Efficient evaluation of machine concepts under hysteresis and inertia influences

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Abstract

This paper addresses the use of machine tool models that are originally created in a commercial FEM-Modelling environment for static and dynamic investigations for the integrated analysis for frictional and inertial effects. A pathway between the FEM-software and a proprietary rigid-body simulation environment has been developed for the dedicated analysis of machine concepts regarding physical design parameters. The resulting model allows specific studies of machine tools, considerably reducing computing and post-processing times. The systematic effects of inertial and frictional cross-talk are explained in detail. Examples of measured cross-talk effects and their correlation with the actual acceleration values are shown.

1 ANSYS to Rigid-Body – Pathway – GUI (Graphical User Interface)

The point of central interest for machine tools is the relative displacement between tool-tip (TCP) and work-piece [1] under special conditions, e.g. loads. Certainly, standard FEM-tools offer a wide range of analysis options, but getting the particular output required in the case of machine tools normally requires time consuming modelling and post–processing effort. In our case, models originally created in ANSYS are transferred into a proprietary analysis rigid-body environment for integrated static and dynamic investigations.

In ANSYS, the physical (stiffness and damping) properties of the coupling elements between the various bodies are defined using dedicated script templates. These coupling definitions are then transferred into the interface – GUI, being completed by inertia and configuration parameters of the various bodies directly derived from the ANSYS FEM (Finite-Element-Method) model.

Figure 1 gives an idea how a machine tool under investigation is represented. Here, physical parameters of the original ANSYS model can be modified if required.

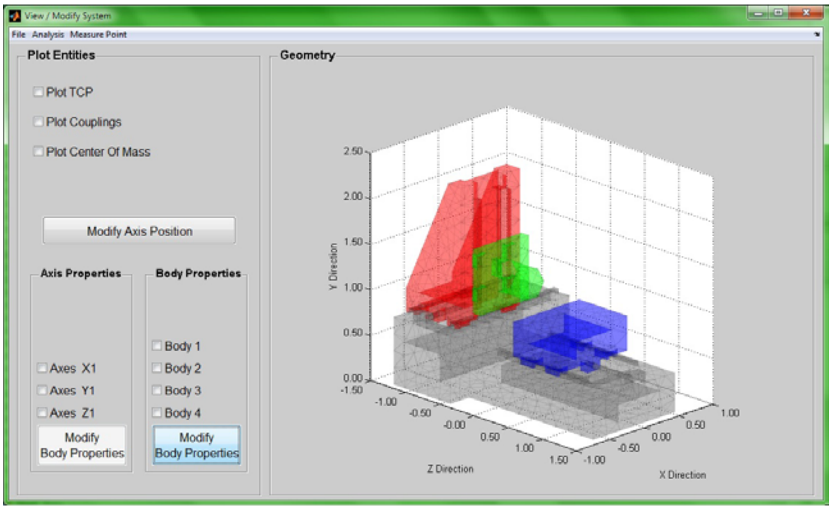

Figure 1: Machine representation in modification window of the interface - GUI

1.1 Frictional and inertial load cases

Based on the kinematic configuration of the machine tool, in addition to gravity and process loads, also kinematic load cases are defined automatically: These kinematic load cases consist of states where single axes are moved with constant velocity or with constant acceleration [2]. In the first group of load cases (constant velocities) frictional forces are derived and applied to the structure. There are two friction models which can be chosen alternatively: A viscous friction model, e.g. with constant friction forces per roller bearing element, and a “dry friction model” where the normal forces on the guideway elements due to gravity are taken into account.

In the second group of load sets drive and inertial forces are derived automatically. The location of closed loop measurement systems is also taken into account via its decisive influence on the resulting positioning properties. As output for all load cases, 3D-deviations at the TCP are obtained and post-processed automatically.

The basic mechanisms of inertial and frictional straightness deviations are explained below: Figure 2 shows the configuration parameters for inertia (top) and for friction (bottom) in 2D. The basic equations for both cases in 2D are given in (1) and (2).

