

Paper:

Advanced Control Strategies for Active Vibration Suppression in Laser Cutting Machines

Berend Denkena, Martin Eckl, and Thomas Lepper

Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

An der Universität 2, 30823 Garbsen, Germany

E-mail: lepper@ifw.uni-hannover.de

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Due to rising energy requirements, the use of low-weight materials is becoming more important, especially in aerospace and automotive engineering. Because of their high strength-to-weight ratio, carbon fiber reinforced plastics (CFRP) are increasingly replacing metals. These materials are usually machined by milling operations. Their main problems are high tool wear, thermal damage, and surface integrity. This paper presents a machine concept and control strategy to substitute milling with laser cutting. Because a high, constant-trajectory velocity is required during laser cutting operations, a highly dynamic machine tool is needed. Conventional machine tools requiring large workspaces are inertial and therefore unsuitable for this task. Thus, a portal machine concept was investigated with an additional laser scanner and lightweight moving components. To increase path accuracy, two control strategies were implemented and analyzed in a multi-body simulation. One approach is to use a frequency-separating filter, while the second is based on estimation of tool center point positioning error using a Kalman filter. An acceleration sensor located near the tool center point (TCP) or the drive current signal can be used as input for the Kalman filter. Both input signals are investigated and compared in this paper. Results presented in this paper show that with these control strategies, highly dynamic trajectories can be realized with high precision.

Keywords: hybrid positioning, laser cutting, vibration control, vibration compensation, Kalman filter

1. Introduction

Due to dwindling energy resources and therefore rising prices for fossil fuels, there is enhanced environmental awareness in our population. Moreover, financial incentives created by environmental standards, imposed by governments, force industry to keep a check over fuel consumption by cars and aircrafts. The engine is optimized and the impact of air resistance is decreased to reduce fuel consumption. Furthermore, the weight of airplanes and cars is reduced. Because the mass of an airplane or car has to be accelerated it has a profound influence on fuel

consumption.

To reduce the mass of parts, as a first step, low-weight metals like aluminum and titanium are used. As a second step the topology of parts is optimized to make them thin in areas with less stress [1]. Even more weight can be reduced with new lightweight materials. Currently, materials like aluminum and titanium are increasingly replaced by carbon fiber reinforced plastics (CFRP). Parts made of metal are machined with conventional turning or milling machines. Process parameters, tool geometry, and machine tools are optimized for these machining operations. Today new CFRP materials are machined with milling operations, too. One challenge is high tool wear during cutting because of the mixture of very hard carbon fibers with the binding polymer [2, 3]. Current cutting tools are worn out after a short period of use. Further, a wide range of different cutting tools is needed for different finished surface geometries.

An alternative processing operation is laser cutting, which is also used for metals. One benefit of laser cutting is that it does not require different tool geometries. Downtime, during which machining cannot be done, can be reduced significantly if tool changes are reduced or abolished. Although laser cutting of CFRP is more flexible than mechanical cutting operations, it is not commonplace in manufacturing. One reason is the lack of sufficient research. Furthermore, the relevant process know-how has not yet been transferred to industry. Because laser cutting is a thermal process, thermal influence on the material at the border zone is a significant challenge. Thermal energy has a negative influence on the mechanical properties of the material and therefore must be reduced. The Laser Zentrum Hannover (LZH) has been working on this topic. Researchers analyzed thermal influence on CFRP for different process parameters in machining. The results presented in [4–6] show that cutting of CFRP with high-power lasers is possible.

Within the project, *CFKLas*, a new machine concept for laser cutting, has been developed. Because laser cutting operations are conducted with higher-trajectory velocities than in conventional mechanical cutting operations, the requirements on machine structure are very high. Although there are no process forces, high accelerations lead to machine vibrations. Thus, the machine has to be stiff and dynamic to realize a constant and high trajectory ve-



locity. Furthermore, the working space has to be large in the case of manufacturing large frames for the aerospace industry. Realizing highly dynamic trajectories may reduce the cost advantage over mechanical cutting operations where the dynamics can be lower, if not a new and cost-efficient machine concept is developed.

To achieve a high and accurate trajectory velocity, two concepts for error compensation have been invented and are presented in this paper. Both concepts take advantage of the highly dynamic axes provided by the laser scanner. In the first concept, a frequency-separating filter divides the trajectory into high- and low-frequency parts. The low-frequency part is the set point signal for the drive motor of the machine, while the high-frequency part is the set point signal for the scanner. In the second concept, a Kalman filter is used to estimate vibrations at the TCP. As the input signal of the Kalman filter, an acceleration sensor located near the TCP can be used as well as the drive current signal. Both approaches are presented in this paper.

2. Hybrid Positioning in a Portal Machine

2.1. The Concept of a Laser Cutting Machine

The aim of this paper is to present a machine concept for manufacturing large aerospace parts. Because the parts are up to 6 m long, a large working area is needed. Typically, large machine tools are designed as gantry machines. These portal machines have a large and heavy bridge with a high inertial mass. Especially in laser cutting operations, the trajectory velocity is high compared with machining operations in metal cutting. To prevent material damage due to the laser cutting process, the cutting velocity must be very high and constant. Therefore, the machine axes have to be accelerated and decelerated very dynamically to ensure a constant trajectory velocity. To make positioning more dynamic than with the conventional machine axes, in addition to the three Cartesian axes, a laser scanner is added which can position the laser focal point in a small space. The scanner is placed at the end of the kinematic chain of the portal machine.

As shown in **Fig. 1**, the laser scanner unit consists of a laser-focusing unit and a scanner head. The focusing unit is used to set the focal point of the laser beam to the surface. To change the distance of the focal point in the focusing optics, the diverging lens is positioned using a linear drive. The distance must be adjusted by rotation of the mirrors because of the varying length of the laser beam, depending on the position of the focal point. Rotation of the mirrors is used to position the laser beam. The correct position of the focal point is important for material removal.

For dynamic positioning of the focal point, the scanner head redirects the laser beam with the help of two mirrors driven by galvanometer drives. Because of the low weight and inertial moment of the mirrors, the scanner is able to position highly dynamically.

