Journal of Machine Engineering, Vol. 9, No. 3, 2009

compensation, regulation
mechatronic system, FEM model

Miroslaw GROCHOWSKI[1]
Adam BUKOWSKI[1]

MODELLING THE COMPENSATION OF MECHANICAL AND NON-MECHANICAL DISTURBANCES IN MECHATRONIC SYSTEMS

In this paper a technique of modelling mechatronic systems is investigated. Such technique is developed in order to build an integrated model of workpiece-fixture system additionally equipped with active deformation compensation system. The analysed type of workpieces is flexible, thin-walled parts under mechanical and non-mechanical loads. The model has been implemented in ABAQUS/Standard FEM code where the behaviour of actuator drives and a control system has been emulated with user subroutines. Model simulation data is confronted against data from experimental investigations and theoretical calculations. Additional multi-domain simulations with the use of temperature-displacement calculation scheme, utilising the analogous deformation compensation method are described. Finally, results are discussed and conclusions are drawn.

1. INTRODUCTION

The appearance of a microprocessor caused the control of mechanical processes to become simpler, more precise and more flexible (“intelligent”). Up to date mechanical means realising the processing of information in machines are being replaced by electronic means; at the same time the reliability of electronic components has become sufficiently high in order to withstand vibrations and other factors of the mechanical environment. This is how a new branch of science has come to life – mechatronics – interdisciplinary area of technical sciences integrating mechanical engineering, machine building, electrical engineering, electronics and information technology [0]. Typical examples of mechatronic objects are hard drive components, modern combustion engines, industrial robots, CNC machine tools, etc. A tendency of introducing mechatronics to the areas reserved for traditional solutions is observed, such as hydraulic manipulators of engineering machines [0] or fixing devices for machining [0]. More and more frequent application of mechatronic

[1] Wrocław University of Technology, Institute of Production Engineering and Automation, ul. Łukasiewicza 5, 50-371 Wrocław, Poland, e-mail: miroslaw.grochowski@pwr.wroc.pl, adam.bukowski@pwr.wroc.pl

systems poses the need of their modelling in order to simulate their behaviour, which is especially helpful during conceptual, verification and optimisation phases of designing.

The authors concentrate on elaborating a technique of modelling flexible workpiece-fixture systems in which mechatronic components take part in correcting workpiece position errors caused by the influence of the external factors. In order to realise such task it has been decided to apply a Finite Element Method (FEM) and the main problem which has arisen was the integration of the FEM with non-mechanical elements and control system models.

An interesting tool for such purposes is Hardware Description Language with the use of which it is possible to implement the FEM [0]. Other means which enables utilising the potential of various programming tools more thoroughly is the application of various intercommunicating programming tools [0, 0]. It is also possible to create computation tools from scratch which allow realising calculations with the consideration of various object behaviour aspects [0].

The selection of a specific method depends on the task complexity level and on which parts of the modelled process pay a decisive role. Because in case of the flexible workpiece-fixture system a decisive role is played by the behaviour of the machined workpiece [0], the authors decided to elaborate a technique of computing the behaviour of the workpiece model with the use of the FEM software (ABAQUS/Standard), whereas the behaviour of the control system has been modelled from scratch with the use of FORTRAN programming language. The communication between individual models is realised by means of user subroutines in ABAQUS/Standard.

Two objects have been chosen as demonstrators of the technique for two types of disturbances – mechanical and thermal factors. In terms of elastic analysis an aluminium alloy beam has been chosen, the results of beam analysis have been verified experimentally as well as theoretically. In terms of coupled temperature-displacement analysis a simple aluminium alloy frame has been chosen, although results were not verified.

2. MECHATRONIC STRUCTURE WITH LINEAR ELASTIC MECHANICAL COMPONENTS

2.1 PROBLEM DESCRIPTION

The analysed workpiece is in form of a fixed-end aluminium alloy beam (Fig. 1) which is deflected by a linear actuator F1. Linear actuator F2 is introduced to adjust the line of deflection in order to achieve constant position of a critical point P. Both actuator drives are stepper motors. A proportional-integral (PI) regulator is used for adjusting the displacement of the compensating actuator F2, based on the error value which is in form of point P displacement – u_p. Correcting value generated by the PI regulator is interpreted as the absolute position of the compensating actuator, expressed in the number of stepper motor steps.

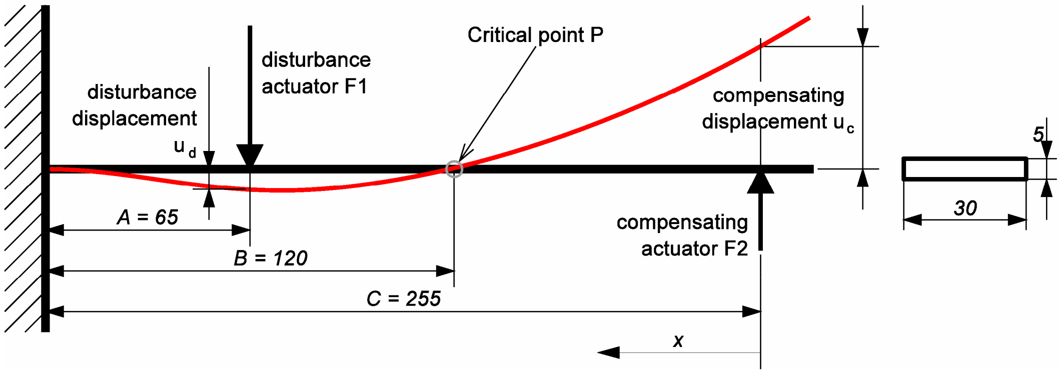

Fig. 1. Schematic view of the demonstration beam.

