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SOLUTION FOR A HIGH DYNAMIC DRIVE SYSTEM

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ABSTRACT

A high dynamic drive system for incremental linear motion that avoids the target conflict between a high motion velocity and high drive torque is presented.

This conflict can be successfully overcome by separating the main function "driving" into subfunctions "driving" "energy storage" and "energy release".

A test bench proofing the concept is introduced. Furthermore, a numerical process model (MatLab-Simulink) for the test bench is presented. This model serves for developing different control solutions for the system.

Finally, applications for the described system are shown and further development goals are defined.

Index Terms - high dynamic drive system, incremental feed drive

1. INTRODUCTION

Research at the Bremen Institute of Mechanical Engineering (bime) on machine tools and especially high dynamic drive principles has shown that there are many applications which demand high dynamic drive systems capable of incremental or step wise motion.

Ballscrew type linear drives are the standard drive solution in most of today's machine tools. The problem of these drives is a target conflict between high motion velocity and high drive torque. The maximum torque can be increased by using an additional gear or reducing the thread pitch. This however results in the controversy of a reduction of maximum velocity.

Other solutions like linear direct drives show high dynamic performances but have weakness with respect to load compliance and maximum thrust force. Aside from that in comparison they are quite expensive.

In the first part of this paper, the concept of a new high dynamic drive system is introduced based on a test bench and first test results are shown. In the second part, the numerical process model of the test bench is explained. Then, the model is compared to the real systems. In the end some applications for the drive system are shown and further development objectives are addressed.

2. PRESENTATION OF THE NEW DRIVE CONCEPT

The idea of the new high dynamic drive system is to use the phase of idleness in incremental feed processes to accumulate energy.
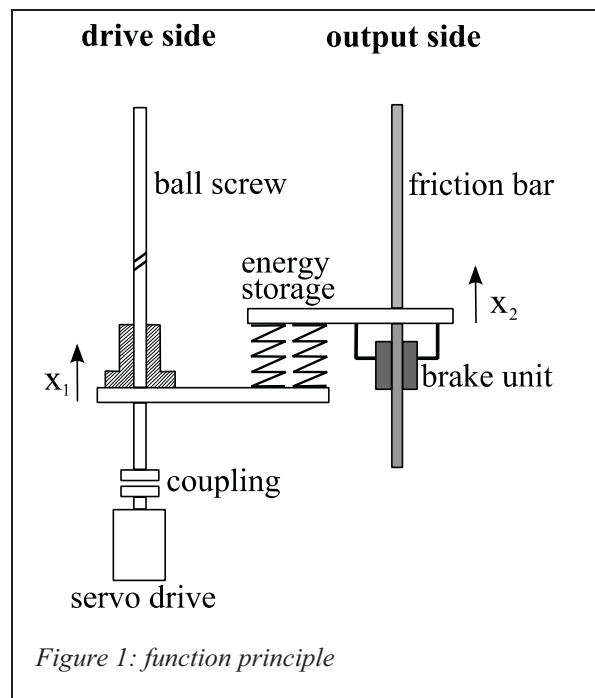

Figure 1: function principle

2.1. Function principle

The system consists of two guided slides, which are connected by a spring mechanism.

The first slide is actuated by a conventional ballscrew drive and a synchronous servo motor. It forms the drive side, which is controlled to keep a constant velocity ($\dot{x}_1 = const.$).

The second slide, which forms the output side, is equipped with a brake. With this, the output side can be blocked at a certain position, while the drive side continues running at constant velocity. In this case the spring is compressed and stores the energy from the drive side.

