Mechatronic Optimization, Analysis and Simulation of Machines

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INTRODUCTION

Generally, various companies are involved in the development of important parts of a machine: The machinery construction OEM – these are often medium-sized companies – have overall system responsibility for the mechanical subsystem and the automation components are provided by companies who offer motors, control systems and drives. The A&D MC business division of Siemens AG, with its portfolio of state-of-the-art automation equipment, is not only in a position to offer intelligent control systems in conjunction with high dynamic performance motors and drives, but it can also support machinery construction OEMs when they are mechanically designing a machine. This is achieved by analyzing and optimizing the complete machine in a focused fashion. This means that manufacturers of machine tools and production machines can use the mechatronic services in the form of machine optimization, analysis and simulation. This article describes the basics of machine simulation. The contents of the article focus on the area of suitable modeling of the complete mechatronic system and less on the detailed description of the individual subsystems.

Every machine tool and production machine is designed so that specific motion is impressed on a mechanical structure. This motion should be as fast as possible with the adequate precision. The second section will show that specific characteristics of the mechanical system to be moved limit the velocity which can be achieved with the specified accuracy.

The third section will show how a mechanical subsystem is modeled using discrete elements. And the fourth section will explain which systems require modeling with finite elements. The procedure which is applied to generate a model is then discussed using examples from industry.

MECHANICAL LIMITS - ACHIEVABLE MACHINE DYNAMIC RESPONSE

The ability to impress the required movement on a mechanical structure is often restricted by the lowest natural frequencies of the mechanical structure. This can be clearly illustrated using a simple system: A load, with mass \( m_L \) is shown in Fig. 1 which is to be moved by a motor with mass \( m_{Mot} \). The coupling between the motor and load is established using a spring with stiffness \( c \) and a damping element with damping characteristic \( d \). The goal of a closed-loop control with this configuration would be to define the force \( F \) as a function over time of the measured quantities, so that the position of the load follows, as precisely as possible, a specified reference characteristic.

The Laplace transformation of \( G_{XLXM}(s) \) with the motor position, \( X_{Mot}(s) \) as input quantity and the position of the load \( X_{Load}(s) \) as output quantity is given by:

\[
G_{XLXM}(s) = \frac{X_L(s)}{X_{Mot}(s)} = \frac{2DTs}{s^2 + 2DTs + T^2}\]

The amplitude characteristic from the associated frequency response characteristic \( G_{XLXM}(f) = |X_L(j2\pi f)/X_{Mot}(j2\pi f)| \) can be seen in Fig. 2. In this case, the parameters are assumed as follows:

\( m_L = m_{Mot} = 1 \text{kg}, \ c = 3948 \text{ N/m}, \ d = 3.77 \text{ Ns/m}. \)
In the vicinity of the resonant frequency \( f_{res} = 1/(2\pi) \sqrt{(cm)} = 10 \) Hz, at which frequency the load oscillates with respect to the motor, a significant increase in the amplitude characteristic can be identified. When the motor moves at frequencies exceeding 10 Hz, then with increasing frequency these act less and less on the load. This can be clearly seen with the negative gradient of the amplitude characteristic with 40 dB/decade. At 33.5 Hz, a value of \( G_{xLxM}(33.5 \) Hz) = –20 dB can be read-off from the characteristic in Fig. 2. When the motor moves with a frequency of 33.5 Hz, in a steady-state condition, it moves the load with an amplitude which is reduced by a factor 10. From 100 % of the complete mechanical energy which the system has with this motion, less than 1 % can be found in the load itself!

Furthermore, the load with increasing frequency can only move with an associated significant deformation of the spring which the system has with this motion, less than 1 % can be found in the load itself! In many cases, it is hardly possible to move the load when the motor is moving with a significantly higher frequency than the resonant frequency with which the load oscillates with respect to the motor (in the example, this is 10 Hz). For movements at higher frequencies, the ratio between the energy which is fed to the load (useful energy) and the total energy is extremely low. The deformation of the spring also significantly exceeds that of the load movement. This would mean that the mechanical components, which are used to transmit the force from the motor to the load, would be destroyed.

The lowest resonant frequency, with which the mechanical system to be moved oscillates with respect to the motor, can be considered, in many cases, to signify the achievable dynamic response of a machine. This is independent of the closed-loop technique which is being used. This is the reason, that for each control strategy, the limits of the selectable control parameters and the achievable control dynamic response are frequently determined by the lowest quenching frequency of the machine.

