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FLEXIBLE COUPLING OF DRIVE AND GUIDE ELEMENTS FOR PARALLEL-DRIVEN FEED AXES TO INCREASE DYNAMICS AND ACCURACY OF MOTION

Productivity enhancement is achievable by increasing the dynamics of machines. This can be accomplished by decreasing the moving mass or by increasing the driving forces. One possibility for increasing the drive forces lies in the use of multiple parallel-acting actuators. Because of the mechanical coupling between the drives, undesirable interference of the actuators occurs. This limits the control bandwidth of the feed axis. Feed axes of machine tools are mostly equipped with linear guides fitted with rolling elements. Since they dispose of just one degree of freedom, small errors in their alignment or thermally induced errors cause constraining loads. In this paper compliant mechanisms are used to mechanically decouple the slide from the guide and drive elements. The application of flexible joints in gantry axes with linear motors and ball screws is investigated. The dynamic behaviour of the feed axis components are modelled using the Finite Element Method. By applying the modal reduction technique, the dynamic behaviour is included in an elastic multibody simulation. With the mechanical decoupling of the drive and guide elements it is possible to increase the control bandwidth of the system. Deviations in motion may also be corrected with parallel-driven feed axis by applying compliant mechanisms.

1. INTRODUCTION

One key aim in the development of machine tools and similar machine systems is to increase their productivity whilst improving or maintaining their accuracy of motion. Amongst other methods, an increase of productivity is achievable by increasing the dynamics of the machines. This can be accomplished by decreasing the moving mass or by increasing the driving forces. One possibility for increasing the drive forces lies in the use of multiple parallel-acting actuators. The drives are mechanically coupled by the moving machine structure. A well-known representative of such systems is the gantry drive. Apart from improving the dynamics of the feed axis, gantry design is commonly used in machine tools because of achieving thermal symmetry and increasing rigidity.

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Additionally, skewing of the Tool Centre Point (TCP) can be avoided or compensated if position dependent process or inertia forces are present. Therefore the accuracy of motion can be improved compared to single-drive servo systems. In highly dynamic machine tools, the gantry architecture is used to realise the “Drive at the Center of Gravity” (DCG) principle [1]. By applying the driving force at this most shock-insensitive position of the moving structure, the excitation of vibrations due to acceleration/deceleration is kept to a minimum. Nowadays the prevalence of gantry type machines for high speed metal cutting has increased. This indicates that the investment costs are at least counterbalanced by the increase of accuracy and productivity. Furthermore, by using parallel-acting actuators the drives can be downsized which also means that the inverter losses may be reduced. Additionally two motors can be used to drive a ball screw axis, which is typically and sufficiently controlled by only one drive [2]. This so called both-side drive has a better dynamical behaviour and the potential to eigenmodes damp that appears with adding a second motor.

Because of the mechanical coupling between the parallel-acting actuators, undesirable interference of the actuators occurs. This limits the possible control loop settings and thus, the control bandwidth. Therefore dynamic trajectory deviations increase. Feed axes of machine tools are mostly equipped with linear guideways fitted with rolling elements. Since they dispose of just one degree of freedom, small errors in their alignment or thermally induced errors cause constraining loads and, therefore, lifetime is significantly reduced. Additionally, friction forces will also increase. The same problems arise if ball screws are used. Thus one has to ensure small tolerances in assembly of the guide and drive elements, in order to avoid mechanical stress.

According to the current state of the art, there are no flexible joints used in commercial machine tools for the purpose of mechanical decoupling of parallel-acting actuators. In order to avoid constraining loads, high accuracy of manufacturing and assembly of machine components is required. For machines with more than one actuator per axis, angular movements about the axis perpendicular to the direction of motion can be realised by using appropriate command values. This correction-functionality is not commonly used in machine tools. In [3] this capability is demonstrated for the correction of motion errors of a parallel-driven slide. The Institute of Machine Tools and Control Engineering (TU Dresden) developed an experimental machine with correction functionalities [4]. In [5] a system for the dynamic sag compensation of a large boring machine is presented. The system comprises a central radial-axial bearing, on which the headstock is rotatably mounted. The rotation of the headstock-carriage is effected by the two Z-drives. A similar system described in [6] uses elastic coupling elements. The drive force for adjusting the inclination of the supporting beam (RAM) is applied by an additional electromechanical actuator. In [7] the dynamics of dual-drive servo mechanisms are investigated with respect to the influence of elastic couplings of the actuators of the gantry. By applying compliant mechanisms the constraining loads of the guiding-system are reduced.

In this paper, flexible joints are used to mechanically decouple the parallel-acting actuators from the guide and drive elements in order to increase the control bandwidth of the system. Additionally, mechanical stress is reduced and deviations in motion can be corrected by using the additional degree of freedom. In gantry-type machine tools with a very high structural stiffness, the first eigenmode (yaw mode) is mostly dominated by the stiffness of the guiding system. This mechanical eigenfrequency limits the control bandwidth of the gantry stage. By applying flexible joints this mode is shifted in the lower frequency range. The increase of control bandwidth is possible because the controller can easily compensate this coupling-frequency (yaw mode) of the mechanical system.

In the following Chapter 2, an overview of control strategies for parallel-driven feed axes is given. The compliant mechanisms are presented in Chapter 3. A feed axis with a single slide in gantry design equipped with ball screws as well as with linear motors is investigated. The corresponding simulation model is briefly described in Chapter 4. Findings from simulations are presented in Chapter 5.