The entire process cycle took 40s and was divided into two stages with different speeds of disturbance changes in which the disturbance actuator realised the pre-defined trajectory shown in Fig. 2. In first half of the cycle, disturbance actuator speed was set to $v = 6.25 \cdot 10^{-5}$ m/s and $v = -6.25 \cdot 10^{-4}$ m/s, while in the second half of the cycle $v = 6.25 \cdot 10^{-3}$ m/s which is the maximum speed achievable by the actuators applied in the experiment.

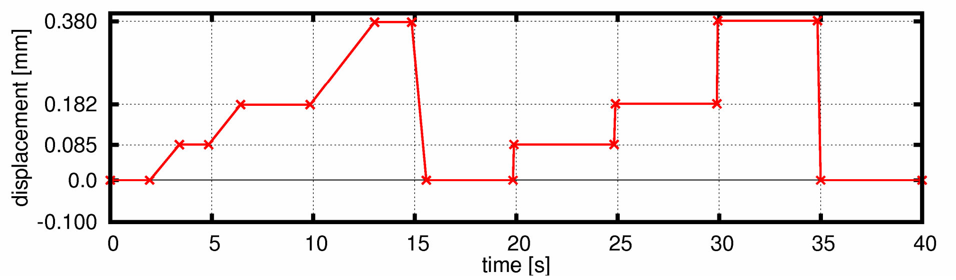

Fig. 2. Disturbance amplitude cycle.

2.2 ANALYTICAL CALCULATIONS

In order to verify the values calculated with the use of FEM and the ones acquired during an experiment, calculations of beam deflections with the use of analytical equations have been conducted. Calculations have been prepared only for the assumed values of disturbance displacements; the behaviour of a PI regulator as well as other aspects of mechatronic components were ignored. In order to calculate the required compensating deflection it is necessary to find an equation of a beam deflection curve, assuming that the disturbance displacement u_d and displacement of critical point $u_c = 0$ are known. Beam deflection curve is described by a relationship valid for small deflections:

$$EI \frac{d^2 u}{dx^2} = M \tag{1}$$

Where: u – beam displacement, x – position on axis of the beam (Fig. 1Fig.), M – bending moment.

Beam is divided into two sections by a concentrated force F_1. Bending moments in these sections of the analysed beam are equal to:

$$M_1 = \frac{1}{EI} F_2 x \tag{2}$$

$$M_2 = \frac{1}{EI}\left(F_2 x - F_1 x(A - C)\right) \tag{3}$$

Where: E – Young's modulus, I – second moment of area, $F1$, $F2$ – external forces, A,C – geometrical dimensions of the beam.

Based on formulas (1, 2, 3), the following formulas are derived:

$$\lambda_1 = \frac{du}{dx} = \int_x M_1(x)dx = \frac{1}{EI}\frac{x^2 F_2}{2} + C_1 \tag{4}$$

$$\lambda_2 = \frac{du}{dx} = \int_x M_2(x)dx = \frac{1}{EI}\left(\frac{x^2 F_2}{2} - \left(\frac{x^2}{2} - (C - A)x\right)F_1\right) + C_2 \tag{5}$$

$$u_1 = \int_x \lambda_1(x)dx = \frac{1}{EI}\frac{x^3 F_2}{6} + C_1 x + D_1 \tag{6}$$

$$u_2 = \int_x \lambda_2(x)dx = \frac{1}{EI}\left(\frac{x^3 F_2}{6} - \left(\frac{x^3}{6} - \frac{(C - A)x^2}{2}\right)F_1\right) + C_2 x + D_2 \tag{7}$$

Input data are disturbance displacement and critical point location, therefore forces F_1 and F_2 must be calculated from the deflection curve equations (6, 7). The solution of above equations requires the definition of integration constants C_1, C_2, D_1, D_2, as well as forces F_1, F_2.

$$\begin{cases} \lambda_2(C) = 0 \\ u_2(C) = 0 \\ u_1(C - B) = 0 \\ u_1(C - A) = u_d \\ \lambda_1(C - A) = \lambda_2(C - A) \\ u_1(C - A) = u_2(C - A) \end{cases} \tag{8}$$

This system of equations has been solved with the utilisation of a Computer Algebra System *"Maxima" (http://maxima.sourceforge.net)*. Acquired theoretical compensation deflections are marked on Fig. 8 and 9.

2.3. OBJECT AND PROCESS MODEL

A modelled stepper motor driven actuator is suitable only for compensating relatively slow changing processes with frequencies up to few Hz. First natural frequency of the considered beam without actuators is about 100Hz and is much higher than working frequencies of other components, in the event of that, the dynamic response of the beam has been neglected and elastic response of the beam has been considered in a quasi-static state. The whole process has been divided into 0.025s time steps.

Object behaviour and PID regulator value are calculated every time step, compensating actuator trajectory is calculated alt the beginning of every actuator movement. The whole calculation procedure is presented on Fig. 3. Mechatronic model is composed of four basic components:

1a. FEM work piece model,
1b. models of actuators and contact areas,
2. Proportional-Integral-Derivative (PID) regulator,
3. stepper motor and its drive control models.

