*Journal of Machine Engineering, Vol. 13, No. 3, 2013*

*machine tools, milling,*
*grinding, boring, chatter,*
*active damping system*

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# AVOIDING CHATTER BY MEANS OF ACTIVE DAMPING SYSTEMS FOR MACHINE TOOLS

Chatter in machining processes is strong dependent on the dynamic compliance behaviour of the machine tool and workpiece. The critical cutting depth where chatter occurs is in inverse proportion to the absolute value of the negative real part of the complex dynamic compliance response function of the machine tool. Therefore, while designing a machine tool, its dynamic behaviour should be investigated and optimized, e.g. by using the finite element method. If further design optimization are not possible, active damping systems might help avoiding chatter of the machine tool. Active damping systems can compensate the dynamic displacements between tool centre point of the machine tool and workpiece by applying dynamically correlated external energy (e.g. compensation forces) onto the machine structure. Next to an explanation of the general idea and systemization of active damping systems, this paper gives examples of recent research activities in the field of active damping systems for machine tools of the Laboratory for Machine Tools and Production Engineering (WZL) of Aachen University, Germany. The main challenges while designing active damping systems for machine tools are carved out.

## 1. INTRODUCTION

Nowadays for simulation purposes, the interaction between machine and process (Process-Machine-Interaction, PMI) is understood as a closed loop. The dynamic behaviour of the machine tool is represented by the forward path of the loop. The feedback represents the influence of the tool centre point (TCP) displacements on process forces, which again affects the machine tool structure. In [1], Altintas and Weck give a decent overview of different chatter stability models for turning, boring, drilling, milling and grinding. Already in the 1950s, Tlusty and Polacek [3] as well as Tobias and Fishwick [4] developed independently from each other the classical chatter theory, which describes the relation between the cutting depth, the dynamic compliance behaviour of the machine tool and the cutting coefficients. In [5] Weck and Brecher state significant stability relevant parameters, e.g. position and mass distribution of machine components, non-linearities, preload and clamping of axes, cutting edge geometry, feed and cutting rate.

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A plurality of approaches to increase the stability limit of the PMI are described in [1], [2],[5] and [6]. Depending on the stability relevant parameter, these approaches can be categorised in process optimization, mechanical design optimization and dynamic auxiliary systems. In case of cutting with geometrically defined cutting edges (e.g. turning, milling), revolution speed dependent chatter stability lobes are commonly used to characterize the stability behaviour of the PMI [5]. The analytical calculation of stability lobes was initially introduced by Tobias and Fishwick [4] in 1958. Dependent on the characteristic of the stability lobes, the cutting process parameters, especially the cutting depth and spindle speed can be chosen to maximize the material removal rate without instabilities [5]. Regarding PMI of machining processes with geometrically undetermined cutting edges (e.g. grinding), Younis [7], Klotz [8], Folkerts [9] and Hennes [10] introduced the todays relevant analytical correlations and equations. Folkerts developed the so-called limiting phase criterion, which can be used to identify critical frequencies and enables specific optimization of the grinding process. Besides the machine tool behaviour, workpiece speed, grinding wheel diameter and feed per revolution are of great importance for stability of grinding processes [9]. Generally, process optimization can be carried out on an existing machine tool without big expense. If process optimization does not have the desired effect or if the machine tool is still in design phase, mechanical design optimization might be applied. Costs of optimization of mechanical design rise with on-going state of the machine tool development. Therefore, it is the main goal to realise an optimal dynamic structure, which combines high stiffness together with low masses, in the early phase of the machine tool development. For this purpose numerical approaches, like the finite element method (FEM) or multi body simulations (MBS) are used to build virtual prototypes, which can be investigated and optimized cost efficiently [5],[10],[12],[13].

## 2. DYNAMIC AUXILIARY SYSTEMS FOR MACHINE TOOLS

Dynamic auxiliary systems can either be planned to be added within the machine tool design phase or if the machine is in operation and show instabilities. The cost of auxiliary systems depend on the type of approach and therefore are related to the complexity of the system. One can find passive and active approaches used in the field of machine tools.