2. CONTROL-PRINCIPLES OF PARALLEL-DRIVEN FEED AXES

The simplest control strategy for parallel-driven feed axes is the parallel synchronous control method. With this method, also known as Master-Master-Control (MMC), the drives receive the same motion command and are controlled independently without considering the motion of the other. This control scheme is widely used in practice and cascaded position and velocity control loops are applied mostly in machine tools. Based on a rigid body model of a gantry-type test setup, an adaptive MMC is derived to minimise the tracking error as well as interaxis offset error [8]. In order to optimise motion accuracy of the MMC in [9] an adaptation of the control gain of the velocity controller of a gantry machine is introduced. Findings from simulations showed that with this method the tracking error can be reduced. The Cross-Coupled-Control (CCC) is also a MMC with an additional crosswise acting controller, which uses the position errors in order to improve synchronisation of the coupled drives. The structure and tuning algorithms of the additional controller can be found in [10]. In [11] a CCC for a SMD-assembly machine in gantry-architecture is used, equipped with two ball screws. Findings from experiments revealed that the synchronous error with CCC compared to independent axis control is significantly smaller. Another control principle is the Master-Slave-Control (MSC). The master drive gets the command values from the interpolator. The reference values of the slave drive are the actual values of the master drive. Because of the unavoidable dead time the system tends to oscillate. One extension of the MSC is the so called Relative-Stiffness-Control (RSC) [12]. In this method the position controller is only employed in the master, and independent velocity loops are used. An additional synchronous controller adds current values to the respective drives to compensate velocity differences between the master and slave drive. In [13] this principle is used to control a linear motor driven gantry-type mechanism. The mechanical coupling is identified to parameterise the synchronous controller. In the same manner in [14] the coupling between two ball screws is modelled and taken into account to design a synchronous compensator. In [15] the above mentioned control principles for coupled drives: MMC, MSC, RSC and CCC are investigated and compared by simulations. Additionally, a good overview on the corresponding control structures is given. The CCC showed a better reference behaviour and the usage of the RSC tends to result in better disturbance behaviour.

There are many more than the presented control architectures for parallel-driven feed axes in existence. However, the practical implementation of advanced control structures with industry standard components is limited, since the access to the machine's control is restricted [9]. In this paper the MMC is implemented for feed axes with linear motors and ball screws in gantry-design. By using compliant mechanisms, the drives are mechanically decoupled to increase the control bandwidth. Therefore one is unaffected by hardware restrictions. In addition no complicated model is required for control design.

3. COMPLIANT MECHANISMS FOR DECOUPLING DRIVE AND GUIDE ELEMENTS OF FEED AXES

Flexible joints and compliant mechanisms are state of the art, especially in precision engineering applications and microsystems. They are used as guidance for precision mechanisms and microelectromechanical systems (MEMS) without friction, backlash and wear [16]. However, only small motions can be achieved from compliant mechanisms. By combining standardised linear guide and drive elements (e.g. linear guideways and ball screws) with suitably designed elastic coupling elements, constraining loads can be reduced and control bandwidth can be expanded.

The elastic coupling elements are designed to obtain low stiffness in the desired direction and maximum stiffness in all other directions. Different types of compliant mechanisms are developed and presented in this paper. They comprise spring steel sheets as an elastic element to ensure high strength. One advantage of spring steel sheets as a main component of the flexible joints is that they can be simply produced, e.g. by laser cutting. In Fig. 1 (left) a rotational joint and its corresponding bending behaviour are shown. Only one relatively thick (t_{sheet} > 1 mm) spring steel sheet as elastic element is used and thus high stiffness c_z in Z-direction is reached.

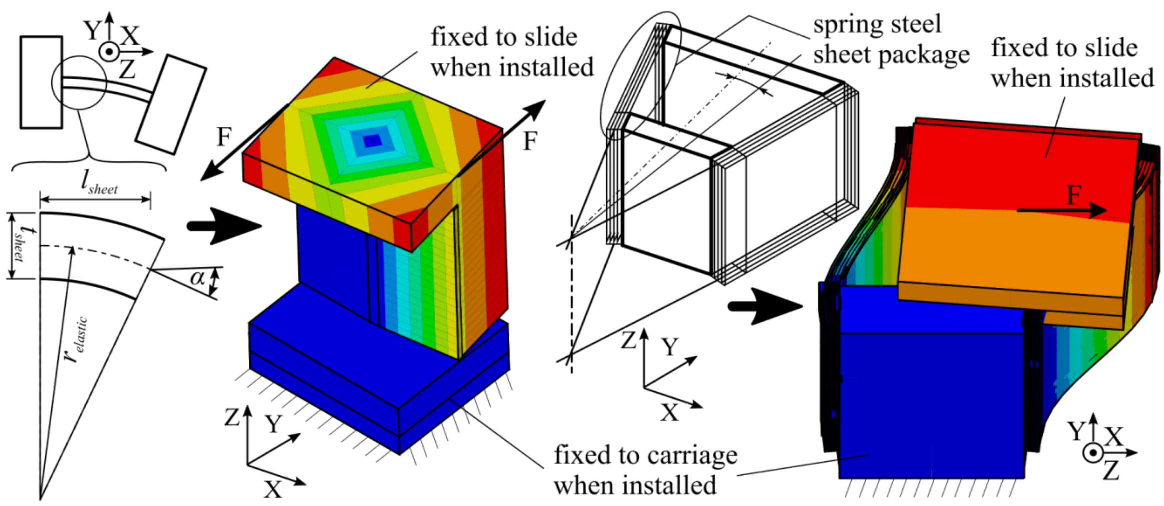

Fig. 1. Bending element used for the rotational joint (left) and trapezoidal arrangement of the spring steel elements for the rotational-thrust joint (right) as well as their bending behaviour