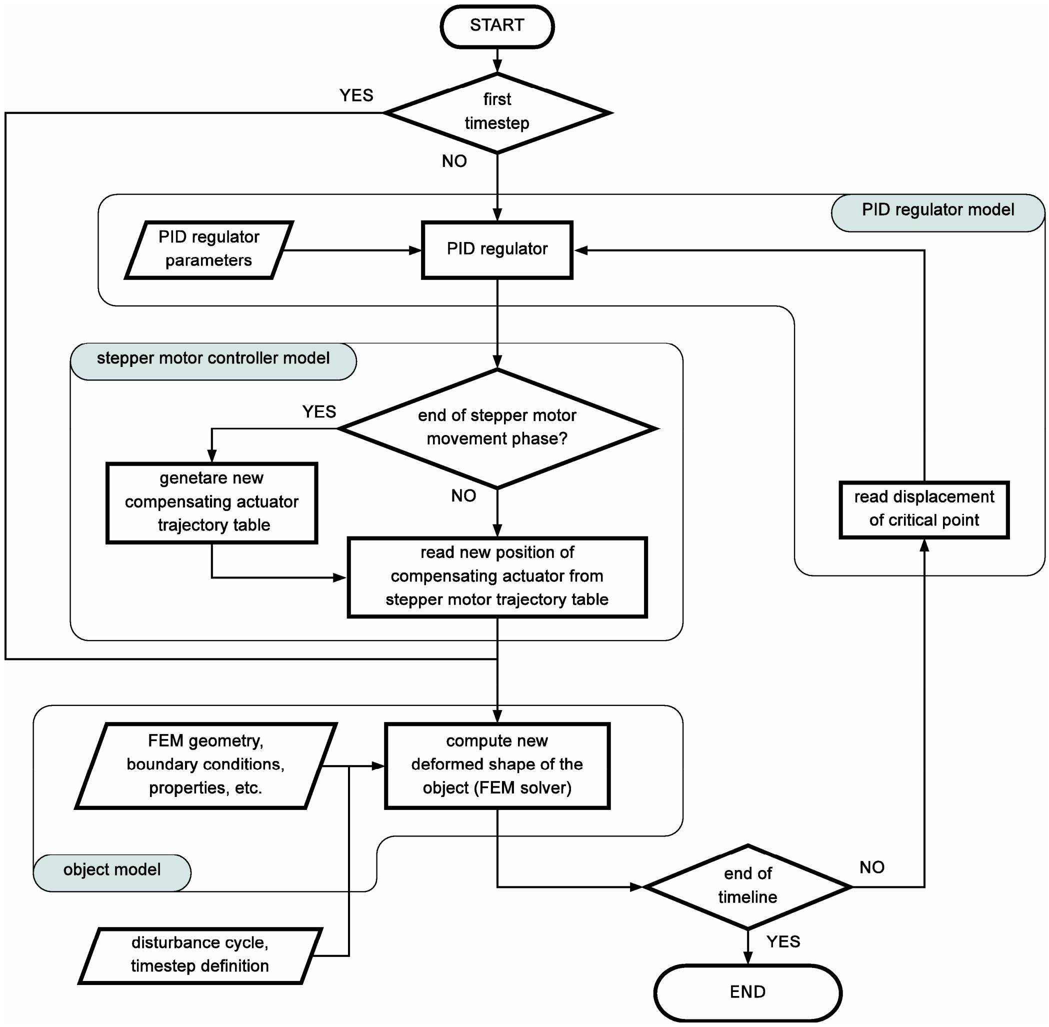

Fig. 3. Flow chart of the simulation procedure

Where components 1st a and 1st b are physically oriented models implemented with the use of the FEM code, 2nd component implementation is analogous to the regulator implementation in industrial applications, and 3rd component is a behavioural model of a stepper motor and its drive controller unit. 1st, 2nd and 3rd components of described mechatronic model are highlighted on Fig. 3.

2.4. FEM MODEL OF WORKPIECE AND ACTUATORS

Workpiece model has been divided into 74 quadratic hexahedral reduced integration elements of type C3D20R. Because of a full symmetric nature of the problem only a half of the analysed workpiece has been modelled (Fig. 4). Direct equation solver and full Newton solution technique have been used, while geometrical nonlinearity has not been taken into account.

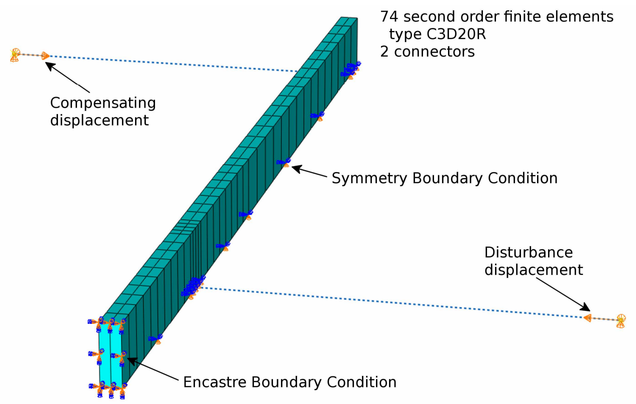

Fig. 4. FEM model of a beam – mesh and boundary conditions

Actuators, modelled as connectors, transfer compression forces only. Disturbance actuator is rigid, compensating actuator is flexible with linear stiffness of k = 75000 N/m at distance of 0.025mm, after exceeding this displacement the connector's stiffness is rigid. Such behaviour represents a very simplified contact model and to some extent emulates the real behaviour of the object.