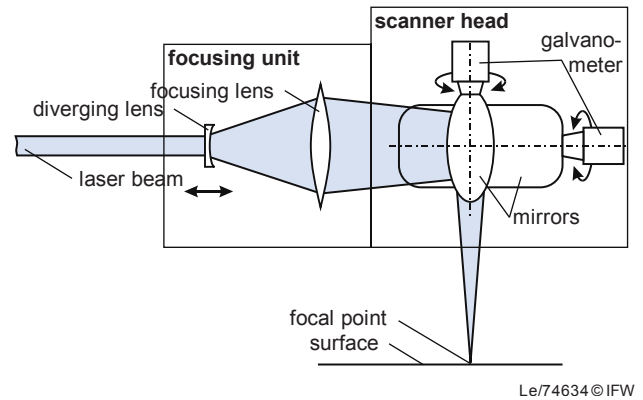


Fig. 1. Basic functionality of a laser scanner unit [7].

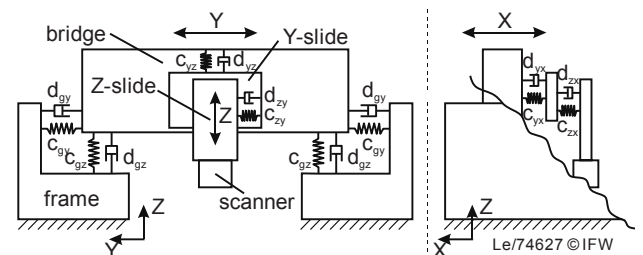


Fig. 2. Multi-body model of the portal machine.

The laser scanner unit is placed at the end of the Cartesian machine axes; thus, the conventional axes position the scanner. The scanner unit has three additional axes, which make positioning redundant. One resulting benefit is the possibility of a hybrid positioning control system. Hence, the positioning unit of a laser scanner can be used to compensate positioning errors of the linear drives. Two different control strategies are described in section 2.3.

2.2. Multi-Body Simulation

In order to design and investigate the two control schemes, a multi-body simulation model of the machine has been developed. The machine is modeled with multiple solid bodies, each of them described by mass, center of gravity, and inertia properties. According to the geometry and kinematics of the machine, the bodies are connected by kinematic joints as well as stiffness and damping elements. In order to simulate machine vibration, the axes are modeled as direct drives and the guides are represented as spring-damper systems (**Fig. 2**). Cascade controls are used to control the drives. The guides typically are the main compliance of machine structures [8].

The mechanical model consists of a base frame, a bridge with Y- and Z-slides, and a laser scanner head. The X-drive is located at the base frame and moves the bridge along the linear guide rails of the frame in the X-direction. The X- and Y-directions are horizontal directions while the Z-direction represents the vertical direction. The base frame is very stiff because it is very thick and is usually made of steel and filled with concrete. Therefore, its influence on positioning accuracy is negligibly small compared with the compliance of the guide rails and guiding

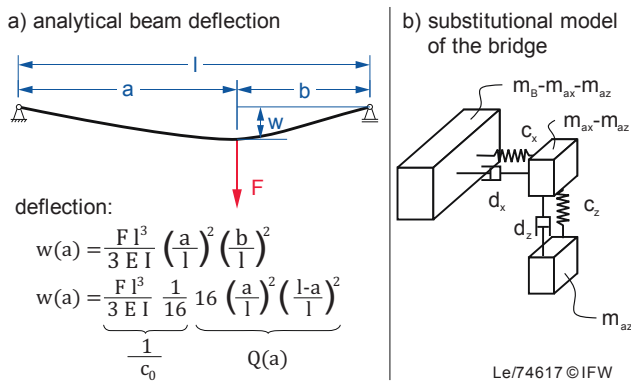


Fig. 3. Analytical beam deflection model.

shoes. To simulate this compliance, the stiffness values from manufacturers' specifications are implemented for both Cartesian directions normal to the X-direction. Measurements on a portal machine with static force, being applied to the guiding rails, confirm these stiffness values. For each direction, the stiffness of the guiding rails and guiding shoes are summarized into one stiffness and damping value. These springs and dampers are added for all three guiding rails in the X-, Y-, and Z-directions. The notation used for stiffness and damping values in Fig. 2 shows the assignment to their respective positions and orientations. The damping and stiffness elements connecting a body with its base are denoted with the respective body's first letter as the first index (i.e., g: gantry, y: Y-slide, z: Z-slide) and the acting direction as the second index. The stiffness and damping elements always act normal to the guiding direction. For example, the Y-slide has springs and dampers acting in the X- and Z-directions.

The deflection of the bridge is modeled within the multi-body model with two additional bodies, added between the bridge and Y-slide. The approach is shown in Fig. 3. The figure shows the position-dependent deflection of a bending beam. This deflection depends on the position (a) of the Y-slide, which represents the force application point. The deflection of the beam can be calculated analytically by the given equations. These are used to parameterize the enhanced multi-body model, which is built up to complement the given model of the machine kinematic. The first eigenmode of the deflection of the bridge in the vertical and horizontal directions are simulated by three bodies linked by springs and dampers. These bodies replace the bridge originally simulated by one body. A mechanical substitutional model of the bridge is presented in Fig. 3b), which shows the three bodies and the springs and dampers. The bridge has vertical and horizontal degrees of freedom representing the deflection in its first horizontal and vertical eigenmodes. The first body is added to represent the horizontal deformation. Thus, the springs are acting in the horizontal direction, which corresponds to the machine X-axis. The vertical deformation is represented by the second body and springs and dampers act in the vertical direction. The vertical direction corresponds to the Z-direction of the

machine. The bridge has a total mass of m_B . To keep the total mass of the bridge constant, the mass of the additional bodies is subtracted from the total mass. The resulting mass of the main bridge is $m_1 = m_B - m_{ax} - m_{az}$. The mass m_{ax} is the modal mass for the first eigenfrequency, representing the horizontal bending mode. Because the modal mass m_{az} , which is associated with the second (vertical) eigenfrequency, is connected to the second body, the mass of this body must be reduced by the modal mass m_{az} . Therefore, the second body has a mass of $m_2 = m_{ax} - m_{az}$. The mass, stiffness and damping parameters are calculated with the help of results from simulations using the finite element method.