In order to adjust the pivot of the compliant mechanisms, trapezoidal arrangement of the spring steel elements is used. The developed rotational-thrust joints are displayed in Fig. 2, right. These flexible joints apply thin (t_{sheet} < 1 mm) spring steel sheets packaged in order to reach minimum stiffness in the transverse direction (bending, where sheet thickness t_{sheet} enters the bending stiffness in the third power) and maximum stiffness in all other directions (tension/compression and shear, where total thickness of the package $n \cdot t_{sheet}$ is proportional to the stiffness value). By variation of the number and/or thickness of the spring steel sheets in the package, the stiffness can easily be adjusted. In Fig. 2 (top) the scheme of the parallel-driven feed-axis with flexible joints is illustrated. The slide (dotted line) is shown from above with the drive and guide elements beneath it. The two investigated variants of the arrangement of compliant mechanisms are shown: variant 1 with a combination of rotational joints as well as rotational-thrust joints and variant 2 with only rotational-thrust joints. These variants could be integrated into machining centres like DMG MORI's *NV 4000 DCG*. As a reference, a model, according to variant 1, but with rigid "blocks" instead of the flexible joints, is used (var. 0, not depicted).

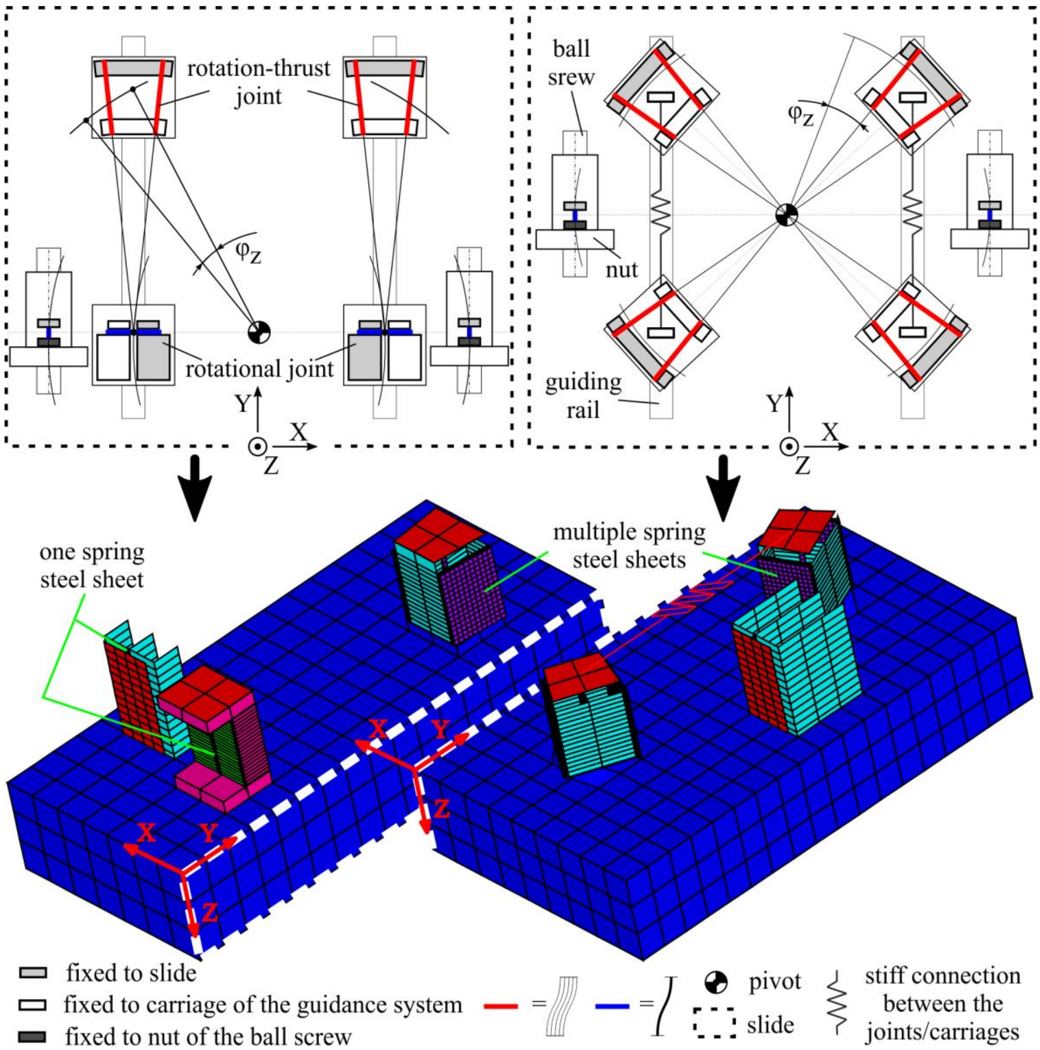

Fig. 2. Compliant mechanisms with rotational-thrust joints and rotational joints – variant 1 (top left) and with rotational-thrust joints only – variant 2 (top right) as well as the corresponding finite element models (bottom)

In Fig. 2 (bottom) the corresponding FE-models of the slide and elastic joints are shown. For better visibility only half of the model is depicted for var. 1 (left) and var. 2 (right).