2.5. PROPORTIONAL-INTEGRAL-DERIVATIVE (PID) REGULATOR IMPLEMENTATION

PID regulator output is calculated based on the well-known following formula:

$$U(t) = k_p \left[u_p(t) + \frac{1}{T_i} \int_0^t u_p(\tau)\,\mathrm{d}\tau + T_d \frac{\mathrm{d}u_p(t)}{\mathrm{d}t} \right] \tag{3}$$

Where: $U(t)$ – regulator output, k_p – proportional gain, T_i – integration time constant, T_d – derivative time constant, t – time, u_p – critical point P displacement.

It has been implemented in FORTRAN 77 programming language with the use of ABAQUS/Standard subroutines DISP and URDFIL. Data exchange between procedures takes place by means of a COMMON block of memory. Integral is calculated using trapezium rule of numeric integration while derivative is represented as a difference between two latest values of the u_p signal.

2.6. STEPPER MOTOR AND ITS DRIVE CONTROL MODELS

Compensating actuator movement is composed of acceleration phase, movement with constant speed and deceleration phase if the distance is large enough, or of acceleration and deceleration phases in case of small displacements. Modelling such behaviour has been realised by preparing a trajectory lookup table of one movement segment (marked by rectangles in Fig. 9 in the Results section), containing these three phases, which describes the position for every time step of this movement segment. Stepper motor controller is called at the beginning of every movement. Trajectory is realised by reading the trajectory lookup table and sending values as the output of DISP subroutine. Declared values of acceleration and maximum speed are same as in experimental investigations and are equal to, respectively: a = 0.05 m/s^2, v_{max} = 6.25·10^{-3} m/s. One actuator step translates to the displacement of 1.25 µm.

In case of disturbance actuator, the trajectory was simplified and did not represent acceleration and deceleration phases (Fig. 2).

2.7. EXPERIMENTAL INVESTIGATIONS

In analogy to the FEM model structure, a test stand has been built. An overview of hardware components used in the test stand is shown in Fig. 5. The workpiece in shape of a beam is horizontally fixed in a vice. Dedicated disturbance and compensation actuators are composed of a stepper motor, a ball screw, a spline shaft with sleeve and an end-effector equipped with a piezoelectric force sensor and a spherical tip.

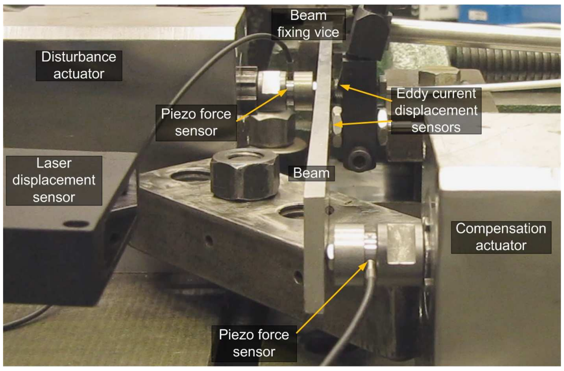

Fig. 5. View of test stand components

A block diagram of the entire measurement and control system hardware is shown in Fig. 6, including sensor amplifiers, data acquisition card and a motion control card. In this figure, elements taking part in the compensation process are distinguished from these used for monitoring purposes and for simulating external disturbances. Stepper motors are powered by means of stepper motor drivers which receive control signals from the motion control card located in the PCI slot of a PC. Beam displacement u_p in a critical point P, as well as disturbance displacement u_d, are measured by means of eddy current displacement sensors. Beam deflection at the end u_c, which is relatively large, is measured with the use of laser triangulation sensor. A data acquisition card, with 16 analogue inputs and sampling frequency up to 250 kHz, is used to acquire analogue signals from the sensor electronics, to convert it into a digital form and to send it via the USB protocol to the PC.

