

A NEW CONSTRUCTIVE SOLUTION FOR A PIEZOELECTRIC TRANSLATION MODULE, DESIGN AND SIMULATION

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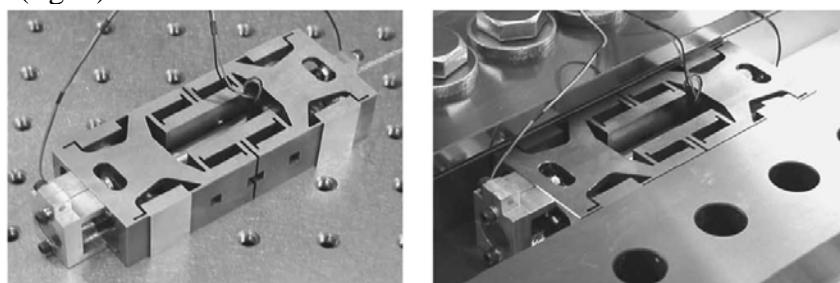
A new constructive solution for piezoelectric translation modules is described. An explanation of the working principle is given. Mechanical resistance and vibration simulations were run as part of the modeling process and results are shown. Simulation results and functional aspects are taken into consideration when compared with other solutions. The design aimed towards offering a solution for large displacement requirements with high force and precise positioning, which implies compensating for high tolerance components, mainly of the guideway.

Keywords: linear, translation, module, piezoelectric, actuation, modal analysis, eigenfrequency

1. Introduction

There are various constructive solutions for piezoelectric translation modules based on stepping techniques. Generally, these are known as a type called “inchworm” and their constructive approach and simulation types are interlinked and based on the aimed purpose of the device.

In 2004, Tenzer and Mrad [1] conceived a design algorithm for piezoelectric drive translation modules for a displacement system with compliant elements (fig. 1).



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Fig. 1. Inchworm type translation module, with compliant structures.

For this structure, the CAD model was simulated in ANSYS and the computed eigenfrequencies are presented in Fig. 2.

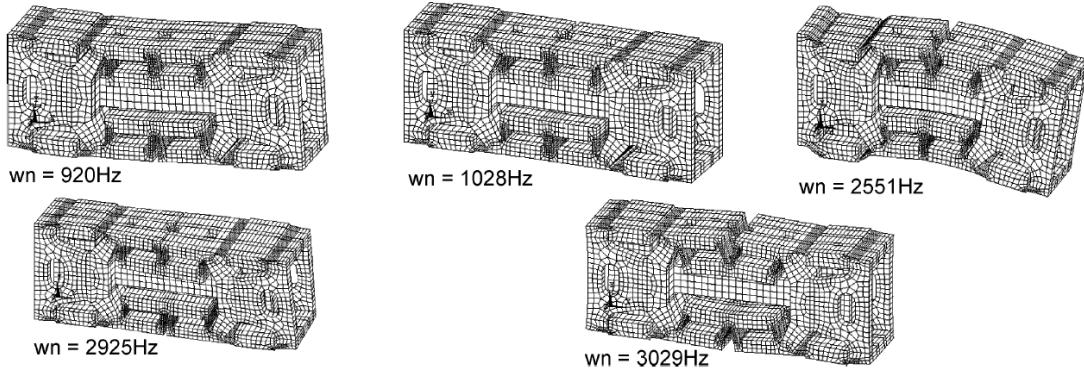


Fig. 2. Finite element analysis for the piezoelectric transmission with compliant elements [1].

In 2005, Li et al. [2] designed, modeled and simulated an inchworm linear positioning module with CATIA and ANSYS software. Fig. 3 presents the mechanical structure of the translation module.

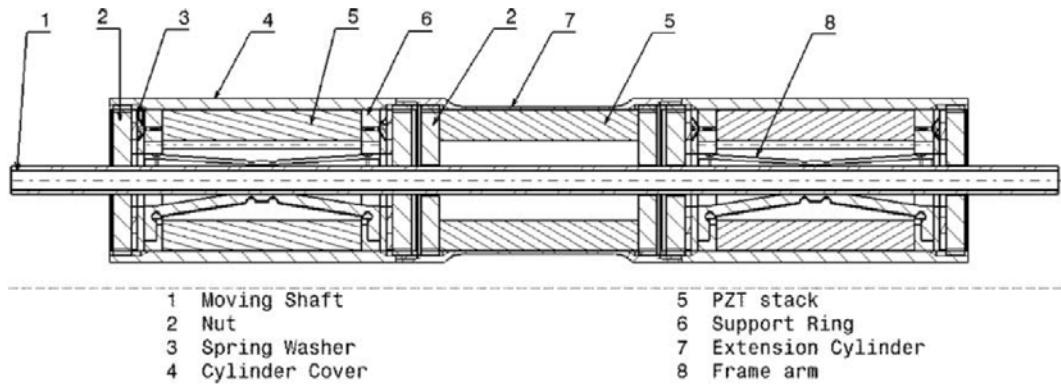


Fig. 3. The structure of the positioning module presented by Li et al. [2].