Fig. 3 illustrates the constructions of the investigated parallel-driven feed axes with linear motors (left) and ball screws (right). From Fig. 3 it can also be seen that due to the mechanical coupling through the slide the linear motors (left) and the ball screws (right) are strongly linked. This coupling is cancelled by the application of flexible joints between the guiding systems, the drives and the slide respectively, compare Fig. 2.

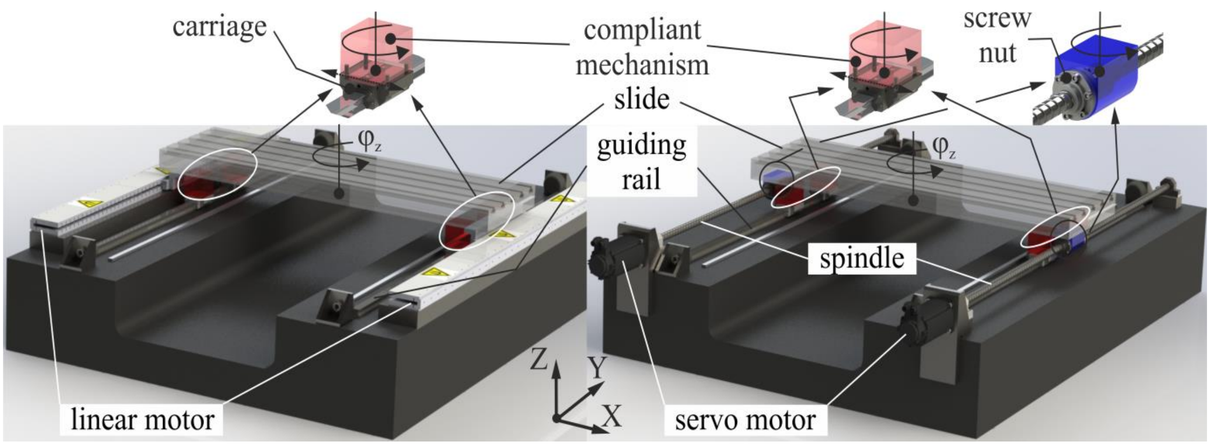

Fig. 3. Construction of parallel-driven feed axes with linear motors (left) and with ball screws (right) as well as the arrangement of the compliant mechanisms

4. MODEL OF THE GANTRY FEED AXES

The dimensions of the observed slide of the gantry (250 kg) are 800 × 450 × 84 mm. The distances between the drives and the guiding rails are 0.68 m and 0.5 m respectively. For the flexible joints applied to the carriage and to the nut (for the gantry with ball screws), 2 mm thick spring steel sheets are used. The rotational-thrusts joints comprise of ten 0.5 mm thin spring steel sheets per package (for var. 1 and 2). The slide, together with the flexible joints, and the threaded shaft (diameter: 32 mm, pitch: 20 mm, length: 2 m) of the ball screws, were modelled with ANSYS®. By applying a modal order reduction technique, the dynamic characteristics of the components were included in an elastic multibody simulation carried out with MATLAB/Simulink®. A rigid base is considered. The axial stiffness of the ball screw contact is 410 N/µm and the stiffness of the guides in X-direction is 720 N/µm. Just viscous friction is considered. The bandwidth of the current control loop is set to be approximately 1 kHz. The implemented PI velocity controller has the following form:

$$G_{PI}(z) = \frac{K_p + K_p \cdot \left({T_0}/{T_n} - 1 \right) \cdot z^{-1}}{1 - z^{-1}}, \tag{1}$$

where: T_0 is the sampling time (125 μs), K_p is the proportional gain and T_n the integral time constant of the velocity controller. Inherent disturbances always exist due to inevitable nonlinear dynamics, such as mechanical coupling [13]. In practice the individual properties of the measurement units, the mechanical components and the servo motors of each side of the gantry are not the same. For this reason parameter deviations are implemented in the model so that coupling effects appear. The corresponding model parameters are listed in Table 1. Aside from the aforementioned, the control loops are identical.

Table 1. Model-parameter deviations for each side of the gantry

	Viscous friction of the guiding system	Viscous friction of the ball screw	Force/moment constant of the motors	Motor coil resistance
Side 1	+10 %	+10 %	+5 %	+7 %
Side 2	-10 %	-10 %	-5 %	-7 %

