Piezo-based Actuator-Sensor-Units for Uniaxial Vibration Reduction in Machine Tools*

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Abstract
Additional piezo-based components in drive trains can significantly improve the dynamic behavior of machine tools. The article presents components that are based on the differential setup. Due to the nonexistence of calculation algorithms the impact of piezoceramic actuators in machinery will usually not be achieved. By using an electro-mechanical analogy for mapping the properties of the piezo actuator, it is possible to establish calculation models that provide both the essential electrical current of the amplifier and the detailed stiffness analysis. To illustrate this the authors explain the actuator-sensor-unit for compensating axial vibration in a feed axis with a ball screw drive. In conclusion, the actuator-sensor-unit has been successfully analyzed and the calculation verified. A further example of an actuator-sensor-unit for a parallel kinematic machine is explained.

Keywords: *machine tool, adaptronic component, piezo actuator*

Introduction

Due to the inconsistency of the criteria, it is very hard to increase the stiffness of a production machine by using classical methods without changing the dynamic properties. Whereas a highly dynamic machine structure has to be light, an increase in the stiffness of an optimized structure with respects to the power flow can only be achieved by adding material. Although lightweight materials improve the system behavior, the effects are in fact limited by the finite value of the E-modulus and the density of the structure material.

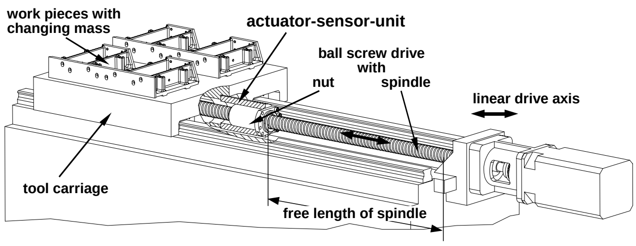

Fig. 1: Typical feed axis with ball screw drive and location of the new adaptronic component

Especially in drive trains, as an important part of machine tool structure, all power transmission parts must have a small mass or moment of inertia in order to ensure the high dynamic. A typical structure element is the controlled feed axis equipped with ball screw drive systems. There are two main demands that have to be met by the structure design. To shorten process time spindle-nut-systems have to move the tool carriage or another machine part with a high velocity and acceleration between two machining operations. To meet this requirement, all transmission parts are designed with a small mass or moment of inertia. However this characteristic conflicts with the second demand: the high precision movement during the machining operation that requires a high stiffness. Depending on the acceleration during positioning operations, respectively the movable work piece mass and the cutting force, the gain-frequency characteristic can be totally different. In spite of the optimizations of the mechanical transmission parts and the positioning control loop, the axial flexibility of the spindle length leads to axial vibration. Continuously growing demands on both precision and machine output intensify the conflict. For above-mentioned reasons, a new active component was introduced in the feed axis, which is placed between the ball screw nut and the tool carriage (fig. 1).

The Differential Actuator Setup

Equipped with a differential setup of piezo actuators, the actuator-senor-unit acts as additional drive in direction of the main drive. The piezo stack actuators with their wide frequency range and high-resolution stroke have a suitable drive characteristic concerning the expected disturbance level. The fig. 2 shows the basic electromechanical principle of the transformation of the actuator stroke to a movement of the feed axis. Taking an offset voltage U_{off} (the half of the nominal voltage) as starting point, the actuators are inversely activated. This leads to a reversal stroke of the tool carriage.

