Advantages and Characteristics of a Dynamic Feeds Axis with Ball Screw Drive and Driven Nut

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Received on January 10, 1999

Abstract
This paper describes the advantages and characteristics of a Dynamic feed axis with ball screw drive and driven nut in comparison with the conventional electromechanical drive. In contrast to the classical drive the ball screw leadscrew is rigidly located at both ends and the nut is driven by a servomotor. The investigation considers two drive variants: the indirect drive with a standard servomotor driving the nut via a toothed belt and the direct drive with a hollow shaft servomotor. The feed system developed at the Institute for Machine Tools and Production Science at the University (TH) of Karlsruhe opens up new prospects for the development of new dynamic machine generations with traverse speed up to 120 m/min and acceleration up to 45 m/s². In addition to the high acceleration and traverse speed, the development program also has the objective of optimised transmission behaviour of the feed kinematic system with positional control. The design features and operating characteristics achieved with the innovative test device are described.

Keywords: Feed-drive, Optimization, Machine tools

1. INTRODUCTION

A simultaneous increase in processing speed and processing quality places high demands on the feed systems of modern day machines. This trend is supported by the considerable developments made in control technology as well as in the development of dynamic drives. In contrast, the mechanical components of the drive sequence are frequently the weakest link when seeking improvement in dynamic behaviour. In the case of electromechanical feed axes, the rotary movement of the servomotor is converted into a linear movement of the table by means of a ball screw. The classical concept of a locating/non-locating NC axis comprises a driven feed screw and a rigid ball screw nut fixed rigidly to the table and can achieve speeds between 30 to 60 m/min with an acceleration of up to 10 m/s². New avenues to overcome this technical limitation have been opened up by the linear drive which promises a significant increase in dynamic behaviour (> 100m/min) thanks to the direct generation of the translational movement. However, the low dynamic rigidity and the significantly greater system costs are especially disadvantageous and mean that the linear drive is not necessarily the ideal solution for all applications [1, 2].
Based upon this background, the incentive to develop a system which can compete in terms of dynamic behaviour and is based on the tried and tested feed technology was given.

2 STRUCTURE OF THE TEST DEVICE

The Dynamic Feed Axis with Ball Screw Drive and Driven Nut is an adaptation of the classical axis concept. In this case the leadscrew is fixed rigidly between two clamping devices and the nut is driven by a servomotor. The thermal longitudinal expansion of the fixed leadscrew is compensated for by using a hollow shaft leadscrew which is cooled from the inside using a cooling medium [3].
Within the framework of the investigations, the two different methods of drive are to be observed. Figure 1 uses a standard servomotor (J_M=0.0066 kgm^2) driving the nut via a toothed belt (i=1, J_{Mech}=0.01 kgm^2). Figure 2 shows a direct drive system of the nut (J_{Mech}=0.0077 kgm^2) with a hollow shaft servomotor (J_M=0.0127 kgm^2). Both dynamic motors of the latest generation provide 120 Nm or 80 Nm to accelerate the table and achieve values of up to 4500 r.p.m. In addition, the mass of the table can be increased to half a ton in 100 kg steps. Digital and cascade controls and a linear scale are used to provide information on position.
Because the leadscrew is rigidly fixed at both ends, the torsion and axial rigidity of the overall system is far greater than that of a located / non-located leadscrew. The lowest rigidity is not on the non-located side as is the case with conventional feed axes, but in the middle of the positioning range which reduces the change in rigidity along the positioning length.