From this structural-mechanical analysis, the first two eigenfrequencies as well as their modal masses and damping constants are known. The modal mass is calculated by Eq. (1) for the first eigenfrequency f_1 in the horizontal direction and the corresponding stiffness c_{0x} .

$$f_1 = \sqrt{\frac{c_{0x}}{m_{0x}}} \Leftrightarrow m_{0x} = \frac{c_{0x}}{f_x^2} \quad \dots \quad (1)$$

As mentioned before, the deflection amplitudes vary with the position of the slides. Thus, the stiffness and damping values have to be adjusted for each position of the slide. According to the deflection, the stiffness rises while moving from the center position to one side. This change is described by the position-dependent variable Q , which has a value from 0 to 1. The center position is defined as Q equals 1.

The stiffness is calculated according to the bending line of the beam (Fig. 3) by Eq. (2) for each position of the Y-slide, represented by the variable a :

$$c_x(a) = c_{0x} \frac{1}{Q(a)}, \quad \dots \quad (2)$$

where the variable c_{0x} is the stiffness value for the first eigenmode in center position. This value is obtained from a finite element simulation, as well. The damping value is kept constant because the impact on the machine behavior is low and does not influence the quality of the control strategy tested with this model. The mass for the deflection in the X-direction is calculated according to Eq. (3) in the same way as for the stiffness. To keep the eigenfrequency of the bending beam itself constant with changing stiffness values, the modal mass depends on the position, too. The modal mass m_{0x} is calculated from the eigenfrequency of this deflection shape and the stiffness c_0 is calculated as shown in Fig. 3. Thus, the additional mass is calculated with Eq. (3).

$$m_{ax}(a) = m_{0x} \frac{1}{Q(a)} \quad \dots \quad (3)$$

The third body of the model presented above is the new base for the Y-slide. A displacement of the body leads to a displacement of the Y-slide. Thus, the Z-slide, moving relative to this Y-slide is displaced the same way.

Compared with the X- and Y-slides, which are considered to be rigid, the Z-slide is considered as a flexible body. This is necessary because the actual Z-slide be-

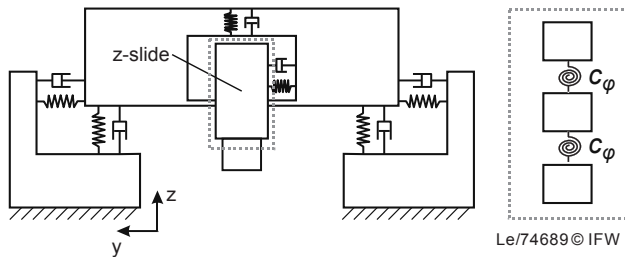


Fig. 4. Multi-body model of the Z-slide.

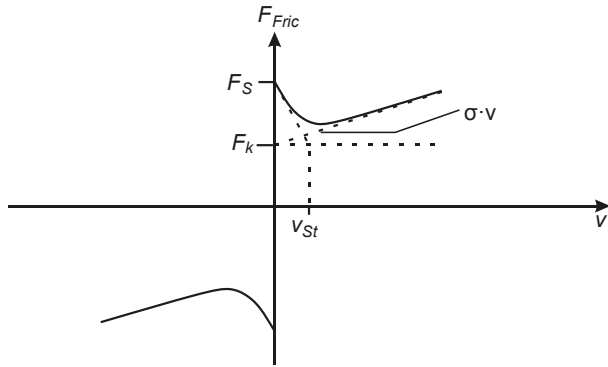


Fig. 5. Stribeck curve [19].

haves like a compliant bending beam. To consider the influence of the compliant bending beam on the vibration behavior of the machine tool, the Z-slide has to be implemented as a flexible body. For this purpose, the Z-slide is modeled by three rigid bodies which are connected to each other by rotational springs (Fig. 4). The stiffness of the springs c_φ is parameterized by a finite element model of the machine tool. This approach is based on Hackelöer and Torres [14, 15].

Each of the rigid bodies has one-third of the mass of the actual Z-slide. The inertia of the three bodies was calculated using CAD software. The stiffness parameter was determined as $c_\varphi = 5 \cdot 10^9$ Nm/rad.

At the end of the kinematic chain a scanner unit with three additional axes, described in section 2.1, is located. Because of the low inertial mass of the mirrors compared with that of the machine structure, the scanner is simulated as infinitely dynamic and without mass or inertia.

In addition to the machine components and joints, guide friction is implemented within the simulation model. This is necessary because guide friction influences the dynamical behavior of the machine.

To consider friction between the guide rails and the guiding shoes, a friction model is implemented within the multi-body model. The friction force is described by the Stribeck curve as shown in [9–12]. Fig. 5 shows the velocity-dependent friction force based on that model. The ordinate shows the friction force and the abscissa shows the velocity.

According to the Stribeck curve, the friction force F_{Fric} decreases after overcoming the solid/boundary friction owing to increasing velocity v until reaching the Stribeck velocity v_{St} . This friction is called mixed friction. Fur-

Table 1. Identified friction forces.

Velocity [m/min]	Friction F_{Fric} [N]
0	650
13	567.6
32.5	698.2
65	1019.4

Table 2. Estimated Stribeck parameters.

