Active component and control design for torsional mode vibration reduction for a parallel kinematic machine tool structure

Reimund Neugebauer, Volker Wittstock*, André Bucht, André Illgen
Fraunhofer-Institute for Machine Tools and Forming Technology IWU Chemnitz/Dresden, Germany

ABSTRACT

The paper reports the holistic development of an active piezo-based component concerning the mechanical design and the control. The active component is used for the reduction of torsional vibrations in a strut of a tripod parallel kinematic machine. By means of this new component the main drawback of the x, y, z-tripod structure can be eliminated. A calculation shows the compliance of the connection between actuators and the adjacent mechanical parts as the most sensitive point of the design. The characteristic values of the piezo actuator were transformed into the active component with the help of design factors. For reducing the structural vibrations a control laws is presented that changes the properties of the electro-mechanical structure, like damping or stiffness. This is possible by a feedback of motion signals, e.g. velocity. The described electro-mechanical model was used for the control design. Experiment results, which are finally presented, show a reduction of structural vibrations.

Keywords: Design factors, piezo-based component, parallel kinematic machine, machine tool, active damping

1. INTRODUCTION

In the early nineties the parallel kinematic machine principles (PKM) became an important field of machine tool research. According to their non-orthogonal arrangement of the three feed axis, each feed axis have to be controlled simultaneously or parallel. Contrary to the classical machine tool with serial feed axis, there is no coincidence of feed axis direction and the workspace coordinates. The main advantages of the PKM is the lightweight structure, which guarantees high dynamic [1]. But this causes also the disadvantage of the missing stiffness of the mechanical structure, that is necessary for precision machining operation. The design rules of traditional high precision machine tools don’t work for that new kind of high dynamic machine tools. The structural vibrations are excited by the inertia force as well as by the process forces while cutting operations. Therefore the application of active or adaptronic components in drive structures can improve the system behavior and precision especially of parallel kinematic machines [2].

A typical example of the structural drawback is a tripod (Fig. 1). This kind of machine has only three struts (or legs) instead of six of the so-called hexapod and realizes motions in x, y, z – direction. According to torsional load on the struts the static and dynamic stiffness of these structures is too small for achieving a considerable precision. Since the direction of the vibrations caused by this torsion does not correspond to the stroke of these struts the main drives cannot compensate them. In order to eliminate this structural drawback, new active elements for compensation were introduced. These elements were placed between the movable end of each strut and the spindle support (end effector) [3]. Based on this idea the “intelligent strut” was built in 2002 [4]. It consisted of the commercial strut actuator system and the laboratory specimen of a new so-called actuator-sensor-unit (ASU). The main item is a piezo-driven rigid state joint that levels out the torsional vibrations. The piezo stroke was controlled by using a simple controlling concept with disturbance correction and considering the piezo actuator state. Compared with the conventional strut the stiffness of the active strut was up to 40 times higher.

2. EXPERIMENTAL TRIPOD MACHINE “3POD”

2.1 Machine layout

For validating the function of ASU mentioned above and other active components the Fraunhofer IWU designed a modular parallel kinematic test bench for x, y, z motion of the HSC (high speed cutting)-spindle for milling (see Fig. 1).

*volker.wittstock@iwu.fraunhofer.de, Fraunhofer IWU, Nöthnitzer Str. 44, D-01187 Dresden, Germany
Due to the modular concept the test bench can be easily reconfigured. Clearly defined interfaces simplify the chances and the enhancements of components or assemblies. Identifying feature of the 3POD is the use of three driven struts with a complete different characteristic. In this way, these three struts cover the most common types, which are currently used in parallel kinematic machines. In respect of the problems with analysis methods at a PKM, the improvements of the machining result concerning dynamic and precision can be directly demonstrated at the 3POD. One side of each strut is fixed to the base by means of cardan joints. The movable ends are also cardanic connected to the spindle support. With respect to the short distance to the spindle support, the active components should be mounted at these ends.