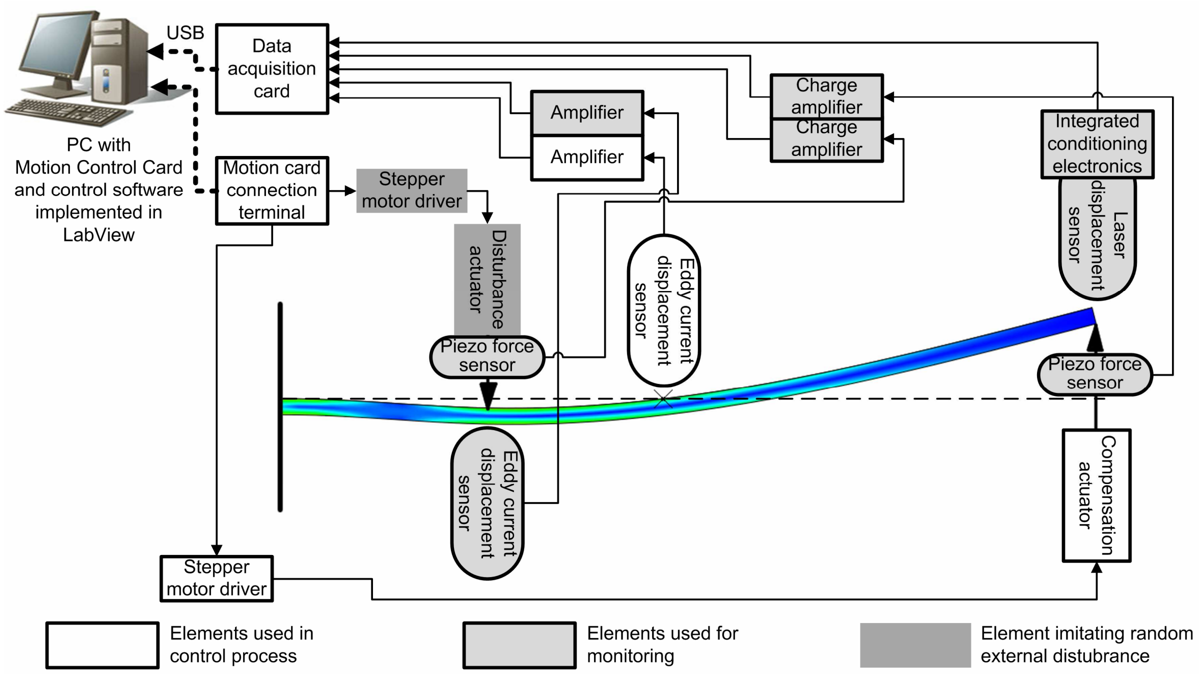

Fig. 6. Block diagram of test stand hardware

Testing procedure is based on applying three values of disturbance displacements by a disturbance actuator with three different speeds (Fig. 2). Control system has been implemented in LabView graphical programming environment based on the regulation algorithm shown in Fig. 7, aiming at the generation of such control signals, as to ensure desired critical point location $u_p = 0$. The value of critical point P displacement u_p is read from the sensor-amplifier system by the data acquisition card and fed to the PID controller function which generates the compensation displacement value, expressed in stepper motor steps. This amount of steps is sent to the motion control block as movement destination. Finally, motion control card is ordered to calculate the trajectory and start the motion. PID function and motion control libraries supplied by the LabView programming environment developer were utilised.

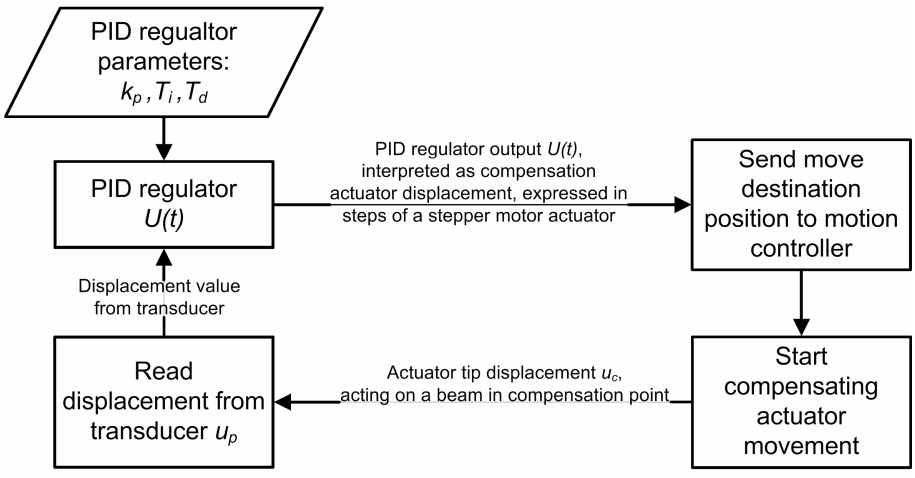

Fig. 7. Regulation cycle block diagram

2.8. RESULTS

As a result of the research work, displacement data from simulations and measurements, visualised in Fig. 8 and 9, have been obtained. On Fig. 8 displacement values calculated by means of FEM simulations are compared to the results obtained by means of analytical methods for small deflections (Fig. 8b) and to the results from the experiment (Fig. 8a). Probable cause of these differences is the influence of beam fixing conditions in a vice. In the models, the beam is fully constrained at the fixed end, and during the experiment some rotation of the beam could occur influencing its deflection curve shape.