In comparison to active systems, passive systems have a dissipative nature and in general a simple design. Four basic setups for passive systems are shown in Fig. 1, left. While the impact and the Lanchester damper are rarely used in machine tools, the vibration absorber and the tuned mass damper (TMD) are more common. Den Hartog [14] and Brock [15] developed an analytical approach for the calculation of the optimal TMD properties: mass $m$, stiffness $k_2$ and damping $c_2$. Main intention of this calculation is a reduction of amplitude amplification at the highest resonance of the to be damped system (called original system) with mass $M$, stiffness $k_1$ and damping $c_1$. When the stabilityenhancement of PMI rather than the reduction of amplification is of interest, the maximum absolute value of the negative real part of the dynamic compliance behaviour of the machine tool has to be minimized. Analytical approaches for the calculation of optimal TMD properties to increase stability are given in [16] and [17]. Passive systems are tuned to only one specific resonance frequency. If the resonance frequency changes (e.g. due to the influence of the work piece weight on the machine table), the passive system might loose its effect. In consequence, semi-active systems have been developed Fig. 1, centre, [18]. Semi-active systems have, like passive systems, a dissipative behaviour, to take energy out of the original system [19]. The adjustable coupling of the auxiliary mass to the original system enable a controlled tuning of the semi-active system by changing the stiffness and/or damping of the connection component. Therefore, the semi-active system can adapt to changes of the resonance frequency of the original system.

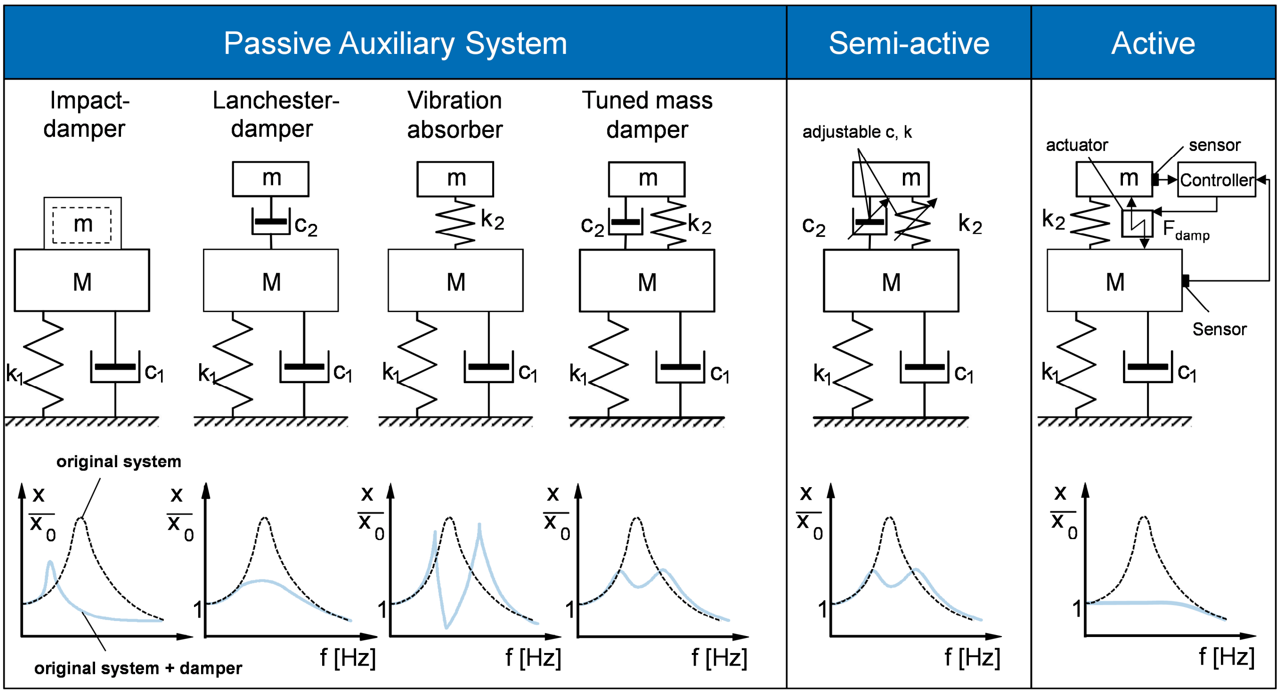


Fig. 1. Passive, semi-active and active auxiliary systems for machine tools [5]