When the brake is released again the energy from the spring is set free and the output side is strongly accelerated. It can reach velocities much higher than those obtained by the ball-type linear drive on the drive side.
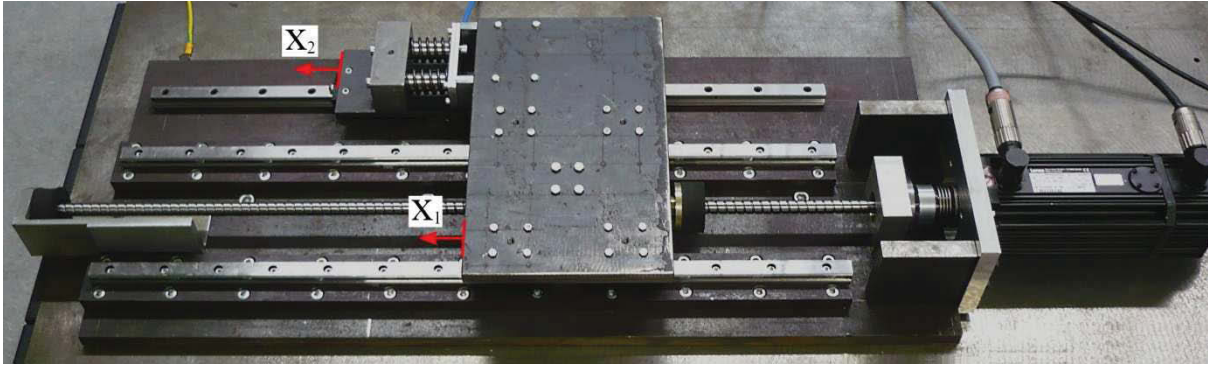
Figure 2: test bench

The reaction force, which acts on the drive side can easily be absorbed by the ball-type linear drive, which is arranged for high drive torque respectively drive force. It does not need to run at high velocity.
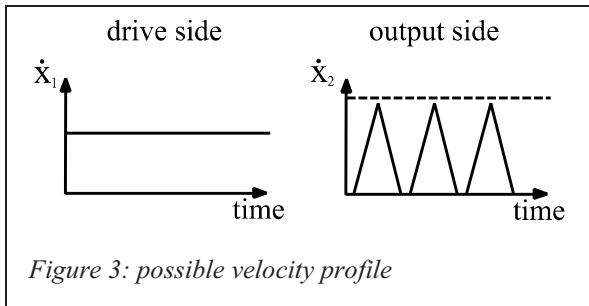

Figure 3: possible velocity profile

2.2. **Test bench**

The test bench can be seen in Figure 2: test bench. It consists of the parts already described in subsection 2.1. On the right the synchronous servo motor is mounted at a flange. It is connected to the ball screw by a coupling. Slide 1 bears the nut that translates the rotational movement to a linear movement. The slide is guided by 2 linear guideways.

Slide 2 is guided by a single guideway. The brake unit is mounted between the two guide blocks. It is a spring-loaded brake element opening with pneumatic pressure. The brake pads actuate against the rail guide.

The connection between the drive side and the output side is formed by two springs. This element has additional functions in two aspects compared to the principle scheme (Figure 1).

The first is that the springs are preloaded. The second going along with the preload is an end stop, which limits the maximum distance between the two slides. Further there is some friction on the guides of the springs working as damping.

In the absence of the preload there would not be any force on slide 2 without a position difference ($\Delta x = x_1 - x_2$). That would cause an oscillation at the output side.

2.3. **Test results**

For a first test the drive side is controlled to a constant velocity of $\dot{x}_1 = 15\ mm/s$ and the brake is switched by a signal generator. The control signal for the pneumatic valve which releases the brake is a square-wave signal. The position of slide two is measured with a laser interferometer.

The results are evaluated in Matlab. The velocity signal and the acceleration signal are generated by numerical differentiation of the position measuring.
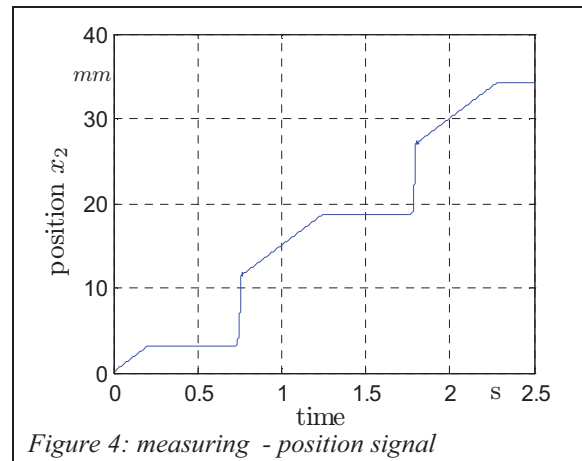

Figure 4: measuring - position signal

In Figure 4, the position signal is illustrated and the general behavior of the mechanism can be seen. At the beginning the brake is inactive and both slides are accelerated synchronously to the constant velocity $\dot{x}_1$.When the brake is activated the position remains constant and the spring is compressed.

