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MITITGATION OF MACHINE VIBRATION BY MEANS OF AN ACTIVE SPINDLE BEARING UNIT

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ABSTRACT

Due to the high cutting speeds and depths of modern milling machines vibrations are generated within the whole machine structure. These vibrations lead to a poor machined surface and an additional load for all machine components. Within the collaborative research centre 368 "Autonomous Production Cells", IFAS is involved in the development of an active spindle bearing unit which is capable of damping vibration without reducing milling efficiency [7].

The special layout of the "three cavity ring" results in a high dynamic and stiffness, little need for space and a possible displacement of five tenth of a millimetre. A zero-leakage design is very important for the performance of the actuator. Reducing the leakage of the actuator will increase the dynamic and stiffness of the drive.

Using a micro controller for controlling the bearing unit leads to a decentralised system with local intelligence, that can be integrated into a milling machine.

KEY WORDS

active hydraulic damping, chatter control, high stiffness and dynamic, piezo servovalve, spindle bearing

INTRODUCTION

High metal removal rates are a main goal of cost-effective manufacture and can be achieved by means of high cutting speeds, high feed rates and a suitable cutting depth. Unfavourable cutting conditions may cause chattering of the milling tool, which leads to poor surface quality on the workpiece and possibly even to tool breakage. Variable speed or active damping of the milling machine spindle represent possible solutions, which may be implemented in order to achieve a higher metal removal rate while maintaining a stable process. The goal is to change the energy state of the whole oscillating system in order to bring the chattering milling process back to a stable state. An active spindle bearing concept is being pursued at IFAS and is described below.

Apart from piezo ceramics and magnetic bearings, other possible candidates for the actuator technology for active bearing systems are hydraulic servo-systems. The high force density and the acceptable stroke distinguish the hydraulic systems from the other alternatives. In addition to this hydraulic drives are easy to control.

CHATTERING

Nowadays the metal removal rate of milling machines is not limited by the motor capacity, but by self-induced vibrations of the whole machine structure, which are called chatter vibrations or chattering. As a result of the milling process every milling machine is deformed by the cutting forces. The grade of deformation is dependent on the static and dynamic stiffness of the structure of the milling machine. Exceeding the maximum cutting depth generates self-induced vibrations with high amplitudes, which lead to a poor surface quality of the machined workpiece (Figure 1) according to the eigenfrequency of the milling machine. Repeated machining of this bad surface results in a dynamic excitation of the milling machine structure. Therefore the second cut causes unsteady cutting forces, which excite a self-induced, regenerative vibration [8]. As this vibration is an additional load for the tool and the machine it can even cause tool breakage and that is why it needs to be reduced and cushioned.

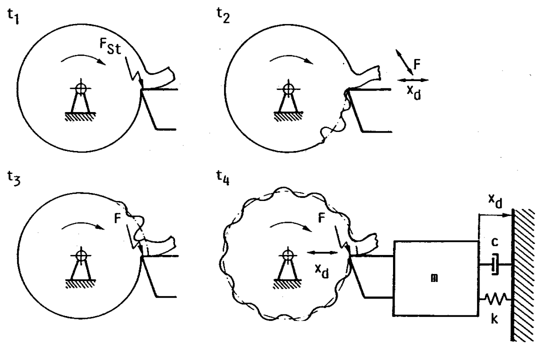

Figure 1 Regenerative chattering

Another type of self-induction takes place in coupled systems. At least two orthogonal eigenfrequencies are that close to each other that their vibrations interact and excite the other system [2]. Figure 2 shows the tip of a milling tool, which stands for an oscillating system with two degrees of freedom, cutting into a workpiece.

In opposition to the regenerative effect the locality coupling can cause chattering with the first cut into the unmachined workpiece and does not need a badly machined surface. There are further reasons for the excitation of chatter vibrations, but they are not that relevant for milling machines because of the high cutting speeds. One main characteristic of chattering is, that it is not dependent on the rotational speed of the machine.