As it can be seen, the structure is compact, simple, but there are no guideways or obvious structures to prevent rotation of the center axis.

In Fig.4 a) it is presented the mechanical structure of the position indexing assembly, made in CATIA and in Fig.4 b) it is presented the simulation with ANSYS software of a half of the compliant element from the position indexing structure.

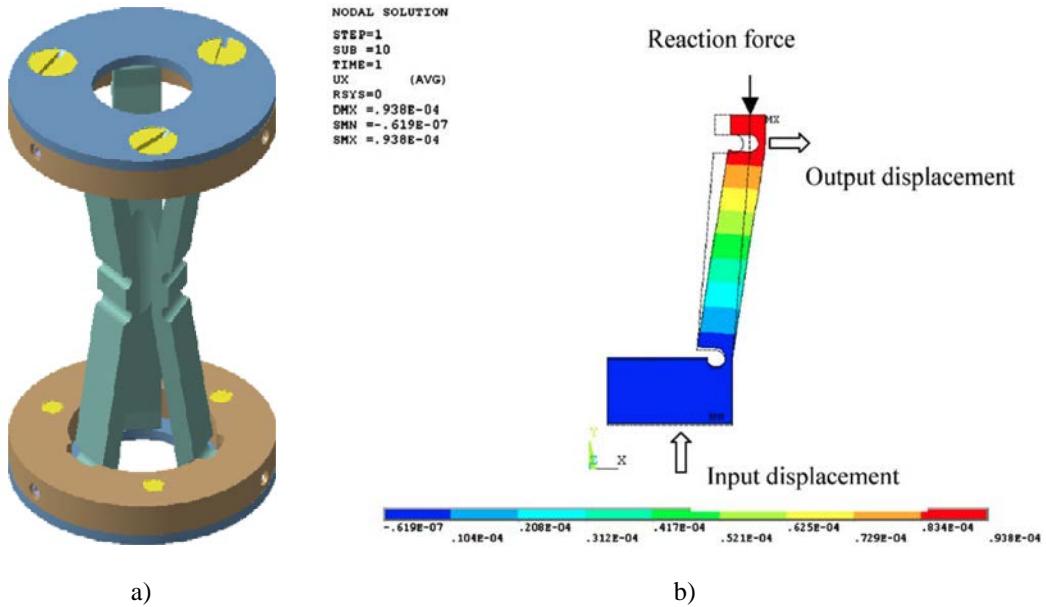


Fig. 4. The mechanical structure of the position indexing assembly: a) CAD model; b) FEA simulation (Li et al. [2].)

In 2009, Kim and Kang [3] developed a self-propelled "cellular" motor (fig. 5) with piezoelectric actuation, also based on the inchworm motion principle.

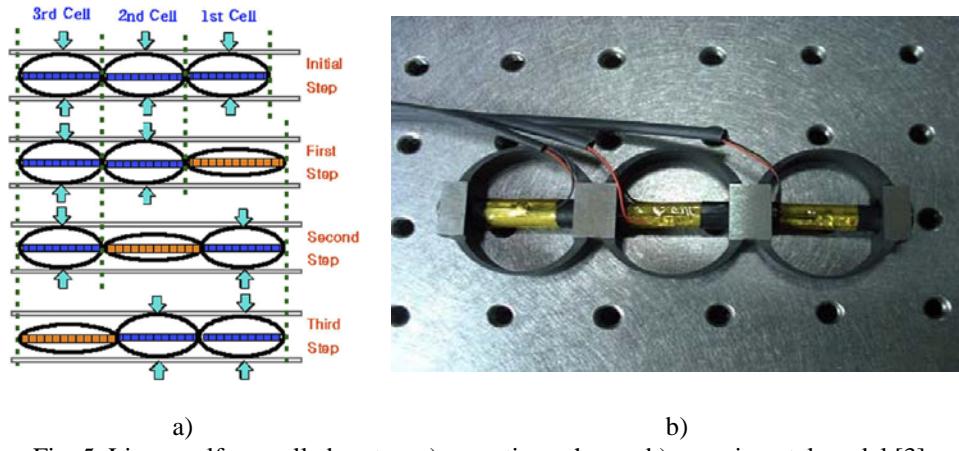


Fig. 5. Linear self-propelled motor: a) operation scheme; b) experimental model [3].

In 2010, Hii et al. [4] made a compact piezoelectric actuator inchworm engine but with very small available displacement. This model also does not seem to prevent rotation.

In 2017, Shao et al. [5] constructed an inchworm in-line linear positioning module presented in Fig. 6 with position indexing on a toothed rack with $5\mu\text{m}$ steps (Fig. 7).

