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Kinematically Coupled Force Compensation—Experimental Results and Advanced Design for the 1D-Implementation †

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Abstract: Typically, the feed dynamics of machine tools are limited to reduce excitations of machine structure oscillations. Consequently, the potential increase in productivity provided by electrical direct drives cannot be exploited. The novel approach of the Kinematically Coupled Force Compensation (KCFC) combines the principles of redundant axes and force compensation to achieve an increase in the machine's feed dynamics. Because the drive reaction forces are directly applied to the machine frame, they cancel out each other perfectly if the relative motion at the Tool Centre Point (TCP) is split according to the mass ratio of the slides. In this paper, the principle of KCFC is introduced briefly and possible improvements in the design of machine structures and control are presented. The results of experimental investigations obtained by means of a 1D-KCFC Test Bed illustrate the effectiveness of the principle. Moreover, a further increase of the compensation quality can be achieved by decoupling the force flow from the machine frame, by means of elastic elements. Finally, an outlook on future research with reference to the 1D-implementation as well as possible applications of the KCFC in highly productive processes is given.

Keywords: linear motor; control; compensation; feed drive; redundant axis

1. Introduction

A further increase in the productivity of motion-guided machines, especially machine tools and handling equipment, can be achieved by means of higher feed dynamics [1]. This is usually accompanied by an increased vibrational excitation resulting from higher drive reaction forces. If vibrations occur, in extreme cases, the required quality of motion and thus the quality of manufacturing and handling, respectively, cannot be guaranteed. With reference to machine tools, various methods for the reduction of vibration excitation resulting from highly dynamic feed motions have been developed and investigated. A comparison of the jerk decoupling [2] (also known as pulse decoupling, see Figure 1a), force compensation [3] (see Figure 1b), floating principle [4] (see Figure 1c), and Kinematically Coupled Force Compensation [5] (KCFC, see Figure 1d) can be found in [6]. These principles facilitate the reduction of drive reaction forces, which may cause an excitation of the machine frame. Since the machine frame forms a multi-mass oscillator, it can be simplified by the lumped mass model with the parameters m_{base}, c_{base} and b_{base}.

Only the floating principle [4] and the KCFC [5] are able to reduce the energy consumption for motion generation with the same or even increased feed dynamics [6]. Both methods use a redundant axis configuration, whereby the relative motion required for the machining process is generated between the two slides moving in opposite directions. In KCFC, the motion is distributed within the control system according to the mass ratio K_{m} in Equation (1), which has to be known as precisely as possible, since it determines the quality of force compensation according to Equation (2).

$$K_{\mathrm{m}} = m_{\mathrm{B}} / m_{\mathrm{A}} \tag{1}$$

$$m_{\mathrm{A}} \times a_{\mathrm{A}} = m_{\mathrm{B}} \times a_{\mathrm{B}} \tag{2}$$

For the Floating Principle, the division of motion inevitably results from the mass ratio. The Floating Principle is superior to the KCFC in terms of energy consumption due to the use of only one drive [6]. In contrast, it requires a very high design effort for realisation in more than one feed direction. What both methods have in common is that they should be used for compact and rigid slides, which are designed according to the "Drive at the Center of Gravity" (DCG) feed principle [7], since they cannot reduce the vibrations of the moving assembly itself. For compliant slides or column structures, mechatronic systems [8], methods of model-based compensation of vibration [9], or active vibration damping, for example using Active Damping Devices (ADDs) [10], may be applied. ADDs also offer the possibility to dampen self-excited vibrations, especially regenerative chatter [10].

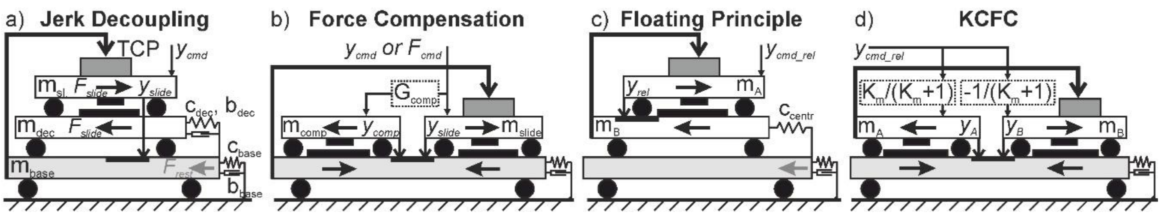
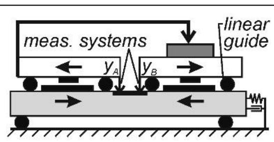
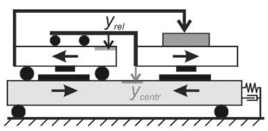
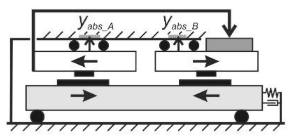


Figure 1. Principles for reduction of vibrational excitation of highly dynamic machine tools [11].

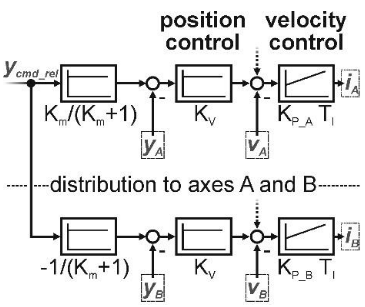
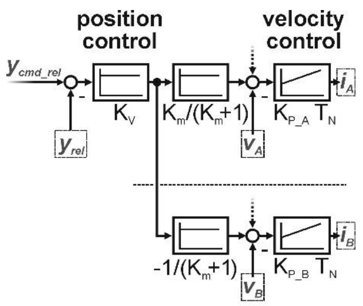
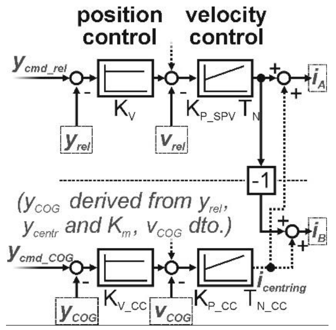
Aside from the reduction of drive-induced vibrations, the redundant axis configuration of KCFC offers the possibility to realise alternative arrangements of the guiding and measuring systems—relative between both slides or relative to an external (frame-independent) reference [5,12]. Possible design modifications are listed in Table 1. The realisation of guiding and measuring systems relative to the base frame is regarded as state of the art (RB in Table 1, left). The relative guidance between the slides (RS Table 1, middle) offers higher rigidity due to proximity to the machining process. A relative measuring system (y_{rel} in Table 1, middle) may enable a reduction of measuring errors (e.g., Abbe errors) and measured vibrations (for instance frame vibrations). In principle, guiding and measuring systems may also be realised based on a frame-independent, external reference RE ($y_{\mathrm{abs\_A}}$ and $y_{\mathrm{abs\_B}}$ in Table 1, right).