Fig. 1. Experimental parallel kinematic test bench 3POD at the Fraunhofer IWU Dresden [5]. The drive direction of the struts and the torsional loads are indicated

Fig. 2. Frequency response of the spindle support in z-directions

2.2 System analysis of the passive system

Though the system of machine structure and active component was concurrently developed as one system, it was not possible to simulate the complete system for sizing the active component. The introduction of smart material in the passive structure would change the whole system. For this reason it was additionally necessary to use assumptions for
dimensioning the active component. Another problem is that the mechanical PKM-structure has different properties at every point of the workspace because of geometrical relation of the struts to each other (angle, length, inertia). In addition to that the disturbance process force can vary in wide range. By using a multi-body system calculation the maximum of the torsional load was identified. Nevertheless, it was not possible to define a maximum of force or moment vs. stroke or twisting that is necessary to be generated by the used piezo actuators. As generally known from other PKM-structures the struts were identified as the weakest elements. In the experiments the spindle support was excited by a shaker in z-direction, whose axis did not cross one of the strut axis. A representative frequency response is shown in Fig. 2. The picks represent the natural frequencies of the torsional mode of each strut. The frequency range lies clearly beneath the common frequency range of machine tool components. As the decentralized controllers of the ASU are operating independently each controller can adapt itself to the strut structure. Thus, the same mechanical design of the component can be used for all struts. The main cause of the yielding of the spindle support is the missing stiffness of the length constant strut. Consequently, this strut 3 will be equipped first with the ASU.

3. DESIGN OF THE ACTIVE COMPONENT

3.1 General design aspects

Active components as part of assemblies are provided for reducing vibrations of the machine structure. As the active components are still developed separately it is necessary to define certain constraints. But the problem is that the active component itself changes the behavior of the whole system. However, a system simulation makes no sense, if no actual and verified model of the active component is available. After the initial experimental verification of the intelligent strut mentioned above (see Sec. 1) the actuator-sensor-unit and power transmission were detected as weakest points of the system. The conclusion was drawn, that first of all, the piezo-based mechanical component respectively the power transmission must be significantly improved.

The active components can either be designed to improve already existing machines respectively substructures or for realizing total new machine concepts, which absolutely need the new functionality (e. g. test bench 3POD). But in both cases the design process could not start from a passive, but optimized component since this structure is too stiff and the entire piezo-generated force is be necessary for deforming the passive metal structure. The force, which remains for compensating process forces, would be too small. With a purposeful designed yielding structure, which is added by an actuator that lies in the flux of forces, an active structure evolves, which can adapt itself to ambient conditions. The flux of forces within the active component is going to be conducted across piezo actuators.

![Principle solution of the redesigned torsional compensation applied for six actuators](image)

3.2 Basic idea of the differential actuator setup

Equipped with a differential setup of piezo actuators the actuator-sensor-unit acts as additional lateral drive to the main drive direction. The piezo stack actuators with its wide frequency range and high-resolution stroke have a suitable drive characteristic concerning the expecting disturbance level. In contrast to the single actuator setup there are less restrictions
with the differential arrangement concerning the location and the direction in the machine setup. But then again the design is more complicated. A kind of clamping part holds the adjustable preload elements at the separated ends of the actuators. This has a positive effect as no additional outside spring has to be mounted for this purpose, which would reduce the actuator performance. With this property the differential setup is ideal for applications in light weight structures like PKM, at them a vertical mass rarely preloads the actuators with compression. The main advantage is that all parameters of the active component are generated in two directions. Piezo-based components need a certain preload to protect the piezo actuators form backlash or tensile stress generated inside of the component under dynamic load. According to the inner structure of the multilayer stack actuator tensile stress leads to its fracture. The essential preload depends on the dynamic load at the active component. The mechanical preload is completed by the so-called electrical preload, which is generated while the offset voltage \( u_{offset} \) is applied. The flux of force is closed inside the component (see Fig. 6 and [6]).

![Explosion view of the CAD-model of the actuator-sensor-unit design for applying three times two pairs of actuators](image)

**Fig. 4. Explosion view of the CAD-model of the actuator-sensor-unit design for applying three times two pairs of actuators**

### 3.3 Basic principle of the 3rd generation

The Fig. 3 shows the basic electromechanical principle of the transformation of the actuator stroke to a rotating movement. Taking an offset voltage (the half of the nominal voltage) as starting point the actuators are inversely activated. This leads to a twisting movement of the assembly. By means of lining up several of these basic elements the free angle of twist can be increased with concurrent constant torsional moment. The single disk-shaped elements are now connected by a new solid state joint consisting of simple beams with the torsional stiffness \( c_{St} \). In this way the undesired deformation in the flux of force, that reduce the efficiency of the component, was set down. The arrangement with three times two pairs of actuators is shown in Fig. 4. It already includes the power supply of the actuators via printed circuit board. The proved outer sleeve as solid state joint with the torsional stiffness \( c_{H} \) and the mechanical interfaces to adjacent components were reused with a few changes to the former generation. All characteristic values like stiffness, damping, forces and inertia can transform by means of the radius \( r_{P} \) either to the actuator stroke or inversely to the twisting (or torsion) of the strut (the stiffness concerning the actuator stroke is generally named with \( k \)).