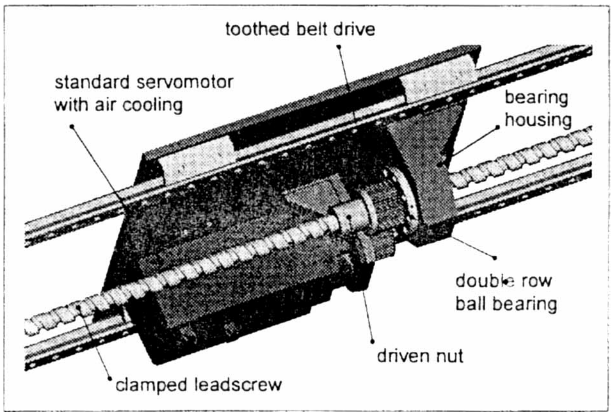


Figure 1: Drive variant with standard servomotor

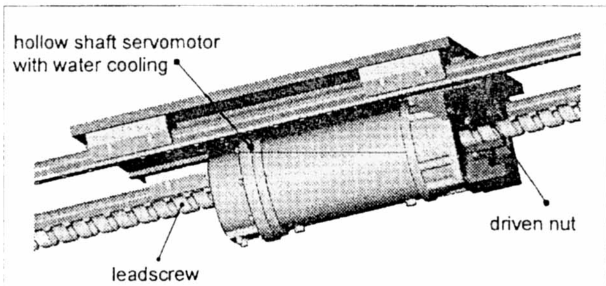
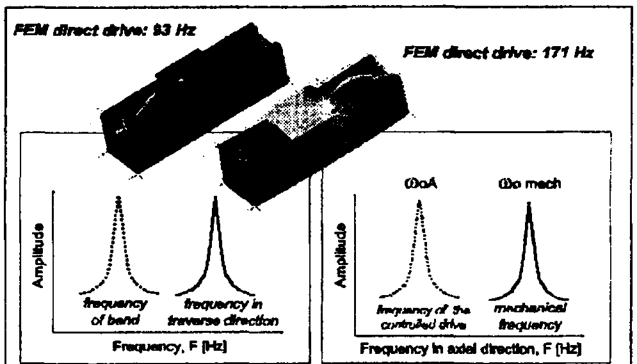


Figure 2: Drive variant with hollow shaft servomotor

In order to achieve a highly dynamic system, care was taken to consequently minimise the moment of inertia when designing the overall rotary drive. Despite the mass of the servomotor which must also be moved, the resulting moment of inertia for a leadscrew length of 2 m is comparable with a driven leadscrew of the same length. If the leadscrew is longer or if it has a greater diameter, the moment of inertia is even more favourable.
The drives used within the framework of the experiments with a diameter of 40 mm and a thread pitch of 40 mm achieved a speed of 120 m/min at a maximum 3000 r.p.m. With a table weight of 100 kg, acceleration values of up to 45 m/s² were achieved using an indirect drive and up to 25 m/s² were achieved using a direct drive.

3 DYNAMIC BEHAVIOUR OF FEED KINEMATIC WITH DRIVEN NUT

In addition to achieving high acceleration and traverse speeds, optimised transmission behaviour of the controlled feed kinematic system is necessary to achieve high dynamic behaviour. To translate the required sequence of movements it is therefore necessary that the axis of movement follows the set positional values as closely as possible. This requirement can only be fulfilled approximately in the real system. This is because of the inherent dynamic behaviour of the cascade control which is made up of P-positional controls and PI speed controls, with its elements of energy translation and mechanical components. During the translation of the motor rotatory movements of the servomotor into a longitudinal movement of the table, pronounced resonance oscillations are caused in the traveller movement. The bending vibrations of the drive rod or the transverse oscillations of the table system can be the limiting factors for the control quality or control stability and the associated track accuracy which can be realised (Figure 3). These are caused either by the bending critical number of revolutions which limits the system or by jerking movements or external disturbances which could occur.



Frequency of the position control	ω_{oL}	0,6 - 0,7 ω_{oA}
Frequency of the speed controlled drive	ω_{oA}	2 - 3 ω_{oL}
First mechanical frequency	$\omega_{o\,mech}$	2 - 3 ω_{oA}
Additional mechanical frequencies	$\omega_{o\,mech+n}$	2 - 3 $\omega_{o\,mech}$

Figure 3: Overview

The dominant resonance in the direction of movement is caused by the resonant frequency of the feed drive with a controlled number of revolutions or the mechanical transmission components. However, only the mechanical resonant frequencies which lie in the vicinity of the feed drive controlled by the number of revolutions have a significant influence on the frequency response. For the relative position of the frequencies to one another, the factors mentioned in Figure 3 must be adhered to [4]. With regard to the maximum number of revolutions and the resulting high traverse speeds resulting from this, the driven nut with a leadscrew which is fixed at both ends gives a direct advantage as it increases the bending critical number of revolutions by a factor of 2.26 compared to a common located/ non-located leadscrew. Even for drives rotating at a great number of revolutions, the system lies within the sub-critical range for travel up to 1.8 m.