Table 1. Implementation of guiding and measuring systems for redundant axis configurations [11].

	Relative to Base Frame (RB)	**Relative between Slides (RS)**	**External Reference (RE)**
arrangement of linear guides and linear measuring systems			

Analogous to relative guidance and measuring, superimposed control loops for the relative position y_{rel} (SP in Table 2) as well the relative position y_{rel} and velocity v_{rel}, respectively (SPV in Table 2), can be implemented [5,12]. These superimposed controllers have to be supplemented by an additional controller for the common motion of the slides in relation to the frame ("centring control" for y_{centr} and y_{COG}, respectively, shown on the example of SPV in Table 2). Centring control acts on the common centre of gravity (COG) y_{COG} of both slides, which normally does not move for KCFC under ideal conditions. The superimposed control system facilitates the separate parameterisation of the control loops for relative motion and centring and may enable a higher control bandwidth. In simulative investigations on control with frame-independent reference ($y_{\mathrm{abs\_A}}$ and $y_{\mathrm{abs\_B}}$ in Table 1, right), it was found that this type of measuring system could not be operated in the stable range [12]. In addition, the external reference structure would be difficult to build.

Table 2. Conventional (A) and alternative controller structures (SP and SPV) for the Kinematically Coupled Force Compensation (KCFC) axis configurations.

Axis Control (A)	Superimposed Position Control (SP)	Superimposed Position and Velocity Control (SPV)
		

In this paper, only the axis-based control (A in Table 2), combined with guiding and linear measuring systems relative to the base frame (RB in Table 1), are considered for the 1D-KCFC with linear motion. The dynamic behaviour (response to set point changes and disturbance reaction) of this combination is similar to that of two independent single axes.

Further possible kinematic designs, especially for the KCFC with more degrees of freedom (DOF), can be found in [5].

KCFC is primarily intended for applications with negligibly small process forces (e.g., micro milling, HSC milling, pick and place applications, etc.), where the excitation caused by drive reaction forces is dominant [13]. The principle aims to reduce dynamic excitations of the underlying machine structure (usually the frame), caused by the drive reaction forces, which may induce vibrations at the Tool Centre Point (TCP) via the machine's kinematic chain. Furthermore, KCFC provides a higher overall feed dynamic for the process compared to a single axis by adding the feed dynamics of the two redundant slides. In comparison to the single axis, the kinetic energy of the motion and thus the electrical and mechanical losses can be reduced by KCFC [6]. An improvement of the control quality may be achieved if the excitation of structural eigenmodes, which are critical for the controller, can be suppressed by the force compensation effect of KCFC.

In [5] and [6] KCFC was introduced and the theoretical basics were given. First simulative results obtained by analysis of a lumped mass model were discussed in [12]. Results of the simulative analysis with an extended elastic Multibody Simulation (MBS) model can be found in [14]. In this paper, experiments with variation of the mass ratio K_m are presented and analysed in the time and frequency domain. In addition to the original paper [11], simulative analyses using the extended FE model from [14] are carried out in order to investigate the combination of KCFC with an additional decoupling of the force-conducting structural components (comparable with jerk decoupling in Figure 1a).

2. Test Bed for the Experimental Investigation of 1D-KCFC

2.1. Mechanical Design

In the case of compact machine tools, the first dominant eigenmodes of the machine frame typically occur in the range of 100–150 Hz. Moreover, there are rigid-body-oscillations of the entire machine on the base in the range of some 10 Hz. The developed 1D-KCFC test bed (see Figure 2a) emulates these dynamic properties. For this purpose, the base frame is mounted with spring steel sheets on a heavy cast plate (foundation). The number of sheets and the coupling to an adjustable bending spring at the front side of the frame (see Figure 2a) allow for the variation of the frame's eigenfrequencies. In the present configuration, the dominant natural frequency in the Y-direction is approximately 26.7 Hz. The design of the two slides as a gantry configuration makes it possible to apply the drive forces in their centre of gravity according to [7]. Highly dynamic ironless linear motors with a peak force of F_{max} = 1400 N (Tecnotion AL3806N, Tecnotion GmbH, München, Germany) are used.