Active systems, as shown on the right hand side of Fig. 1, compensate the dynamic displacements between the TCP of the machine tool and the workpiece, by applying dynamically correlated external energy (e.g. damping forces $F_{damp}$) onto the machine structure [19]. In general, an active system consists of mechanical and an electrical components and therefore are understood as mechatronic system. While mechanical components encompasses the structure of the active system and its coupling to the machine structure, the electric component comprises the controller, amplifiers for sensors and actuators and filter. Sensors and actuators serve as interface between mechanic and electric components. If sensors integrated into actuator or structure are used for autonomous improvement of mechatronic structures, those systems are called adaptronic. Example for adaptronic structures are the so-called smart materials, e.g. piezoceramics or memory shape alloys [6],[23].

Main advantage of active systems over passive and semi-active approaches is a higher power density [5]. This advantage leads to the main disadvantage of active systems. Due to the closed loop control of the active system, they can become unstable. Hence, applying the optimal compensation strategy needs a bigger effort in comparison to e.g. semi-active systems. That leads to three major challenges for developing active damping systems: the mechanical design, the control strategy and suitable actuators and sensors [22]. The mechanical design of the active system needs to be robust, reliable, suitable for production and assembly and has to fit to the given design space. Next to stability, the controller design has to be robust against environmental changes and provide the necessary dynamic. Finally, actuators and sensors have to meet the requirements regarding forces, travel ranges, energy supply and bandwidth. The complexity of active systems leads to the need for methods to save effort and costs for active damping system development.

Recently, Manoharan formulated a method for active system development for increasing productivity of milling processes by use of virtual prototypes which comprise the machine tool and the active system with its actuators, sensors and controller [23]. Fig. 2 summarizes the basic steps proposed by Manoharan, which easily can be adapted to different active systems. In the beginning, a metrological investigation of the dynamic behaviour of the machine tool is necessary, to identify the weak spots of the machine. Here, the direct frequency response function (FRF) of the dynamic compliance behaviour give an insight into critical resonance peaks where negative real parts occur. A modal analysis visualizes the vibration patterns to identify the weak spots. Finally, cutting tests can be used as benchmark to identify the maximum cutting depth at given process parameters. If the weak spot of the machine tool is successfully identified, a basic strategy for compensation of the weak spot can be developed. To verify the feasibility of this strategy a rough analytical estimation of necessary forces, displacements, stiffness and bandwidth of the active system should be conducted.