### 4. MODELING OF PIEZO-BASED COMPONENTS IN DIFFERENTIAL SETUP

#### 4.1 Electromechanical analogy for piezo actuators

Considering all condition and the geometry of multilayer stack actuators according to Fig. 5(a) the actuator stroke can described with the following simplified equation:
\[
\Delta l_p = s_{33} \cdot \frac{I_p}{A} \cdot F_y + d_{33} \cdot n_p \cdot u_p
\]  
(1)

(with \(s_{33}\)-elastic compliance coefficient, \(I_p\)-actuator length, \(A\)-active area of piezo ceramic material, \(d_{33}\)-charge constant, \(n_p\)-number of layers, \(u_p\)-applied voltage, \(t_p\)-ceramic layer thickness, \(E\)-electric field, \(T\)-tension). The equation couples the stroke and the generated force. The maximum force \(F_B\) is reached by blocking the actuator, in contrast to that, the displacement is zero while no external force is applied.

The electromechanical equations of state lead to the two-port model for discrete actuators. This model has been used e. g. in the field of electro acoustic to convert mechanical systems into electric circuit elements. But the usage of analogies leads always to certain compromises and restricts the general validity. For the mechanical engineering the mechanical system itself must become the focus of attention, so that the converting of electric quantities into mechanical one is practical. Assuming that a voltage controlled amplifier is used the calculated results must include a statement of the required electrical current \(i_p\) and with them also a statement of the required total power of the amplifier. Due to the capacitive characteristic of the actuators, the electric current is caused by the change of the voltage amplitude as well as by external disturbance of the active component. The mechanical stiffness \(k_p\) of an actuator, which can be derived from the blocking \(F_B\) force and the free stroke \(\Delta l_p\), covers total values of both, the active layer and the passive electrode layers.

\[
\alpha_{33} = \frac{d_{33} \cdot A}{s_{33} \cdot t_p}
\]  
(2)

with the relations for voltage to force:

\[
F_i = \alpha_{33} \cdot u_p
\]  
(3)

capacity to electric stiffness:

\[
k_{el} = \alpha_{33}^2 \cdot \frac{1}{C_p}
\]  
(4)

resistance to damping coefficient:

\[
d_{el} = \alpha_{33}^2 \cdot R_i
\]  
(5)

Resultant, the change of the coordinate \(x_{el}\) is proportional to the electric charge and leads to electric current:

\[
i_p = \dot{q}_p = \alpha_{33} \cdot \dot{x}_{el}
\]  
(6)

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Fig. 5. Geometry of the used piezo stack actuator (a), electromechanical transducer model of a piezo stack actuator as converter (b) and its reduction for quasi-static calculation (c)
Using these equations the input for simulation and the design of amplifier can be derived.

Basically, there is a difference between material characteristic and actuator parameters. This fact among other things can be explained by the electrodes, which are made of thin layers with a smaller stiffness than the ceramic. The comparison between the calculated parameters of the ideal actuator geometry and real actuator structure shows a discrepancy, which cannot be ignored. If the dimensions of actuator geometry are chooseable, the equations (2) to (6) are be applicable. But for the majority of applications of mechanical engineering there are only few different geometries of commercial stack actuators available. In this case, the real characteristic values of actuators are used, completed by those of the piezo material. With static calculations, the model can be simplified by eliminating the electric stiffness $k_{el}$ and with replacing the inner piezo force $F_i$ by the blocking force $F_B$. The active actuator stiffness $k_p$, which is measured at the short-circuited actuator, can be taken from the catalogues, too. Due to the combination of both theoretical and measured input values, the calculation results has to be carefully rated concerning the scope.

$$x_1 = \frac{F_{S} \cdot (1 + z_p) \cdot (1 + z_A \cdot z_p) + z_p \cdot [F_{i12} \cdot (1 + z_A \cdot z_p) + F_{i11} \cdot z_A \cdot (1 + z_p)]}{k_p \cdot z_{FKG} \cdot (1 + z_p) \cdot (1 + z_A \cdot z_p) + z_p \cdot [1 + z_A + 2 \cdot z_A \cdot z_p]}$$  \hspace{1cm} (9)

