Th. Thümmel, J. Woyke
TU München

SIMPACK FEMBS Simulation of a Moulded Pantograph Mechanism

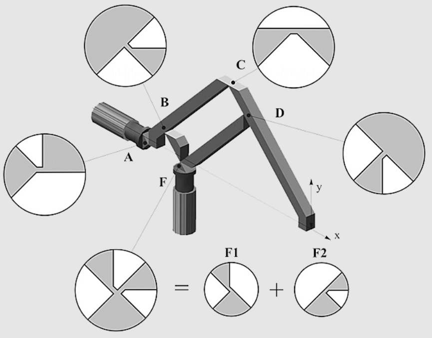

Figure 1

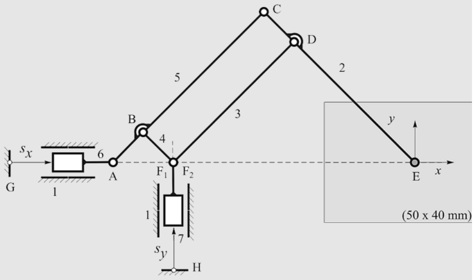

Figure 2

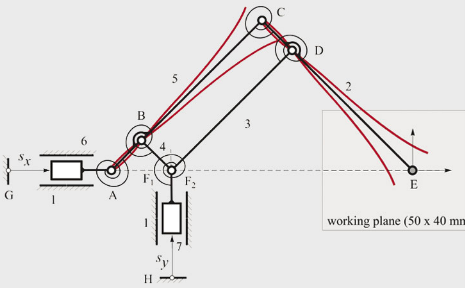

Figure 3

A miniature pantograph mechanism of uniform material, including large-deflective hinges , has been designed to assemble portable electric devices, e.g. mobile telephones. High adaptability in its functionality and usage, as well as a high-speed of up to 20 cycles per second are required. Positioning errors are to be expected due to the inertia forces, which cause the mechanism to deform elastically.

The design of the mechanism has necessitated the use of an advanced tool in its virtual prototyping. With the help of the SIMPACK interface to FEA programs, FEMBS, selected eigenmodes, calculated in ANSYS, were imported into SIMPACK, where a surface mount job was successfully modelled.

OBJECT OF INVESTIGATION

Small, portable electrical and electronic devices and computer accessories contain minute devices of less than a few square-millimetres located on electric circuit boards. However, most of the mm-sized devices are often assembled by huge surface mounting systems in the dimension of a metre.

A new surface mount system composed of parallel arranged miniature manipulators has been proposed. One miniature manipulator consists of a moulded two degree of freedom pantograph mechanism, which involves hinges, providing large deflections, (A, B, C, D, and F) and links; both of these components are made of the same material, i.e. PP (Polypropylene), as shown in Figure 1.

However this type of construction introduces new problems regarding positioning and vibrations due to nonlinear kinematics and dynamics, caused by large deflections of the hinges, elastic bending of the links and varying inertia forces. Building a model, that includes the pantograph's exact dynamics and kinematics, could create the ability to compensate positioning errors by input motion control.

Figure 2 illustrates the dimensions and simplest rigid-body linkage model of the pantograph mechanism, with the motion of the system defined via the following global transfer functions

$$x = -4\ s_x$$
$$y = \ 5\ s_y$$

MODELLING THE PANTOGRAPH MECHANISM

An enhanced model of the pantograph is built up in SIMPACK as shown in Figure 3. The hinges A to F are modelled as revolute joints combined with non-linear torsion spring-damper elements.

A static FEM analysis calculated the non-linear spring properties using elastic ANSYS-models for the hinge. The non-linear stress-strain-curve for polypropylene provided the elastic material properties. By fixing one side of the hinge and displacing the other, the reaction torques were be determined. The FEM calculation resulted in the non-linear torsion spring characteristics of the hinges, approximated as a polynomial of 5th order.

As a first estimation an approximately constant damping characteristic was available from experiments performed at Tokyo Institute of Technology P&I Lab, in the Professor Horie Group. In these experiments thin beams including one hinge capable of large deflections was given an initial deformation. The response function was then measured allowing the damping ration to be calculated.

In order to get the modal parameters of the pantograph, a modal analysis with a maximum of 300 Hz was performed using ANSYS. Fixed drives were assumed as boundary conditions, which meant the linear actuators were not displaced.

This delivered four eigenfrequencies and corresponding eigenforms, two in transverse, i.e. normal to the working plane, and two bending eigenforms. In the mechanism itself it is unlikely that there are excitations in the transverse direction, and therefore the two bending eigenforms at 94 Hz and 205 Hz were investigated, see figure 4.

The first approach investigated the two beams (link 2 and link 5) in ANSYS with the boundary conditions matching the

eigenmodes of the pantograph parts. The results were then exported into SIMPACK via the interface FEMBS.
An elastic multi-body system model shown in figure 3 was built, which delivered the eigenfrequencies of 106 and 239 Hz.

VERIFICATION BY A SPECIAL MOTION TASK

Figure 5 shows two chosen trajectories of the tool point E when surface mounted. Trajectory A is a very simple motion, where the mechanism is used to pick and place an object. Trajectory B is an advanced version of trajectory A, designed to reduce the dynamic displacement error of output point E. Trajectory B provides the required motion, but without the corners seen in the motion of trajectory A. Nevertheless, input functions s_x and s_y for both trajectories include sinusoidal connections in the time domain (no jumps in velocity and acceleration). The rigid linkage model, see figure 2, would deliver the reference without dynamic displacement error and vibrations.
The frequencies of these jobs are both 6.67 Hz i.e. the cycle is 150 ms. The input functions can be designed in many ways, however, the comparison trajectories A and B are sufficiently simple to allow the dynamic characteristics of the deviations of the tool point E to be investigated.
Figure 6 shows the dynamic displacement error of the output point E from the reference trajectory shown in Figure 5.
In the future, the elastic multi-body system will be verified experimentally. An iterative improvement of the multi-body system with FEM and experimental methods will also follow.
The import of mode shapes of elastic bodies does increase the accuracy obtained when calculating a mechanism's motion.

More information at
Thomas Thümmel, Joseph Woyke
Lehrstuhl für Angewandte Mechanik,
Technische Universität München
www.amm.mw.tum.de
in cooperation with:
Tokyo Institute of Technology
P&I Lab, Prof. Horie Group

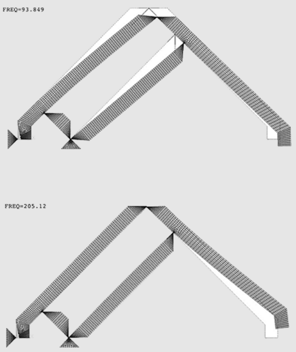

Figure 4

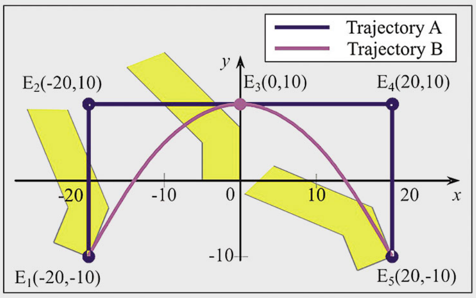

Figure 5

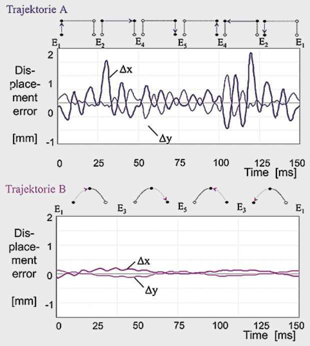

Figure 6