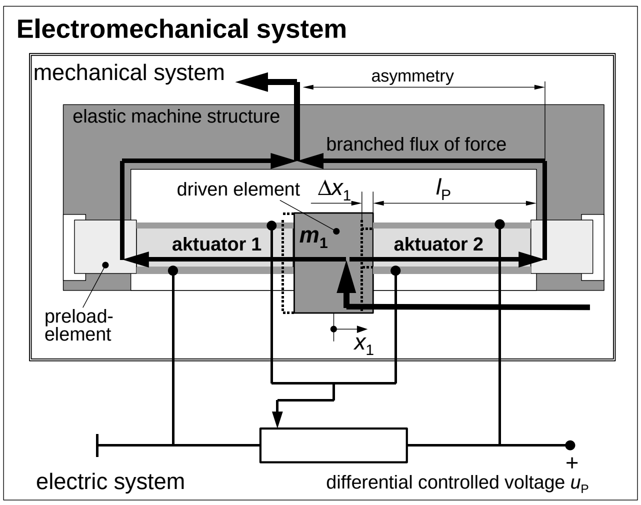

Fig. 2: Electromechanical scheme of the differential setup of piezo actuators

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Piezo-based components need a certain preload to protect the piezo actuators from 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 is applied. The flux of force is closed inside the component [1].

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 with horizontal movements as well as 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.

Aspects of Sizing Active Components

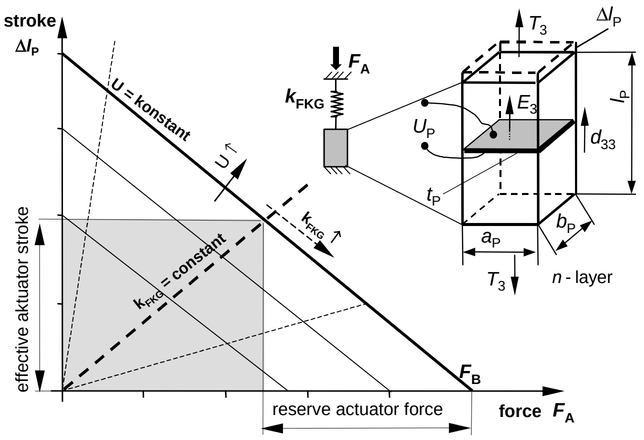

Fig. 3: Geometry of a d_{33}-multi layer actuator and characteristic curve of the dependence between the stroke and the generated force

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, 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. But in both cases, the design process could not start from a passive, but optimized component since this structure is too stiff and a considerable part of the piezo-generated force is needed

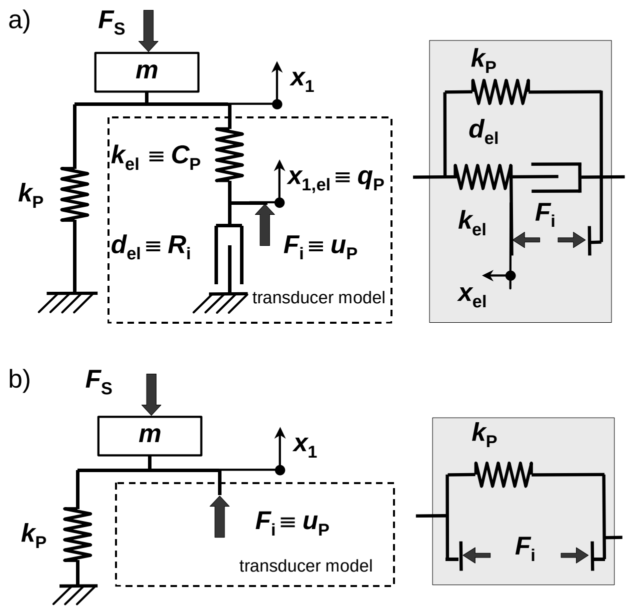

Fig. 4: Electromechanical transducer model of a piezo stack actuator as converter (a) and its reduction for quasi-static calculations (b)

to deform the passive metal structure. The force, which remains for compensating process forces, would be too small. With a purposeful designed yielding structure, supplemented by an actuator that lies in the flux of forces, an active structure evolves, which can adopt itself to ambient conditions. The flux of forces within the active component is going to be conducted across piezo-actuators.

The parallel mechanical structure serves as solid state joint with the stiffness k_{FKG} and assures the transfer of forces and torques, which do not lie in the effective direction of the actuator. Only the control of the actuator stroke for leveling out the structure deformation, which belongs inevitably to the active system, guarantees a stiffness that amounts a multiple of the passive stiffness of the original system. The ideal external behavior of an assembly can be accomplished by active compensation with integrated distributed actuators. In a first step, the actuators are placed in a separately concepted component that can be assembled into different exisiting machine structures.

