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THE FINITE ELEMENT ANALYSIS OF HIGH PRECISION POSITIONING SYSTEM

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Abstract: The paper deals with design and computation of precise positioning mechanism. The aim of mechanism is to transfer applied load without any change of position of parts in question. The stabilization of position is assured by elastic elements connected to reductor which is adjusted into correct position by an actuator. The functionality of mechanism and its stiffness and strength characteristics were checked by the finite element method. The computations were accomplished for prescribed displacement of reductor. On the base of this type of loading the behaviour of the structure was evaluated.

KEYWORDS: mechanism, reductor, elastic element, actuator

1 Introduction

There exists a group of extreme precision positioning systems working in dimensions of micrometers, but all have common property – they serve for positioning of small masses [1-5]. Such systems are used in laboratory equipment, microscopes, machining centres, and so on, where manipulations with masses in order maximally of several tents of kilograms are necessary. The paper deals with design and analysis of high precision positioning mechanism proposed for manipulation with masses of several hundreds of kilograms. The mechanism is designated for the positioning of given machine part in one prescribed direction. The behaviour of mechanisms under loading was checked by the finite element method using linear elasticity theory. The numerical simulation was realized on high precision positioning system [6-8], the model of which is given in Fig. 1.

The stand consists of massive frame 1 in which mass 2 is included, actuator 3 connected with elastic member 4 that serves as mechanical transformator of movement. This member is joined with elastic hinge to auxiliary frame 5. The mass is connected also by second elastic hinge 7 on bottom part of elastic member 4. The movement of mass due to movement of actuator 3 and transformation of this movement through elastic member 4 is measured by high precision measurement system 6. The members 4 and 7 are made of steel 34CrNiMo6.

The chemical composition of the steel 34CrNiMo6 (WNr 1.6582) can be seen in Tab. 1, the mechanical properties then in Tab. 2.

Tab. 1 Chemical composition % of steel 34CrNiMo6 (1.6582)

C	**Si**	**Mn**	**Ni**
0.3 - 0.38	max 0.4	0.5 - 0.8	1.3 - 1.7
P	**S**	**Cr**	**Mo**
max 0.025	max 0.035	1.3 - 1.7	0.15 - 0.3

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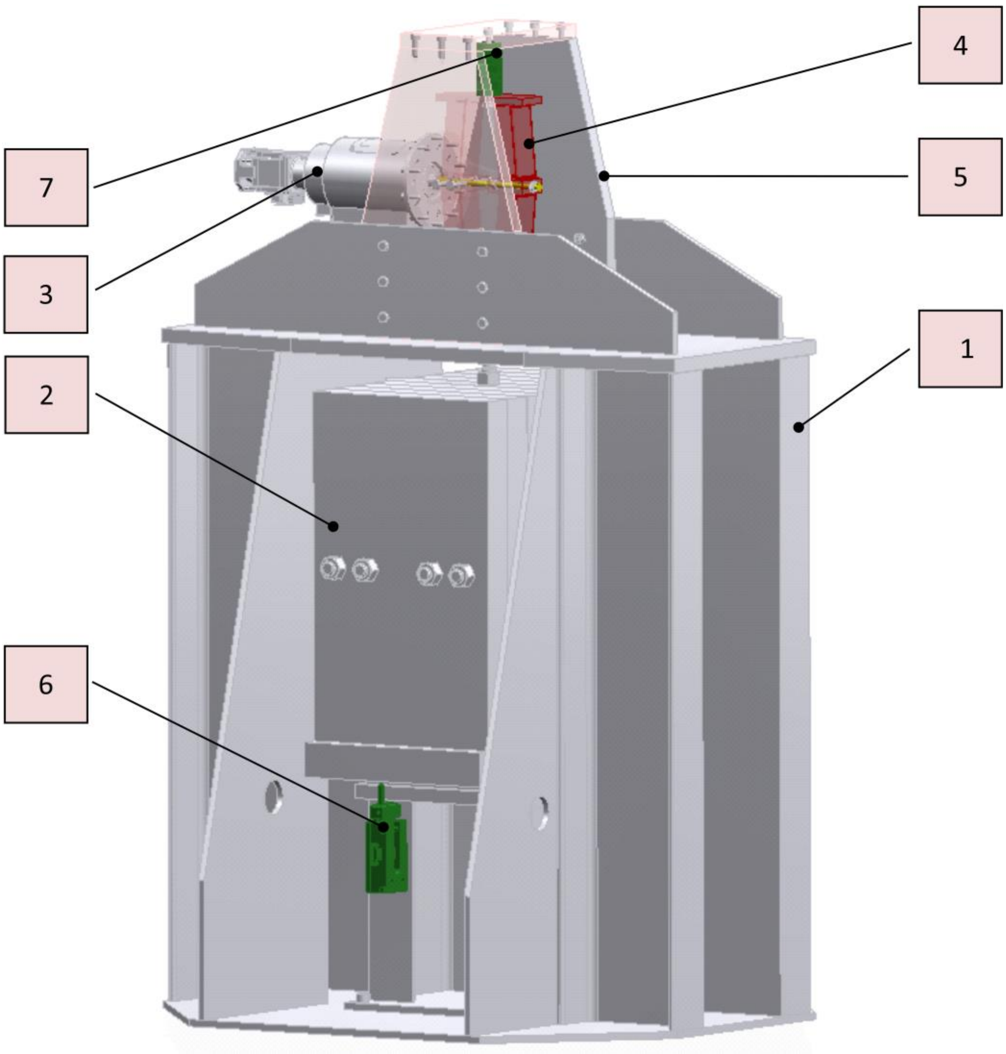