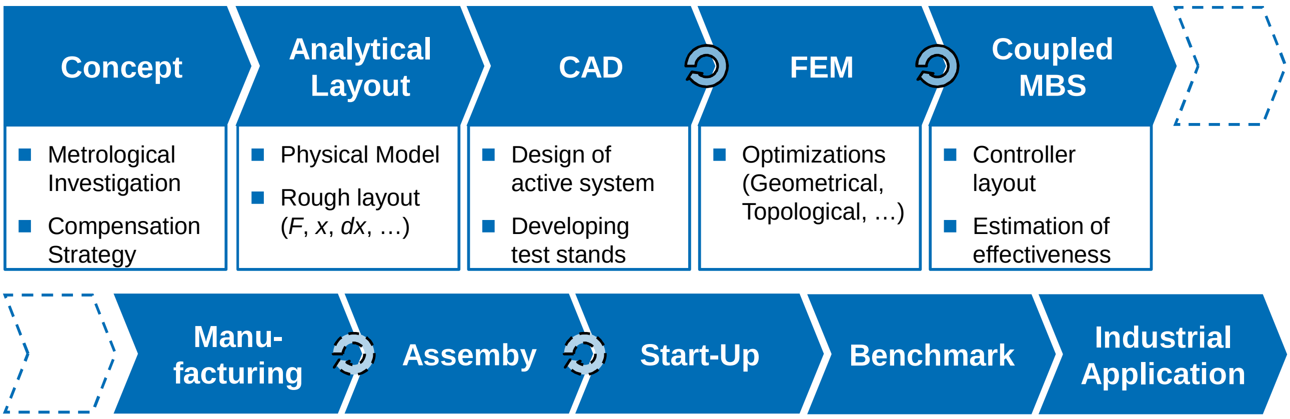


Fig. 2. Method for designing active systems for machine tools

The following design phase has several back-loop steps until the optimal setting and design is found. FEM simulations and/or MBS are suitable tools to build a virtual prototype of the machine tool and the interaction with the active system. While flexible bodies can be implemented into the MBS environment, controller, actuators and sensors are coupled by a Computer Aided Control Engineering (CACE) interface within the MBS. The resulting virtual test stand can be used for optimization of geometry or topology of the active system mechanical or the controller design [11]. The final design of the virtual prototype will be manufactured, assembled and started-up. Further optimization of design and control are kept within small limits and therefore are not cost intensive. After integration of the developed active system into the original structure, the final benchmark, e.g. by comparison of the FRF or cutting test with and without the active system, gives an insight into the potential of the developed active system. Depending on the results of the benchmark an industrial application can be pursued.

## 3. STATE OF THE ART OF ACTIVE SYSTEMS FOR MACHINE TOOLS

In the past years, several active systems have been developed and investigated. Neugebauer, Denkena and Wegener review in [6] developments in mechatronic systems for metal cutting and forming machine tools. 2011 Hesselbach published several results of the German Research Foundation (DFG) in the field of smart structures for machine tools [20]. Further active approaches are presented in [1],[5] and [19]. For active systems in grinding processes, [21] gives a brief overview of the current state of the art.

One can distinguish active systems which are added to the machine tool structure and systems, which are integrated into the structure of the machine tool. As an example, Manoharan developed in [23] two active systems, where one is structure integrated and the other is mounted onto a machine table of a milling machine. In case of the structure integrated active system, the so-called structure integrated compensation modules (SICM), piezoelectric actuators are positioned in the indirect flux of force to compensate the bending vibration of large overhanging structure components, e.g. z-sliders of portal milling machines Fig. 3a). The application of additional compensation forces counteracts the occurring TCP-vibrations and therefore damp the resonance magnification of the bending mode. After an analytical rough layout, detailed FE-Simulations were carried out to optimize the coupling of active system to machine tool as well as the eigenfrequency of mechanical components of the active system. The controller was optimized by the use of a flexible MBS of a portal milling machine, where controller, actuators and sensors where modelled in an MATLAB Simulink® environment and coupled through the CACE interface. Manoharan developed and manufactured a demonstrator for verification and further investigation of this compensation strategy. Two low voltage piezoelectric actuators induce the compensation forces, while strain gauges, glued to the piezoceramic, measure the displacement of the actuators. This displacement signal is used as control variable. The double integration of the signal of the acceleration sensor provides the actual displacement of the TCP of the demonstrator. By using this double integrated signal as set value for the displacement of the actuator, the deflection at the TCP could be reduced. The test stand successfully proved the potential of the SICM for use in portal milling machines. The resonance amplification of the bending mode could be reduced by a factor of ten. In addition to that the negative real part of the complex dynamic compliance behaviour of the test stand could be reduced and therefore the estimated process stability increased. The results of this work are summarized in [24]. In a current research project, founded by the German Research Foundation (DFG), the WZL integrates this structure integrated concept into an existing portal milling machine in cooperation with the company Heinrich Georg GmbH. Main goal of this project is the expansion of the control by an additional axes, for damping the bending oscillation in two horizontal directions. In this case it will be possible to benchmark the developed SICM with the help of cutting tests, which will be performed with and without the active system. Next to the estimated productivity enhancement, an economical benchmark will be possible to holistically observe the machine behaviour.