5. SIMULATION RESULTS

5.1. GANTRY FEED AXIS WITH LINEAR MOTORS

In this section the results from the investigation of the feed axis with linear motors are presented. Since linear direct drives are used as actuators, no mechanical transmission elements exist. The obtained dynamic behaviour of the compliant mechanisms and the slide from numerical analyses in ANSYS® were firstly compared with the elastic multibody simulation. Since no other elastic bodies are present, this can easily be done in ANSYS® by elastic mounting the flexible joints of the gantry system in accordance to the stiffness of the guideway. For the feed axis with linear motors, no flexible joints to decouple the drive components are necessary. The dynamic characteristics observed in the multibody simulation matches with findings from ANSYS®. In Fig. 4 the eigenmodes obtained in ANSYS® are also illustrated in the Bode diagram of the mechanical system. Without flexible joints (var. 0) the first eigenmode of the feed axis is a yaw mode (see Fig. 4). This mode is most prevalent and originates from the rigid body rotation of the gantry constrained by the stiffness of the guiding system. This frequency can be manipulated by applying compliant mechanisms between the carriages of the linear guides and the slide. Because of the significant reduction of the rotational stiffness around the Z-axis $c_{\varphi z}$ (var. 2) and the stiffness in X-direction (var. 1) of the system, the decoupling of the guide elements from the slide is achieved. As can be seen from Fig. 4, the frequency of the first eigenmode (yaw mode) of the system without flexible joints is decreased from 225 Hz to 8 Hz and 13 Hz for variant 1 and 2 respectively. By using flexible joints, the overall stiffness of the systems is slightly reduced. Therefore the frequency of the pitch mode at 670 Hz is reduced to approximately 410 Hz for both variants with compliant mechanisms, as shown in Fig. 4. For variant 1 an additional eigenmode at approximately 270 Hz occurs. The system without flexible joints and variant 2 have a similar mode in the same frequency range.

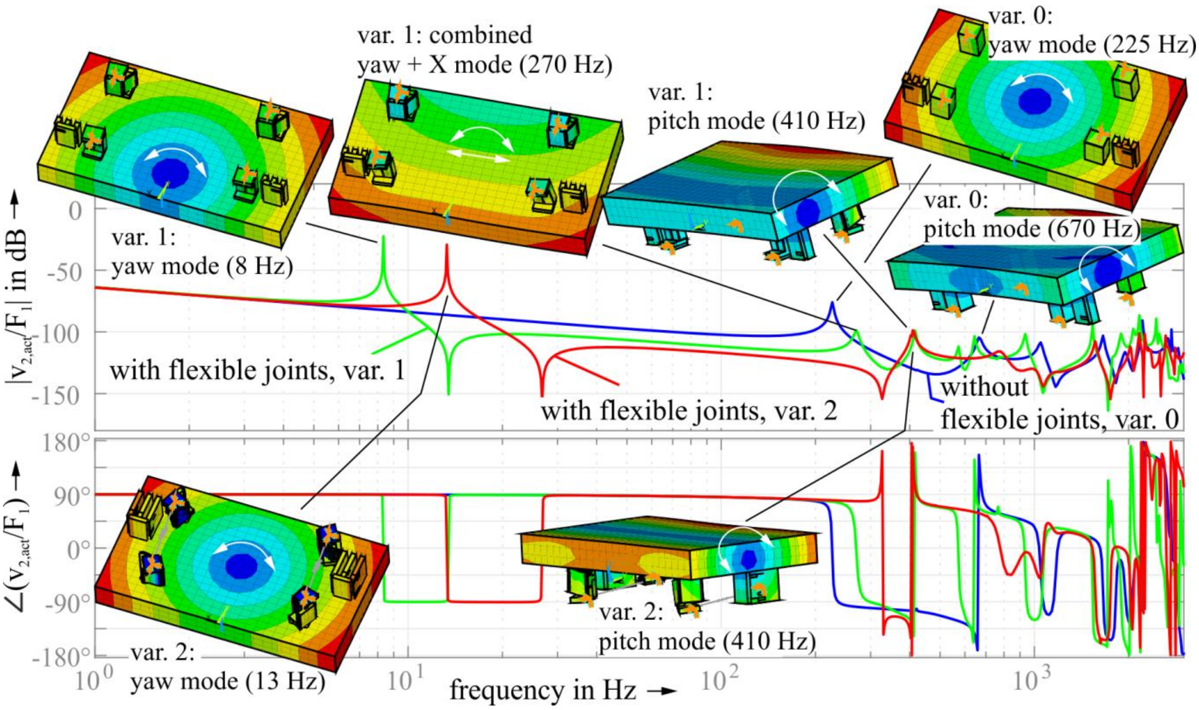

Fig. 4. Bode diagram of the dynamics and the corresponding modes for the different variants (linear motors)

However, for these systems no such eigenmode appears in the frequency response diagram since these systems are symmetric and the mode is a vibration which is perpendicular to the direction of motion and therefore cannot be excited. The frequencies of the rotational modes around the X-axis are filtered by using notch filters with appropriate centre frequencies (410 Hz for variant 1 and 2, 670 Hz for the system without flexible joints; compare Fig. 4) in order to increase the control parameters. Since the gantry-system is still coupled by the control loops, tuning of the parameters of the controllers in the frequency domain is challenging. Therefore controller tuning is performed in the time domain heuristically. Hereto a velocity step is used to investigate the system response. The gains K_p of both controllers are set the same and are increased until they reach the ultimate gain values, at which the actual velocity has stable and consistent oscillations. In order to verify the stability limits, the open loop behaviour of the coupled drives with the so obtained ultimate gain is investigated. For this purpose the loop of controller 1 is closed (reference velocity is set to 0) whilst the loop of controller 2 is kept open and vice versa. In order to find the critical value of $T_{n,crit}$, the ultimate gain $K_{p,crit}$ was halved and the mentioned procedure was repeated. This was carried out for all variants in the same manner. From Fig. 5 it is clear that the application of compliant mechanisms does not affect the stability behaviour of the gantry axis. It can be seen that the resonances at low frequencies for the variants with flexible joints do not limit the adjustable controller settings. It is also evident that the critical values of $K_{p,crit}$ (Fig. 5, left) and $T_{n,crit}$ (Fig. 5, right) are found to be precisely the same using time domain analysis since no gain or phase margins can be observed by regarding Fig. 5. Because of the mechanical decoupling of the drives, the ultimate gains $K_{p,crit}$ of var. 1 and var. 2 are significantly higher and the critical integral time constants $T_{n,crit}$ are much smaller than those of the system without flexible joints. With flexible joints, one drive can easily follow the commanded velocity almost independently for a wide frequency range whilst the other drive holds its velocity. It is coupled through the higher eigenmodes of the system. However, these modes lie in a higher frequency range and therefore can be filtered out. If the stiffness of the moved structure is reduced, the influence of other modes has to be considered and could limit the control bandwidth of the system. By improving the construction of the gantry system, exciting of the pitch mode can be avoided. The point of the driving force input just has to be moved to the “pivot” of this mode. For this case, and especially for var. 1, it is expected that the control bandwidth could be further increased. The integral time constants of the velocity controllers were set to $T_n = 2 \cdot T_{n,crit}$.