Generally, a closed-loop cascade control is used for machine tools and production machines [GHW2000]. The gain factor of the closed-loop position controller – the so-called \( K_v \) factor – is considered, in the machine tool sector, to be the quality parameter of a machine. In [HT1997], an inter-relationship is specified between a possible \( K_v \) factor and the lowest natural frequency.

The highest \( K_v \) factor (in \( \text{m/(mm-min)} \)) which can be selected for the closed-loop position control of an axis is approximately one tenth of the lowest natural frequency of this axis (in Hz). This relationship applies if the natural frequency lies below 100 Hz and the position controller sampling time is 1 ms. For higher sampling times, the dynamic response is already limited for lower natural frequencies due to the computation time of the position controller. A theoretical derivation of these inter-relationships is provided in [Tr1975a].
CLOSED-LOOP CONTROL AND MODELING OF A GRINDING MACHINE AXIS

Many grinding machines have a heavy workpiece, mounted on a table which is moved in the horizontal axis using a spindle and spindle nut. In this particular example, the spindle is directly connected to the motor through a coupling and the spindle nut, mounted on the table, moves together with the table without rotating (Fig. 4).

Often, the dynamic response of a machine is limited by such an axis or the associated natural frequencies are manifested in the form of chatter marks on the workpiece. This means that a significant potential for improving the complete machine can be evaluated by specifically analyzing the weakest axis. For this particular mechanical structure, discrete elements (springs, masses, damping elements) can be used to generate a model. The mechanical subsystem, corresponding to the model in Fig. 4, can be defined if the translatory motion of the table, spindle nut and workpiece is converted into rotational movement.

In the example here, it is assumed that a mass of 9 tons is to be moved for a spindle pitch of 10 mm with the following moments of inertia: Tachometer, motor, coupling, spindle and table have the following values respectively 0.025·10⁻⁴kgm², 430·10⁻⁴kgm², 50·10⁻⁴kgm², 350·10⁻⁴kgm² and 228·10⁻⁴kgm². The stiffness values are assumed to be as follows:

- \( c_{\text{Tach}} = 2800 \text{ Nm/rad} \)
- \( c_{\text{Mot}} = 258000 \text{ Nm/rad} \)
- \( c_{\text{Kupp}} = 36000 \text{ Nm/rad} \)
- \( c_{\text{Sp},\text{tor}} = 90000 \text{ Nm/rad} \)
- \( c_{\text{ax},\text{tor}} = 600 \text{ Nm/rad} \)

Two obvious quenching frequencies can be identified in the amplitude characteristic of the speed control loop (Fig. 5), which, according to Section 2, are characteristic for the behavior of the axis to be moved. The lowest quenching frequency is at 25 Hz which is obtained from the resonant frequency with which the mass of 9 tons to be moved oscillates with respect to the motor, corresponding to the axial stiffness. The next higher quenching frequency is at 142 Hz, and is essentially defined by the coupling stiffness and the spindle moment of inertia. However, the table, oscillating at 25 Hz with respect to the motor, defines the system behavior which will become clear in the following. The transient response is analyzed using this example, i.e. the response of the axis when external disturbances occur, for example, machining forces.

Two speed controller settings will be further investigated. A comparison is made in Fig. 6 of the speed control loop with the two parameter assignments: For setting (B), a high speed controller gain \( K_v = 40 \text{ Nms/rad} \), \( T_n = 5 \text{ ms} \) with an appropriate integral action time \( T_n \) was selected; and for setting (A), a lower gain \( K_v = 12 \text{ Nms/rad} \), \( T_n = 15 \text{ ms} \) was set, to match the lowest quenching frequency.

The behavior of the controlled system at the workpiece is of interest. The table deflection for a step-like disturbing force of 500 N is shown for both controller settings in Fig. 7. For the parameterization (B), the system responds with a violent 25 Hz oscillation of the table. On the other hand, for the adapted setting, this oscillation decays significantly faster. The better transient response, in spite of the lower controller gain factors, can be explained using the root locus characteristic of the speed control loop.

The root locus characteristic is shown in Fig. 8, which applies for an integral action time of 5 ms (setting B). The PI controller gain is so high that the two poles, which originate in the open circle, have almost been shifted to the zeros. This means...
that the speed-controlled system contains a conjugated, complex pole pair with a low damping of approx. 4% and a frequency of approx. 25 Hz. This poorly dampened pole pair can hardly be noticed in the frequency response characteristic of the speed control loop (Fig. 6), as the zeros of the open circle remain in the closed circle. From the perspective of the motor encoder, this weakly dampened pole pair is almost compensated by the zeros close to it. As a result of the higher-level position control loop, the pole pair damping can only be increased to 5%, whereby the associated natural frequency decreases to 24 Hz.