Variable	Name	Predicted value
F_k	Kin. friction force	360.2932 N
v_{St}	Stribeck velocity	0.1637 m/s
δ	Form factor	1.0555
σ	Viscosity factor	608.2003

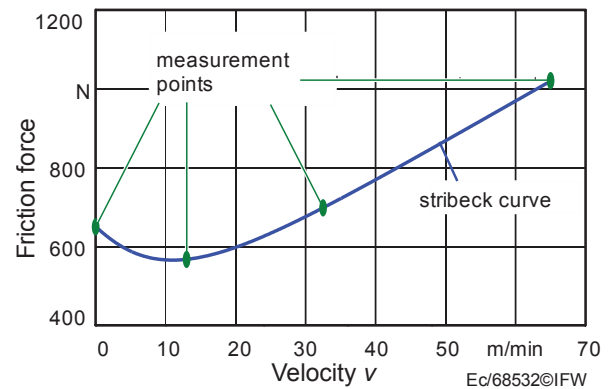


Fig. 6. Parametrized Stribeck curve.

ther increase of the velocity leads to asymptotic increased friction force which is called fluid friction. Eq. (4) shows the relation between friction force on the ordinate and velocity at the abscissa.

$$F_{Fric}(v) = \text{sign}(v) \left(F_k + (F_S - F_k) e^{-\left(\frac{|v|}{v_{St}}\right)^\delta} + \sigma |v| \right) \quad . . \quad (4)$$

This relation is defined by the static friction force F_S , the kinetic friction force F_K , the Stribeck velocity v_{St} , the form factor δ , and the viscosity factor σ . The static friction force F_S is measured by the drive current needed to overcome this static friction force while the machine axis is not moving. The measured drive current is multiplied by the motor constant K_M to achieve the acting force of the drive. The other four parameters are detected by moving the actual machine tool with different velocities. Based on the drive currents and the motor constant K_M , the friction force related to each velocity can be determined. Table 1 shows the friction forces related to the different velocities identified on a gantry machine axis.

To identify the unknown parameters of the Stribeck curve, the measured forces are used in a parameter study based on the genetic algorithm [13]. Table 2 presents the estimated values for this measured data.

Figure 6 shows the Stribeck curve belonging to the identified values presented in Table 2 as well as the mea-

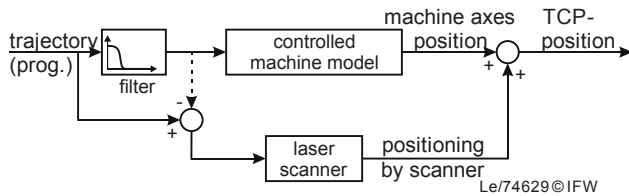


Fig. 7. Frequency separation method.

sured friction force for different velocities.

This friction model described by Eq. (4) is implemented in the multi-body simulation of the gantry machine tool.

2.3. Control Strategies

Owing to large machine components, the dynamic behavior of these moving machine parts is very inertial. Furthermore, measurements show that low stiffness and excitations of eigenfrequencies of the structure lead to positioning errors of the TCP. Thus, a hybrid positioning system was designed using a laser scanner. This laser scanner realizes three additional but redundant degrees of freedom. These are used to implement two hybrid positioning strategies, described in this section. The aim of these control strategies is the reduction of positioning errors and trajectory errors using the scanner head. One strategy is designed to use a filter avoiding dynamic impacts to the machine axis and the other predicts trajectory errors with the help of a Kalman filter and compensates these errors with the positioning unit of the laser scanner. The control strategies are described in sections 2.3.1 and 2.3.2. In this paper, the multi-body simulation generates measurement data used for evaluation of these approaches. Thus, the control strategies described in this paper can be applied to the kinematics of any real Cartesian machine without the need of a parameterized multi-body simulation.

2.3.1. Frequency Separation Method

The first approach, presented in Fig. 7, is designed to reduce machine excitations by filtering the trajectory before it is given to the machine axes. For this frequency separation method, the programmed trajectory is filtered for each axis separately. A low-pass filter eliminates dynamic excitations in the trajectory to smooth the axis trajectory. This avoids excitation of resonances of the machine structure. The filtered signal is taken as the programmed trajectory for the machine axes which are numerically controlled by an adjusted algorithm including the filter. The difference of the filtered and programmed trajectories is given to the laser scanner unit. Using this strategy, the TCP is the sum of the real machine axes positions and the position of the laser scanner. While the machine axes position depends on the kinematic of the machine and their dynamic behavior, the scanner is simulated as infinitely dynamic. Thus, the scanner output position is the programmed scanner position in consideration of the limited travel range. The cut-off frequency of the filter has

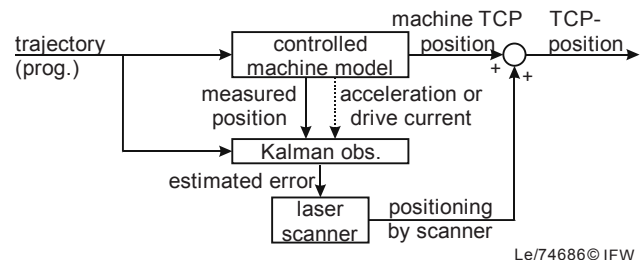


Fig. 8. Error estimation approach.

to be set up with a frequency which is low enough that the machine is able to follow this low-pass filtered trajectory. At the same time the frequency has to be high enough that the difference between programmed and filtered trajectories is not larger than the limited workspace of the scanner which compensates the high-frequency component of the trajectory. The filter is adjusted such that the TCP position, as the sum of machine position and scanner position, is able to follow the programmed trajectory. Because of the fact that this strategy does not get any information about the machine itself, it is not able to compensate positioning or trajectory errors. This strategy reduces impacts on the machine that lead to trajectory errors and controls the scanner in order to realize the high dynamic movements filtered out from the machine trajectory.

2.3.2. Error Estimation

The second approach is not designed to reduce impacts to the machine but to consider machine behavior and use this information for error compensation. The aim of this approach is a machine model estimating the position of the machine for a given trajectory. In this approach a Kalman filter is used to predict machine behavior. By this prediction, the positioning error is calculated and the scanner is set to compensate this error with a fast and very dynamic system suitable for this task. In this case, one Kalman filter is set up separately for each machine axis. This is a valid approach for Cartesian machines because the effect of axis coupling is very low.

As shown in Fig. 8, each Kalman filter uses three input signals to estimate one output. One input is the programmed machine trajectory. Furthermore, the linear scale built into the axis is used to identify the machine position. Use of a third signal makes prediction of the machine state even more precise. Thus, two different approaches for this third signal are shown and compared. In one approach, an acceleration sensor is used and the other approach uses the current signal. The output of the Kalman filter is the estimated machine position error.