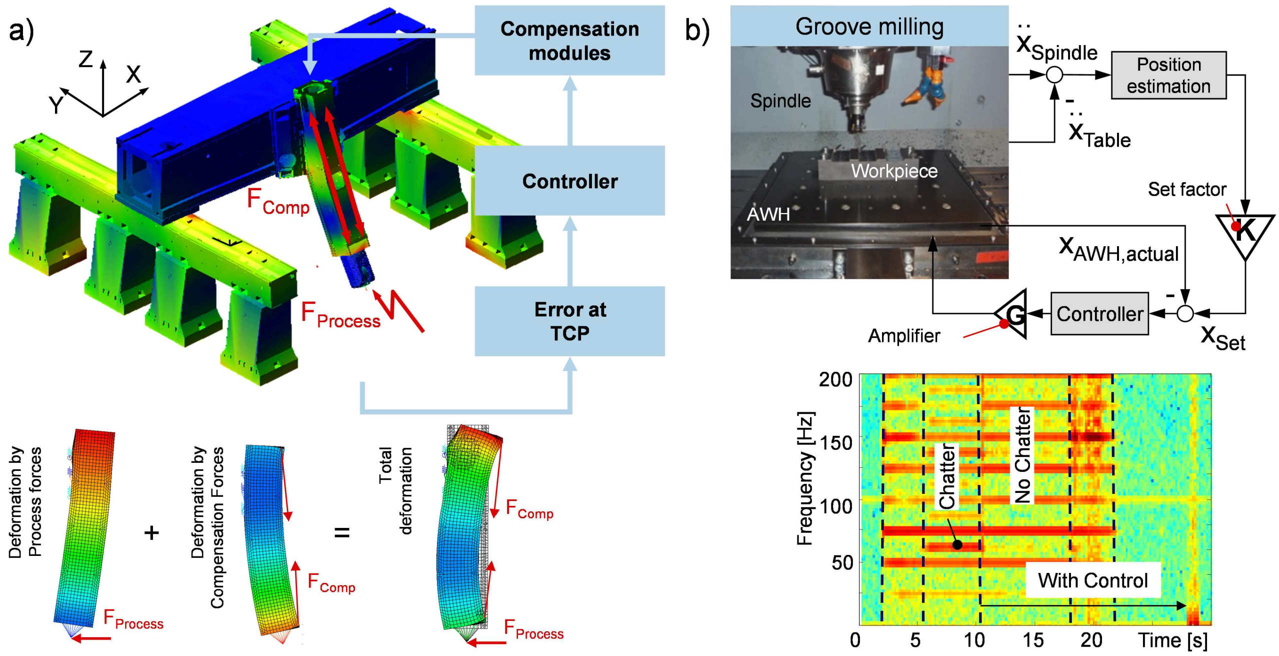


Fig. 3. Structure integrated compensation modules [23] – a), Active workpiece holder [23] – b)

The second active system Manoharan developed in [23] is an active workpiece holder (AWH), which can be mounted to the machine tool as an auxiliary active system Fig. 3b). This AWH comprises two piezoelectric actuators, who drive two axes arranged in series. Those translational redundant axes at the workpiece side enable a highly dynamic positioning of the workpiece. Undesired displacements between TCP and workpiece can be compensated. Manoharan optimized the design of the AWH by the use of a FE-model. Monolithic flexures for guiding the axes were at first analytically laid out and finally optimized with the FE-model to guaranty a high translational flexibility in the guided direction and high stiffness in the other five degrees of freedom. The total shape of the AWH has been topologically optimized, to increase the eigenfrequencies. Besides the two piezoelectric actuators the AWH uses one length gauge for each axis to measure the actual position of the AWH-axis. To get the relative deflection between TCP and machine table (basically the positioning error), the position of the TCP and the machine table is estimated out of the acceleration signal from position estimator, which uses several low- and high-pass-filter to perform a double integration of the acceleration signal in the concerning bandwidth of the AWH. The calculated positioning error is used as set signal for the actual deflection of the AWH-axis, which is controlled by a simple PI-controller. Manoharan mounted the assembled AWH onto a machine table of a milling machine. His comparison between the FRF with and without AWH yields a complete compensation of the critical resonance amplification. Further on, a negative real part could be avoided. In cutting test Manoharan showed, that the critical cutting depth with activated AWH could be increased by 50 % depending on the revolution speed [25].

Piezoelectric actuators are broadly used in active systems for machine tools. Isermann observed in [26], that piezoelectric actuators have on the one side a high power density and a excellent dynamic behaviour. Besides that, piezoceramics offer a high stiffness and therefore do not induce an additional weak spot in the machine tool structure, if integrated e.g. in the direct flux of force. On the other side, piezoelectric actuators lack in positioning range. The commonly used piezo stack actuator provides only 2 ‰ travel range of the total stack length. Moreover, piezoceramics are sensitive against shear forces, which has to be considered while designing active systems. In recent researches, Brecher, Bäumler and Brockmann have shown, that especially in large scale machine tools, new hydraulic actuator concepts are a promising alternative to the piezoelectric actuator. In [27] Brecher et al. presented an hydraulic actuator, controlled by a high dynamic servo valve, which was designed especially for the SICM, presented above. This new actuator design overcomes former disadvantages of hydraulic actuators, e.g. leak oil, nonlinearities and low dynamics, by using topological optimized steel membranes as pressure chamber sealing and piston guidance instead of using gliding gaskets Fig. 4a. Furthermore an analytic model of the actuators behaviour was introduced, to be included into a coupled MBS of the earlier mentioned test stand for the SICM. With this model, it was shown, that the hydraulic actuator might at least have the same damping effectiveness as that of the piezoelectric actuator. In fact it is assumed, that because of the longer travel range of the hydraulic actuator in comparison to the piezoelectric actuator, the damping effect will increase further. After manufacturing and assembling of the actuator, first test were used to verify the actuator model. The static and dynamic behaviour of the physical actuator very well correlates with the behaviour of the actuator model.