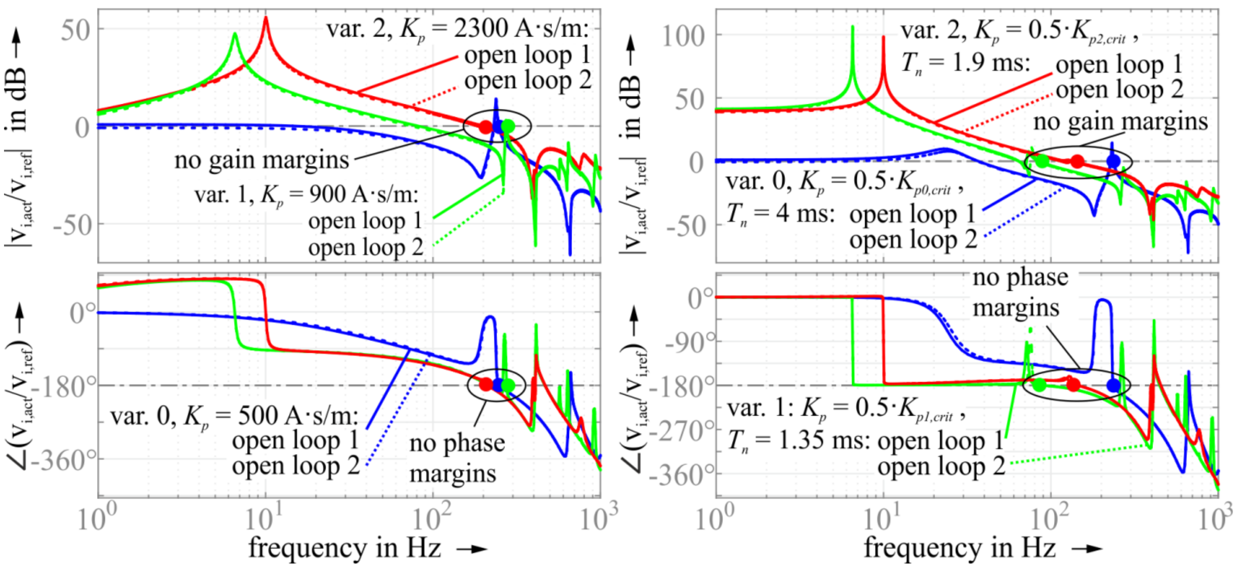

Fig. 5. Bode diagram of the open loop transfer functions of the velocity controller for the ultimate gains of K_p (left) and the critical values of T_n (right) for each variant

With these parameters, the closed loop behaviour of the velocity controller, shown in Fig. 6, left, is achieved. From Fig. 6, left can be seen that the amplitude of the closed loop transfer function is considerably high. Therefore the amplitudes of the closed loop are restricted by a static limit of 1.2943 dB. This corresponds to a PT_2-system with a damping of 0.5 to achieve a small overshoot [17]. In order to realise this restriction for variant 0 the integral time constant T_n was increased to 28 ms and K_p was kept the same. For comparability T_n was set as well to 28 ms for the variants with compliant mechanisms, and the gains were reduced to decrease the amplitude. Although the gains of variant 1 and 2 were reduced, the values for K_p are still significantly greater than which of variant 0 (see Table 2). This leads to a wider bandwidth of the closed velocity loop as can be seen in Fig. 6, right.

It should be noted that optimisation of the control parameters is not the intention of this paper. Just the influence of the compliant mechanisms on the achievable control parameters of the gantry system is of interest. The proportional gains of the position controller were tuned to realise that the amplitude of the closed loop not exceeds 0 dB.

The Bode diagram of the closed position loop is shown in Fig. 7. It is important to note that velocity feedforward control was not active. Also, the control bandwidth of the position controllers for the variants with compliant mechanisms is much larger. This implies that dynamic trajectory deviations will decrease and the disturbance behaviour is improved compared to the gantry system without flexible joints. The achieved control parameters for all variants are listed in Table 2.