When the brake is released again slide 2 is strongly accelerated and moving at a very high velocity until it hits the end stop of the springs. At this point the velocity of slide 2 is synchronized with the velocity of slide 1. The slide is moving again with the constant velocity $\dot{x}_1$ until the brake is activated and the incremental cycle begins again.

The line with the constant gradient $\dot{x}_1$ can be easily seen. That means that the overall velocity is still influenced by the velocity of slide 1, but it is much

better utilized because the drive does not need to be stopped anymore.

Figure 5 shows a more detailed view of a single step. In addition to the position the velocity and the acceleration are shown. It can be seen that slide 2 can reach velocities and accelerations higher than the maximum possible which the ballscrew drive permits.

At the end stop, slide 2 is stopped very abrupt which causes high peaks in the acceleration graph. The impact of the two metal parts results in a short vibration, which can be recognized very well in the velocity graph.
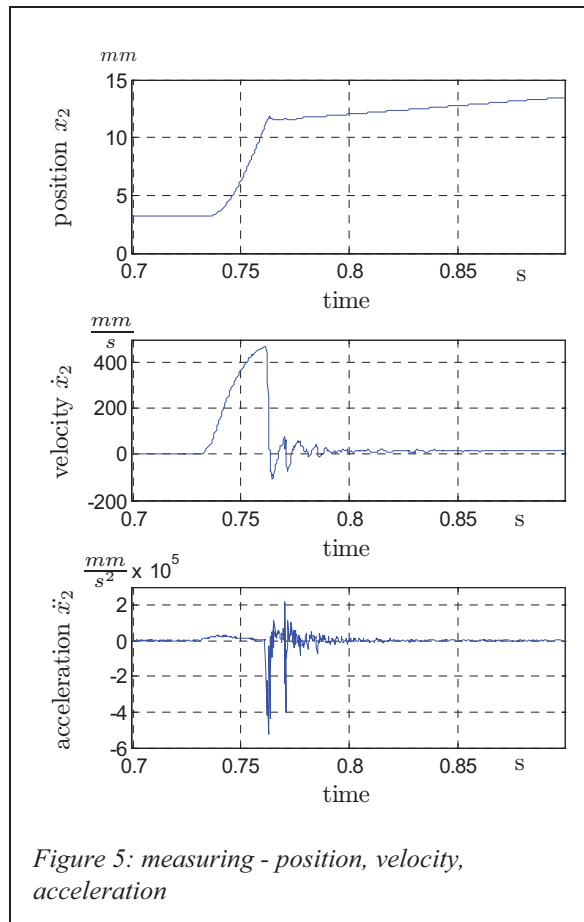

Figure 5: measuring - position, velocity, acceleration

3. NUMERICAL MODEL

A numerical process model of the test bench was developed in MatLab-Simulink as basis for the development and testing of different control structures for the system. The motor is velocity controlled and uses a cascaded PI-PI-controller structure.

3.1. Model equations

The synchronous servo motor is modeled with a simplified model after [1]. The mass of slide 1 is transformed to an equivalent moment of inertia and summarized with the moment of inertia of the motor. The friction of the caged ball linear guides is negligible. Nevertheless, there were made some tests with a coulomb and viscous friction model for verification of this assumption.

The position x_1 of slide 1 can be gained by integration of the dynamical equilibrium of the moments.

$$\left(J_{motor} + M_{slide,red}\right) \cdot \dot{\omega} = M_{motor} - M_{spring,red}$$

Slide 2 can be considered as a simple moving mass and the position x_2 can be gained by integration of the dynamical equilibrium of the forces.

$$m \cdot \ddot{x}_2 = F_{spring} - F_{brake}$$

As the two slides are coupled by the spring the spring force appears in both equations of motion.