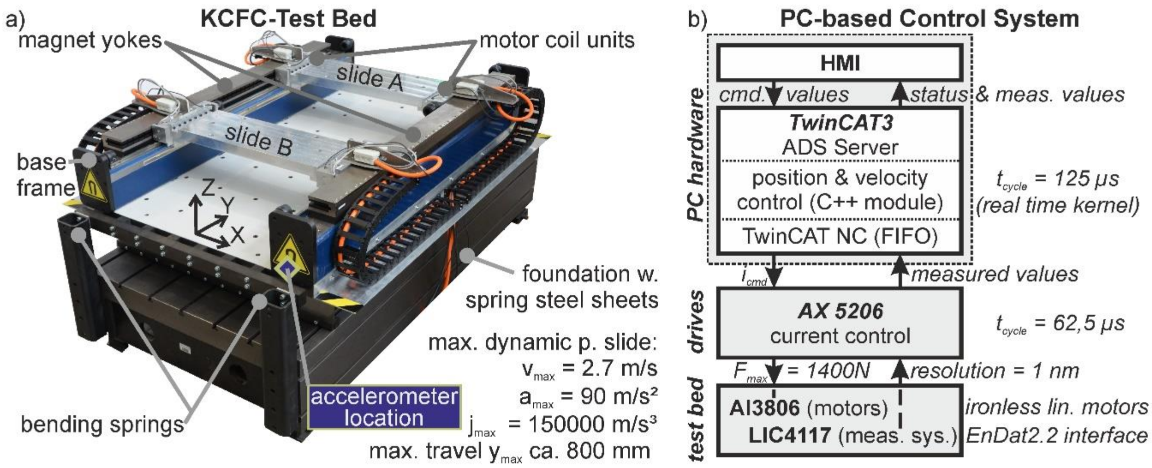


Figure 2. Mechanical design (**a**) and control system (**b**) of the 1D-KCFC test bed [11].

2.2. Control Concept for the 1D-KCFC Implementation

In order to implement the KCFC-specific superimposed controller cascades shown in Table 2, the control of the test bed was implemented within the PC-based control system TwinCAT3 from Beckhoff (see Figure 2b). The trajectory generation and data recording are carried out by the HMI-software. In the real-time capable control core provided by TwinCAT3, own controller cascades (compare [5,12,14]) have been integrated as code in C++. The motor current is controlled by the drive control unit (Beckhoff AX5206, Beckhoff Automation GmbH & Co. KG, Verl, Germany), which also evaluate the high-resolution absolute optical measuring systems (Heidenhain LIC4117, DR. JOHANNES HEIDENHAIN GmbH, Traunreut, Germany) via an EnDat2.2 interface.

2.3. Experimental Characterisation and Modelling of the 1D-KCFC Test Bed

During the assembly process of the test bed, extensive measurements using a frequency analyser PULSE Modal 3560C (Brüel & Kjaer, Bremen, Germany) were performed for later adjustment of the dynamic behaviour of the simulation model. Initially, the eigenmodes and eigenfrequencies of the freely suspended subassemblies (slide and frame) were measured with five accelerometers in each assembly step. Due to the high frequency resolution of 31.25 MHz, damping ratios could be determined precisely.

The modelling of the mechanical components of the test bed was carried out in ANSYS® (version 18.2, Pittsburgh, PA, USA). Geometry description, meshing, solution, and data export were programmed as APDL (ANSYS Parametric Design Language) script. Meshing was realised with linear hexahedral elements (SOLID185) and shell elements (SHELL181) for the spring steel sheets.

Rigid coupling elements (MPC184) were used for load distribution at the coupling nodes of slide and frame. The distributed application of drive forces to the linear motor was carried out as a boundary condition with the RBE3 command. In order to facilitate the model adjustment, assembly joints were parameterised with separately adjustable modulus of elasticity for normal and tangential direction. On the basis of the experimental modal analyses, a successive model comparison of the FE models was carried out.

Finally, modal substituted systems were derived from simulative modal analyses of frame and slide by means of a modal reduction. These reduced order models were integrated into the block simulation in MATLAB/Simulink® (version R2017a, Natick, MA, USA) and finally parameterised with the damping coefficients determined experimentally. The elastic structure components were connected by movable spring-damper elements facilitating a continuously position-variable elastic multibody simulation. This elastic MBS model was supplemented with simplified models of the drives and the controller cascades to enable a holistic simulative analysis of the 1D-KCFC test bed [14].

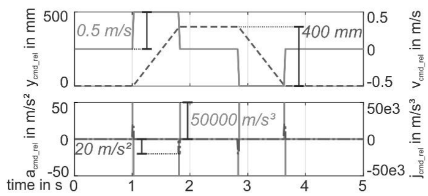
3. Experimental Investigation of the Effectiveness of the KCFC on the Test Bed

3.1. Experimental Procedure and Evaluation in the Time Domain

A highly dynamic motion profile (see Table 3, right) was used for the experimental investigation of the 1D-KCFC. As a reference case, this motion profile was realised by a single slide (parameter set I in Table 3). The parameter set IIa represents the KCFC with a mass ratio of $K_m = 1$ and the set IIb shows the reference case where only slide B moves ("half travel range"). The parameter sets IIIa and IVa show the KCFC with mass ratios unequal to one. Though the gain factor K_{P_B} of the velocity control loop has been adapted to the changed mass m_B, in sets IIIb and IVb, the gain factor K_{P_B} was not adjusted to induce deviations between the control loops of slide A and B. For all axes, the position controller gain was set to K_V = 20 1/s with 100% velocity feed-forward. The integral time constant in the velocity controller was set to T_I = 3 ms.

Table 3. Parameter sets (left) and motion profile (right) for the experimental investigation of 1D-KCFC with Axis Control (A) according to Table 2 [11].

Par. Set	m_A	m_B	K_m	K_{P_A}	K_{P_B}	Motion Profile (Relative Motion)
I	not used	30 kg	1	50 As/m	50 As/m	
IIa	30 kg	30 kg	1	50 As/m	50 As/m	
IIb	not used	30 kg	1	50 As/m	50 As/m	
IIIa	30 kg	38 kg	1.26	50 As/m	63.3 As/m	
IIIb	30 kg	38 kg	1.26	50 As/m	50 As/m	
IVa	30 kg	51.3 kg	1.71	50 As/m	85.5 As/m	
IVb	30 kg	51.3 kg	1.71	50 As/m	50 As/m	



All other parameters of the test bed and the drives can be found in the Appendix A (Table A1). In all experiments, the frame vibration was measured with an accelerometer at the base frame (for sensor location, see Figure 2a).