Fig. 1 View to the test stand

Tab. 2 Mechanical properties of steel 34CrNiMo6 (1.6582)

Diameter d (mm)	< 16	> 16 - 40	> 40 - 100	> 100 - 160	> 160 - 250
Thickness t (mm)	< 8	8 < t < 20	20 < t < 60	60 < t < 100	100 < t < 160
Yield strength Re (MPa)	1000	900	800	700	600
Tensile strength Rm (MPa)	1200 - 1400	1100 - 1300	1000 - 1200	900 - 1100	800 - 950
Elongation A (%)	9	10	11	12	13
Reduction of area Z (%)	40	45	50	55	55
Toughness CVN (J)	35	45	45	45	45

2 Simulation of high precision positioning system

The simulation of high precision positioning system (Figs. 1, 2 and 3) which is based on elastic member 4 was accomplished by the finite element method. The positioning system is loaded by its weight and the fixation of structure is realized on the top plate. The self-weight of the structure acts in vertical direction (downstairs).

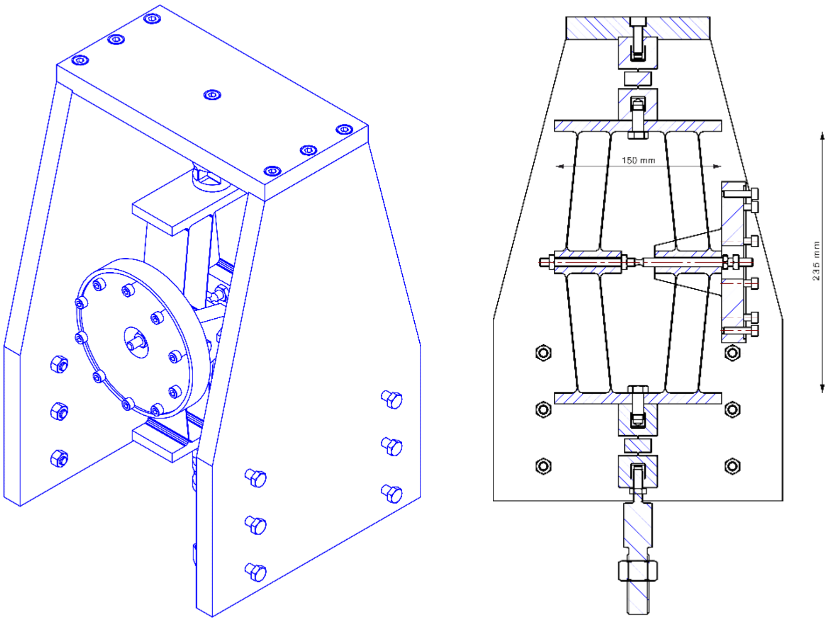

Fig. 2 Model of stand

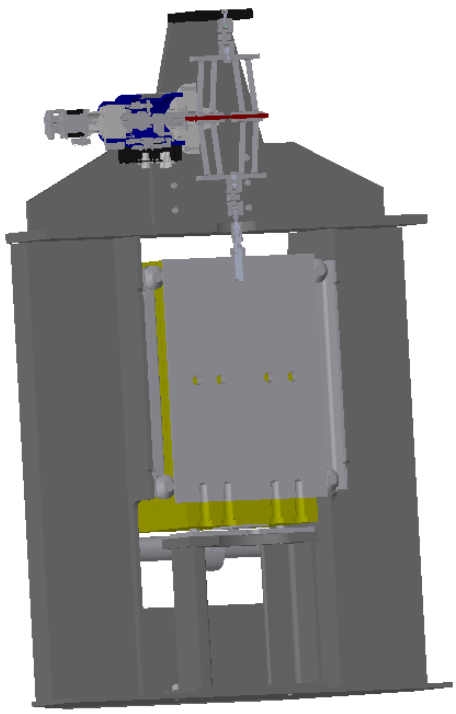