Table 2. Critical proportional gains and integral time constants (index: *crit*) and tuned parameters (no index) of the velocity and position controllers of the gantry system with linear motors

	$K_{p,crit}$ in A·s/m	$T_{n,crit}$ in ms	K_p in A·s/m	T_n in ms	K_v in 1/s
Variant 0	500	4	250	28	100
Variant 1	900	1.35	450	28	150
Variant 2	2300	1.9	700	28	220

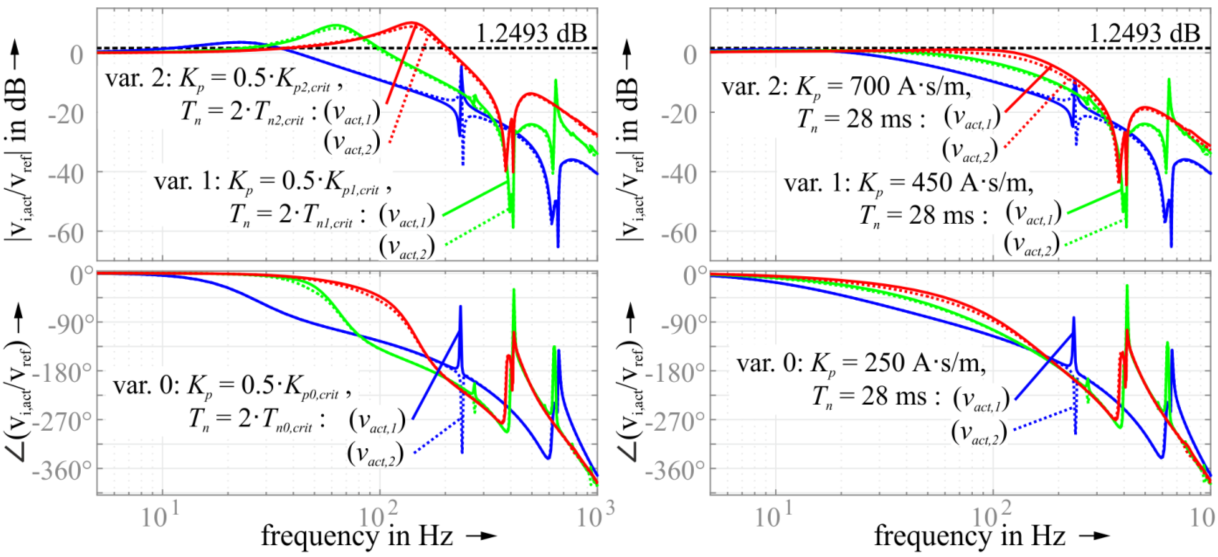

Fig. 6. Bode diagram of the closed loop transfer functions of the velocity controller for tuned values of T_n and K_p (left) and for adjusted values of T_n and K_p to the amplitude by a static limit of 1.2493 dB (right) for each variant

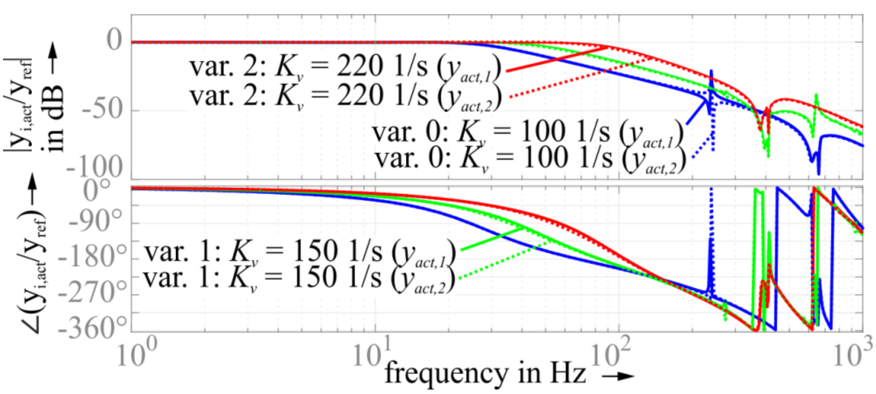

Fig. 7. Bode diagram of the closed position loop for each variant (linear motors)

5.2. GANTRY FEED AXIS WITH BALL SCREWS

The dynamic characteristics of the feed axis with ball screws, illustrated in Fig. 8, are obtained by applying a torque τ to one screw of the gantry and observing the velocity of the other side. The frequencies are shifted in a lower frequency range because of the presence of the ball screws, the drive couplings and the bearings. It is evident that with flexible joints the frequencies of the yaw modes are also significantly reduced.

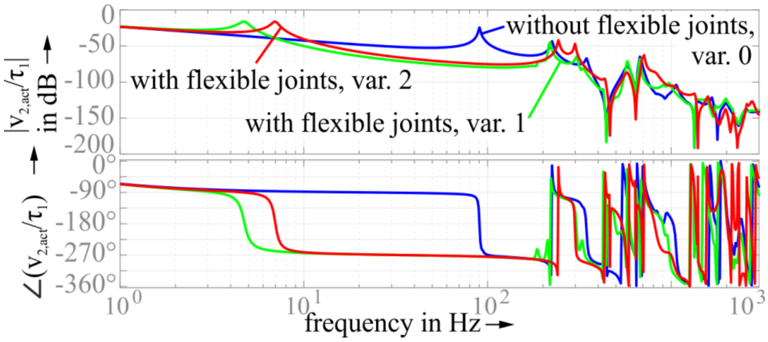

Fig. 8. Bode diagram of the mechanics of the different variants (ball screws)

In order to obtain the control parameters of the twin drive system with ball screws, the procedure described in section 5.1 is used. The reference behaviour is illustrated in Fig. 9. In Table 3 the achieved parameters of the control loops with and without flexible joints are compared. The possible control loop settings can be significantly increased when compliant mechanisms are used. Therefore the control bandwidth can be expanded as seen in Fig. 9, thus leading to a decrease in dynamic trajectory deviations.