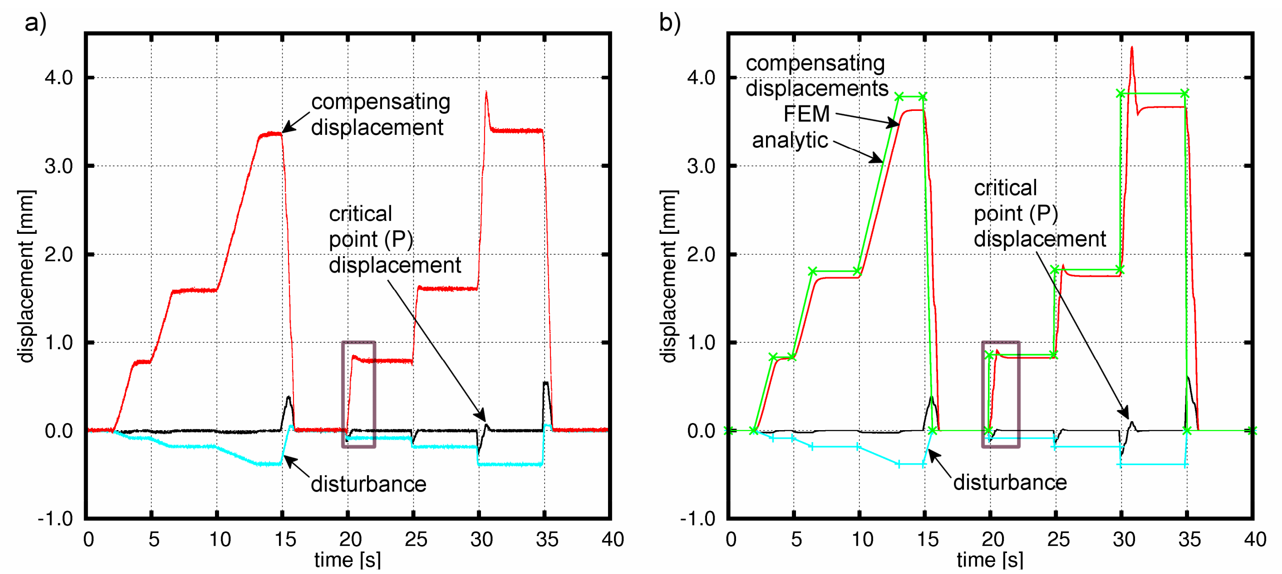

Fig. 8. Cumulative plot of displacements in three characteristic points on the beam;
a) experimental results, b) analytical and FEM simulation results

Fig. 9 shows in detail the results obtained during first rapid disturbance marked by rectangles on Fig. 8. The appearance of error (*critical point (P) displacement*) causes immediate generation of the correcting signal by the PI regulator (*PI regulator value*). This correcting value is sent to the stepper motor controller which starts the movement. At the beginning of the first movement phase a certain delay is observed which represents the cancellation of play between the beam and the actuator tip. As the correcting value increases the actuator realises s-shaped movement segments, marked by rectangles on Fig. 9a and b, composed of acceleration, constant speed movement and deceleration motion phases. A certain overshoot of all signals is typical for PI regulators which are set to react quickly to the appearance of error.

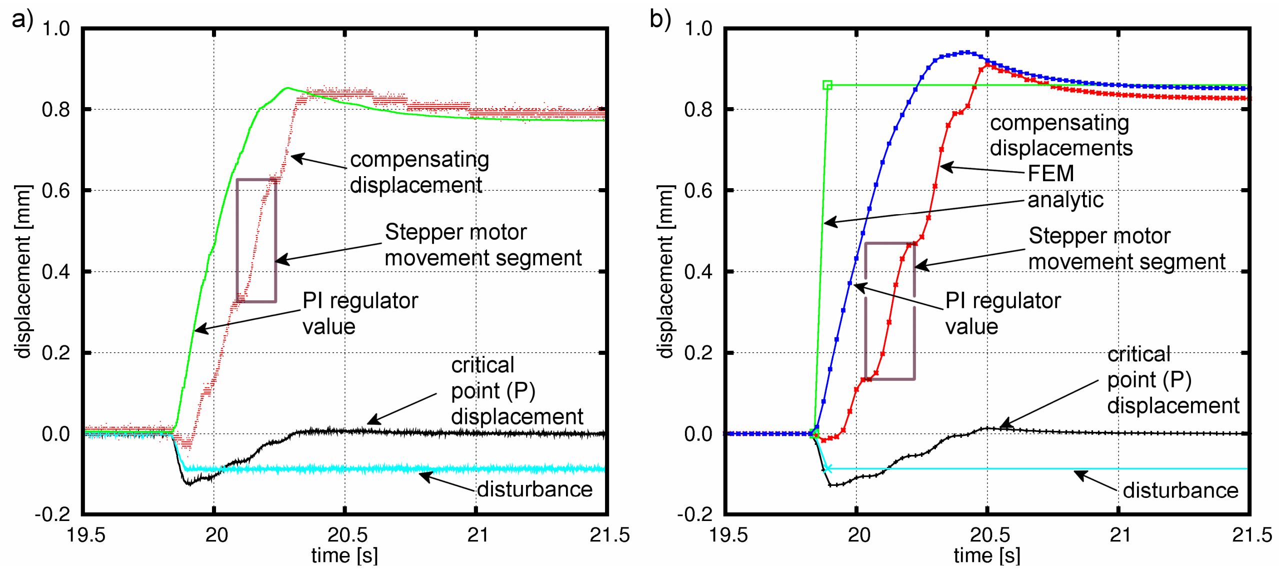

Fig. 9. Detailed plot of displacements in three characteristic points on the beam during first rapid disturbance;
a) experimental results, b) analytical and FEM simulation results

The comparison of experimental and FEM calculated results shows good correlation between values of displacements and especially very good correlation between dynamic responses of mechatronic components. A delay to the change of the real actuator movement observed at the top of the plot from Fig 9a is caused by play of the actuator's ball screw. Such phenomenon has not been taken into consideration in the FEM model which causes the difference between the calculated and measured values. The difference of steady-state values between the PI regulator value and compensating displacement during FEM simulations has been caused by the simplified contact model (see section 2.4).

During investigations of the model different meshing variants have been checked, including very coarse and very fine meshes, 1st and 2nd order finite elements, as well as 1D and 2D elements. In all cases in which the model has been prepared according to rules of good practice, the results were very similar. Very good results in scope of the calculation time have also been achieved.

3. MODELLING MECHATRONIC STRUCTURE WITH LINEAR COUPLED THERMO-MECHANICAL COMPONENT

In order to define possibilities of the utilisation of the elaborated calculation scheme, the analysis of a simple frame subject to the influence of a variable heat source acting on one of the frame surfaces has been conducted. The main structure of the model is the same as in the previously described investigation, but some modifications of objects have been introduced and described in the next subsections. The results of the analysis have not been experimentally verified.