The linear scales measure the relative distance between the two bodies connected to the corresponding joint in the direction of the axis. This is not the real position because the position of the preceding body may change owing to thermal, static, or dynamic effects. Thus, an acceleration sensor close to the TCP measures acceleration. The acceleration is a good characteristic to identify dynamic vibrations, leading to positioning errors. In contrast to the

linear scales, the acceleration sensors give absolute signals for TCP behavior. Thus, the estimation accuracy can be increased significantly.

To measure the acceleration close to the TCP position, an acceleration sensor mounted to the machine structure is used. This leads to additional purchase and maintenance costs for the machine tool and makes this approach impractical on any machine without an integrated acceleration sensor. Thus, an alternative approach described in this section replaces the acceleration signal with the current of the drives. The current of the drives must be measured in any case to control the drives; thus no additional sensor is needed. As described before, the friction force depends on the velocity of the drives and therefore has a nonlinear effect on the current. However, nonlinearities cannot be predicted by a Kalman filter if an extended Kalman filter is not used. This extended Kalman filter is able to determine nonlinearities but needs a higher effort for implementation and requires more computing power for control. Thus, the extended Kalman filter is not used for this approach. In order to predict the part of the current signal that leads to acceleration, the actual current signal is reduced by the part that is necessary to overcome friction. That part is calculated from the velocity signal using the friction model. In this way, the current signal can be used instead of the acceleration signal in the model identification process and as an input signal for the Kalman observer, which is necessary for the second control strategy. The need for a friction model makes this approach more complex in modeling and identifying the state space model but removes the need for an acceleration sensor. In the following section the control strategies are described using the acceleration sensor.

2.3.3. Identification and Use of Machine Model

As mentioned earlier, the Kalman filter needs a machine model to predict the dynamic behavior of the machine structure, drives, and control loops for an excitation while moving the machine axes to the position on the programmed trajectory. For identification of this model, the machine parameters used for setting up the multi-body model are not available. Instead, this approach is designed to generate a state space model from measurements received by a real machine. This model is typically expressed in state space, which makes system identification easier and does not require knowledge of physical relations.

Figure 9 shows the approach used for the estimation of the state space model for one axis. Input to the Kalman observer and, therefore, for the model, is the programmed trajectory u . Output of the model includes the measured signals (linear scale y_{lin} and TCP acceleration y_{acc}). Furthermore, the TCP position error (difference of actual position and set value) has to be estimated and is included as an additional output, y_{err} . To identify the real position of the machine, an external measuring system is used. This real machine position y_{TCP} is necessary for system identification. During compensation this values is solely esti-

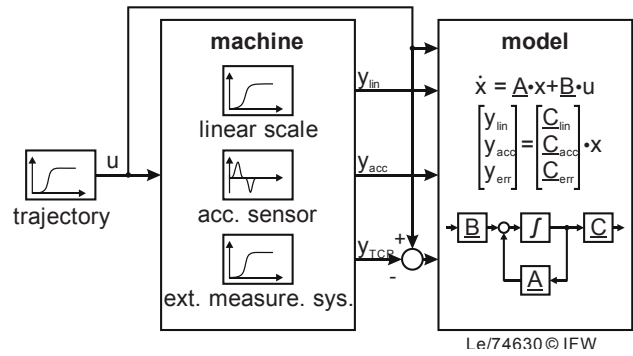


Fig. 9. Approach for creating a state space model of the machine.

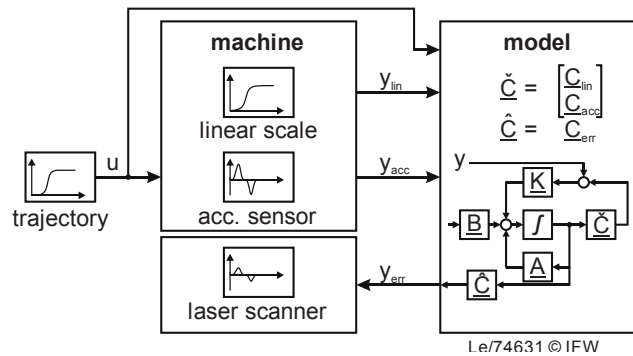


Fig. 10. Error prediction and compensation with a laser scanner.

mated by the Kalman filter. To excite a wide range of frequencies in the identification, the system is excited with a step trajectory in the relevant direction.

The estimation process can be done by different mathematical methods. An appropriate method is prediction error minimization (PEM) [16].

The identified matrices A and B describe the behavior of the machine while matrix C describes the output of the Kalman filter. For the identification process the C Matrix is set to put out all three measured values (y_{lin} , y_{acc} , and y_{err}).

Figure 10 shows the Kalman filter and feed forward loop for compensation of positioning errors caused by dynamic effects. The Kalman filter uses the state space model described earlier. For error compensation, several C matrices are identified. To feed the measurement signals back to the actual predicted system state, the output matrices C_{lin} and C_{acc} are used. The estimation output matrix C_{err} is defined for the compensation output. This matrix defines the error as the difference of the estimated position of the machine and the system input. For compensation this difference is not given as a measured value from an external measuring system, as for model creation, but is estimated and used to control the position of the laser scanner. More information on Kalman filters [17, 20] and state space models [18] can be found in the literature.

As explained before, the approach shown in this paper is developed using a multi-body simulation model, but it is shown that this approach is transferable to a real machine. With external measurement equipment, the posi-

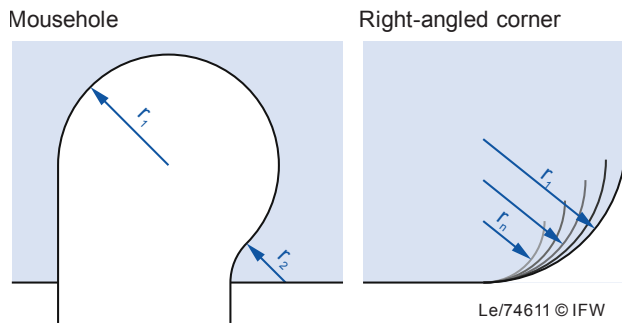


Fig. 11. Mousehole and edge radius with different radii.

tion of the machine can be determined for model identification. Thus, this model can be used in the same way as described in **Fig. 9** for any real machine.