Another electrohydraulic active system which can be applied directly to the oscillating machine structure was presented by Schulz in [28]. The hydraulic actuator consists of a double action hydraulic cylinder moving the auxiliary mass. A conventional servo valve controls the oil flow rate. The hydraulic capacity has been minimized in order to increase the actuator dynamics. For further investigations of the actuator behaviour the exterior diameter of the piston can be varied Fig. 4b). In his research, Schulz compared different types of actuators regarding their deployment in an active system using inertial forces. He concludes, that servohydraulic drives offer bandwidth, large maximum inertial forces and a compact design space, necessary for machine tool structures. If using servohydraulic drives for active systems, in general the dynamic behaviour of the servo valve is limiting the bandwidth of the system to approximately 200 to 300 Hz, depending on the valve. Schulz showed, that with a servo valve with its cut-off-frequency at 300 Hz, the inertial actuator has a positive influence on the systems damping up to 400 Hz. Faster movements of the piston only increase the power consumption without a further positive effect on the systems damping. Schulz mounted his inertial system on two large scale machine tools. Cutting tests have shown, that this actuator raises in both cases the critical cutting depth by a factor of five, depending on the revolution speed.

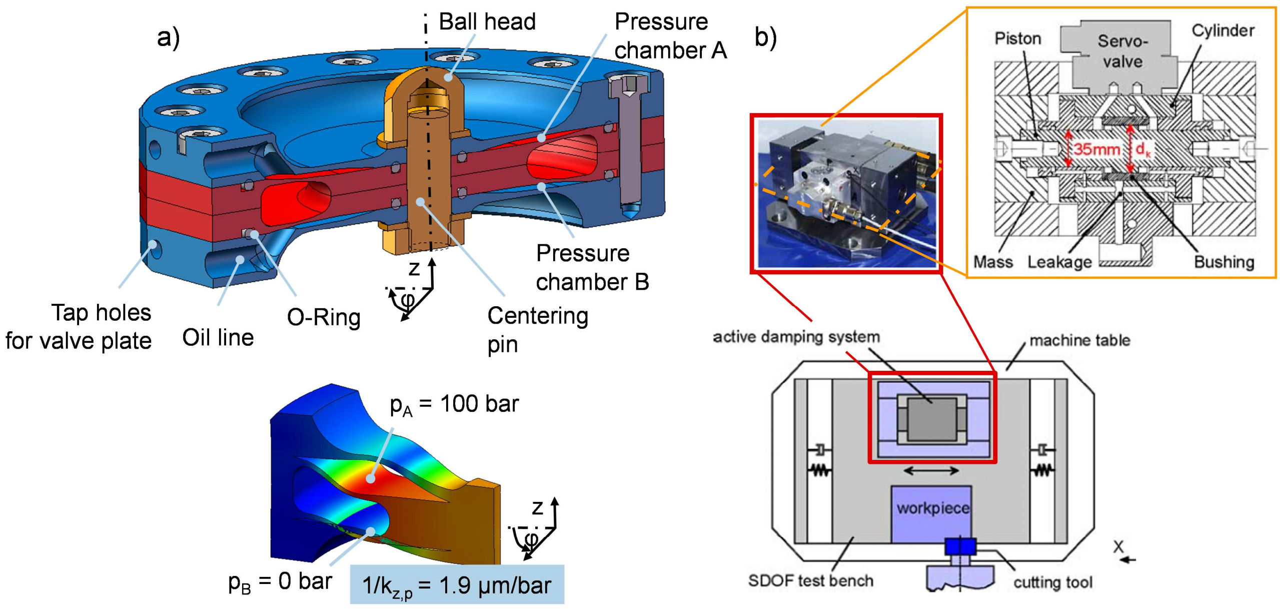


Fig. 4. Membrane based hydraulic actuator [27] – a), Active tuned mass damper [28] – b)