3.1 Identification of the mechanical transmission elements

To determine the transmission behaviour, the impulse responce of the feed kinematics are recorded using an acceleration recorder on the table. Figure 4 shows the frequency response of the direct drive with a hollow shaft servomotor for three different table positions with a table mass of 140 kg. The leadscrew has been fixed at both ends and pretensioned with 100 µm. The extreme and middle positions have been selected for the table positions. With a dominant resonant frequency of 170 Hz, the direct drive displays significant one-mass oscillation behaviour. The associated oscillation curve is shown in Figure 3 using the finite element method. The axial oscillation of the table system via the stiffness of the spring of the ball screw becomes obvious. By changing the axial rigidity only slightly via the leadscrew length, the transmission behaviour of the overall structure is totally independent of the table position. By increasing the additional mass to 150 kg or 310 kg, the dominant resonant frequency is reduced to 130 Hz or 110 Hz (Figure 5). The effect of an increase in the table mass therefore is to reduce the resonant frequency by approx. 30% or 35% and at the same time to reduce the amplitude with the additional mass. The comparison of these feed kinematics with a leadscrew which has only been rigidly fixed at one end (comparable with a located / non-located leadscrew arrangement) is shown in Figure 6.

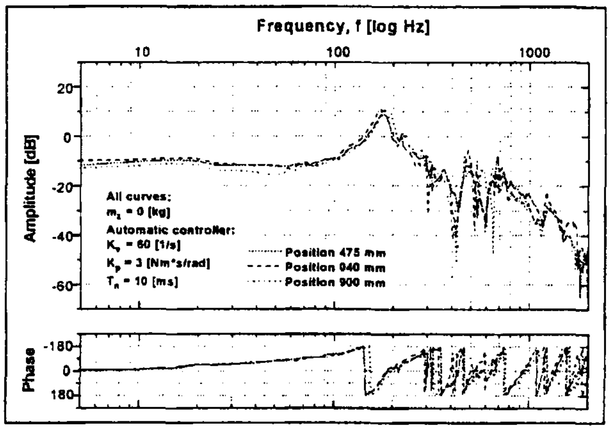


Figure 4: Response to Force: Direct drive

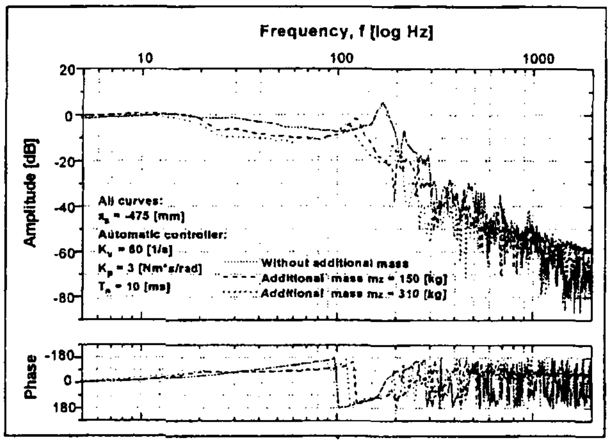


Figure 5: Response to Force: Direct drive

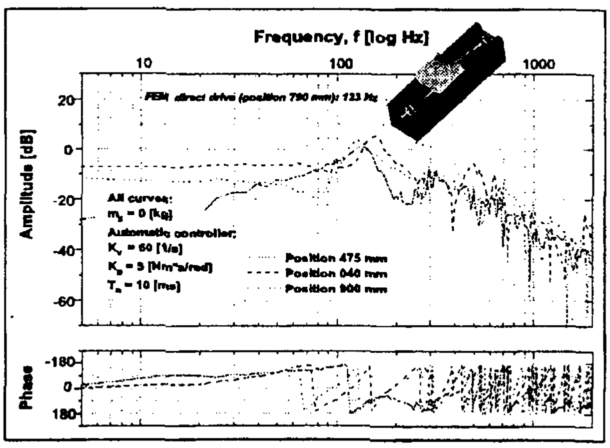


Figure 6: Response to Force: Direct drive

By having one non-located leadscrew end, the resonant frequencies shift into a range between 120 to 150 Hz depending on the table position. The lowest frequency is found at the non-located end and increases continuously towards the located end of the leadscrew. The transmission functions of the overall kinematics differ greatly for the different table positions.