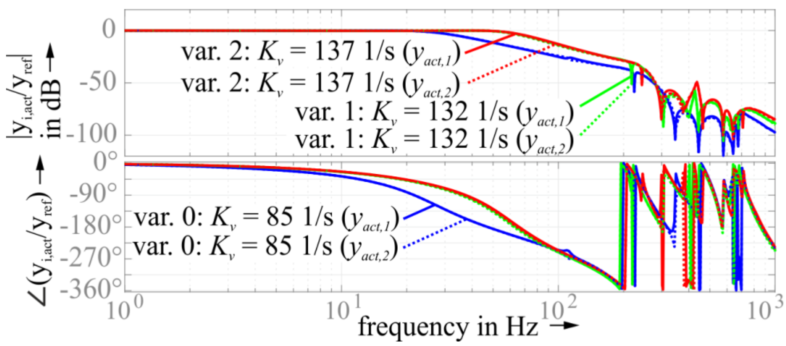

Fig. 9. Bode diagram of the closed position loop for each variant (ball screws)

Table 3. Critical proportional gains and integral time constants (index: *crit*) and tuned parameters (no index) of the velocity and position controllers of the gantry system with ball screws

	$K_{p,crit}$ in A·s/m	$T_{n,crit}$ in ms	K_p in A·s/m	T_n in ms	K_v in 1/s
Variant 0	215	5	107	27.5	85
Variant 1	690	3.5	185	27.5	132
Variant 2	580	3.25	192	27.5	137

5.3. CORRECTION FUNCTIONALITY OF GANTRY FEED AXES WITH FLEXIBLE JOINTS

Multiple parallel-acting drives in a feed axis can be used to realise correction functionalities. Fig. 10 shows the frequency behaviour if a rotation φ_z around the Z-axis of the slide is required. With flexible joints a dynamic rotation of the slide is possible.

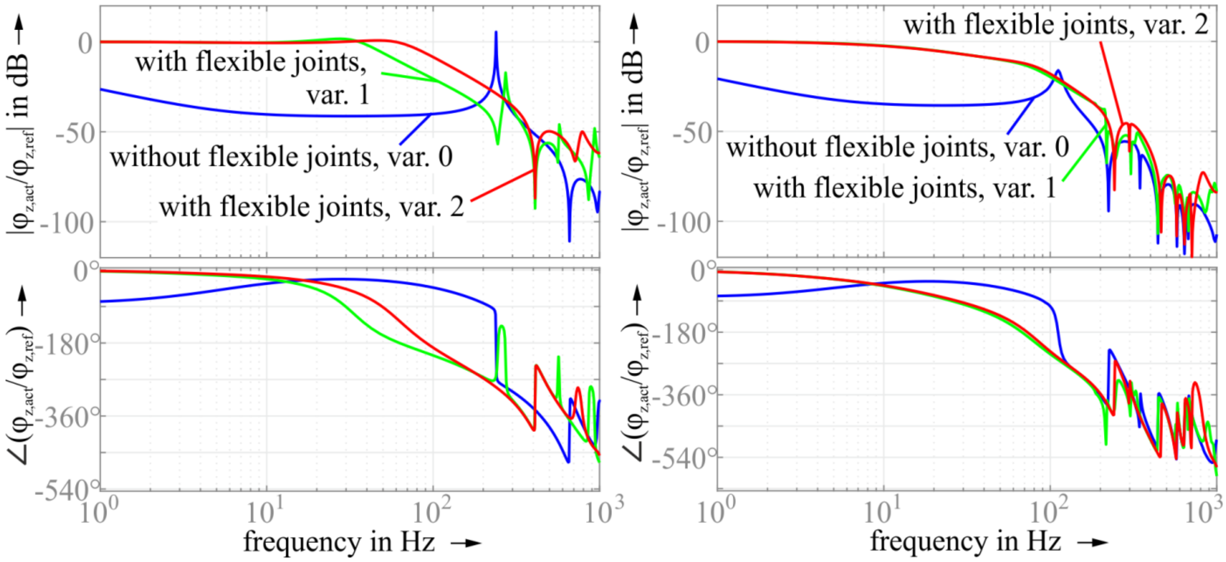

Fig. 10. Bode diagram of the yawing behaviour of the feed axis with linear motors (left) and ball screws (right)

Compared to the gantry system with ball screws shown in Fig. 10 (right), the gantry system with linear motors achieves a wider bandwidth (see Fig. 10, left). It clear that without the compliant mechanisms, the yaw mode is excited; compare Fig. 10 with Fig. 4 and 8. It has been demonstrated that parallel-driven feed axis with flexible joints can be applied with excellent results to correct deviations in motion. Since in this case two drives are available the additional degree of freedom can be used to compensate yaw deviations.

6. SUMMARY

By using compliant mechanisms to decouple the drive and guide elements from the slide of a gantry-type feed axis, the control bandwidth could be significantly increased. This could be shown for gantry systems with linear motors or ball screws. Findings from simulations demonstrated that the increase of control bandwidth of the feed axis with linear motors is greater than with ball screws. With flexible joints the proportional gains of the position and velocity controllers can be more than doubled for the investigated feed axis with linear motors. With the ball screw axis, the proportional gains can be increased by more than one and a half times. Therefore the reference and disturbance behaviour are greatly improved for gantry systems with compliant mechanisms and it was shown that parallel-driven feed axes with flexible joints can be used to correct deviations of motion.

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