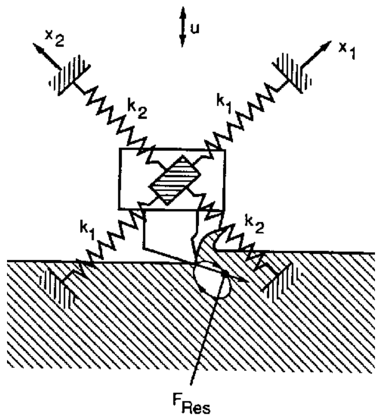

Figure 2 Locality coupling

To maintain a stable milling process, these vibrations need to be sensed, identified and controlled and thus eliminated. Therefore the self-induced vibrations need to be separated from the separate-excited vibrations, like the vibration, which is caused by the first contact of the cutting edges of the milling tool with the workpiece. After identifying the chatter vibrations a suitable principle for reduction needs to be implemented. Normally the cutting depth, the cutting speed or the rotational speed is reduced, which reduces the metal removal rate of the machine.

But in case of the active spindle bearing, which is developed at IFAS, the front bearing of the spindle is controlled and seated hydraulically to provide high forces for the milling process. The oscillating deformation of the milling machine geometry is compensated by the oscillation of the spindle bearing system. Without reducing any of the basic cutting parameters, the metal removal rate can be maintained at a high level.

THE DESIGN OF THE ACTUATOR

According to the restrictions for the layout of machine tools the most important design features for the active spindle bearing are high static and dynamic stiffness, low need for space, a leakage-free principle and, to compensate the deformation of the headstock, a possible displacement of ± 0.5 mm.

The conventional spindle bearing system consists of two angular ball bearings in the front of the spindle, that are preloaded in an O-arrangement, and a preloaded angular ball bearing in the rear. The active spindle bearing device also uses the conventional bearing in the rear of the spindle. Whereas the conventional bearings in the front are fitted into an

active bearing ring. The bearing ring is designed as a “three-cavity ring” (Figure 3) to achieve a better homogeneity of the forces on the perimeter.
The active ring is controlled by three sickle-shaped, hydrostatic cavities which are pressurized by three servo-valves – one for each cavity. The possible displacement out of the center of the active bearing ring needs to be ± 0.5 mm. The enclosure contains also three hydrostatically balanced sealings, which separate the space between the enclosure and the active ring into the three cavities. These sliding pads are arranged on the perimeter with an angle of 120° between each other. To maintain a constant gap between sliding pad and active ring and to prevent the pad from loosing contact with the ring, four springs and a constant pressure are applied to the back side of the pad.