The different coefficients of the inner force terms $F_{i,n}$ show the influence of the component asymmetry $z_A$. The blocking force has exactly the value, which is necessary to push the positioned component back to the original zero position ($x_1 = 0$). Therefore the equation is set to zero and transformed to the outside disturbance force $F_S$. The offset voltage,
applied at both actuators, leads to an offset moving of the coordinate $x_{\text{off}}$. The real blocking force results from the average of the direction-dependent values:

$$F_{B,\text{eff}} = \frac{z_A(1 + z_A + 2z_p z_A)}{(1 + z_p)(1 + z_p z_A)}, \quad F_{\text{eff},max}$$

The efficiency of the force and stroke transmission is primarily characterized by the serial stiffness $k_{\text{An}}$ of the adjacent components respectively of the design factor $z_p$. The different dependence of both values, blocking force $F_B$ and free stroke $\Delta l_p$, from this factor is shown in the diagram of the Fig. 7. Consequently, a well-designed power transmission consists of elements in the flux of force that are as stiff as possible. The main advantage of the use of design factors is that totally different active component are comparable.

![Diagram](image)

**Fig. 7.** Different dependence of blocking force $F_B$ and free stroke $\Delta l_p$ from the compliant serial piezo support (design factor $z_p$)

Considering all elements of the component and the transformation by means of the radius $r_p$ the torsional stiffness of the actuator-sensor-unit is:

$$c_{\text{ASE}} = r_p^2 \left( \frac{2n_A}{n_{\text{sch}}} \cdot \frac{k_p \cdot k_{\text{An}}}{k_p + k_{\text{An}}} + \frac{k_{\text{Sh}}}{n_{\text{sch}} + k_{\text{H}}} \right)$$

According to the small value of $k_{\text{Sh}}$, this term can be neglected. The static dimensioning of the component also encloses the calculation of the necessary preload for achieving a backlash-free system, which is described in detail in [7].

### 4.3 Dynamic modeling

For the dynamic calculation and the inclusion in combined machine dynamics – closed loop control models the reduction to two external and one internal degrees of freedom is necessary. The model reduction is effected by summarizing the serial element and the actuator values. In contrast to the symmetric setup, this summarization of asymmetric elements leads to a deviation in the calculating results. With a optimized designed component this disadvantage is less serious since the inclusion in the overriding system structures is simpler. Due to the missing static preload force in the dynamic model, the dynamic force is modeled in the simplified model as alteration from zero. The equation of motion can be solved only in the frequency range because of the missing mass at the coordinate $x_2$. The velocity of the internal coordinate $x_{2,\text{el}}$ can be converted to the electric current and the phase by using the electromechanical analogy. For calculations in the time domain it is necessary to use the state-space model established in control engineering. The Fig. 10 shows the simplification from the whole component model to a reduced model, which can be used in overriding machine simulations.

## 5. CONTROL DESIGN

### 5.1 Active damping using pole placement

The behavior of a dynamic system is determined by the poles in the complex plane. By means of inducing of forces, which are proportional to suitable kinetic quantities like position, velocity and acceleration, the poles can be specifically moved and thus, the system behavior will be influenced as desired. This fact is illustrated in Fig. 8. The example shows
the system damping of a single degree of freedom (DOF) system. The system is described by a linear differential equation

\[ m\ddot{x}(t) + d\dot{x}(t) + cx(t) = F_i(t) \]  

After converting into the frequency domain and restructuring the equation

\[ G_s(s) = \frac{X(s)}{F(s)} = \frac{1}{c} \cdot \frac{1}{T^2s^2 + 2DTs + 1} \]  

denotes the description of the transfer function as it is commonly used in control engineering. \( T \) describes the resonant frequency and \( D \) the damping of the system. With a feedback of a proportional velocity signal as shown in Fig. 8, the transfer function of the closed loop control circuit is

\[ G_{wa}(s) = \frac{G_s(s)}{1 + G_s(s) \cdot k_{DVF} \cdot s} = \frac{1}{c} \cdot \frac{1}{T^2s^2 + \left[ \frac{2DT + k_{DVF}}{c} \right]s + 1} \]  

Now, the damping term is the summation of the natural system damping and the additional damping induced by a suitable actuator. All other parameters of the of the closed loop remains unchanged in comparison to the open loop.

![Fig. 8. Single DOF-system (a) and closed control loop with velocity feedback (b)](image)

This attempt of a solution shows several advantages:

- The actuator-sensor-unit (ASU) can be operated autonomously. Thus, no communication with the overriding machine control is necessary and the integration in existing systems is possible with less effort.