In Figure 3, the results of the experimental investigations are presented. In the case of variants with KCFC, the distribution of motion according to the mass ratio K_m is clearly recognisable by the distribution of the relative motion y_{cmd} to the slide motions y_{cmd_A} and y_{cmd_B}, according to Equations (3) and (4).

$$y_{cmd\_A} = (y_{cmd\_rel} \times K_m)/(K_m + 1) \tag{3}$$

$$y_{cmd\_B} = (-y_{cmd\_rel} \times 1)/(K_m + 1) \tag{4}$$

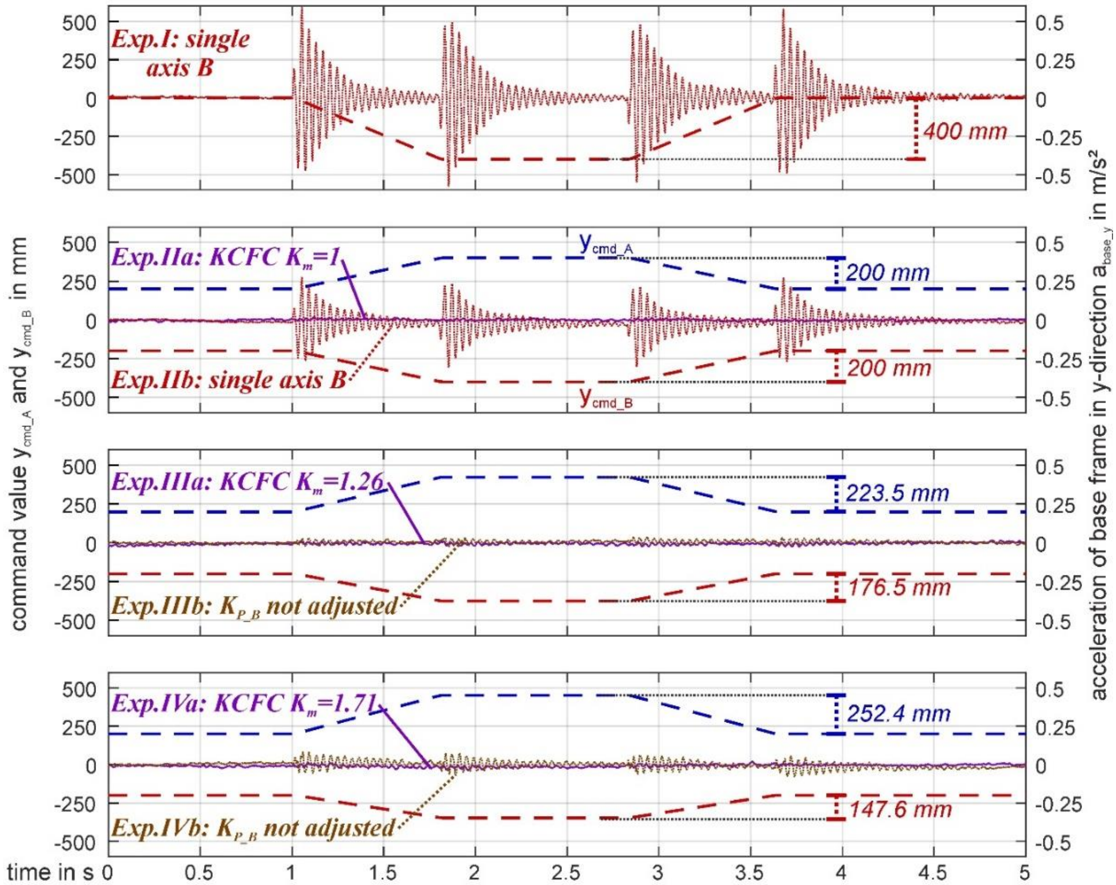


Figure 3. Experimental results for the parameter sets I to IVb as shown in Table 3 [11].

For the uncompensated cases in set I and set IIb the frame oscillation of 26.7 Hz is clearly visible. This corresponds to the frame vibration as it would occur during operation of a comparable single axis. When generating the relative motion according to the KCFC-principle (set IIa, IIIa, and IVa), the frame vibration is not excited, even under mass ratios unequal to one. However, if the control loops are not matched to each other (K_{P_B} not adjusted), a slightly visible excitation of the frame vibration occurs in set IIIb while the excitation is clearly visible in set IVb. Accordingly, the KCFC is robust against small parameter deviations of the moving mass in the range of less than 20% (Set IIIb).

Basically, it is possible to calculate and monitor the uncompensated (residual) forces based on the measured motor currents, assuming a constant motor force constant K_{Mot}. Similarly, the frame acceleration may be observed by means of an accelerometer. In processes with unknown or variable moving masses, it is conceivable to identify these by means of an identification algorithm, e.g., based on the algorithm used in [15]. Alternatively, it would be possible to derive the mass ratio from the machining simulation in the CAM system and store it in the NC program. Basically, K_m can be continuously adjusted in the controller during the execution of the motion program. Extended KCFC-algorithms including the identification and continuous updating of the mass ratio will be the subject of future research.

Since the velocity amplification factor K_V = 20 1/s is relatively low for a linear direct drive, it has to be increased for future experiments. The low value of 20 1/s is presumably due to the implementation in the PC-based control system of the test bed, where unconsidered delay may occur, as a result of the implementation and data transfer within the PLC-program (see Figure 2b). To achieve a higher control bandwidth, the entire cascade control (see A in Table 2) can be shifted into the drives' integrated controller. This is not possible for the advanced control concepts SP and SPV.

3.2. Evaluation in the Frequency Domain

The analysis of the recorded acceleration–time curves in the frequency domain also confirms that the KCFC does not excite the dominant frame eigenmode at 26.7 Hz. This analysis has also shown that there is only a slight excitation of higher frequency eigenmodes (see Figure 4). Presumably, the energy, brought in by the motion profile according to Table 2, is too low for a significant structural excitation. In this respect, further experiments, making use of the full dynamic potential, especially the acceleration, of the test bed (compare Figure 2a), will be carried out in future.