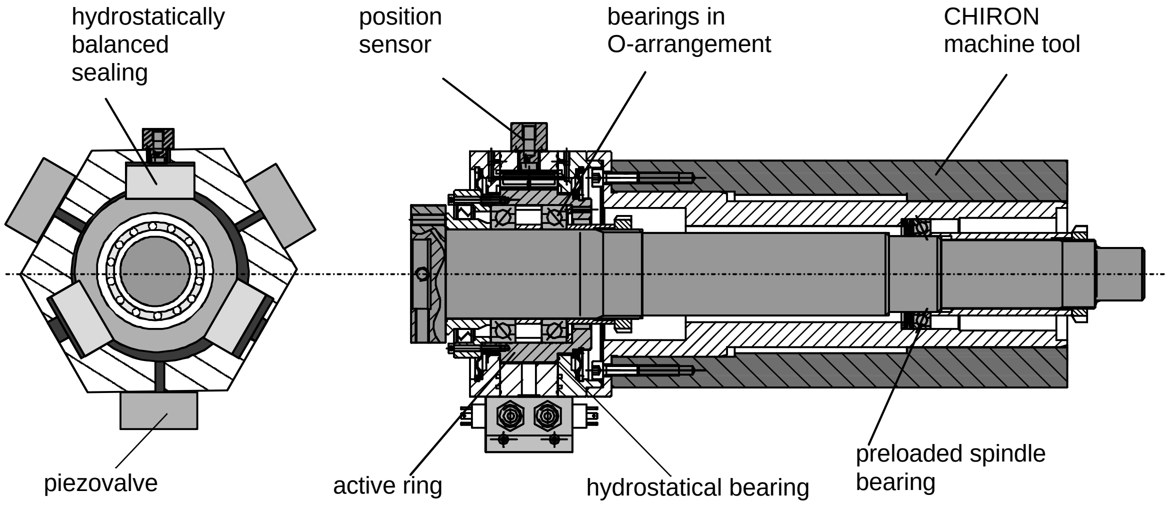

Figure 3 Principle of the active spindle bearing

Beneath the control of the radial position of the active bearing ring, the ring needs to be seated axially in the enclosure. Therefore three hydrostatic pockets are implemented on each side of the active ring to carry both, the cutting forces during the milling process and the weight of the spindle. Furthermore this axial bearing reduces the friction for any radial displacement.
Three piezo-electrically driven servo-valves, which have been developed at IFAS for this purpose, can be used to set the pressure in the cavities [4]. The valves are illustrated more exactly below. As the spindle and the valves are both prototypes, it is not recommended to test all prototypes in one test arrangement. This would only result in numerous repairs and the project might lose its focus. It is therefore legitimate to separate the prototypes in order to solve specific problems separately. For this reason, the spindle unit is driven by conventional and well-tested MOOG servo-valves whereas the piezoelectric servo-valve-prototype is tested in another arrangement. Since the stiffness and thus the hydraulic eigenfrequency of the drive is much higher than the dynamic of the valve, faster valves will increase the drive performance.
The large piston area of the ring segment and the low dead volume result in a calculated radial stiffness of 8000 N/µm for the hydraulic bearing, which corresponds to 30 times the stiffness of the used ball bearings. The high stiffness causes a high eigenfrequency of the drive thus the slower valves limit the dynamic of the drive. For this reason a simple single loop controller suffices to control the position of the active ring [6].
The current position of the active bearing ring is determined by the position of the sliding pads. Three sensors, which operate according to the eddy-current measuring principle, are aligned to the back sides of the sliding pads and detect their positions. The eddy-current sensors were also developed specifically for this purpose at IFAS.
The principle of a hydrostatic bearing comes along with an unavoidable leakage. Because of this, in addition to the position control of the bearing ring, the pressures in the three cavities need to be regulated. Therefore a pressure sensor measures the pressure in each of the working cavities.
Because of the large piston areas of the hydraulic cavities force is lead smoothly and evenly into the bearing ring. Figure 4 shows the active spindle

bearing built into the headstock of a CHIRON milling machine.

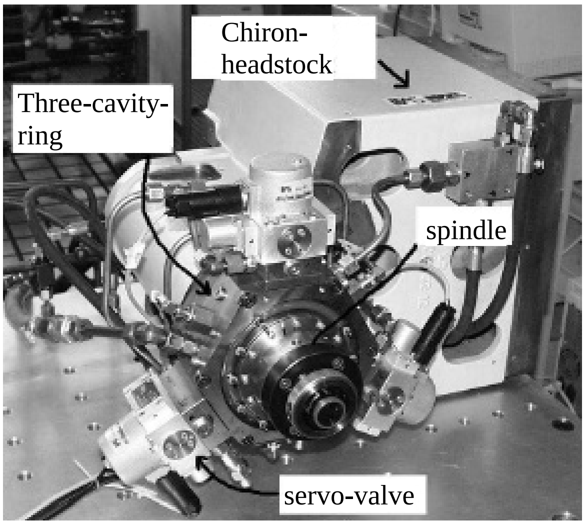

Figure 4 Photograph of the active spindle mount

PIEZOELECTRIC SERVO-VALVE

The piezoelectric servo-valves are used for pressurizing the three cavities of the active bearing ring. In order to control the chattering, high cut-off frequencies (-3dB) of up to 500 Hz in the small-signal range are required. As the hydraulic system of the active spindle is a very fast-responding one, the dynamic of the system is limited by the servo-valves and therefore servo-valves with high cut-off frequencies become key components.

High quality servo-valves are mostly pilot controlled and have a high level of dynamic behavior as well as a high accuracy in positioning the main spool, because of the spool position feedback [4].

Figure 5 shows a sectional view of the piezoelectric servo-valve, that was developed at IFAS. The pilot section of the servo-valve is designed as a flapper-nozzle system. The spool is moved by modifying the control pressures on the end faces of the spool.

The pilot section is supplied via the hollow main spool, which also houses the constant nozzles for the pilot section. The two flapper-nozzle units, which are permanently connected to the piezoelectric bending actuators, create the variable nozzles, which set the control pressures.

A piston with a diameter of 1 mm on the back side of the piezoelectric unit compensates the static force of pressure on the front side of the actuator. In this way only the pulse force of the flow against the flapper has to be overcome during movement of the main spool.