- For setting up the controller only a little knowledge of the structure and the parameters of the control path are necessary. There is no need of a detailed model of the control path. Only one free parameter has to be set.

- According to the simple structure a very simple and practical realization of the controller is possible.

- On condition that sensor and actuator constitute a collocated pair, the stability can be guaranteed [9].

The disadvantage of this approach is that there is no influence on the frequency response outside of the resonance.

### 5.2 Setup and used hardware

Piezo actuators are electrical capacities whose power requirements rise in a linear fashion at a constant level of voltage when the triggering frequency rises. This is the reason why in dynamic operation it is necessary to provide sufficient amplification power for the piezo actuators used. The control signals are generated with the XPC-Target program package built into the MATLAB®/Simulink® that supports prototypical implementation of control algorithms in real time. The block-oriented model created on a desktop PC (the host PC) is transformed into an executable code with a C/C++ compiler. All of the model’s signals and parameters are written in one variable that acts as the interface between the desktop PC and industrial PC (target PC) that processes the control model in real time. After D/A-conversion, the controlled voltage is available per actuator at the input of the voltage amplifier. The sensor signals are read into the target PC and processed after A/D-conversion. A graphic operator interface with built-in safety functions and limits is used for controlling.
5.3 Active damping of torsional vibration

The described control algorithm is applied at the actuator-sensor-unit (ASU), which was mounted at strut 3 (see Fig. 1). Especially this strut has structural drawbacks. The relatively high yielding of the tube made from aluminium can be levelled out by the active component with only a little increase of the strut mass.

The twisting of the strut is detected by strain gauges applied to the tube and the ASU. The angular velocity is deduced by a derivative controller. However, this kind of controller strongly amplifies high frequent interference signals (e.g. measurement noise). Therefore, a lowpass is added to the controller. The following controller structure has been proven

\[ G_{DVF}(s) = k_{DVF} \frac{T_k s}{(f_k s + 1)} \]

As the controller has only two free parameter the setup is relatively simple. The parameter \( T_k \) denotes the time constant of the controller. The cut-off frequency follows as

\[ f_k = \frac{1}{2 \cdot \pi \cdot T_k} \]

Below this frequency the controller acts as differentiating element and provides as a result a velocity proportional control signal. In the range above \( f_k \) the controller acts as a lowpass and avoids launching high frequency noise into the control loop. The second parameter, the gain \( k_{DVF} \), is set up iteratively at the controlled system.

![Fig. 9. Advanced mechanical model of the actuator-sensor-unit for the control design](image)

Fig. 9 shows the advanced model of the ASU built in the strut 3 of the 3POD, which uses the reduced electromechanical model. This overriding model was reduced to the torsion of strut 3 with only a few degrees of freedom and considering the transformation with the radius \( r_p \). Consequently the coordinates are named \( x \) for translation and \( \varphi \) for rotation respectively torsion. The disturbance moment \( M_S \) is equivalent to the disturbance \( F_S \) and acts against the mass moment of inertia \( J_S \). The other two struts are not active struts yet.

6. EXPERIMENTS

Each design cycle is concluded by an assurance of properties. According to the guideline [10], there is a difference between verification and validation. In case of the ASU the verification includes all experiments, in which the functions of the component itself were inspected. The validation leads to a higher level of testing. The question has to be answered, whether these functions of the ASU are useful for the overriding system. Especially with the validation, the integration of different domain-specific design will be tested.

6.1 Verification of quasi-static characteristic values

For the verification of the mechanical calculation a separate test bench was designed, which is suitable for loading the component with a pure torsional moment. The transformation between the generate moment and the measured force is
realized by means of a lever. The measurement of the free twisted movement is possible without any restriction. Contrary to that, the measured blocking force is negatively influenced by the friction in the second bearing (Fig. 10a).

The torsional stiffness can be measured in a static test as well as in a free-vibration test. There were no big difference to the calculated value of 46.2 Nm/mrad. The free twisted movement is possible in frequency range up to 300 Hz without any resonance picks.