Regarding the magnitude of translatoric displacements at the TCP the following factors have to be considered:

- amount of inertia F_a or friction F_f forces due to acceleration, mass and friction
- offset of the inertia ΔY_{Fa} or friction ΔY_{Ff} to the driving force
- offset ΔZ_{TCP} of the slide's centre of gravity to TCP in direction of motion
- stiffnesses $k_{Y,i}$ and $k_{A,i}$ of the guideway elements (bearings e.g.)
- offsets of the guideway elements ΔZ_i (quadratic influence!)
- tilt-stiffness $k_{rot,A}$ of the guideway-system orthogonal to direction of motion

$$EYZ_{inertia} = \frac{F_a\, \Delta Y_{F_a}\, \Delta Z_{TCP}}{k_{rot,A}} = \frac{M\, a\, \Delta Y_{F_a}\, \Delta Z_{TCP}}{\sum_i k_{Y,i} \Delta z_i^2 + \sum_i k_{A,i}} \quad (1)$$

$$EYZ_{fr.} = \frac{F_f(div)\, \Delta Y_{F_f}\, \Delta Z_{TCP}}{k_{rot,A}} = \frac{\mu(div)\, F_P\, \Delta Y_{F_f}\, \Delta Z_{TCP}}{\sum_i k_{Y,i} \Delta z_i^2 + \sum_i k_{A,i}} \quad (2)$$

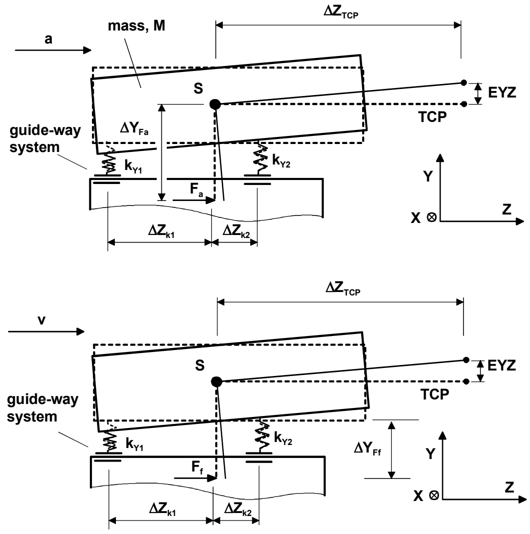

Figure 2: Schematic of the inertia (top) and friction (bottom) tilt effects on the TCP

2 Measurement examples

In measurements, the frictional [3] and inertial phenomena shown in Figure 2 can frequently be found. As illustration for the proportionality of the crosstalk to the actual axis acceleration, the acceleration values and the corresponding straightness deviations for a set of positioning movements over 100mm with varying acceleration settings are shown in Figure 3.

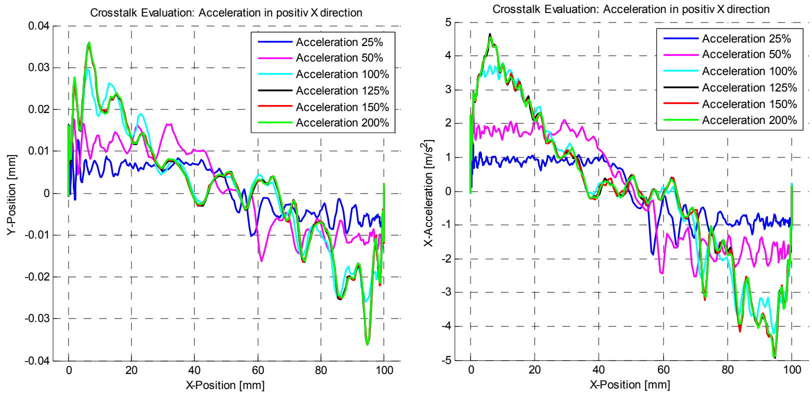

Figure 3: Crosstalk and acceleration on a set of positioning cross-grid measurements

3 Conclusion

This paper shows how a pathway between a FEM-software and a proprietary rigid-body simulation environment allows efficient specific studies of machine tools, by reducing computing and post-processing times. This work aims at representing the measured correlation between cross-talk and acceleration by integrating inertial and frictional effects in a rigid-body model of a machine tool.

This work has been funded by the Confederation's innovation promotion agency (CTI) of the Swiss Federal Office for Professional Education and Technology (OPET).

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