Modeling of Piezo-based Components in a Differential Setup

Piezo ceramic actuators are dielectric material, which generate a charge when an external mechanical force is applied (sensor effect). In contrast, the application of an electric field leads to a deformation (actuator effect). Both effects are grounded on the atomic dipoles structure and are concurrently effective [2], [3]. This coherence is described with electromechanical state equation on condition that material is orthotropic. The equations couple the electric behavior of a capacitor with the behavior of an elastic body with space coordinates. The complete tensor description is commonly simplified. Often the equations of state of the strain S and the electric flux density D are used. The quantity of converting is expressed by the piezoelectric constant $d_{n,j}$ with the unit [m/V] respectively [C/N]. There are three possible directions for applying an electric field respectively for feeding the voltage u_P, whereas the direction 3, thus E_3, is mainly used. This electric field E_3 causes an elongation in the same direction (d_{33}-mode).

Considering all conditions and the geometry of multilayer stack actuators according to fig. 3, the actuator stroke can be described with the following simplified equation:

$$\Delta l_P = s_{33}^E \cdot \frac{l_P}{A_P} F_B + d_{33} \cdot n_P \cdot u_P \tag{1}$$

(with s_{33}^E-elastic compliance coefficient). The equation couples the stroke and the generated force. In addition, the diagram of fig. 3 shows the dependence of both, stroke and generated force, form the external prestressing spring k_{FKG}. 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.

Electromechanical Analogy for Piezo Actuators

The electromechanical state equations 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 ones 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.

Considering the isomorphology, the working frequency range up to approximately 200 Hz and the usage of a voltage controlled amplifier the analogy shown in fig. 4 can be applied. Taking into account the movable mass of the whole, the mass of the actuator can be neglected. According to this theory of an electromechanical converter, the electric capacity C_P is equivalent to the compliance without mass, in the following called electric stiffness k_{el}, the inner resistance R_i to the damping coefficient d_{el} and the voltage u_P to the inner piezo force F_i. Taking into account the state equation, the quantities are coupled by the coefficient [4]:

$$\alpha_{33} = \frac{d_{33} \cdot A_P}{s_{33}^E \cdot t_P} \tag{2}$$

with the relations for voltage to force:

$$F_i = \alpha_{33} \cdot u_P \tag{3}$$

capacity to electric stiffness:

$$k_{el} = \alpha_{33}^{\ 2} \frac{1}{C_P} \tag{4}$$

resistance to damping coefficient:

$$d_{el} = \alpha_{33}^{\ 2} \cdot R_i \tag{5}$$

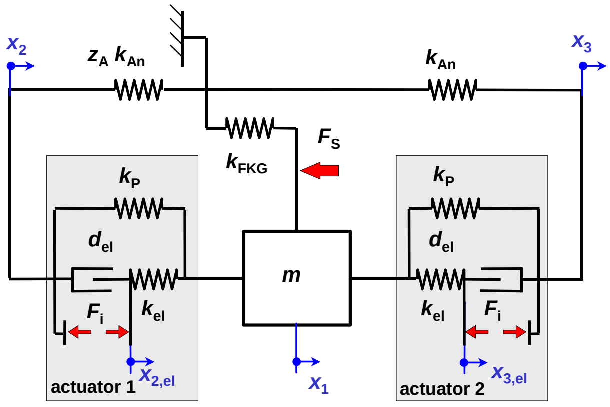

Fig. 5: General electromechanical model of the differential setup of piezo actuators

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

$$i_P = \dot{q}_P = \alpha_{33} \cdot \dot{x}_{el} \tag{6}$$

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 selectable, the equations (3)...(6) are 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 mechanical 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 have to be carefully rated concerning the scope.

