpiece and TCP is shown in the initial model and after first and second sudden jump optimization.

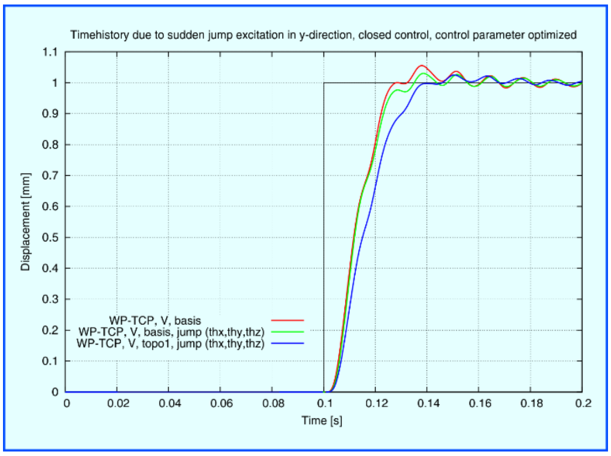

Figure 15: Initial response to sudden jump in Y direction (red) and after first (green) and second optimization of controller parameters (blue)

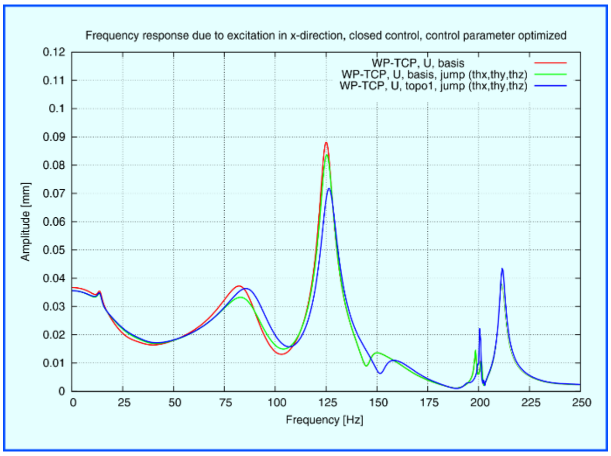

Figure 16: Comparison of frequency response in X direction for initial model (red) and after first (green) and second (blue) sudden jump optimization

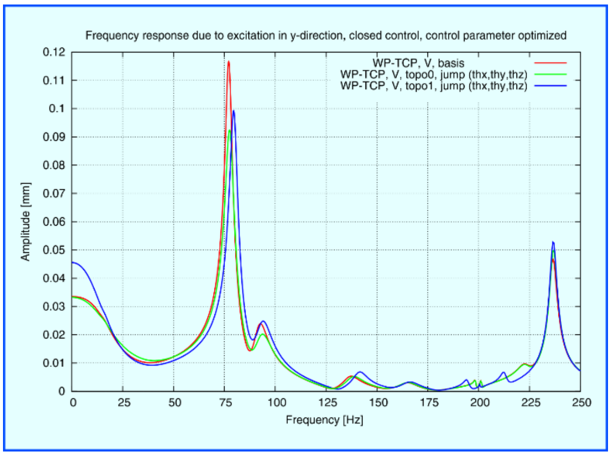

Figure 17: Comparison of frequency response in Y direction for initial model (red) and after first (green) and second (blue) sudden jump optimization

10. Conclusion

Optimization of machine tools is a complex task. The main challenges are:

- The consideration of several loading cases like frequency response and sudden jump analysis reveal conflicting targets, which limit the improvements by optimization. Other load cases like sine jump or circularity test could also be important.
- Lightweight design based on topology optimization is also in conflict with controller settings, because a lighter machine has a modified dynamic behaviour. For acceptable run times, topology optimization with a high number of elements in the design space requires the availability of sensitivities also in time domain.
- The size and weight of the workpiece should not be forgotten. A high weight of the workpiece will of course have a high influence on the dynamics of X axis slide and table.

Beside these challenges, optimization is a promising tool to improve dynamics of a machine tool including the optimization of controller settings. In addition to topology optimization, shape optimization could also be a candidate for improving machine dynamics.

11. References

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