The frequency reply of the indirect drive without servomotor and toothed belt transmission has been shown in Figure 7 for the extreme positions of the table. The dominant resonant frequencies are around 170 Hz. By loosening one of the ends of the leadscrew, the frequencies are shifted by up to 20 Hz on the non-located side. The influence of the leadscrew system has been identified. The systematic construction of the feed axis makes the change in the transmission behaviour clear (large Figure 8). In the first step all frequencies are pushed down due to the addition of the motor with its additional mass. The subsequent tensioning of the toothed belt shifts the first dominant resonant frequency from 141 Hz to 128 Hz. The second frequency remains virtually unchanged.

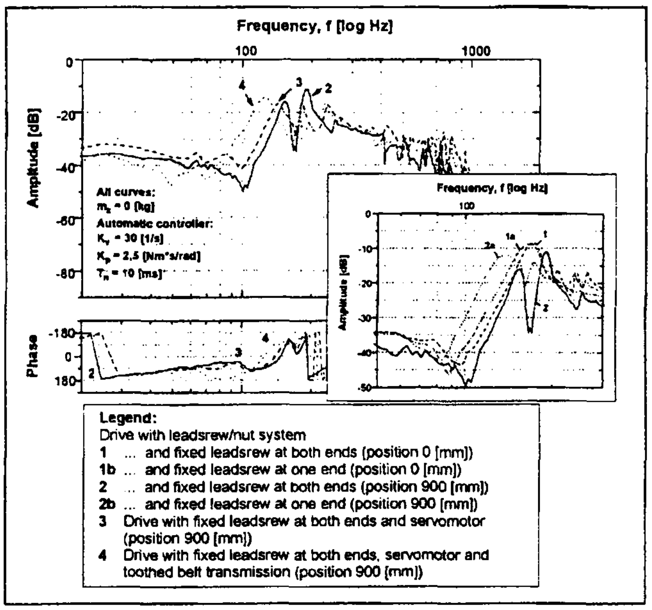


Figure 7: Systematic construction of the indirect drive

The transmission behaviour of the indirect drive with a table mass of 120 kg in the case of different table positions is shown in Figure 8. The extreme positions have two dominant resonant frequencies, 128 Hz and 171 Hz, the middle position has an additional resonant frequency at 98 Hz. To identify the oscillation curves the FEM calculation was used at table position 475 mm. The two lower resonant frequencies result from bending oscillations of the leadscrew and the drive in the vertical and horizontal direction which in turn result in a table movement in the axial direction.

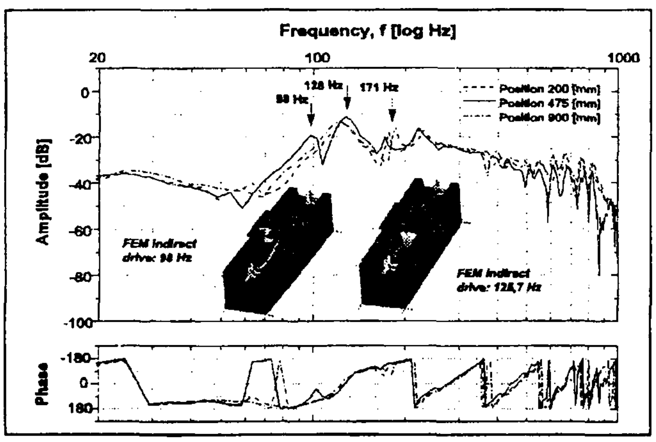


Figure 8: Response to Force: Indirect drive

In contrast the upper resonant frequency is equivalent to a pure axial oscillation of the table via the axial spring rigidity of the leadscrew system. By changing the table position from the middle position, the resonant frequency at 128 Hz changes downwards for both new positions, whereas the frequency at 171 Hz shifts upwards. The transmission behaviour of the overall structure in this area is virtually independent of the table position. This system behaviour can be explained by the fact that when bending is greatly influenced by the extension of the free swinging length, the frequencies fall. By assuming an extreme position, the axial stiffness of the leadscrew is increased, i.e. the frequencies increase with equal mass. This makes it clear that in contrast to the direct drive, the indirect drive system is influenced additionally by the arrangement of the moment of the drive and the increased bending flexibility of the construction. However, both drive variations have high mechanical resonant frequencies in common which are mainly independent of the table position. A virtually linear transmission behaviour with low parameter fluctuations is created whereby the element which sets the limitations is significantly improved.