The end stop of the spring is modeled as a very rigid elasticity. So the spring force can be described by the following equation.

$$F_{spring,1} = c(\Delta x) \cdot \Delta x - d(\Delta x) \cdot \Delta \dot{x}$$

The rigidity and the damping change with the conversion of the position difference.

$$\Delta x = x_1 - x_2$$

The preload of the spring is modeled by a force feed through. That means that until the preload is reached a force that guarantees a synchronous movement of the two slides is applied on slide 2.

$$F_{spring,2} = \frac{\ddot{x}_1}{m_2}\left(+F_{friction,2}\right) , \ F_{spring,1} < F_{preload}$$

$$F_{spring,2} = F_{preload} , \ F_{spring,1} \geq F_{preload}$$

The total spring force is the sum of these two components.

$$F_{spring} = F_{spring,1} + F_{spring,2}$$

In the actual configuration the brake is either working at the maximum braking force or it is released. So there are three possible states that need to be described.

The first is a released brake which means there is no braking force applied on slide 2.

$$F_{brake} = 0$$

The braking force is limited to the maximum Force of the brake. Under that limit the braking force compensates the load.

$$F_{brake} = F_{spring} , \ F_{spring} < F_{brake,max}$$

Over the limit the force is held at the maximum value.

$$F_{brake} = F_{brake,max} \cdot sign(\dot{x}_2),$$
$$F_{spring} \geq F_{brake,max}$$

3.2. **Realization in Matlab-Simulink**

The equations described in the preceding subsection were implemented in a Simulink block diagram and the model was tested with a test signal according to the one used on the real system.

During the first tests it was found that for the case differentiation of the brake, the zero-crossing detection of the appropriate Simulink-blocks must necessarily be activated. Otherwise the inaccuracy in the calculations leads to negative velocity when the brake is activated and the slide velocity $\dot{x}_2$ should be zero.

In the first graph of Figure 7 the results of the model are compared to the measured ones on the test bench. For the exact analysis one single increment is considered. In the second graph the absolute error can be seen.

The maximum error appears when the end stop is hit. It is smaller than 0,4mm. If one compares the graphs in Figure 7 with those from Figure 5, it can be determined that the behavior of the velocity and the acceleration from the model corresponds to the real system. But again it becomes clear that the main error can be found in the modeling of the end stop, which is strongly simplified.
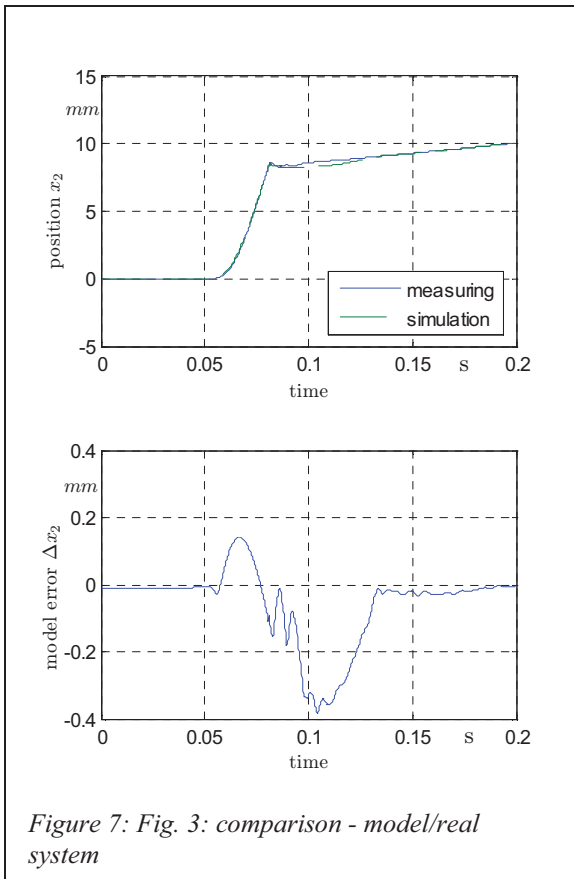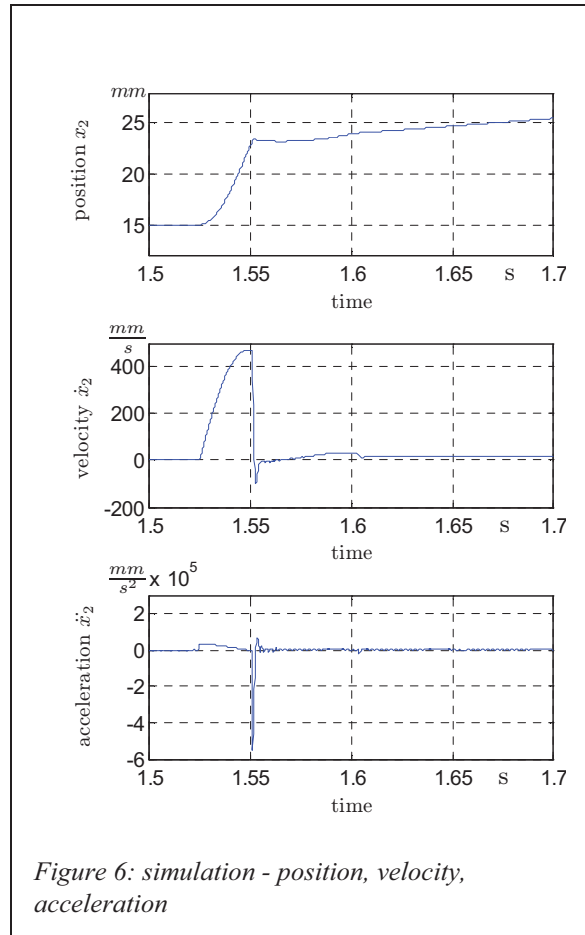