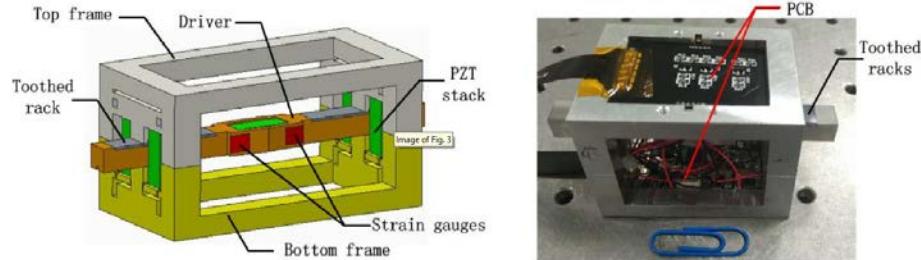


Fig. 6. Inchworm in-line linear positioning module: a) CAD design; b) experimental model [5].

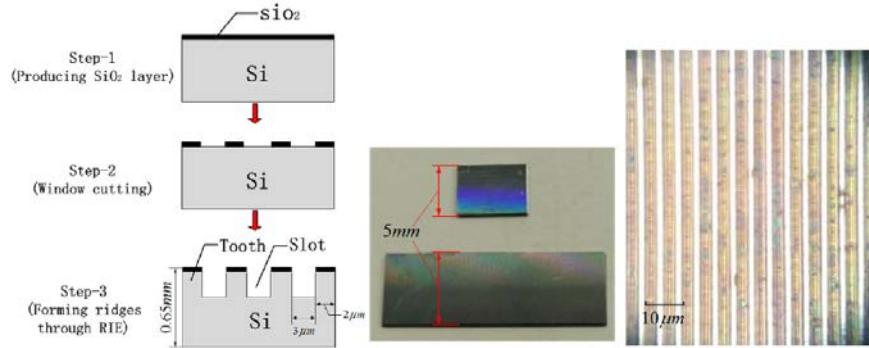


Fig. 7. Toothed rack manufactured with SCREAM procedure (Single Crystal Reactive Etching and Metallization) [5].

In this case, due to the non-linear behavior of the piezoelectric elements, a model of the feed actuator has been made to ensure proper teeth engagement and good indexing of the position. Compared to the Preisach model, the Prandtl-Ishlinskii (PI) model was preferred because of its continuity and the possibility of its reversal.

Regarding the constructive solutions presented, modeling and simulation are closely linked to the destination of the studied object: if systems with large displacement domains are studied, emphasis is placed on the constructive solution, its conception and finite element simulations. Modeling based on material resistance calculations can also be used to determine for example the deformations of the compliant mechanisms.

The finite element method is preferred because both static and dynamic simulations can be made (if necessary) to detect the resonance frequencies of mechanical structures, for example. The method is advantageous because multiple constructive variants can be quickly tested.

2. Design of the novel piezoelectric translation module

The aim of the paper was to obtain a piezoelectric translation module based on the inchworm movement type which could work on guideways of large tolerances. When discussing terms such as “large”, tolerances of several μm in case of small displacement guideways are compared with tolerances of long commercial guideways that can easily go up to ten times larger.

Thus, a commercial linear slide (TKVD30-G3) with appropriate plane surfaces (required in order to provide a suitable surface for braking) and linear ball bushing (KWVE30-B-S-G3-V1) were selected as appropriate representatives for commercial guideways systems and the translation system was conceived based on this type of guiding system.

The concept of the structure that is implementing the movement is shown in Fig. 8.

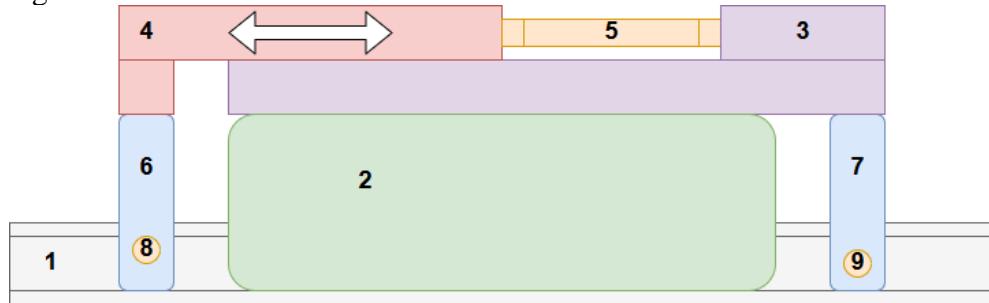


Fig. 8. Diagram of the new concept of linear positioning with piezoelectric actuators.

In the diagram in Fig. 8 the numbered components are: (1) the linear slide, (2) the linear ball bushing, (3) the support (connected to the linear ball bushing, which together with the piece (4) forms a linear guide, and the braking assemblies (6) and (7) containing two opposing actuators (8) and (8') as shown in Fig. 9.

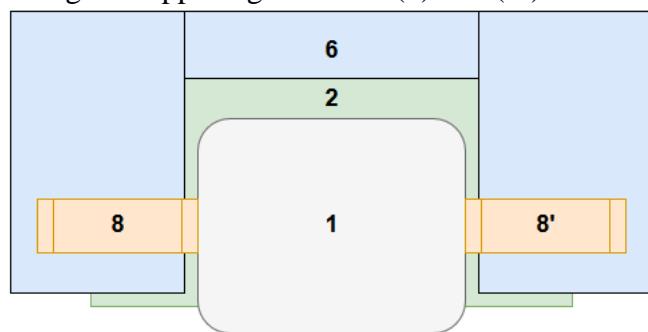


Fig. 9. Simplified diagram of the braking assembly concept

The linear positioning module can move using the “inchworm” movement pattern, by successively grasping and releasing the linear slide and appropriately

changing the distance between the barking assemblies using the displacement of piezoelectric actuator (5).