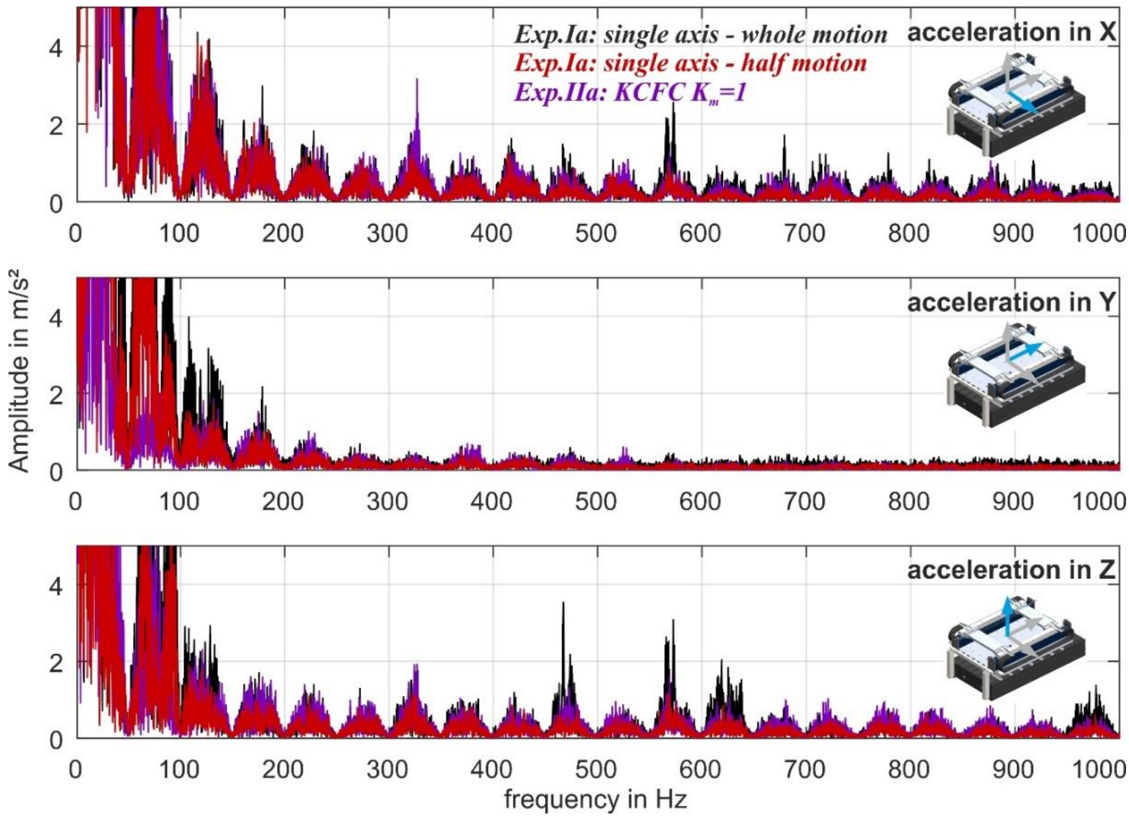


Figure 4. Frequency response of the measured acceleration–time-profiles (compare Figure 3).

Figure 4 shows some higher frequency eigenmodes. In the range from 400 to 700 Hz, some modes are less excited by the KCFC than by the motion of a single axis. Exactly the opposite is true for an eigenmode between 300 and 350 Hz. This corresponds to the frame natural frequency at 351 Hz (masses of the slides not considered, see Figure 5a), which represents the bending of the frame on the spring plates. Considering the force flow in the frame, it becomes clear that the force closure takes place far outside the neutral axis of the base frame. This results in a bending moment, which in turn can excite the frame. Thus, the force flow of the force compensation, even with the collinear drives of the 1D-KCFC test bed, can lead to an undesired structural excitation. In order to counteract this problem, an extended design for the force-conducting parts of the 1D-KCFC test bed was developed, which will be presented in the following section and evaluated based on simulative investigations.