Preumont distinguishes in [29] two different control strategies for disturbance rejection: feedback and feedforward control. If the disturbance of the system can be measured, feedforward control can be an alternative to feedback control. With the knowledge of the disturbance signal a secondary disturbance can be applied by the active system, which cancels the effect of the primary one. For this control strategy, a system model is not necessary. Instead, a filter which adapts to extract the main disturbance frequency and filters the rest of the signal.

The principle of feedback control is based on a comparison of the reference input and the actual output of the system. The error between reference and the actual value is passed into the controller and than applied to the actual system, which reacts due to this input and again generates an actual output. The controller has to be design in such a manner, that the closed loop is stable and meets the necessary properties (e.g. bandwidth, stationary accuracy). This approach involves a low-dimensional mathematic model of the actual system.

As the dynamic behaviour of the machine tool strongly depends on the actual position of the feed axes as well as on the workpieces behaviour, the controller needs to be robust against those changes. That implies the model to be either adaptable or robust as well. In active systems a collocated control systems should be pursued. Collocated systems, are control systems, where the actuator and sensor are connected to the same degree of freedom. [29] shows, that undamped collocated control systems have alternating poles and zeros at the imaginary axis. For lightly damped structures, like machine tools, the poles move slightly more into negative real direction, but remain in alternating succession. This means for the frequency response function a 180° phase lag for each imaginary pole and again a 180° phase lead for each imaginary zero and therefore the phase will always be between 0° and -180°. In the Nyquist plot, this is displayed as a set of nearly circle shape forms (one for each mode) below the real axis. Now, in case of an active control of this system with collocated actuator and sensor, the open loop frequency response has only positive real values regardless the gain of the controller and therefore is stable for many single-input single-output (SISO) control systems. Those control systems are robust.

In [30] Schauerte investigates different control strategies for active systems by means of a mechatronic drilling tool. This structure integrated active system enables a position control of the drilling tool cutting edge as well as a position control of the guiding pads, e.g. for out of round drilling operations, Fig. 5a). For his investigations, Schauerte used a state space model of the piezo actuated drilling tool verified by measurements. Next to simple PI-controller, he investigates a Notchfilter control, an internal model control, a feedforward control and an optimized state space controller with Kalman-Filter. Although the state space controller with Kalman-Filter reached the broadest bandwidth for a position control of the cutting edges, Schauerte advises the application of a Notch-Filter control, which reached nearly the same bandwidth as the state space controller, but is much easier to implement and therefore achieves the best rate between benifit and effort. Besides SISO control strategies, multiple input multiple output (MIMO) control strategies are necessary if there is a variety of control variables.