Quasi-static Calculation of Characteristic Component Values

Considering the characteristic component values blocking force F_B and free stroke Δl_P, the piezo stack actuators of different suppliers are comparable since these values are measured properties of the actuators and not only of the material**.** For transferring this characteristic to the active component, the compliance of the mechanical structure has to be included. The mechanical model in fig. 5 generalizes the effects by introducing design factors as relation between the mechanical stiffness of the metal elements and short-circuited actuator. Concerning the solid state joint with the stiffness k_{FKG}, it can be written:

$$z_{FKG} = \frac{k_{FKG}}{k_P} \tag{7}$$

and for the serial stiffness of the adjacent elements:

$$z_P = \frac{k_{An}}{k_P} \tag{8}$$

In many applications, an asymmetric component design can be avoided. There is also the ratio z_A between the different values of the serial stiffness. Furthermore, the possibility to size the component values free stroke (twisting) and transmitted force by varying the numbers of serial ($n_{Sch} > 1$) and parallel ($n_A > 1$) actuator groups is taken into consideration. But for a better overview, these coefficients and the transformation to the torsions are not shown in the following equation. The free stroke of the differential setup follows from the above:

$$x_1 = \frac{F_S \cdot (1+z_P) \cdot (1+z_A z_P) + z_P\left[F_{i2} \cdot (1+z_A z_P) + F_{i1} \cdot z_A (1+z_P)\right]}{k_P\left[z_{FKG}(1+z_P) \cdot (1+z_A z_P) + z_P(1+z_A+2z_A z_P)\right]} \tag{9}$$

The different coefficients of the inner force terms $F_{i,n}$ shows 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_{off}. The real blocking force results from the average of the direction-dependent values:

$$F_{B,eff} = \pm \frac{z_P(1+z_A+2z_P z_A)}{(1+z_P)(1+z_P z_A)} \cdot \frac{F_{i,max}}{2} \tag{10}$$

The efficiency of the force and stroke transmission is primarily characterized by the serial stiffness k_{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 Δx_1, from this factor is shown in the diagram of fig. 6. Consequently, a well-designed power transmission consists of elements in the flux of force that are as stiff as possible.

Considering all elements of the component, the stiffness is:

$$k_{ASE} = \frac{2n_A}{n_{Sch}} \cdot \frac{k_P \cdot k_{An}}{k_P + k_{An}} + \frac{k_{St}}{n_{Sch}} + k_H \tag{11}$$

It is also possible to define a ratio between the inner piezo force F_i and the outside disturbance force F_S from the equation (10). The optimal ratio $z_{S,opt}$ indicates the maximum outside disturbance force that can be compensated by the component:

$$z_{S,opt} = \frac{F_i}{F_{S,opt}} = \frac{2\,(1+z_P)(1+z_P z_A)}{z_P\,(1+z_A+2\,z_P z_A)} \tag{12}$$

The static dimensioning of the component also encloses the calculation of the necessary preload for achieving a backlash-free system.

The main influences are the dynamic loads from disturbance and mass forces as well as the difference in temperature, which leads, according to the negative coefficient of expansion, either to a release of the preload or to an overload of the actuators. The calculation is similar to the standardized dimensioning of a prestressed bolt joint. The upper limit for the total load of the

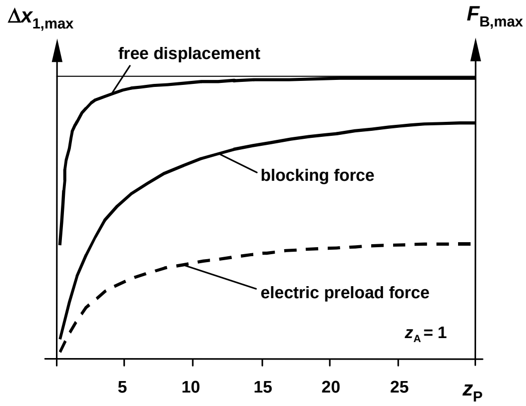

Fig. 6: Different dependence of blocking force F_B and free stroke Δl_P from the compliant serial piezo support (design factor z_P)

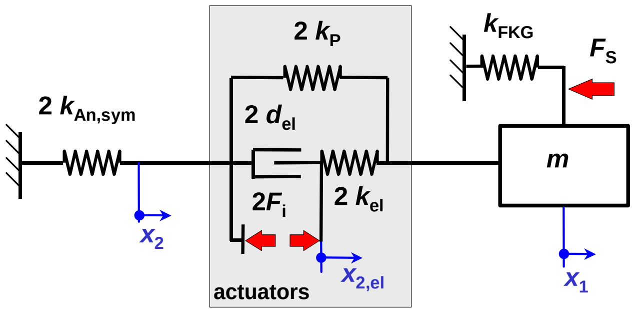

Fig. 7: Simplified model for dynamic calculation

actuator is the depolarizing strength of the piezoceramic material (100...150 MPa).