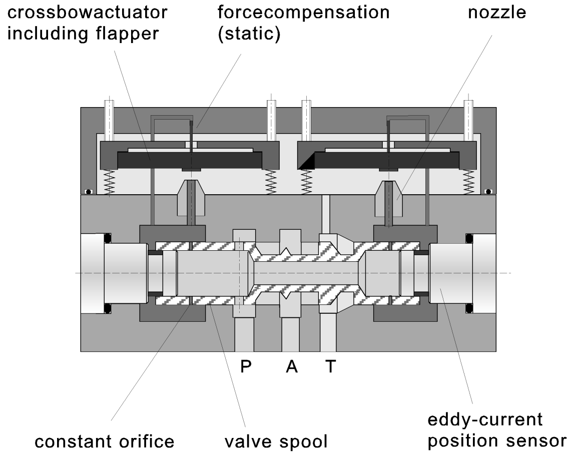

Figure 5 Sectional view of the servo-valve

In order to achieve a maximum pressure difference on the faces of the spool and therefore a maximum force for the movement, both bending actuators are always driven in opposite phases.

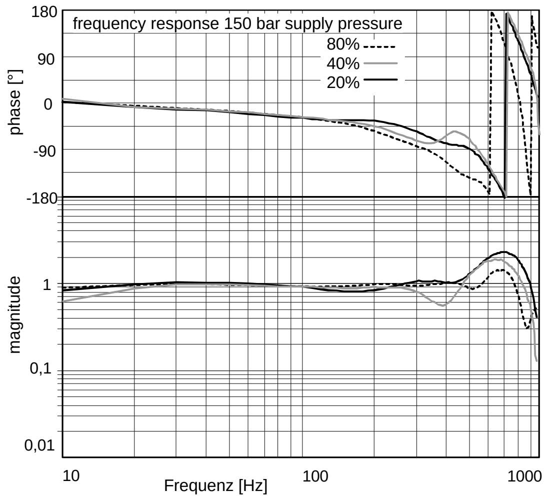

Figure 6 Frequency response of the valve spool in the position control circuit

The displacement transducer, which is based on the eddy-current measuring principle, is required for the subordinate position control loop and is integrated into the valve spool. The end faces of the spool are reduced by stoppers, that house the electrical windings for the displacement transducer. The reduced faces limit the force acting on the spool as well as the required pilot flow to move the spool. The diameter of the stoppers can be modified in order to meet the specific restrictions [4].

The spool valve travels ± 1 mm and the nominal flow rate is 44 l/min. The pressure signal function of an initial prototype version was measured and indicates a gradient of 4000 bar/mm at zero point and is similar to the gradient of high-quality servo-valves. The gain in volume flow is about 80 l/min/mm [4]. Figure 6 shows the closed-loop frequency response of the valve with a position-controlled spool. An analog PI controller was used to close the loop. The cut-off frequency (-90°) was measured at about 550 Hz with 20 % amplitude.

CHARACTERISTICS OF THE HYDRAULIC DRIVE

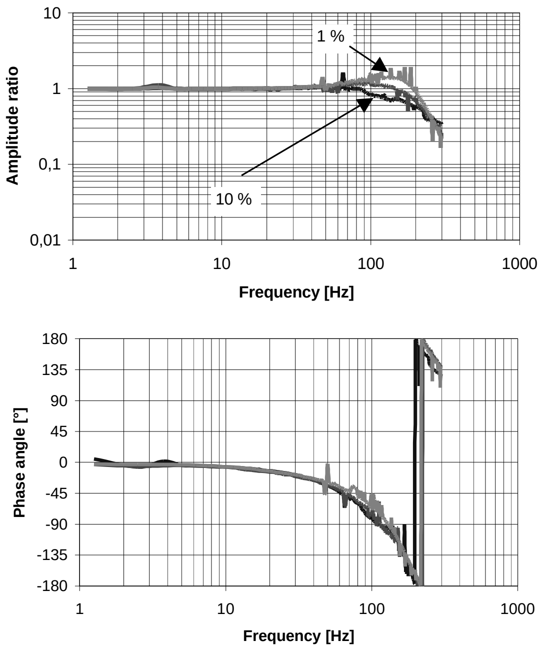

Figure 7 Dynamic of the drive

Figure 7 shows a satisfying dynamic behavior of the drive of the active ring in a bode-plot of the closed-loop step response for various signal levels. It is examined while an analog PI-controller closed the position-control loop and conventional servo-valves (approximately 200 Hz cut-off frequency) were applied to the drive.
The cut-off frequency (-3dB) is reached within 200 Hz and the –90° of the drive reaches 120 Hz at low signal level. This shows, that the drive dynamic is close to the valve dynamic. It does not totally reach the valve dynamic because with the small cavities, which are used for the drive, even the smallest unavoidable leakage leads to a pressure drop in the cavity. Therefore the valve-spool has to open further than the controller signal requests in order to pressurize the cavities satisfactorily [1].