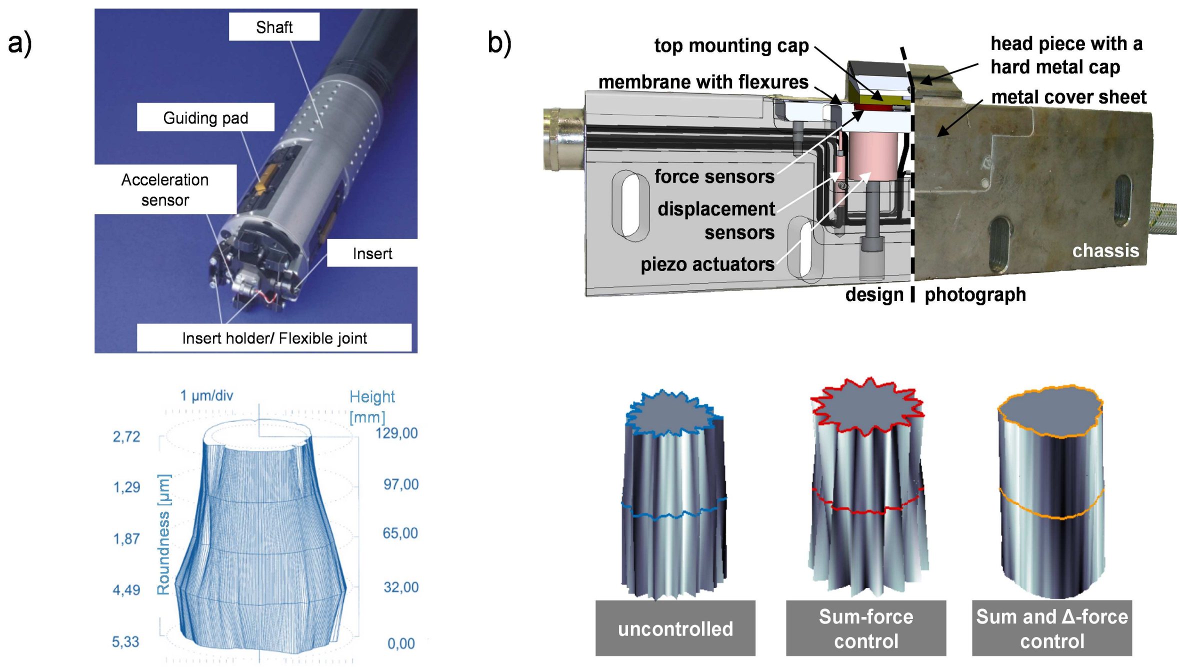


Fig. 5. Mechatronic drilling tool [30] - a), Active support blade [31] - b)

Schauerte investigates in [30] the so-called noninteracting MIMO-control for controlling the position of the cutting edge and the position of the three guiding pads of the drilling tool. He successfully compensates any displacements, which could have destabilized the drilling operation. Surface measurements of the drilled holes show a roundness error of 3 to 6µm.

In Fig. 5b) an active support blade for stabilizing the process of the centreless plunge grinding is shown. This active system lies in the direct flux of force. To except any additional weak spot introduced by the active system, the active support blade design was carefully worked out. In [31] a force driven control of the workpiece position is proposed to compensate the geometrical roundness error and process instabilities occurring in centreless plunge grinding. Goal of the control is a constant, non-oscillating grinding force. Therefore, two force sensors are positioned close to the cutting process, within the active support blade. Piezoelectric actuators generate the necessary compensation forces. The flexures are optimized to eliminate shear forces, which could damage the piezoactuators. Brecher et al. implemented a complete PMI simulation of the centreless plunge grinding process at a centreless grinding machine. The active system was modelled into this PMI simulation to optimize its performance. Later, in grinding tests on the machine tool Brecher et al. showed the decrease of roundness error due to a use of the active support blade.

## 4. SUMMARY AND OUTLOOK

This paper worked out the three main challenges while developing active damping systems for machine tools: the mechanical design, the controller and suitable actuators and sensors. The case studies given in this paper show, that the current state of the art already has possibilities to face those challenges. With all the mentioned approaches, chatter could be avoided. Today, there exist a plurality of active systems with a specific design regarding weak spots of machine tools. Main requirements for the mechanical design are robustness, reliability and design space. In further researches, an evaluation of suitability for industrial applications of active systems in machine tools should be promoted. Especially the early integration of active systems into the machine tools design should be investigated. With help of active systems machine productivity can be exploited. Particular in cases where design and/or process optimization have no further positive effect or are not possible, e.g. boring tools, robots or other mechanical structures with a concept specific weak spot, active systems might be the solution. The controller design have to be stable with the necessary bandwidth and robustness against parameter variations of the machine tool. Easy to implement single input single output controllers e.g. the application of Notchfilters have proved their suitability in the latest researches. While in the past, piezoelectric actuators dominate the field of active systems, nowadays, reviving hydraulic actuators with high dynamic servo valves is discussed especially in large scale machine tools, where large travel ranges are necessary.

*ACKNOWLEDGEMENT*

*The authors like to thank Alexander Guralnik for his effort in contributing to this paper. Different research presented in this paper are sponsored by the German Research Foundation (DFG) in the priority program 1156 'Adaptronic for machine tools' (BR 2905/11-2), the project 'Simulation und Optimierung des dynamischen Verhaltens* *von Spitzenlosschleifmaschinen' (BR 2905/41-1) and the project 'Strukturintegrierte Kompensationsmodule für Portalmaschinen' (BR 2905/49-1).*

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