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 an 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 $x_{2,el}$ can be converted to the electric current and the phase by using the electromechanical analogy (eq. 6). For calculations in the time domain, it is necessary to use the state-space model established in control engineering. The fig. 7 shows the simplification from the whole component model to a reduced model, which can be used in overriding machine simulations.

Systematic Design of the Actuator-Sensor-Unit for Reducing Axial Vibration

Designing is an iterative process of arranging and calculating part dimensions, in which the engineer intuitively follows his ideas. At a certain state of the rough design, however, a method has to be

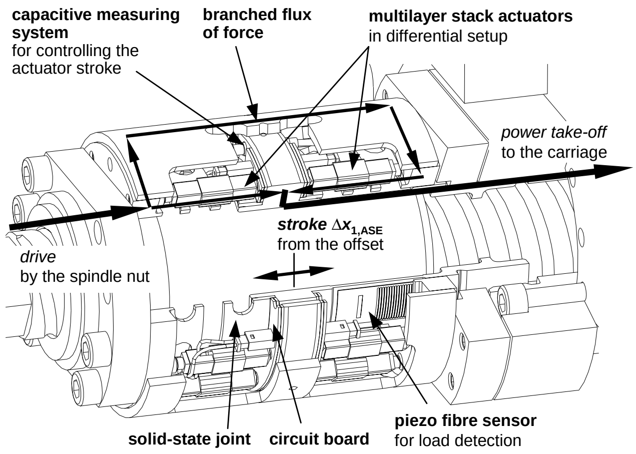

Fig. 8: Detailed design concept of the actuator-sensor-unit

applied to handle all the different criteria of the design. For this purpose, the V-model of the macro cycle according to the VDI guideline 2206 "Design methodology for mechatronic systems" can be used [5]. After the first laboratory specimen of piezo-based components, the design process was then split into sub-functions. The main issues in this state were the mechanical interfaces to the adjacent components and the improved power transmission as mentioned above. The axial actuator-sensor-unit was created for two sizes of ball-screw drives. In the second model, designed for the size 50, some more geometric constraints existed, because this component was already intended for tests in an industrial feed axis test bench.

Above all, the component design depends on the amount of piezo actuators, which are necessary for reducing the vibration, as well as the length, respectively the stroke of these actuators. Beside the possibilities of simulations tools, the simultaneous engineering also requires rough calculations as a starting point. Especially with the actuator-sensor-unit, the number of actuators for the first draft was derived from the inertia force, which appears while the tool carriage is vibrating in its natural frequency. The excitations of the vibration can be e.g. the acceleration force during a positioning movement. In the worst case, the piezo actuators have to generate this force to avoid vibrations.

	actuator type	PI Ceramic: 7x7x36 mm³		
	nominal voltage amplitude	$u_{P,p}$	[V]	50
	number of parallel actuators	n_A		8
	number of serial differential setup	n_{Sch}		1
	component stiffness (passive)	k_{ASE}	[N/µm]	753
design factors	asymmetry	z_A		6.2
design factors	serial stiffness of actuators	z_P		1.56
design factors	stiffness of solid-state joint	z_{FKG}		0.31
design factors	serial stiffness of actuators of the reduced (symmetric) model	$z_{P,asym}$		3.14
design factors	relation inner force to disturbance	$z_{S,opt}$		1.31
	max. free displacement	$\Delta x_{1,ASE}$	[µm]	±14.2
	blocking force	$F_{B,ASE}$	[kN]	12.8 / 8.62

Tab. 1: Characteristic values of the actuator-sensor-unit for reducing axial vibrations for a ball srcew drive of size 50

Concerning the stroke of the differential setup, it has to be considered that there is an inner and an outer closed positioning control loop. The function of the inner control loop of the decentralized actuator-sensor-unit is only a fine positioning, whereas the outer control provides the overall (a kind of rough) positioning. Therefore, the actuators do not need to compensate the compliance of the elastic spindle each time. As the piezo actuator stroke depends on the generated force (fig. 3) as well as on the serial stiffness, only rough estimations are possible which have to be evaluated in a subsequent simulation of the whole system.