Figure 6: simulation - position, velocity, acceleration

Figure 7: Fig. 3: comparison - model/real system

4. OBJECTIVES AND POSSIBLE APPLICATIONS

The position graph in Figure 4 serves only as a first test and for the understanding of the system. The objective for the position graph is illustrated in the graph in Figure 8, which is the result of a simulation.

In this case slide 2 is stopped by the brake before it hits the end stop. The velocity profile consists of short peaks. To reach this behavior all the process parameters like for example the spring stiffness, the spring travel and the velocity on the drive side need to be well coordinated.

The shown motion pattern is typical for incremental processes. Exemplary for such processes winding machines working in the step pack mode or incremental forming processes like the rotary swaging examined in [2] can be named.

In all these cases there are two alternating phases. A phase in which the feed is idle and the process takes place and another phase in which the process is idle and the feed needs to change configuration very fast.

To optimize these processes it is very important to minimize the time in which the machining is idle. That means that the feed drive must be able to generate high acceleration and velocity. But during machining it must also show high load compliance.
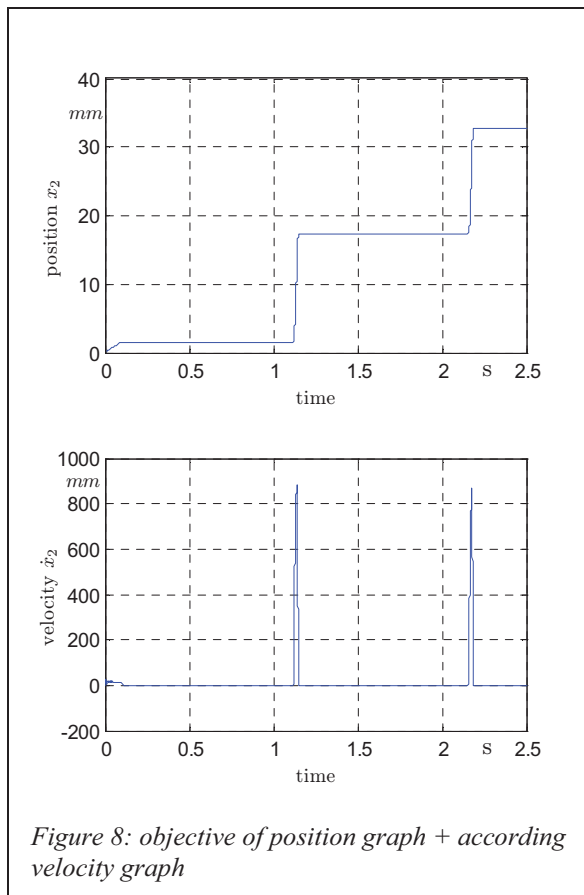
Figure 8: objective of position graph + according velocity graph

The presented drive solution can sustain high processing forces with the brake. With the energy storage device, it fits the dynamic requirements although it uses a conventional ballscrew linear drive.

The development objective is to develop a control that secures the exact positioning. For other applications it is also thinkable to optimize the velocity profile. On the mechanical side mainly the brake can be optimized.

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