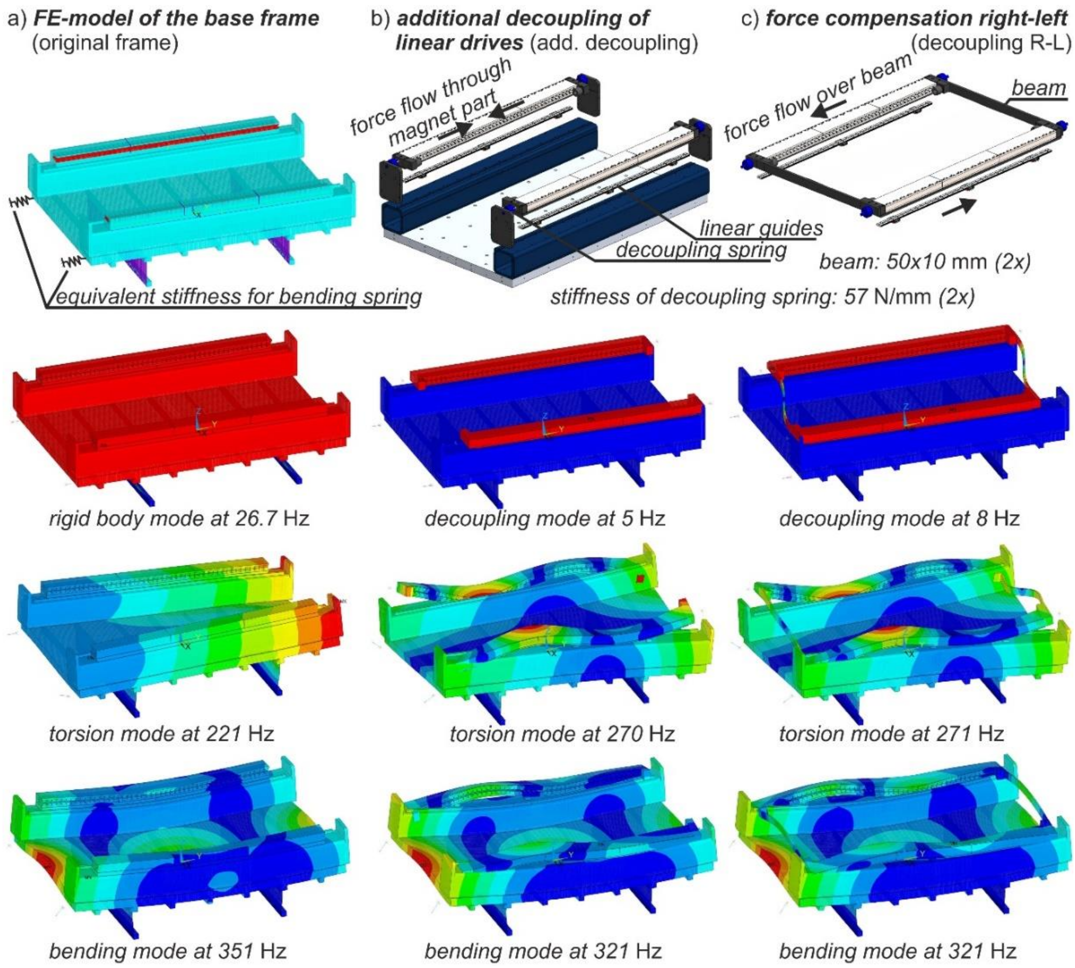


Figure 5. FE modelling of the 1D-KCFC test bed (**a**), variant with decoupling of the linear drives (**b**), and variant with decoupling for force transmission from right to left (**c**), each with selected eigenmodes.

4. Advanced Design Concepts for Optimal Force Flow in KCFC Arrangements

Figure 5a illustrates the FE modelling (see Section 2.3 and Figure 2a) of the 1D-KCFC test bed as well as three selected eigenmodes. The characteristic rigid-body mode at 26.7 Hz occurs in all frame variants at slightly different frequencies. In general, the aim of the KCFC is to reduce excitation of this frequency. The bending mode at 351 Hz is also visible in Figure 4 in the range between 300 and 350 Hz, but is shifted towards lower frequencies, due to the masses of the mounted slides.

In Figure 5b, the concept of the decoupling of the force-conducting linear motor secondary parts is depicted. These are mounted on linear guides and elastically coupled to the base frame via springs (e.g., rubber springs). The resulting mechanical low-pass filter (see decoupling mode at 5 Hz) ensures that high-frequency force components are only transmitted to the frame in an attenuated form. In addition, the force flow is concentrated in the secondary part of the linear motor, so that ideally, only tensile and compressive forces are transferred.

If it is not possible, e.g., because of the arrangement of the process, to realise a collinear force flow, the variant according to Figure 5c can be used for force transmission from the left to the right side of the test bed (decoupling mode at 8 Hz). Here the decoupling also acts as a mechanical low-pass filter, but on the resulting bending and torsion moments about the Z-axis.

In order to validate the concept of decoupling, the frequency responses for three specific load cases for the original frame (compare Figures 2a and 5a), for the frame with decoupling (Figure 5b), and for the frame with force transmission via beams (Figure 5c) were determined using harmonic analyses in the FE environment. A uniform damping ratio of $D = 0.05$ for all modes was assumed for all variants investigated. The forces are distributed over a large number of FE nodes. The vibration amplitude is measured at the location of the accelerometer (see Figure 2a). This far-outside measurement location was chosen as a representative because a large vibration amplitude at this position would typically be transmitted via the slides to the TCP.

Figure 6 depicts the vibration amplitude in Z-direction for the case of ideal force compensation (comparable with measurements shown in Figure 4). The vibration amplitudes for the variants with decoupling over a wide frequency range, but especially for the bending mode of 351 Hz, are significantly lower than for the variant without decoupling. This mainly illustrates the effect of the concentration of the force flow in the secondary parts of the linear motor.

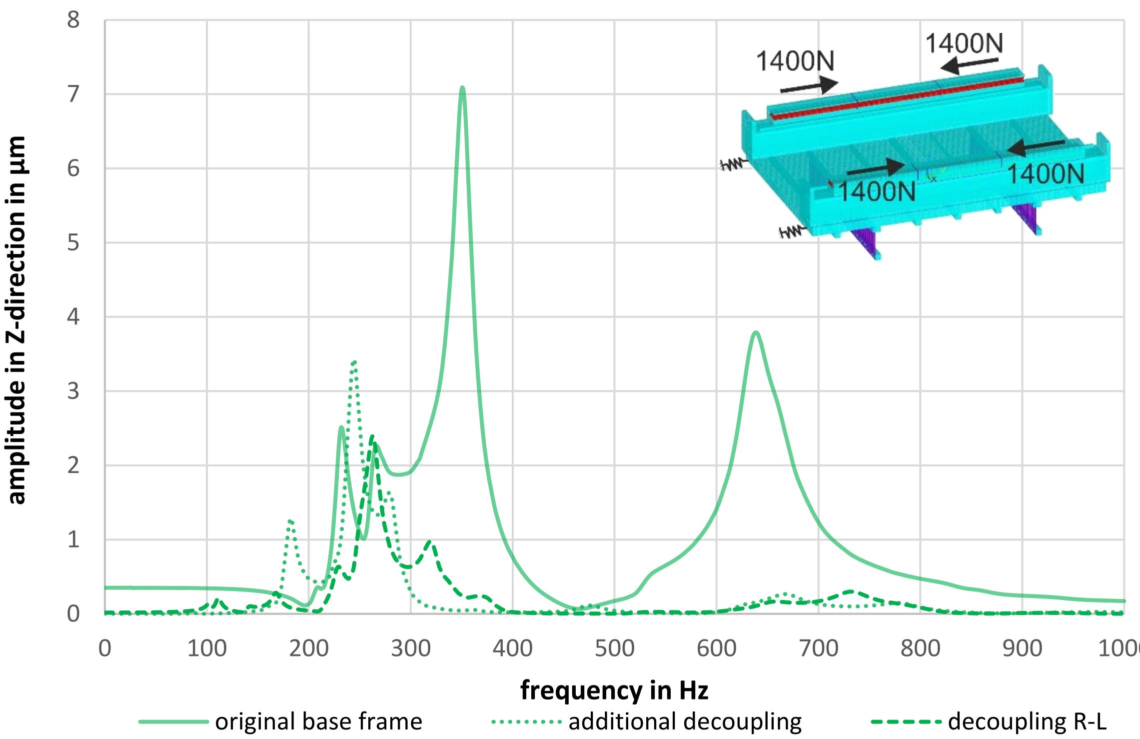


Figure 6. Frequency response of the frame vibration amplitude in Z-direction (at the location of the accelerometer, see Figure 2), excited by counteracting identical drive forces (ideal compensation).