The advantage of the above-mentioned procedure is that first tests can be carried out in a short time and provide experimental data. In this way, the second design cycle already leads to the component of the actuator-sensor-unit shown in fig. 8.

Quasi-static Verification of the Component

The first test was the verification of the free stroke x_1 of the unit. For the following quasi-static measurements, the component was blocked by a rigid frame construction. Despite the rigid frame, the total stiffness was very small since the compliant load cell was included in the flux of force. The stroke – blocking force diagram is shown in fig. 9. During these tests, the actuator-sensor-unit was acting against parts with different compliance. The end points of the measured curve of compliance were extrapolated by a straight line. The intersection points of this line with the diagram axis correspond with the blocking force F_B and free stroke x_1. The measured value of the free stroke meets the calculated result. In contrast to earlier tests, the differences concerning the blocking force of only 11 % indicate a correctly calculated serial stiffness of the actuators. The blocking force was not verified in both directions. The reason for that was the high effort that had required a stiff construction for measuring the generated tension.

Verification in a Feed Axis

For this test, the actuator-sensor-unit was mounted in an industrial test bench of a feed axis. The aim was to analyze the dynamic behavior of the actuator-sensor-unit while moving a mass of 2100 kg. The feed motor was held at a certain position by the control loop since the mechanical blocking with a brake was not possible. Because of that, the measured results represent the behavior of the whole electromechanical feed axis system and not of the single component analysis.

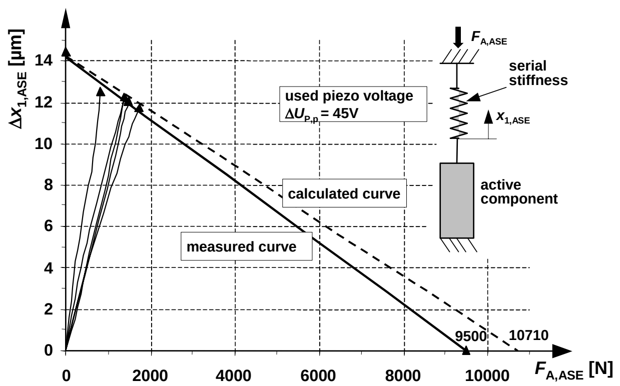

Fig. 9: Measured stroke-blocking-force diagram of the actuator-sensor-unit

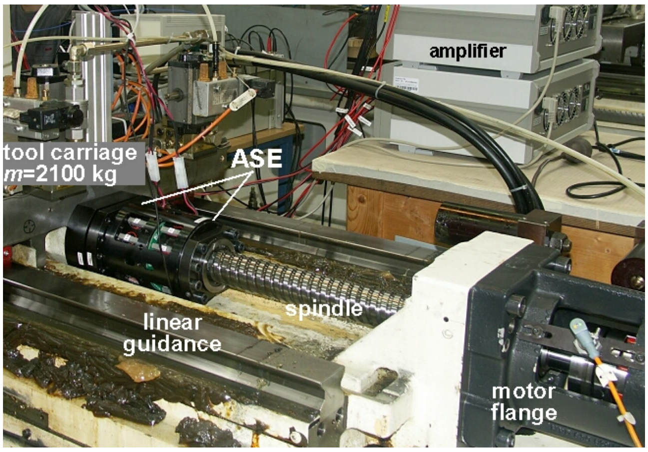

Fig. 10: Test bench with mounted actuator-sensor-unit (shown without the additional measuring setup)