Fig. 3 View to the test stand – cross-section

The model, resulting self-weight force and boundary conditions are seen on the left side of Fig. 4. On the right side is given meshed model of the positioning mechanism. Number of computations were accomplished in order to find the basic behaviour of structure, especially its stiffness.

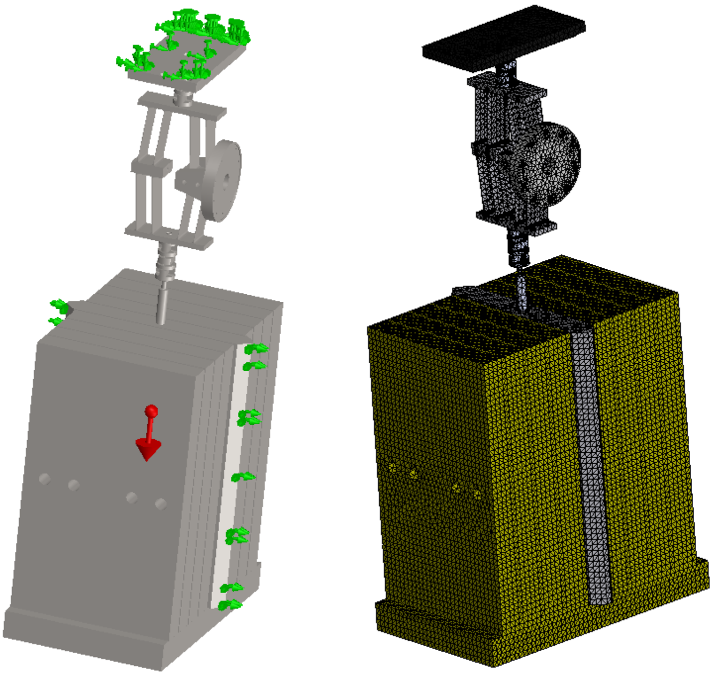

Fig. 4 The boundary conditions and the mesh of finite elements on the model

In Fig. 5 is given field of equivalent von Mises stresses for the structure under self-weight loading. The results have shown that the most loaded members of the stand are the elastic elements. In Fig. 6 are given displacements of points in direction X. Maximum displacement is on the bottom wedge of weight. Due to loading, the reductor is moved together with flange for its fixation. Accordingly, on the right side of this figure are given displacements in direction Y.