2.4. Evaluation of Control Strategies

To analyze the control strategies presented in section 2.3 test trajectories have been defined. The aim of these trajectories is to have a high impact on the machine structure to analyze dynamic machine effects. One difference to metal cutting is the required constant-trajectory velocity over the whole trajectory. This is necessary for good surface quality in laser cutting. The test trajectories, presented in **Fig. 11** show a “mousehole” and a right-angled corner with different edge radii. The trajectory of the mousehole is deduced from the machining process of an aerospace frame. Mouseholes are pockets in aerospace frames which are necessary for good mounting methods. These mouseholes are represented with two different radii. The right-angled corner leads to a good excitation of the machine structure. As one axis accelerates, the other axis decelerates while the direction of the programmed trajectory changes. Thus, in Cartesian machines two axes are excited at the same time. The simulation will show how well these vibrations in two different directions can be compensated with the methods presented in this paper and whether the necessary trajectory accuracy can be ensured.

To analyze the capabilities of the two methods, a trajectory with a right-angled corner is simulated in the multi-body simulation. The methods are analyzed with different parameters such as edge radii, trajectory velocities, and cut-off frequencies of the low-pass filter in the frequency separation method. In the first step, the radius of the right-angled corner is varied from 2 mm to 20 mm. The resulting trajectories, simulated using the frequency separation method with a cut-off frequency of 3 Hz, are presented in **Fig. 12**. This figure shows the resulting simulated trajectory of the machine structure without the laser scanner using a dashed line. The solid line represents the simulated trajectory of the TCP with use of the laser scanner. The programmed trajectory velocity is kept constant at 21 m/min and is plotted with a black dotted line. It is shown that especially small radii excite the machine structure and lead to vibrations of the machine. The simulated trajectory is overdriving its end-position in the X-

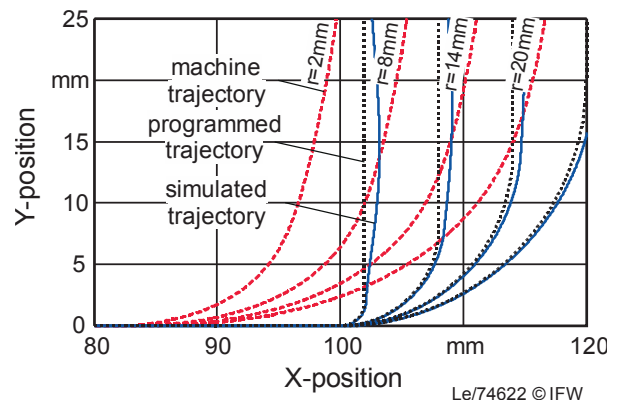


Fig. 12. Frequency separation method with a cut-off frequency of 3 Hz.

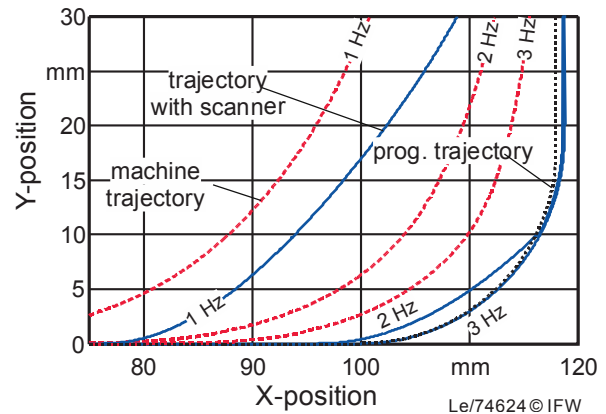


Fig. 13. Frequency separation method with varying cut-off frequencies.

direction due to compliance in the guiding rails and the resulting dynamic vibrations. The impact to the machine and, therefore, the position error, is lower for larger radii.

Besides the radius, the trajectory velocity also has a high impact on the trajectory accuracy. Experiments with laser cutting show that trajectory velocities up to 50 m/min may be necessary for the CFRP machining process. Especially with the frequency separation method, these high velocities cannot be realized with high accuracy for right-angled corners with small radii. In this approach the cut-off frequency is an important setting. While high cut-off frequencies result in higher excitation of the machine structure, low cut-off frequencies make the machine very inert to rapid changes. Thus, a large positioning offset results in poor trajectory accuracy if the offset is larger than the limited workspace of the laser scanner. The results for three different cut-off frequencies are presented in **Fig. 13** for a constant-trajectory velocity of 21 m/min. The dashed line shows the position reached with the kinematics of the portal machine. The programmed trajectory, a right-angled corner with a radius of 18 mm, is plotted as a black dotted line. The solid line shows the trajectory of the focus position of the laser with the help of the laser scanner.

Figure 13 shows that the maximum working range of

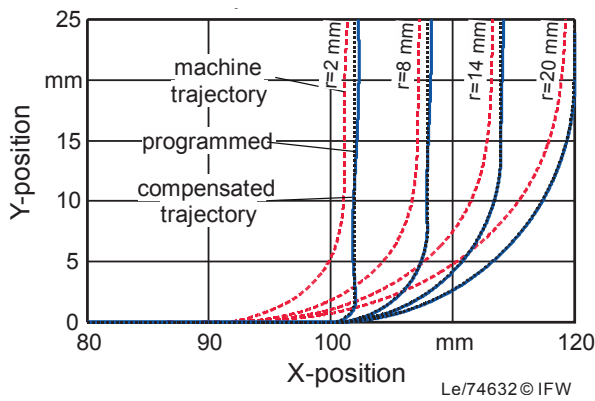


Fig. 14. Error estimation method with a Kalman filter.

the laser scanner limits the accuracy for low frequencies. Filtering the reference position with a 1-Hz low-pass filter leads to a large distance between reference and filtered trajectories. If this difference is larger than the limitation of the scanner then it cannot compensate the large trajectory errors of the machine. The limited workspace of the laser scanner is the reason for the large trajectory error using a 1-Hz low-pass filter with the frequency separation approach. Reducing the cut-off frequency leads to a trajectory with high position error because the dynamic part of the trajectory is filtered out. With a higher cut-off frequency the trajectory filtered out is reduced and, therefore, the machine is excited more.