The root locus characteristic, which is obtained in conjunction with the adapted controller parameterization (A), is shown in Fig. 9. The root locus characteristic has a different curve type as a result of the higher integral action time. However, what is important is that the gain is significantly lower than in case (B). This means that the critical pole pair has a damping of 14% for a natural frequency of 28 Hz. This can be increased to 20% by the position controller, whereby the associated natural frequency is reduced to 25 Hz.

The behavior of the system at the workpiece is decisive for grinding. However, when the disturbing forces are applied as input quantity and the workpiece position as output quantity, there is no zero close to the pole pair being discussed – and only the associated damping is decisive. For the adapted setting (A), this is significantly greater than for setting (B). This can be correspondingly identified in the characteristics with respect to time in Fig. 8.

Summarizing, it can be said that the lowest quenching frequency of 25 Hz, which is especially caused by the high moved mass and the axial stiffness of the spindle, restricts the achievable dynamic response and this is characterized in the transient response. Such a critical frequency cannot be resolved with mechanical measures without using another drive concept. In fact, the critical frequency can only be increased, which from experience, is only possible with some restrictions. When excited, a higher natural frequency would result in oscillations with a lower amplitude thus resulting in an improved workpiece surface. For example, in order to increase the frequency to achieve approximately 35 Hz, if the mass was to stay the same, it would be necessary to double the stiffness!

If a linear drive would be selected to move the axis instead of a drive with a ballscrew, then the present cause of the 25 Hz oscillation would be eliminated. However, the use of linear motors demands a harmonized concept which must be backed up using FEM analysis. Otherwise, there would be the danger that oscillation types which do not have a limiting effect for drives with ballscrew (see-saw oscillations of the table, oscillation of the workpiece with respect to the table, coupled oscillations) would result in problems which would have a negative impact on the workpiece quality.

**ANALYSIS OF A MILLING MACHINE**

In this section, the investigation of a machine will be discussed whose mechanical structure is completely different to the machine which was discussed in the previous section: To start, in
order to evaluate the overall machine dynamic response, it is necessary to analyze two axes. One of the axes is assigned to two motors (gantry group), so that a total of three mechanical drive transmissions have to be considered. Furthermore, before the analysis, it is not possible to make a definitive statement regarding the oscillation types associated with the lowest natural frequencies of the machine. Furthermore, oscillations are also possible which are extremely difficult to mathematically define. The drive mechanical transmissions to be investigated cannot be analyzed independently of one another, as they mutually influence each other – i.e. they are mechanically coupled.

In order to take into consideration such secondary conditions, a model must be generated based on finite elements. This means that the model has a high number of degrees of freedom, from which the decisive system characteristics can be numerically determined.

A typical milling machine and a finite element model of a machine having a similar mechanical design are illustrated in Fig. 9. Several machine elements are designated in the model which will be referred to in the following text. The model corresponds to that of a real machine but is changed so that it does not contain any specific manufacturer know-how.

The milling head is moved along the z axis using a slide which can move on a spindle sleeve. The associated z drive is not explicitly taken into account in the model as the ratio of the stiffness involved to the masses moved in the z axis, in comparison to the other axes, is quite high. The spindle sleeve moves in the y axis using a ball screw which is driven by a synchronous motor (motor 3) through a gearbox. However, the spindle nut is held stationary. Guide rails support the motion of the spindle sleeve with respect to the transverse beam. The complete transverse beam is moved in the x axis as gantry group using two additional spindles which are driven by motors 1 and 2 through directly-coupled ball screws.

To start, the natural frequencies and eigenmovements of the finite element model are calculated. As already discussed in detail, the low natural frequencies are of extreme significance. These are assigned to the axes whose direction of motion is involved with the associated oscillation type. The axis, in whose movement of direction the oscillation type is effective with the lowest natural machine frequency, is the axis which limits the dynamic response of the machine.

The natural frequency should also be determined for fixed motors, as these are characteristic for the motion of the machine for a specified (impressed) motion of the motor. The motors can only move the machine at frequencies which are lower than the lowest calculated natural frequency, corresponding to Section 2. Correspondingly, for higher frequencies, the natural frequencies of the moving motors are of significance. In this particular example, the lowest natural frequency for fixed motors is 23 Hz. The oscillation types, which are associated with the natural frequency, only influence the y axis. This is the reason that the y axis is analyzed in the following.