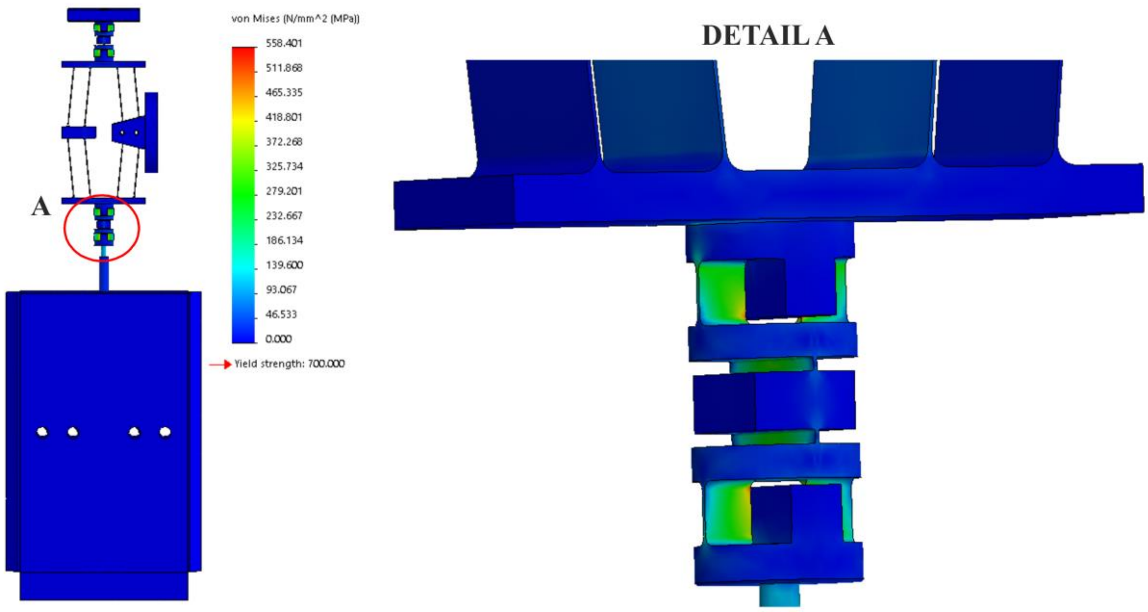

Fig. 5 Field of equivalent von Mises stresses on the model

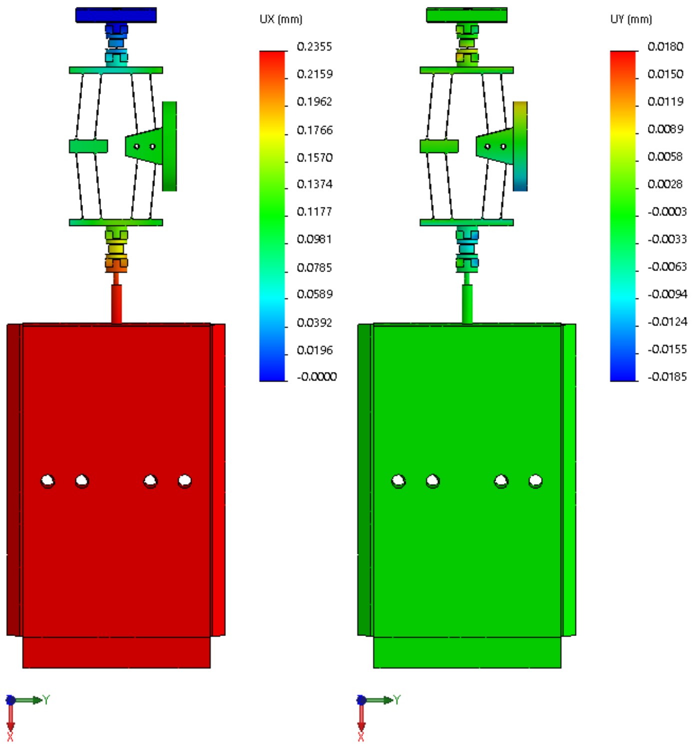

Fig. 6 Fields of displacement of points in X and Y direction, respectively

In Fig. 7 is given field of equivalent von Mises stresses for the structure loaded by self-weight and prescribed displacement of reductor in Y direction by 0.5 mm in one direction and 0.5 mm in the opposite direction. loading. The results have shown that again the most loaded members of the stand are the elastic elements. In Fig. 8 are given displacements of points in direction X and Y, respectively. Due to prescribed displacement in Y direction, the maximum the maximum displacement is smaller than in previous case.