3.2 Identification of the transmission behaviour with the help of the skip function of the revolution control circuit

Figure 9 shows the responses from both drive systems against time in response to an interruption which is generated by a standardised set value skip of the revolution control circuit from 0 to 5 m/min. The type of interruption is therefore equivalent to e.g. the forces during milling or turning if the cut is interrupted. In turn, the acceleration signal is recorded on the machine table. The oscillation decay times of both drives are approx. 80 ms which is quite considerable for tooling machines. This excellent dynamic system behaviour is characterised by the fact that the transmitting moments of the driven nut are absorbed by both ends of the leadscrew. The high axial and torsion rigidity of this leadscrew arrangement therefore makes a direct translation of the rotatory motor moments into axial acceleration possible without excessive storage of energy e.g. by torsion of the leadscrew. The associated translation behaviour makes the greater damping of the indirect drive clear. This means that the additional components in the drive sequence result in increased friction. The direct force transmission between the drive and the machine table and therefore the linear behaviour is therefore lost to a certain extent. The variation of the parameters of the speed control is shown in Figure 10 using the indirect drive as an example. By increasing the Kp value and the integral part (1/Tn) the phase reserve of the transmission behaviour increases. For the resonant frequency of the speed controlled drive with flange mounted mechanical parts this means that the resonant frequency shifts from 30 to 50 Hz. The dominant resonant frequency of the direct drive is 15 Hz, which is significantly lower.

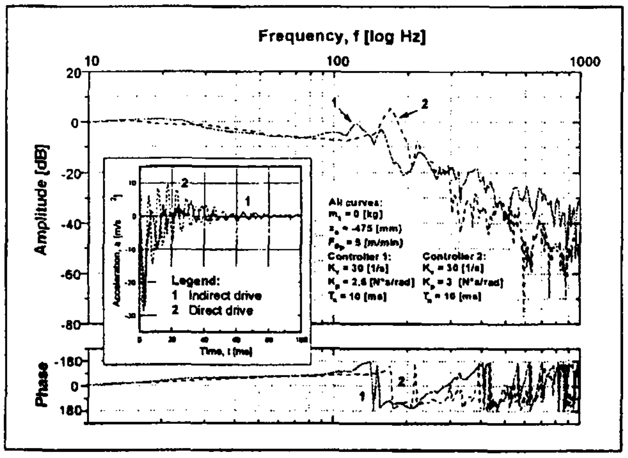


Figure 9: Skip response of both drives

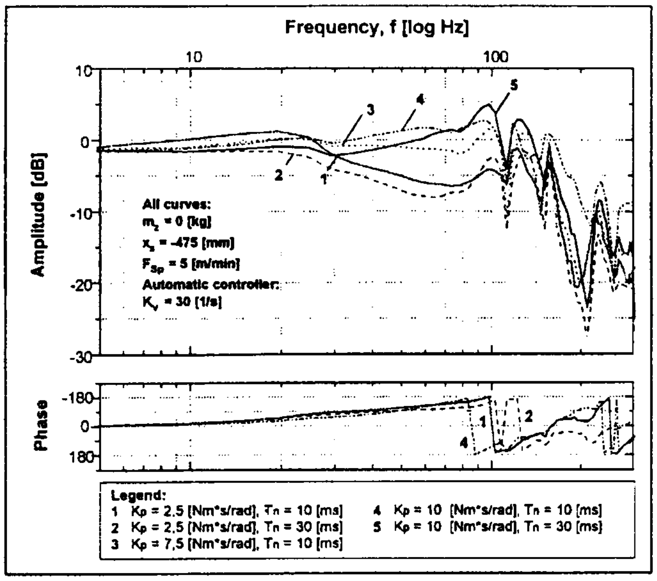


Figure 10: Variation of Kp and Tn: Indirect drive

3.3 Identification of the positioning performance of the position control circuit

The positioning performance of the position control circuit at the axis level is characterised by the difference between the set and actual positional values which is also described as the position error. The size of the position error is proportional to speed in a stationary state and inversely proportional to the velocity gain factor Kv. The increase of Kv not only results in the reduction of the position error, but also in the destabilisation of the positional control circuit which in turn results in oscillations in the set positional value of the table (Figure 11).