Figure 7 shows the response to the load case of the force compensation from the right to the left side of the frame. The force is transmitted here in the form of a moment about the Z-axis, which acts on the entire frame. Therefore, the vibration excitation is significantly lower over a wide frequency range. However, the modes of decoupling at 5 Hz and 8 Hz, respectively, are clearly visible, since the energy from the excitation is concentrated in this frequency range. Within the actual realisation of such a force closure via a bending moment, it must be examined whether the eigenmode of the decoupling is critical for accuracy at the TCP or the controller, respectively. If, for example, it is a quasi-rigid body mode of the entire frame, this is not the case. Alternatively, decoupling can be implemented with highly damping elements (e.g., fibre-reinforced plastics or rubber springs). In this case, however, it must be ensured that the generated heat can be dissipated during continuous operation.

The decoupling of the force-conducting parts also offers the potential to reduce the effect of uncompensated residual forces that arise, for example, because of parameter uncertainties. Figure 8 depicts the frequency response of the vibration amplitudes in the X- and Z-direction for a corresponding load case. The representation for the variant with decoupling R–L (see Figure 5c) was omitted, since it provided comparable results with the additional decoupling (see Figure 5b). As in Figures 6 and 7, the vibration amplitude is significantly reduced over a wide frequency range. However, a stronger excitation takes place in the range of the natural frequencies of the decoupling of the linear drives at approximately 5 Hz, which could be safely reduced by larger damping values, as the residual forces are low.

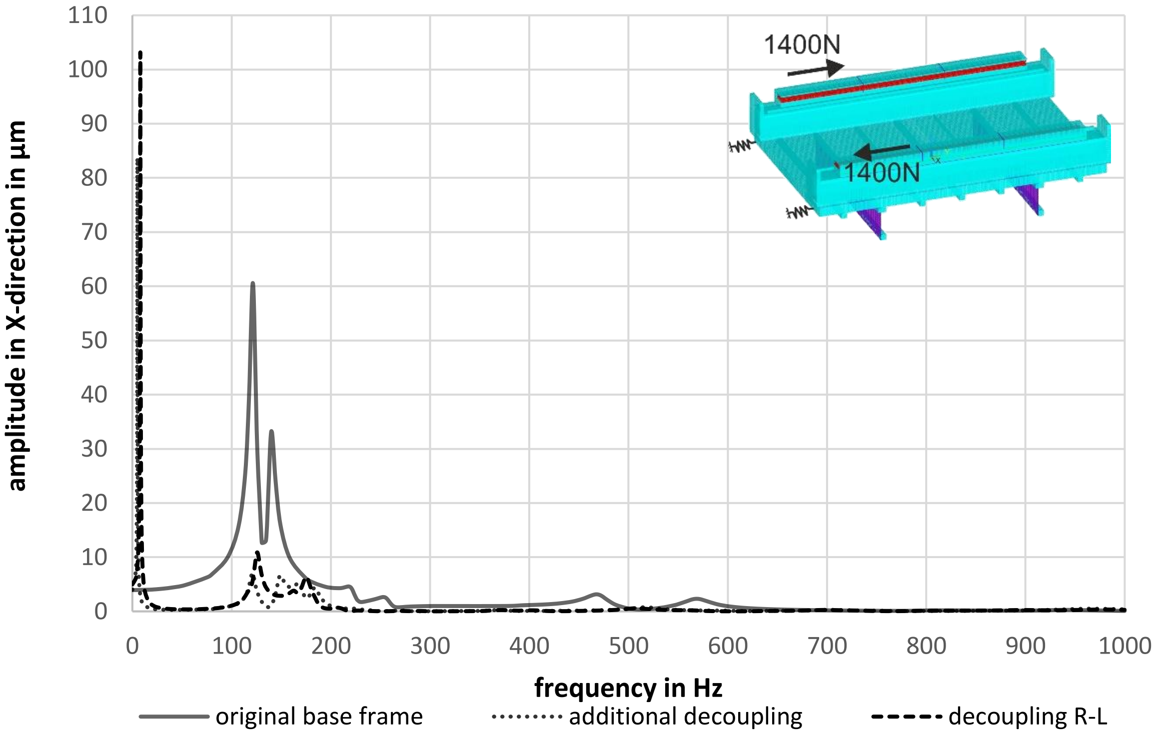


Figure 7. Frequency response of the frame vibration amplitude in X-direction (at the location of the accelerometer, see Figure 2), excited by identical drive forces acting in opposite directions on the right and left sides (compensation right–left).