The braking assembly (6) is comprised of two piezoelectric actuators arranged symmetrically with respect to the linear slide (1), inside a housing. An electrical signal (-30...+150V in case of SA070742-2BE [6]) ensures a displacement of up to $70\mu\text{m}$ for each actuator, to a total of $140\mu\text{m}$ which should be enough to compensate any guideway large tolerance. The two braking systems are similar.

The selected actuator provides the largest displacement in the series and a blocking force of 1800N. A displacement $70\mu\text{m}$ is excellent but it is possible that due to the high capacity ($10\mu\text{F}$) a larger power source will be necessary in order to achieve high frequency stepping.

All actuators must be pre-tensioned and disc-springs were chosen for their mechanical stiffness (large force for small deformation) and compact form. In this regard, discs with 10 mm outer diameter, inside diameter of 5.2 mm and thickness of 0.5 mm were selected from catalogs, being best suited because their dimensions similar to those of the piezoelectric actuator and it is possible to modify the force of the springs by serial/parallel mounting.

The concept was modeled in CAD using SolidWorks design software and the outer structure can be seen in Fig. 10. The modular construction ensures a simple fabrication process.

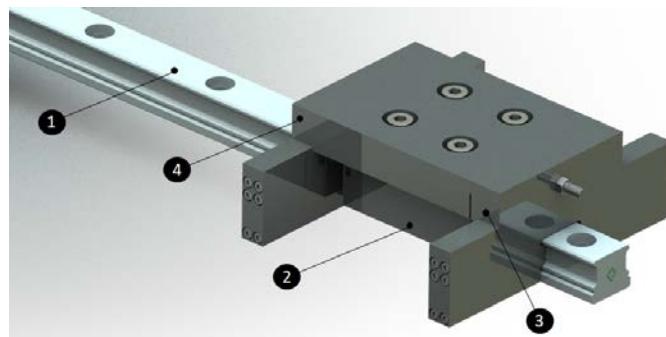


Fig. 10. The CAD 3D model of the piezoelectric translation module with piezoelectric actuators.

In Fig. 10 the following components can be seen: the linear slide (1), the linear ball bushing (3), the horizontal translation assembly covered by a lid (4), as well as the two braking assembly bodies.

In Fig. 11, the piezoelectric translation module is presented in transparency, with no lid. External enclosure of the braking assembly is not shown, to reveal the braking assembly: piezoelectric actuators, braking pistons and pre-tensioning disc springs are visible. In the upper body region, we can

identify the advance actuator and a group of disc springs that pre-tension it and ensure the return of the advance plate when the advance actuator is discharged.

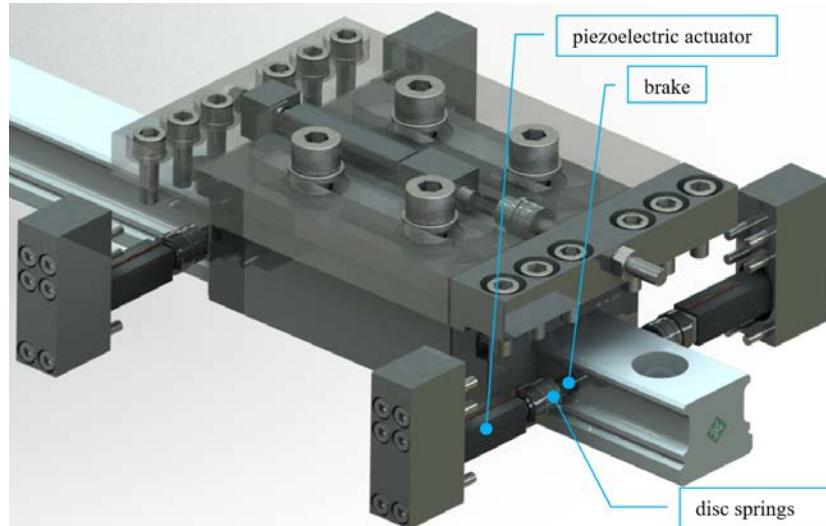


Fig. 11. The 3D CAD view of the inner parts of the piezoelectric translation module.

This solution was designed with scalability in mind. The functional components can easily be adapted to various dimensional types of linear slides with plane surfaces that can be used for braking. The work involved is generally limited to changing only a few dimensions, correlated with the size of the track and the size of the linear ball bushing. Constructive simplicity, scalability and simple adjusting of the play were considered when the solution was chosen.

3. FEA simulation of the piezoelectric translation module

Using the SolidWorks Simulation module, static and dynamic simulations were run to check for unitary efforts, deformations, and eigenfrequencies. The elements on which this analysis was carried out were the brake piston and the main body of the braking assembly.