The amplitude characteristic, shown in Fig. 10, is calculated from the finite element model with the motor torque as input quantity and the speed of the third motor as output quantity. The natural frequency of 23 Hz can be recognized as the quenching frequency in Fig. 10. Furthermore, in addition to the next quenching frequency at 25 Hz, there are resonant frequencies at 24 Hz, 28 Hz, 43 Hz and 74 Hz.

If the oscillation types are now considered, which are associated with the disturbing natural frequencies, their mechanical causes can be identified, and in turn, mechanical design improvements derived. The oscillation type, associated with 23 Hz and 25 Hz, which occurs when the motor rotors are locked, is characterized by a rotational movement of the complete spindle sleeve in the y-z plane (“pitching oscillation”). The resonant frequencies at 24 Hz and 28 Hz occur for the moving motors and indicate an oscillation type similar to that at 23 Hz. However, the motors also move. At 43 Hz, the machine has eigenmovement, where the transverse beam oscillates in the y axis and the side panels buckle significantly. The oscillation types described are shown to the right in Fig. 10. The eigenmovement, which is associated with the quenching frequency at 23 Hz and 25 Hz and the resonance points at 24 Hz and 28 Hz is especially disturbing. This limits the achievable dynamic response when moving in the y axis. When excited by machining
forces, chatter marks occur on the workpiece.

Using the appropriate control techniques, an attempt can be made to avoid these chatter marks by damping the low frequency oscillations. However, this is only possible depending on how energy can be drawn from the critical oscillation as a result of motor motion. For the oscillation type involved, this is only possible up to a certain extent. As far as the motor encoder and direct measuring system are concerned, which are mounted on the transverse beam, the disturbing frequencies can be dampened by adapting the controller parameters. However, oscillations can always occur at the tool without there being a clear feedback to the measuring systems. This is made visible, as an example, in the behavior of the axis when a velocity step is entered (Fig. 11): The velocity is almost monotonous at the two measuring systems. However, an oscillation with approximately 25 Hz is visible at the tool and where a frequency of 74 Hz can also be recognized.

The amplitude characteristic from the compliance-frequency response characteristic (deflection \( \delta y \) at the tool for force \( F \) at the tool) of the closed-loop-controlled \( y \) axis is illustrated in Fig. 11. In addition to the low static stiffness (approx. 2 N/mm), there are several obvious resonant points which correspond to the eigenmodes discussed. If the machining force occurs at a frequency which corresponds to a significant visible resonant frequency of the machine in the compliance-frequency characteristic, then this results in poor workpiece quality. Various pressure angle frequencies are typical when milling, which are dependent on the material being machined. When machining aluminum, these lie in the vicinity of 300 Hz, for steel, approximately 80 Hz, and for titanium, approximately 20 Hz. The machine being investigated is neither in the position to machine titanium nor steel as the resonant points lie in the critical frequency range. However, it should be possible to machine aluminum, as no resonance occurs in the vicinity of 300 Hz.

Finally, it is necessary to increase the lowest quenching frequency by making mechanical modifications. As the associated natural oscillations indicate, when the rotary motion of the spindle sleeve occurs at the lowest natural frequencies, the spindle sleeve is supported by the stiffness of the linear guide with a lever arm. This lever arm is obtained by the location of the guide shoe at the spindle sleeve. It would be relatively simple to increase the stiffness by moving the guide shoe to the outside. However, this would reduce the amount of working space. Another possibility, but involving significantly higher costs, would be to move the spindle sleeve using a twin drive (refer to e.g. [Tr1975b]). Two motors with
two spindles would then be required, whereby the force would have to be introduced into the spindle sleeve at a lower and an upper position. With this arrangement, a rotary movement could be counteracted through the drive mechanical transmission shaft without restricting the working space. The mechanical design improvement should, in the next step, be evaluated using the model, and if required, additional measures drawn up to eliminate the problems which limit the machine performance.

SUMMARY
This article has shown that a machine tool or production machine is a mechatronic system which can be optimized, taking into consideration all of the subsystems. Control-related measures alone are not sufficient, as was clearly shown in Section 2: Specific mechanical properties limit the speed and accuracy of the motion control of a machine. Two application examples were used to show how a machine can be modeled. The limits of control-related measures were indicated and mechanical design steps were explained in order to address the causes of disturbing machine properties. Overall, it has been made clear that the procedure for machine tool and production machines, based on mechatronic methods, is of considerable significance: Development times are shortened and weak points can be identified and eliminated early on so that all of the prerequisites are fulfilled to bring innovative machines faster to market.

LITERATURE
[GHW2000]

[HT1997]

[Tr1975a]

[Tr1975b]