According to fig. 10, the integrated capacitive measuring system of the component was completed by two additional, separately mounted laser measuring systems. The piezo actuators were fed with sinusoidal voltage up to a frequency of 200 Hz. Figure 11 shows the amplitude response for a piezo voltage of $u_{P,p} = 20$ V. The curves have three important resonance peaks. The two upper peaks are caused by the compliant support parts of the measuring devices, but the first peak meets exactly the eigenfrequency of the positioning control loop of the feed axis. The most important result is that the actuator-sensor-unit generates the displacements $x_{1,ASU}$ and $\Delta l_{P,2}$ up to 200 Hz without any resonance. The measuring by the capacitive pick-up of $\Delta l_{P,2}$ has the best quality. The slight decrease of this stroke is caused by the also decreasing voltage of the applied amplifier with increasing frequency. That means both masses in the current test stand setup, the motor inertia and the carriage mass represent a two mass oscillation system, in which the actuator-sensor-unit acts as a kind of excitation (shaker). At certain frequencies, the carriage mass of 2100 kg react as a seismic mass, at which the active component can support. At this frequency range, the motor control loop is not able to hold the position.The difference between the piezo stroke and the component stroke refers to the asymmetric structure.

The conclusion was drawn that the electromechanical design of the active component operates in the system environment of the feed axis in the designated way. The verified component is a prerequisite for further tests of the decentralized control loop.

The second prerequisite for a well-designed adaptronic system is a control that is operating autonomously from the superior machine tool control. This issue will be the object of the next development cycle. The usage of the collocation of actuators and sensors will be analyzed, which makes it possible to apply the so-called low authority control algorithms. With this attempt at a solution, the stability of the control loop will be significantly increased. The initial operation can start without any time-consuming system identification.

Further Example of an Actuator-Sensor-Unit

The second example of an active component is intended for a parallel kinematic machine (PKM) [6], [7]. The struts of an x,y,z-Tripod are loaded with torsion. Since the integration of the actuator-sensor-unit was considered from the beginning of the design process of the 3POD, this adaptronic component for

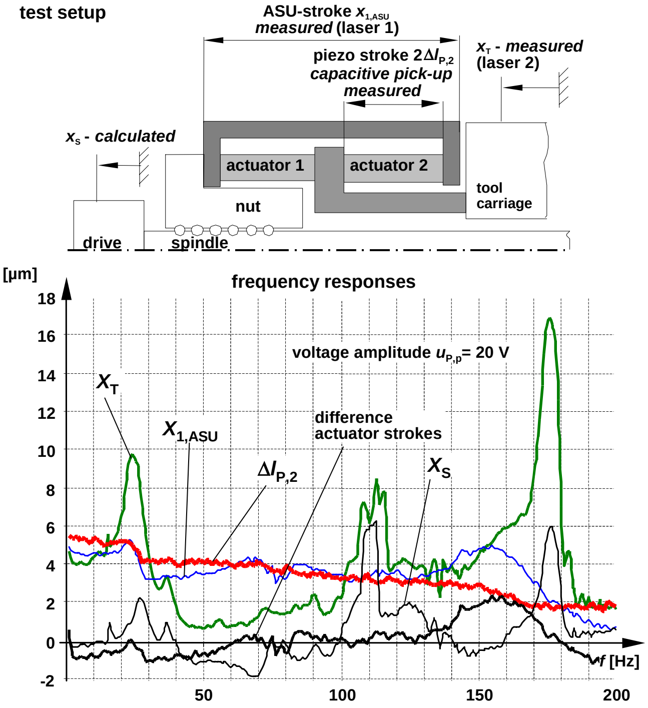

Fig. 11: Measured and calculated frequency response of different coordinates of the feed axis test bench

torsional compensation belongs to the group of components that makes new systems feasible [8], [9]. For this purpose, it is necessary to transform the actuator stroke to a rotating movement. In contrast to the common feed axis, the directions of the main drive and the actuator-sensor-unit do not correspond with each other. Although it is important to note that PKM are new systems. The integration of active components from the beginning of the design process makes solutions possible that are not possible without them.

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