The brake piston has been simulated only statically and the index body both static and dynamic as these were deemed to be the relevant ones in this case.

The features of the mesh network for the brake piston (figure 12) were: Solid Mesh with 4 Jacobian points, element size 0.43 mm, tolerance 0.02 mm, total nodes 68881, total tetrahedral elements 46972.

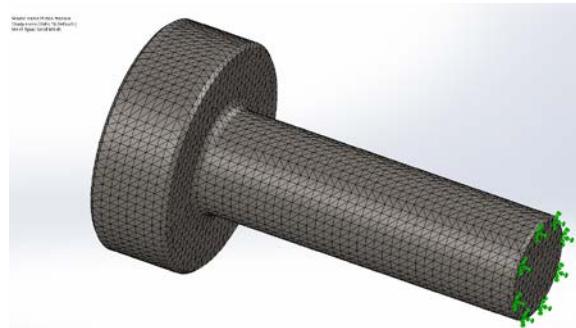


Fig. 12. Mesh network for the brake piston.

In Fig. 13 a) is presented the distribution of mechanical stresses and in figure 13 b) the deformations for a load of 1800 N, which is the blocking force of the piezoelectric actuator. The objective of this simulation is to check the piston resistance.

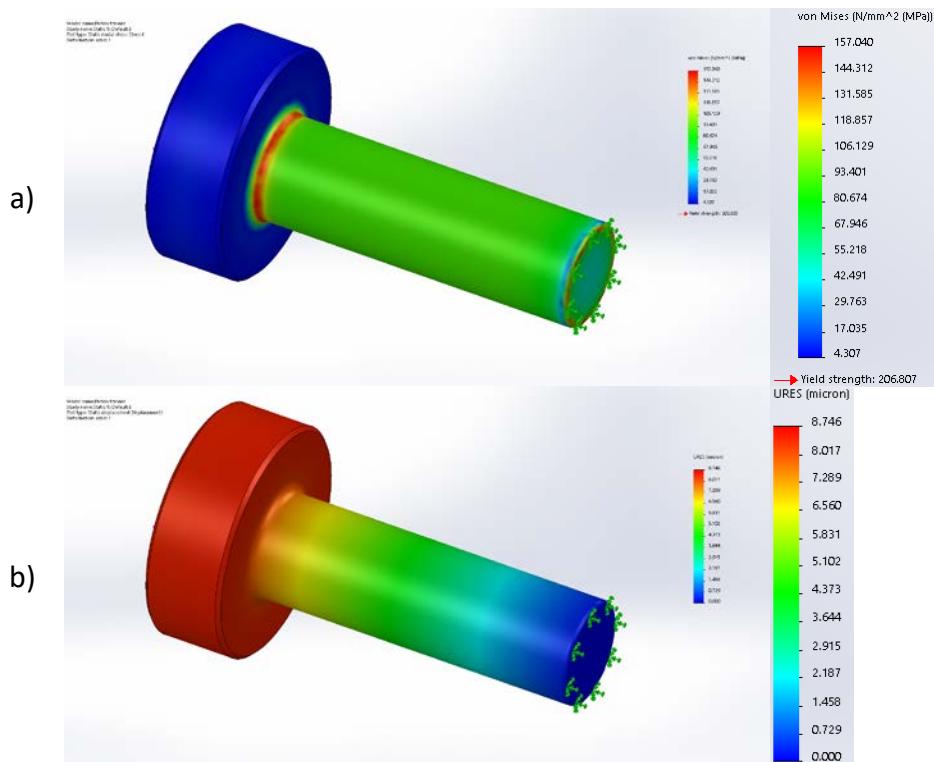


Fig. 13. The simulation results for 1800N load: a) the mechanical stresses in the piston body (with enlarged scale); b) the deformation of the piston body (with enlarged scale)

The maximum load (fig. 13 a)) on which the brake piston is subjected is 157.040 MPa while the material used for the simulation (AISI 304) has a yield strength of 206.807 MPa, thus the body does not suffer plastic deformation.

The maximum deformation (Fig. 13 b)) was 8.746 μ m, but this can only occur at the blocking force of 1800 N. The real load will be lower, since the capable force decreases with actuator displacement and only after contact is achieved the force begins to increase. The force will naturally vary in accordance with the actual size of the gap but it can be maximized by careful assembly and tuning of the braking assemblies in relation to the linear slide.

The features of the mesh network for the main body part of the braking assembly, (fig. 14) were: Solid Mesh with 4 Jacobian points, element size 1.76855 mm, tolerance 0.0884276 mm, total nodes 97502, total tetrahedral elements 63205.