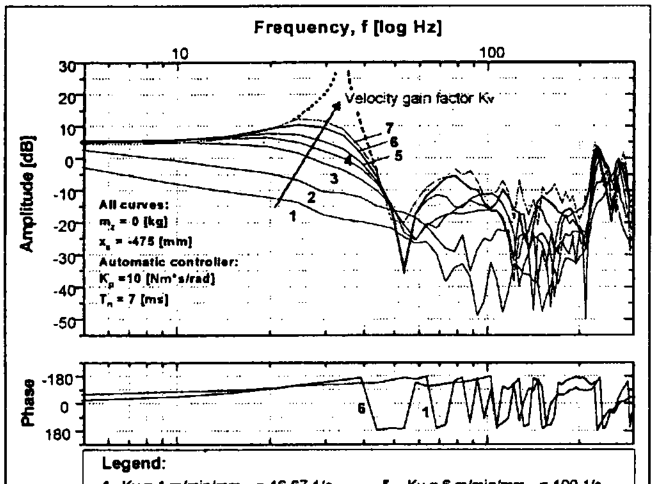


Figure 11: Variation of Kv: Indirect drive

Therefore positioning without overshoot becomes the limit for the value of velocity gain factor. For a specific positioning process this fact means that the traverse duration is significantly reduced without having to forfeit any accuracy. Figure 12 shows this clearly for both drives, with a positional jump of 400 mm and a speed of 80 m/min. After a time of 0.5 s, the table reaches its position window of 2 µm. The variation of speed amplification Kv makes the limit for positioning without overshoot very clear. The values arrived at lie between 60 to 75 1/s. Despite the low resonant frequencies of the direct drive, similar high values can also be achieved. The known factors of the resonant frequency between speed and positional control are virtually eliminated by the low non-linearity in the transmission behaviour of the direct drive.

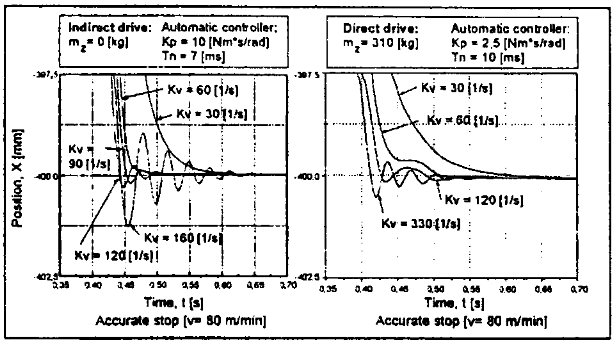


Figure 12: Variation of Kv: Accurate stop of both drives

4 SUMMARY

The feed system opens up new possibilities for the development of new dynamic machine generations based on the tried and tested technology of the ball screw drive. With traverse speeds of up to 120 m/min achieved at high acceleration and system costs comparable to those of a conventional feed axis, this technology represents an interesting alternative to existing systems. Due to the high mechanical resonant frequencies which are mainly independent of the table position, the frequently limiting element is improved significantly. This results in virtually linear transmission behaviour with low parameter fluctuations. In order to be able to achieve even better control dynamic behaviour in the future, it is necessary to reduce the inertial mass of the motors and the mechanical components which in turn increases the resonant frequency of the controlled drive. The thread pitch of the ball screw drive and the translation ratio of the drive sequence influence these greatly. Whilst only the first measure can be realised in the case of a direct drive, both measures are applicable to the indirect drive. A reduction in the thread pitch also reduces the traverse speed as a consequence. By increasing the number of revolutions of the drive speed which can be achieved without problems by increasing the bending critical number of revolutions, means that in turn, high traverse speeds are possible. In addition greater positional control processes can improve the positioning performance of the optimised transmission distance. The performance of this process is characterised mainly by the parameter stability of the transmission distance [5].

5. REFERENCES

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