![Fig. 10. Actuator-sensor-unit built in a uniaxial test bench for simulation disturbances of the 3POD (a) and twisted angle – blocking moment diagram with experimental results in comparison to the calculation (b)](image)

The twisted angle – blocking moment diagram is shown in Fig. 10b. The value of the blocking moment is the result of the extrapolation of the end points of measured curve of compliance. In these tests the actuator-sensor-unit is working against parts with different compliance. The intersections points with the diagram axis correspond to the blocking moment $M_B$ and free twisted angle $\Delta \phi_P$. The measured value of the free torsion meets the calculation result. The big differences concerning the blocking moment indicates a serial compliance, which is much bigger than expected, as well as the problem of the test setup. Despite the significant improvement to former component designs, the serial stiffness remains still the weakest point of the design.

![Fig. 11. Experimental set up of the 3POD (a) and measured frequency response of strut 3 concerning twisting of strut 3 (b)](image)

6.2 Verification of control design

The ASU was mounted into the length constant strut of the 3POD. The excitation of cutting forces as disturbance is simulated in the experiments at the 3POD by means of a shaker, which is mounted off-center underneath the spindle.
support. The shaker generates sinusoidal forces. A laser measuring system detects the position in z-direction where the shaker force is induced. Fig. 11a shows the experimental setup and the strut 3. The first resonance at approx. 13 Hz is damped by the ASU. Compared to this, the influence of the controller at the second resonance is smaller because of the kinematic (see Sec. 2). The yielding of both other struts leads also to a tilting of the spindle support. For this reason each strut ought to be equipped with an actuator-sensor-unit. But the concept and the design of the torsional compensation is verified by these experiments.

6.3 System validation during cutting operation

With the validation, the active damping was tested during cutting operations of the 3POD. Fig. 12a shows the experimental setup of the 3POD. Measuring signals were taken from the strain gauge at strut 3 concerning twisting, which is the variable to be controlled, and from acceleration sensors, fixed to the spindle support in z-direction. The function of the ASU can be evaluated by using both signals. The milling cutter, mounted in the spindle, was put into operation in such a way, that there was a considerable torsional load at strut 3. The spindle speed was adjusted to the eigenfrequency of the strut 3. Nevertheless, the roughing cutter generated a wideband disturbance, which influenced the other struts too.

Fig. 12. Experimental set up of the 3POD (a) and exemplary time signal of the controlled and uncontrolled twisting of strut 3 (b)

Fig. 13. Comparison of twisting of strut 3 during the positioning movement (a) and amplitude of the z-displacement of the spindle support derived from the integrated acceleration signal (b)
The spindle support was moved in x-direction with constant speed and depth of cutting. The control was switched on and off alternately after 10 mm feeding. The Fig 12b illustrates the random disturbance at strut 3 caused by the milling process in comparison with the controlled state of the strut while cutting at the same point. In addition to that, the effect of the active damping can be demonstrated during the positioning movement of the spindle support. The inertia force excited torsional vibration of the strut. The Fig. 13a shows, that the amplitudes can be clearly reduced. With those, the function of the ASU was validated concerning the subsystem “strut 3”. The requirement of the next development cycle is, in case of this subsystem, the comparison of different control designs.

Since the ASU is not able to reduce the vibrations of all directions, only a partial validation concerning the system “3POD” is possible. But also the acceleration in z-direction was clearly reduced. The two times integrated acceleration-time-signal delivers the displacement. The Fig 13b shows the FFT of these signals. The significant reduction up to 17 dB at the resonances of the strut 3 indicates the influence of the active damping of the ASU upon the whole system.

7. SUMMARY AND OUTLOOK

The paper reports the development of an actuator-sensor-unit for experimental machine tool, which has passed through different states from the single component to a whole system. The actuator-sensor-unit was built in a strut of a parallel kinematic machine for compensating torsional vibration. The final test during cutting operations is the result of several iterative design cycles of both mechanical and control engineering. The quality of the mechanical design can be evaluated by using design factors. This specific improvement of the mechanical properties was the prerequisite of the success of the control strategy. The active damping was introduced in the system by using a velocity feedback controller. The decentralized controller was validated by measuring the twisted angle of the strut. With the chosen strategy only the resonances of one strut can be influenced.

Further work will also deal with the model based control design methods such as LQG control. First results are reached on the test bench (not presented in this paper). These results could not directly transferred to the parallel kinematic 3POD since the dynamic of the strut depends on the specific integration in the mechanical environment and of the actual position of the spindle support. Thus, the emphasis will be laid on the derivation of an appropriate state space model or a set of state space models for different operating points of the 3POD and the control design. The further research will focus on the introduction of the second and third active strut into the experimental machine tool, which will reduces the vibration amplitudes of these struts and consequently of the spindle support. Thus, the machining quality can be improved further.

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