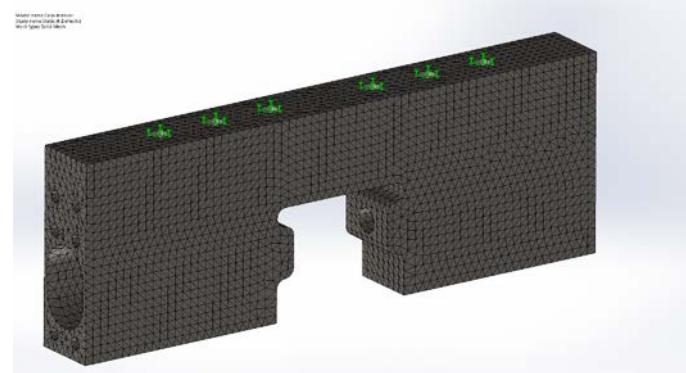
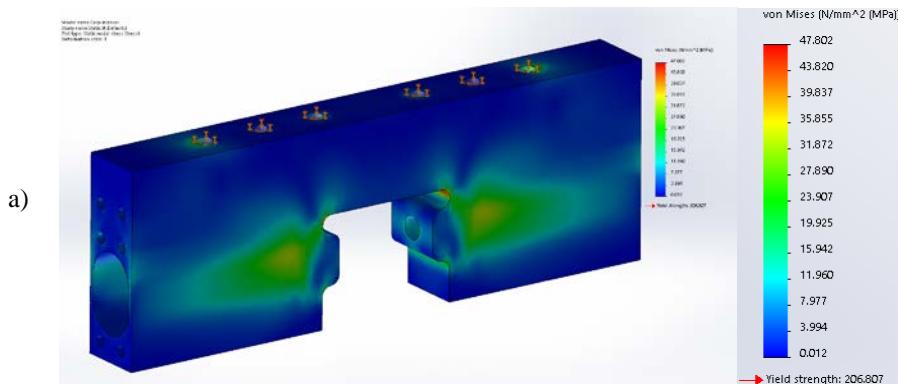


Fig. 14. Mesh network for the main body part of the braking assembly

In the case of static stresses (Fig. 15 a)) for the braking assembly main body, the maximum unitary effort was evaluated at 47,802 MPa with the yield strength limit at 206,807 MPa, thus without plastic deformation.



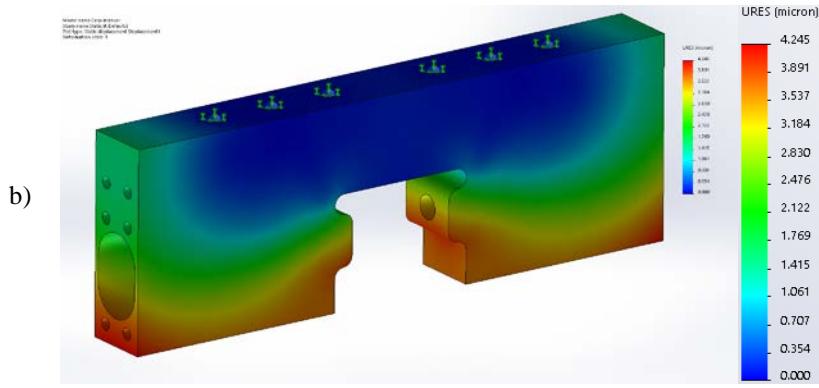


Fig. 15. The simulation results for 1800N load: a) the mechanical stresses in the braking assembly main body; b) the deformation of the braking assembly main body

Maximum index deformation of the braking assembly main body (Fig. 15 b)) was evaluated at $4.245\mu\text{m}$, with the same discussion as above regarding the actual maximum force.

In order to find the eigenfrequencies, a structure consisting of the main body for the braking assembly and the two side caps was used. These were treated as a common body fixed with screws at the top. The mesh network is presented in Fig. 16.

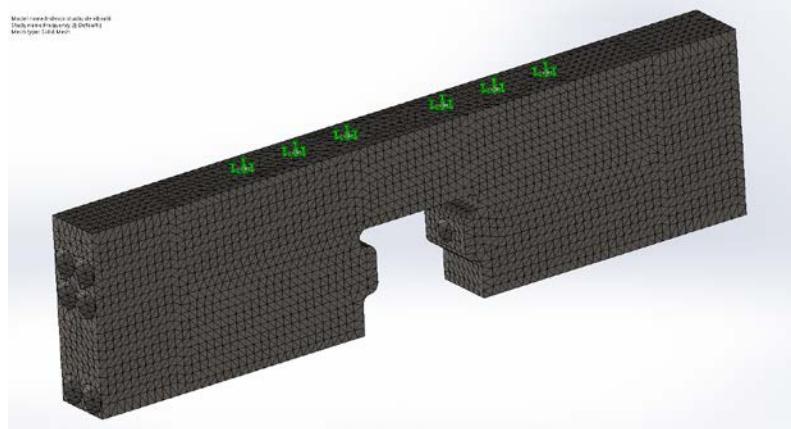


Fig. 16. Mesh network of the indexer housing to simulate its own vibration modes.

The first 4 eigenmodes are presented in figures 17 – 20 and in Table 1 are listed the first 14 eigenfrequencies of the braking assembly housing.