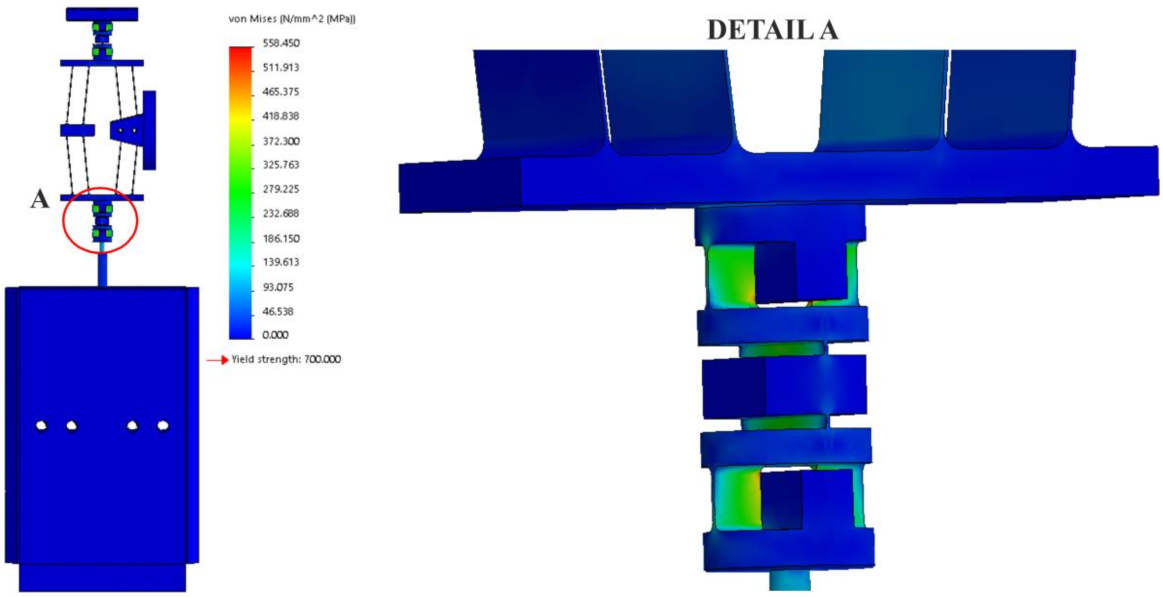

Fig. 7 Field of equivalent von Mises stresses on the model

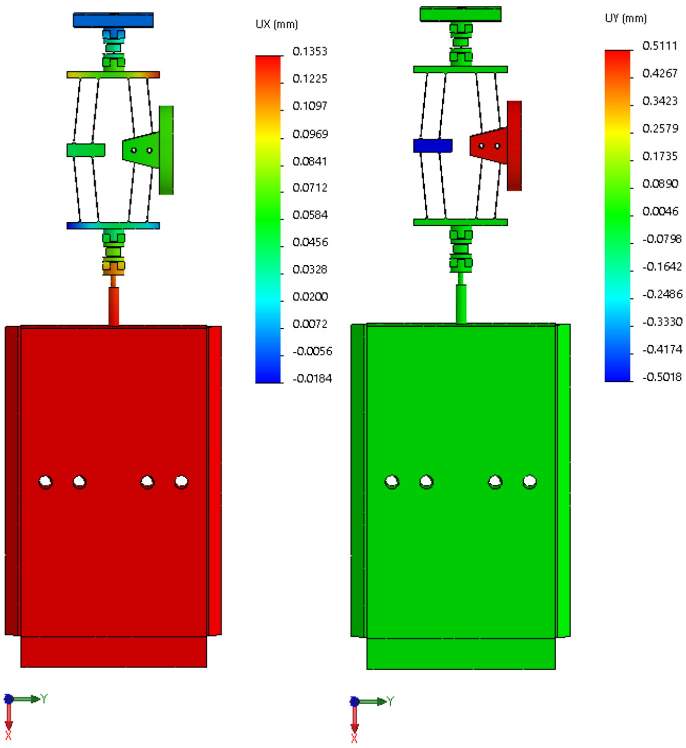

Fig. 8 Fields of displacement of points in X and Y direction, respectively

In Fig. 9 is given field of equivalent von Mises stresses for the structure loaded by self-weight and prescribed displacement of reductor in Y direction by 5 mm in one direction and 5 mm in the opposite direction. loading. In Fig. 10 are given displacements of points in direction X and Y, respectively.