Figure 14 shows the resulting trajectories simulated using an error estimation method. The trajectories of the machine without a laser scanner (red dashed line), the TCP position with error compensation using the laser scanner (blue solid line), and the programmed trajectory (black dotted) are presented in the same color scheme as in **Fig. 12**. The process parameters are the same as for the frequency separation approach. This example shows that compensation with a Kalman filter has more potential than the frequency separation method. While the frequency separation method avoids unwanted vibrations of the machine structure, the error estimation method estimates the error and compensates resulting vibrations. The results also show that right-angled corners with radii as small as 2 mm can be manufactured with good accuracy at a trajectory velocity of 21 m/min. This velocity was used for laser cutting operations in the tests done by the Laser Zentrum Hannover. The laser scanner compensates resulting vibrations from the machine structure very well.

For the error estimation approach, the trajectory velocity and the maximum working range of the scanner are important. The impact of the trajectory velocity on trajectory accuracy is plotted in **Fig. 15**.

With increasing trajectory velocity, the trajectory of the machine (red dashed) deviates from the programmed trajectory (black dotted). For the plotted radius of 18 mm a trajectory velocity of 50 m/min is possible. The resulting error of the machine can be estimated by the model and compensated with the laser scanner.

The limits of both control strategies have been ana-

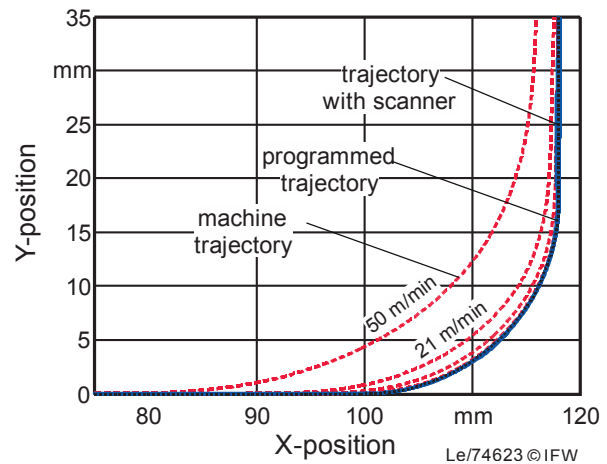


Fig. 15. Error estimation method with varying velocities.

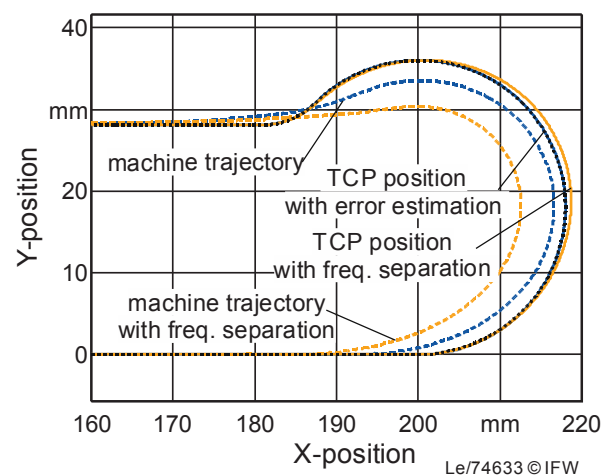


Fig. 16. Simulated trajectories while cutting a mousehole.

lyzed for different settings on a right-angled corner. A tool path for a real part from aerospace could be a mousehole used in frames in aerospace industries. This trajectory is modeled in the multi-body simulation using both control strategies. The resulting trajectories are plotted in **Fig. 16**.

The trajectory velocity is 21 m/min and the radii, defined in **Fig. 11**, are 18 mm and 9 mm, respectively. The tool path, plotted in yellow, represents the trajectory of the machine (dotted line) and the TCP (solid line) with the help of the frequency separation method. The yellow dotted line represents the part of the trajectory realized with the machine axis. The solid line represents the final tool path as the resulting sum of the laser scanner and machine part. While the machine axes are excessively inertial and thus drive a smaller radius than programmed, the trajectory including the laser scanner is close to the programmed tool path. The simulated resulting tool path includes small vibrations, which shows that the machine was dynamically excited even though the tool path was low-pass filtered. These vibrations cannot be compensated by the frequency separation approach. With the help of the error estimation approach these vibrations can be

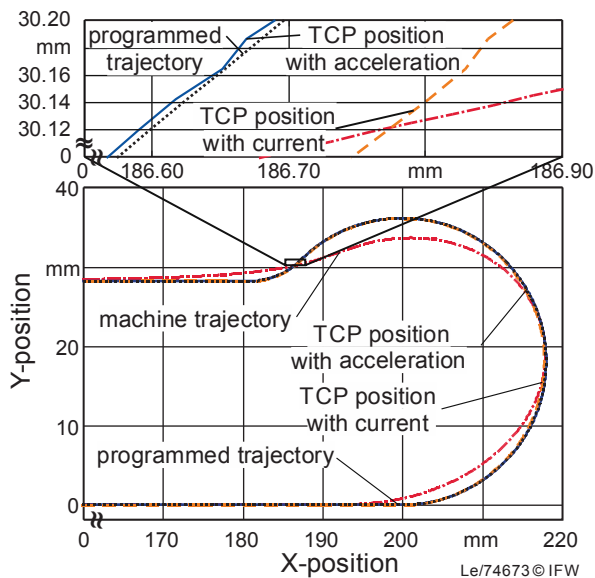


Fig. 17. Simulated trajectories for both compensation signals.

compensated. The Kalman filter can estimate the positioning error and use this information for compensation. This is shown in Fig. 16 by the blue lines. The dotted line represents the path of the machine without compensation or additional control strategies. The solid line shows that vibrations from the machine as well as positioning error are compensated by the laser scanner. The comparison shows the benefit of error prediction using the Kalman filter. Trajectory errors are much lower than without error prediction. For this programmed mousehole the maximum trajectory error is 60 μm while the filtered trajectory has a difference of 870 μm . Thus, the frequency separation method leads to a tool path which is outside of machine tolerance for this part.