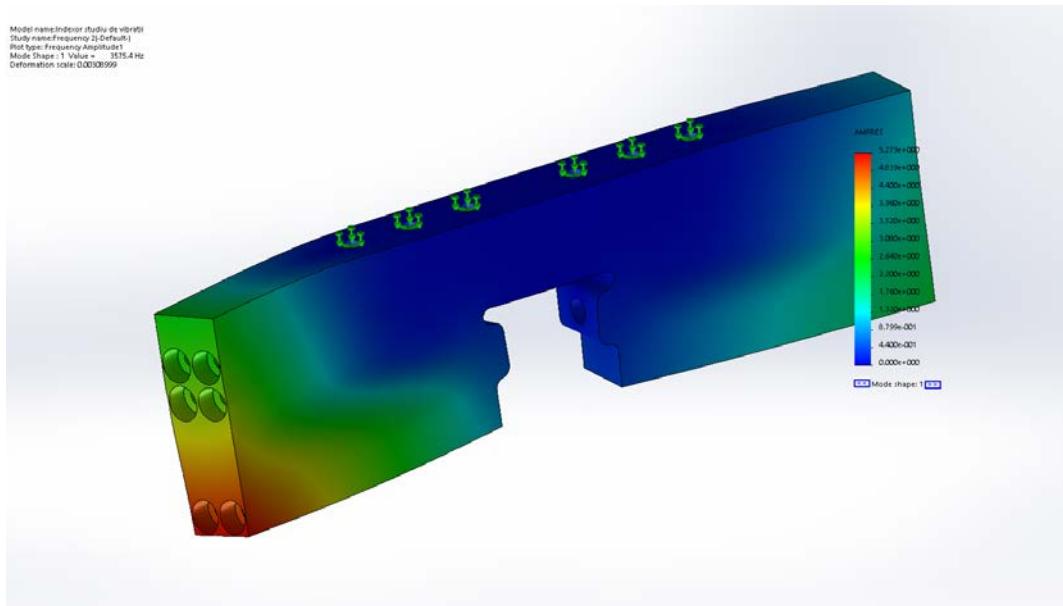


Fig. 17. First eigenmode

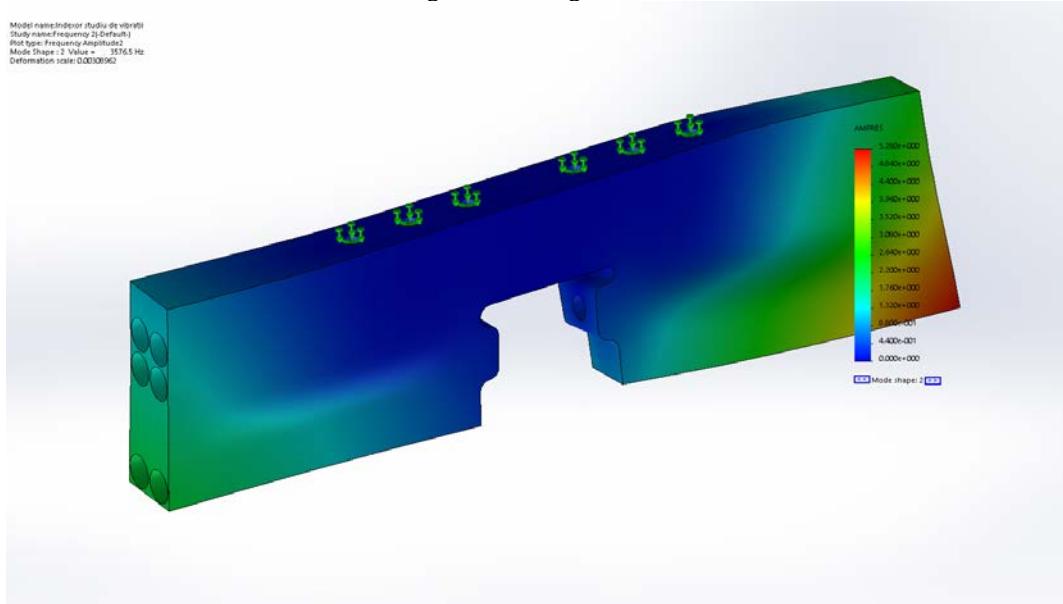


Fig. 18. Second eigenmode

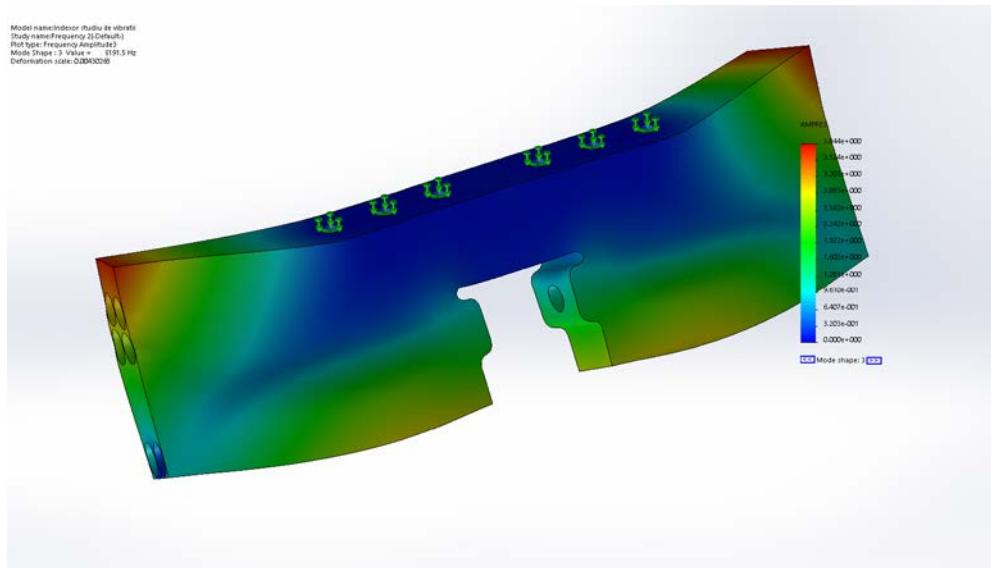


Fig. 19. Third eigenmode

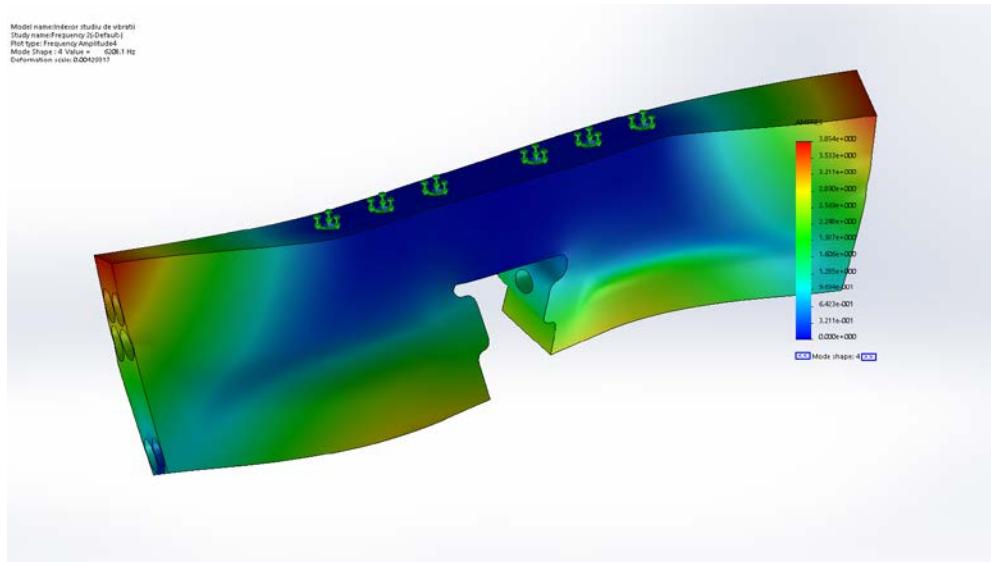


Fig. 20. Fourth eigenmode

Table 1

The first 14 eigenfrequencies of the braking assembly housing

Mode No.	1	2	3	4	5	6	7
Frequency (Hertz)	3575.4	3576.5	6191.5	6208.1	6355.6	6356.3	11255.0
Mode No.	8	9	10	11	12	13	14
Frequency (Hertz)	11277.0	16744.0	16785.0	17032.0	17032.0	21778.0	21794.0

From Table 1 it can be seen that natural vibration frequencies are above 3.5 kHz. Considering that the initially used power amplifier was designed for less bandwidth and for actuators with a capacity of ~ 630 nF (compared to current capacity much higher than 10 μ F), it can be assumed that the module will achieve the resonance frequency.

Comparing these results with the results published in 2004 by Peter [1], where a first eigenmode was found for 920Hz, the current structure has a much higher natural resonance frequency but as mentioned, the high capacity of the actuators (20 μ F in parallel configuration for simultaneous control) could limit the actual actuation frequency to a much lower value. This value could be found if a specific power supply (used for control of the translation module) impedance and power would be known. In any case, the actual achievable frequency will be measured when after the fabrication and assembly of the module.

The static tests have shown that the parts are capable of withstanding the maximum level of forces generated by the actuators.

6. Conclusions

The 3D CAD modeling and simulation performed using SolidWorks Software, for a novel linear piezoelectric translation module was presented. It has been found that the model has several constructive advantages compared to the other models, and through simulation a much larger first eigenmode frequency was found suggesting the capacity for higher positioning speeds but the selected actuator with its large capacity could prevent this. A high bandwidth power supply would be required in order to achieve a high speed.

As a concept, the new model may be bulkier than some of the other model presented for comparison, but it has several distinctive advantages:

- it can be used for large displacements,
- it can be mounted directly on commercial linear guidance systems,
- it can be easily adapted for different typological dimensions of linear slides and bushing assemblies,
- used for linear guides of precisely known tolerances, the necessary travel of the piezoelectric actuators could be less, and piezoelectric actuators with larger capacitance could be selected, increasing the stepping speed towards the first eigenfrequency.

The novel model will be further studied in the future phases of the research.

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² “Actuation, analysis and control for increasing the energy efficiency of industrial equipment and installations used in electrical engineering, mechatronics and fluidics”