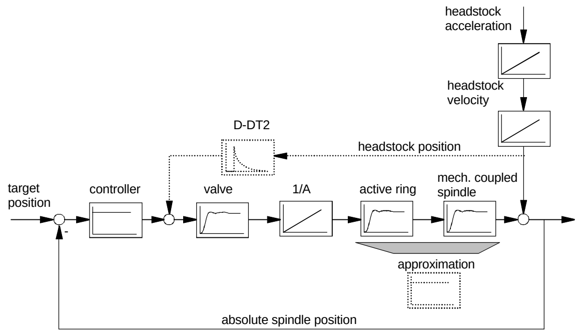

Figure 8 Position control with feed-forward control

As the drive is of a higher dynamic than the valve, the valve dynamic limits drive performance. If the system were linearly approximated, it could be characterized by a series of a servo-valve with a PT_2-characteristic, an integrator (I), the active bearing ring itself (PT_2) and the spindle, fixed to the ring via the conventional bearings (PT_2). Reducing this complex system down to a PT_2-I does not affect the system dynamic in the range where it is normally driven. This is because the transmission behavior of the active ring and the spindle-bearing-arrangement can be approximated by proportional elements (Figure 8). Their eigenfrequencies are by far higher than the dynamic of the remaining system. Sensing the acceleration of the headstock and filtering the signal through a D-DT_2-filter before adding it to the controller output could eliminate the influence of the headstock movement by means of a feed-forward control. This is because the filtered headstock-position and the control path (valve → cylinder → mechanic) show an inverse characteristic to one another.

SUPPORTED FURTHER DEVELOPMENT BY MEANS OF SYSTEM SIMULATION

While the prototype of the active spindle bearing is investigated, the drive is also analysed within the simulation environment of DSH*plus* [3]. The simulation of the drive is a first important step to gain experience with dynamic behaviour of the drive. In figure 9 the simulation model of the active unit is introduced. For the simulation of the three-cavity-ring, which is not part of the library of DSH*plus*, the movement of the active ring is limited to one coordinate and simulated as a standard cylinder drive.
Thus **chamber 1** is supplied by **valve 1** (3-3-way-servovalve), whereas **chamber 2** is supplied by **valve 2** and **valve 3** one half each. The **current value** of the position of the active ring is compared to the **target value** and the resulting **control deviation** is causing a input signal for **valve 1** and multiplied with “-0.5” for **valve 2** and **valve 3**. By this an excitation in vertical

direction is approached (figure 3). The **pressure source** supplies a **system pressure** of 200 bar. The displacement of the drive is controlled by a simple proportional controller. The valves are parameterised according to the manufacturers data, the cylinder and the remaining elements according to the design data. Line lengths and other capacities have similar dimensions as within the real part.

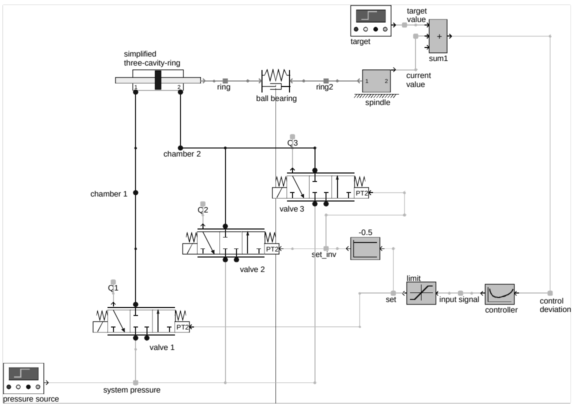

Figure 9 The three-cavity-ring as a model in the DSH*plus* environment

Working with a simulation tool gives the opportunity to evaluate modifications of the drive, without modifying the prototype. In spite of the fast results using the simulation environment, results cannot be adopted offhand, but need to be investigated on the prototype afterwards [5].

CONCLUSIONS

A high-dynamic three-cavity hydraulic cylinder drive is presented in this paper. It shows promising dynamic characteristics in the test-arrangement and will be investigated during a real milling process to prove its ability to damp chatter vibrations.
Another final goal will be to test the combination of feedback and feed-forward control to overcome chattering during milling operations. With all software and controller-design being tested good, the controller will be implemented on a microprocessor to enable the active spindle bearing system to remove chattering from process independently – as necessary for an autonomous production cell.

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