This comparison shows that machining of this part is not possible without any control strategies compensating machine errors. Without the highly dynamic laser scanner these cutting speeds and radii cannot be manufactured using this machine.

2.5. Error Estimation with Use of Current Signals

As described above, the error estimation approach was realized with two different measurement signals for error prediction. In this section both approaches are compared to show their benefits and limits. To compare the suitability of both input signals for the error estimation approach, a state space model was identified for each signal and drive direction. To show the benefits for the customer, the mousehole cutting process was analyzed.

Figure 17 shows the simulated tool path for both signals. The programmed trajectory is plotted in black and has a radius of 18 mm. The trajectory velocity is kept constant at 21 m/min. The machine trajectory (red dash-dotted line) shows that the machine is not as dynamic as required for this laser cutting process. The positioning error is 19 μm . The trajectory of the focal point of the laser compensating with the help of the current signal leads to

a maximum deviation of 190 μm . This is 10 times higher than the deviation achieved with the error prediction with help of the acceleration sensor.

This comparison shows that the result achieved with help of the additional acceleration sensor is much better than that using the current sensor. Compared with the uncompensated machine trajectory with a maximum error of 2.5 mm, the achieved result is still very good.

However, the approach using the current signal is cheaper because it does not require an additional acceleration sensor.

3. Summary

A new control strategy for laser cutting machines has been developed. To realize a dynamic and low-inertial system, a hybrid positioning system is designed. In addition to the inertial machine axes, a highly dynamic laser scanner is used for positioning tasks. The benefit of this scanner is the possibility of highly dynamic positioning of the laser focus point. The positioning of the focal point is limited only by its workspace. While the scanner is simulated as infinitely dynamic without inertia or dead time, the dynamic behavior of the machine is simulated in a multi-body simulation. The model contains the kinematics of the machine as well as a cascade control for each axis actuating the corresponding axis. This model is enhanced by multiple bodies representing the first two bending modes of the bridge.

Two control strategies have been presented and analyzed in terms of trajectory accuracy as well as resulting performance in the machining process. In one approach the programmed trajectory of the machine is low-pass filtered. The second approach uses a Kalman filter to predict and compensate positioning errors. As the input signal, the drive current can be chosen as well as the acceleration signal received from an additional sensor close to the TCP. The advantage of using the drive signal is that no additional costs are required since the necessary signal is already present within the machine control system. The advantage of the use of the acceleration signal is higher accuracy, as the investigations in this paper have demonstrated.

Both strategies have been applied to a trajectory used for cutting mouseholes. The best results were achieved with the error estimation approach, assuming a good model for the dynamic behavior of the machine. To obtain this model an approach is described for measuring and predicting the model with integrated position sensors. An additional acceleration sensor is added close to the TCP position, improving the accuracy of the model and therefore the error estimation in the Kalman filter. It is shown that high trajectory velocities necessary for laser cutting can also be realized with good precision for small radii.

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Name:

Berend Denkena

Affiliation:

Full Professor, Production Engineering and Machine Tools, Director, Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

Address:

An der Universität 2, 30823 Garbsen, Germany

Brief Biographical History:

1987-1992 Research Engineer, Institute of Production Engineering and Machine Tools, University of Hannover
1992 Doctorate in Engineering at the Faculty of Mechanical Engineering, University of Hannover, Doctor Thesis "Wear of Ceramic Cutting Tools"
1992 Thyssen Production Systems, Essen, Design Engineer
1993-1995 Head of Standards Engineering and Systems Analysis, Thyssen Production Systems Inc., Auburn Hills, USA
1995-1996 Head of Machining Center Development, Thyssen Production Systems, Ludwigsburg "Hüller Hille GmbH"
1996-2001 Head of Engineering and Turning Machine Development, Gildemeister Drehmaschinen GmbH, Bielefeld
2001- Full Professor (W3) of Production Engineering and Machine Tools, Director of Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

Main Works:

- G. Byrne, D. Dornfeld, and B. Denkena, "Advancing Cutting Technology," Annals of the CIRP, Vol.52, Issue 2, pp. 1-20, 2003.
- R. Neugebauer, B. Denkena, and K. Wegener, "Mechatronic Systems for Machine Tools," CIRP Annals – Manufacturing Technology, Vol.56, Issue 2, pp. 657-686, 2007.
- B. Denkena, F. Hackelöer, and C.-C. Neuber, "5-DOF Harmonic Frequency Control Using Contactless Magnetic Guides," CIRP J. of Manufacturing Science and Technology, Vol.2, Issue 1, pp. 21-28, 2009.

Membership in Academic Societies:

- International Academy of Production Engineering (CIRP), Fellow
- German Academic Society for Production Engineering (WGP), Member
- German Research Foundation (DFG), Fachkollegiat für Spanende Fertigungstechnik, Fellow



Name:

Martin Eckl

Affiliation:

Research Engineer, Machines and Robots, Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

Address:

An der Universität 2, 30823 Garbsen, Germany

Brief Biographical History:

2009- Research Engineer, Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

Main Works:

- B. Denkena, M. Eckl, and D. Brouwer, "Development of a Multiple Degree of Freedom Knee Disarticulation Prosthesis with Active Leg Length Variation," Modern Mechanical Engineering, 2014, 4, pp. 207-221, DOI: 10.4236/mme.2014.44020
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Name:

Thomas Lepper

Affiliation:

Research Engineer, Machines and Robots, Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

Address:

An der Universität 2, 30823 Garbsen, Germany

Brief Biographical History:

2012- Research Engineer, Institute of Production Engineering and Machine Tools, Leibniz Universität Hannover

Main Works:

- K. Litwinski, M. Eckl, T. Lepper, and B. Denkena, "Hybrid Positioning for Large Laser Cutting Operations of Carbon Fiber Reinforced Plastics," The 15th International Conference on Precision Engineering, Kanazawa, Ishikawa, Japan, pp. 350-354, 2014.
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