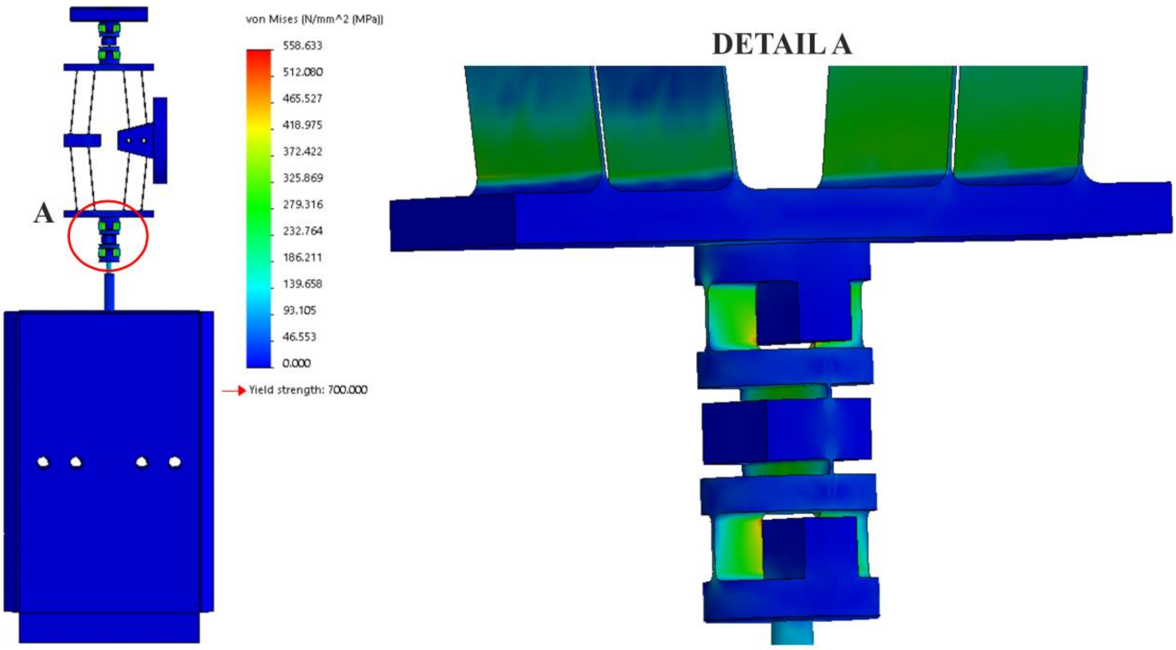

Fig. 9 Field of equivalent von Mises stresses on the model

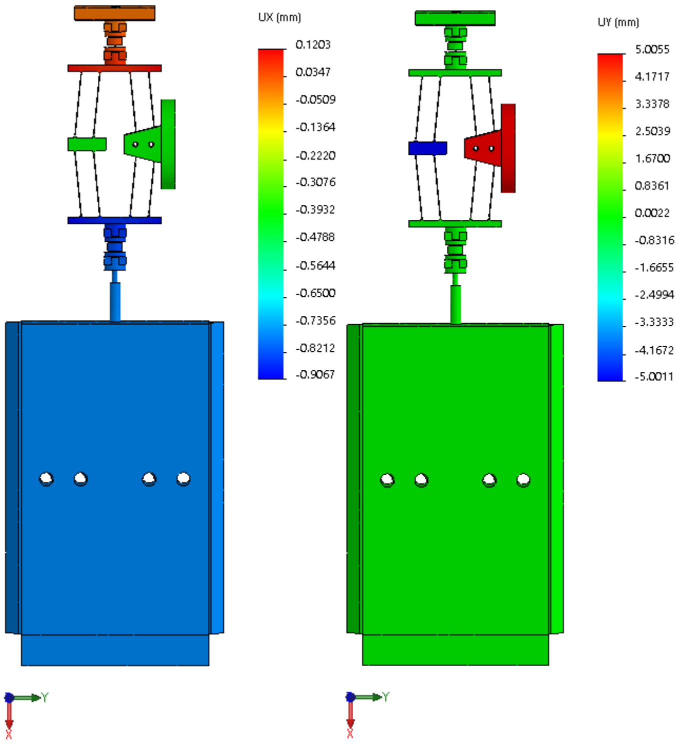

Fig. 10 Fields of displacement of points in X and Y direction, respectively

Besides of above described simulations were accomplished computations for other prescribed displacement in direction Y. All simulations are summarized in Tab. 3. Here, has to be mentioned that the maximum values of von Mises stress do not cross approximately 560 MPa which is value under yield stress of the material 34CrNiMo6 (WNr 1.6582) of such stressed parts. All displacements in direction X in Tab. 3 depend on prescribed displacements given in column one of the table. These vary from 0.5 mm to 5 mm in positive and negative direction of axis Y.

Tab. 3 Dependence of maximum displacement in direction X on prescribed displacement in direction Y

Prescribed displacement in Y direction (mm)	**Displacement of bottom surface of weight in X direction (mm)**
0	0.2355
0.5	0.1353
1	0.0355
2	-0.1644
3	-0.3643
4	-0.5641
5	-0.7639

CONCLUSION

In the paper is described stress and deformation analysis of high precision positioning equipment. The computations were accomplished for different types of loadings including self-weight of structure and prescribed displacement of parts due to movement of actuator. The structure was loaded in elastic range using isotropic materials. The main interest of authors was focused to deformation and stress characteristics of given elastic joint, i.e. displacements, strains, stresses. From the computations can be stated fact that the most stressed parts of structure withstand the applied loading. The results of numerical simulation of elastic members were consequently used for realization of kinematical analysis of movement of supporting system of heavy objects. The aim of such analysis was to verify range of movement of reference points on positioning axes of the object. The problem of this analysis was that this system is statically undetermined and accordingly the stiffnesses of joining elastic members have to be considered for the computations. Besides of stress analysis, the authors simulated influence of temperature to the positioning precision. However, these simulations are not included in the paper.

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