3.1. OBJECT DESCRIPTION AND ITS FEM REPRESENTATION

Object is represented by a simple, flexible frame disturbed by a heat source acting on one surface of the frame (Fig. 10).

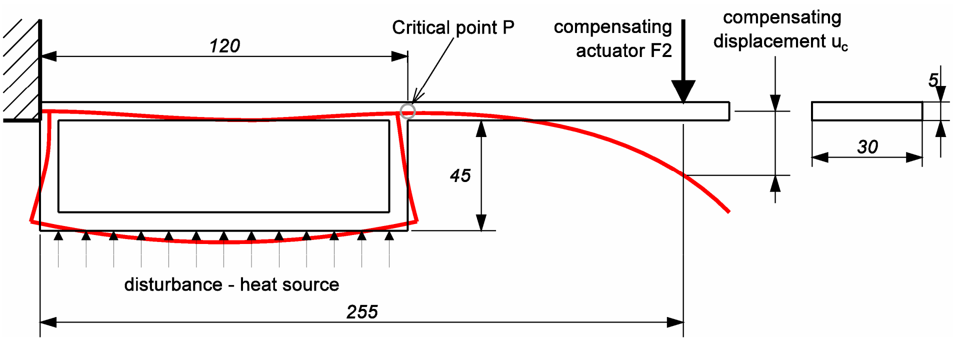

Fig. 10. Frame shape workpiece model

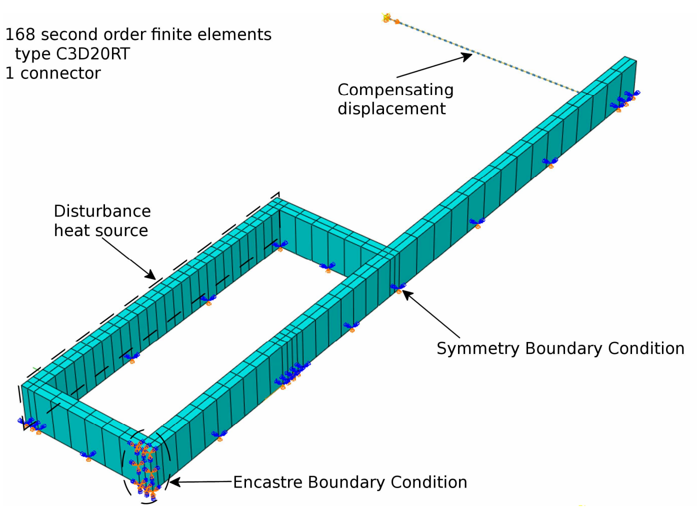

Fig. 11. FEM model of the frame - mesh and boundary conditions

An energy supplied to the object is at the level of 10 kW/m^2 which causes supplying of 36 W of energy in heating process and no energy in cooling process. Cooling is realised only by dissipation of thermal energy by means of radiation to the ambient environment, radiation coefficient was set to 1. Energy is supplied in a cycle presented on Fig. 12a, b.

FEM model of the frame is based on 168 second order coupled thermo-elastic finite elements type C3D20RT. A connector which models compensating actuator is suitable to transmit tension and compression forces, contact area is modelled by the same means as in the previously described investigation but is also suitable to transfer tension and compression forces.

3.2. CALCULATIONS AND RESULTS

The whole process of the frame analysis takes 800 seconds and is divided into 0.5s time steps.