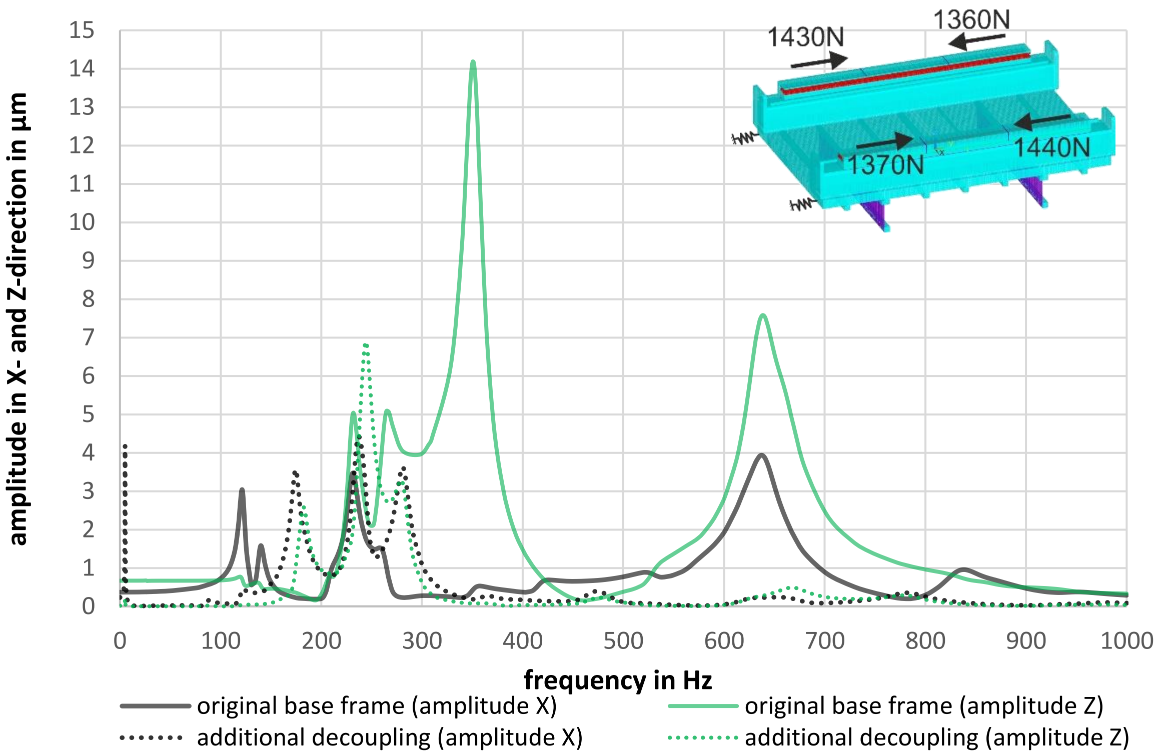


Figure 8. Frequency response of the frame vibration amplitude in X- and Z-direction (at the location of the accelerometer, see Figure 2), excited by opposing, non-identical drive forces (residual forces).

In conclusion, the concept of additional decoupling of the force-conducting parts of the KCFC is very promising, especially for frame structures susceptible to vibration. For further research, the existing 1D-KCFC test bed will be modified for additional experimental investigations accompanied by more detailed simulation-based analyses, considering the actual damping values, especially for the decoupling elements (e.g., rubber springs), and including the slides mounted on the frame.

5. Discussion

In this paper, the principle of KCFC was introduced briefly, and possible design changes for corresponding machine structures and the implementation of the control system were discussed. Mechanical design and control implementation were detailed on the example of the 1D-KCFC test bed. Based on experiments on this test bed, the compensation effect of the KCFC was demonstrated. It turned out that control loops, particularly the velocity gain K_{P}, have to be matched as much as possible in order to achieve a satisfactory compensation effectiveness. This might be achieved by automated identification of the mass ratio K_{m}.

Furthermore, based on the experimental results assessed in the frequency domain in combination with additional simulative analyses, it could be revealed that a further increase in the compensation quality of motion systems using the KCFC principle can be achieved by separating the force flow from the accuracy-relevant machine frame.

6. Conclusion and Outlook

The performed experimental and simulative investigations lead to the following conclusions:

- KCFC is quite complex and costly compared to competing principles, but seems to be suitable for processes with the highest dynamic requirements;
- The redundant axis arrangement of the KCFC enables the implementation of various structural and control concepts, which may permit an improved adaptation of the machine system to high dynamic processes with negligible process forces;
- The compensation effect of the 1D-KCFC could be verified by the experiments presented in this paper. It could be proven that the method is sufficiently robust against small parameter deviations of the moving masses (<20 %). In case of higher parameter deviations, the parameterisation of the control must be adapted (velocity gain K_{P}), which may be done by means of identification algorithms;
- With the presented approach of mechanical decoupling of the force conducting parts, a further improvement of the compensation quality seems possible, even with unconsidered parameter deviations or in a non-collinear drive arrangement.

In the near future, further investigations on the 1D-KCFC, considering additional decoupling of the linear motors, also taking into account the quality of motion achievable at the TCP, will be carried out. Experimental investigations on disturbance behaviour of KCFC axes with higher values for K_{V} will also be performed.

Beyond that, based on the findings obtained from experimental and simulative analyses of the 1D-KCFC test bed, small KCFC motion systems with at least two or more DOF (e.g., 2D-KCFC) will be developed in order to advance into the field of ultra-high dynamic motion for highly productive processes with limited travel range ($<100 \times 100$ mm^2, e.g., wire bonding process). In these highly productive applications, the effort for KCFC drive arrangement can be justified, especially if the KCFC principle is exclusively adopted for those parts of the process chain with the highest dynamic requirements.

7. Patents

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Appendix A

Table A1. Additional parameters of drives and control system of the test bed.

Parameter	Variable	Value	Parameter	Variable	Value
position control cycle	T_{pos}	125 µs	acc. feed-forward	K_{acc}	0.00
velocity control cycle	T_{vel}	125 µs	current limit	I_{lim}	11.3 A
current control cycle	T_{curr}	62.5 µs	voltage limit	U_{lim}	325 V
curr. control prop. gain	K_{I_curr}	84.0 V/A	motor resistance	R_{mot}	7.9 V/A
curr. contr. integr. const.	T_{I_curr}	0.80 ms	motor inductivity	L_{mot}	0.014 Vs/A
vel. feed-forward factor	K_{vel}	1.00	motor force constant	K_{mot}	124 N/A
vel. LP-filter	T_{vel}	0.00 ms	back-EMF constant	K_{EMF}	124 Vs/m

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