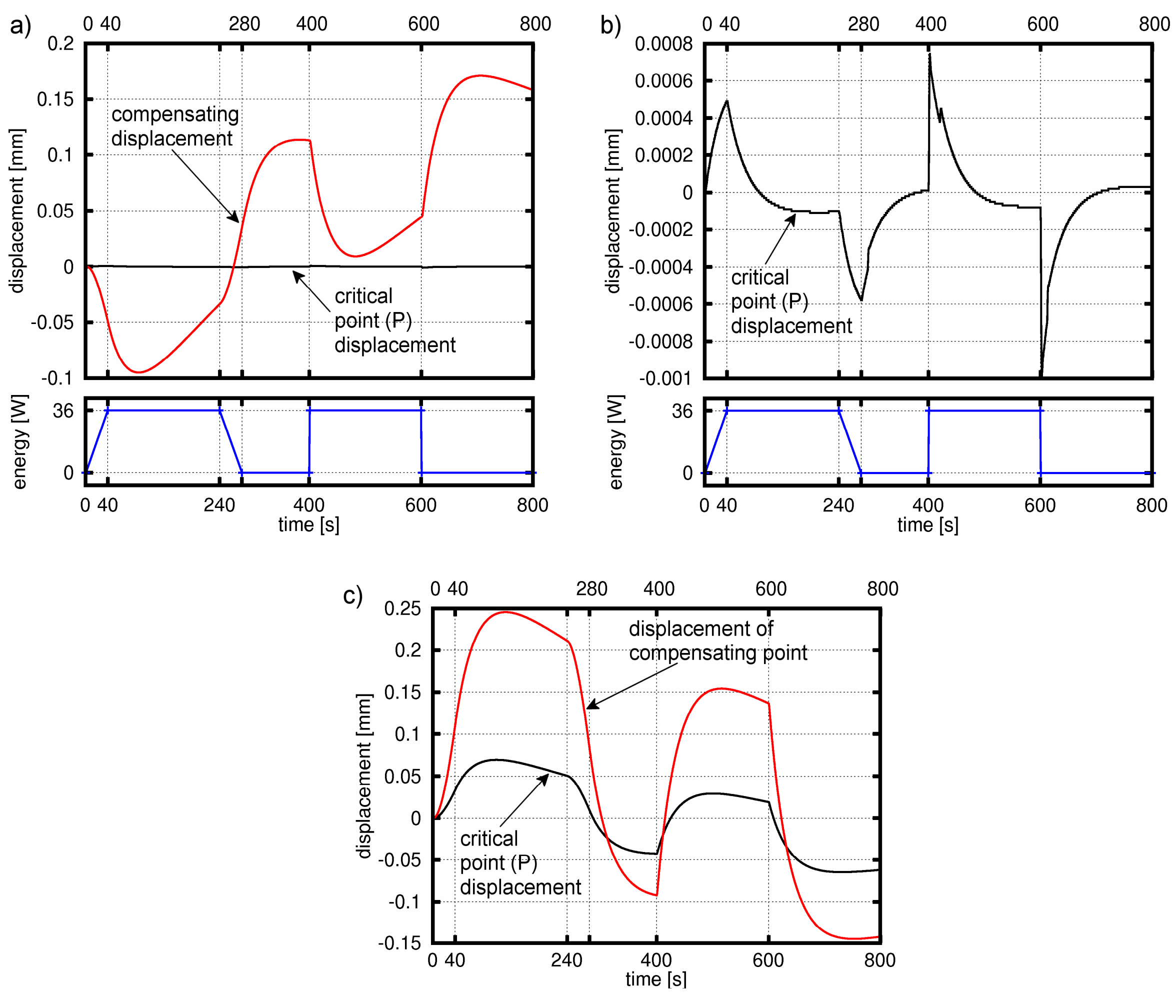

Fig. 12. Critical point displacement, compensating point displacement and disturbance energy of the analysed frame; a) with compensation, b) with compensation – scaled plot showing only critical point displacement, d) without compensation.

The calculations have been carried out both for the case with active compensating functionality and with the compensation turned off. The results shown on Fig. 12a, b present the effectiveness of the compensating algorithm operation in comparison to the cycle without compensation (Fig. 12c). In case of no compensation the compensating connector was not fixed. In case of applying compensation, maximum displacements of the critical point P do not exceed 1µm, while in case of no compensation maximum deflection in point P from the setpoint reaches the value of 70µm. In the presented case the effectiveness of compensation operation is limited mainly by a relatively long time step, equal to 0.5s.

4. CONCLUSIONS

The presented technique allows building models of mechatronic components with its main advantage being the possibility of full utilisation of computational capabilities of commercially available FEM analysis software. The presented technique is also significantly independent from the possessed software therefore can be applied by any user capable of utilising an advanced FEM system. The advantages of the presented technique are ease of modelling the behaviour of the mechanical structure taking into account factors having influence on it, as well as the possibility of implementing any characteristics of the applied non-mechanical elements. Disadvantages are the need of full implementation of the behaviour of the remaining elements of the analysed system, as well as, as in the presented implementation, lack of possibilities of analysing the system which is subject to mechanical vibrations.

Long-term goal of the authors is the creation of the analysed model of the fixture-workpiece system equipped with active actuators and control systems which allow compensating workpiece deformations caused by the external forces. Realised investigations are a basis for assuming that with the use of the presented technique it is possible to model such objects, having in mind that main problem which remains to be solved is the implementation of few interacting actuator-regulator systems.

ACKNOWLEDGEMENT

Hereby work was realised as a part of the AFFIX project (http://www.affix-ip.eu/), which is a part of the EU 6th framework program. The work was supported by Polish Ministry of Science and Higher Education (390239).
Calculations have been carried out in Wroclaw Centre for Networking and Supercomputing (http://www.wcss.wroc.pl), grant No. 109.

REFERENCES

[1] BUKOWSKI A, GROCHOWSKI M, JĘDRZEJEWSKI J., 2008, *Hybrid Modelling of Machine Tool Body with Cooling Component*, Journal of Machine Engineering, 8/1/84-93.

[2] GAWRYSIAK M., 1997, *Mechatronika i projektowanie mechatroniczne* (in Polish), Bialystok, Wydawnictwo Politechniki Białostockiej.

[3] HILLER M., (1996),*Modelling, simulation and control design for large and heavy manipulators*, Robotics and Autonomous Systems, 19/167-177.
[4] JAKOVLEVIC M., FOTIU P. A., MRCARICA Z., DETTER H., 2000, *A system-level simulation of complex multi-domain microsystems by using analogue hardware description languages*, Sens and Actuat, 82/30-39.
[5] JĘDRZEJEWSKI J., KOWAL Z., KWAŚNY W., MODRZYCKI W., 2004, *Hybrid Model of High Speed Machining Centre Headstock*. Annals of the CIRP, 53/1/285-288.
[6] KLAASSEN B., 1997, *A method for tightly coupled thermal-electrical simulation,* Mechatronics Journal 28 239-245.
[7] NEE A. Y. C., SENTHIL KUMAR A., TAO Z. J., 2000, *An intelligent fixture with a dynamic clamping scheme*, Proc Instn Mech Engrs, 214/183-196.
[8] SIEBENALER S. P., MELKOTE S.N., 2006, *Prediction of workpiece deformation in a fixture system using the finite element method*, International Journal of Machine Tools & Manufacture, 46/51-58.