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АНАЛІЗ ДИНАМІКИ 3Д-ПРИНТЕРА З МЕХАНІЗМОМ ПАРАРЕЛЬНОЇ СТРУКТУРИ

Розгляуті особлиовсті динамічного аналізу 3Д-принтера з механізмами параельної структури, моделювання динамічних процесів систем твердих тіл в ADAMS, числове моделювання контакту з виокристнням IMPACT функції та створення моделей для частотного аналізу .

Ключові слова: *аналіз динаміки, симуляція систем з твердих тіл, 3Д-принтер, паралельная кінематика, числова оптимізація.*

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DYNAMIC ANALYSIS OF A 3D - PRINTER WITH PARALLEL STRUCTURE MECHANISMS

The focus is on some features of dynamic modelling of 3D printer with mechanisms of parallel structure, modeling dynamic processes in ADAMS, numerical modeling of contact and using the IMPACT function for contact modeling, model building for vibration analysis and modal analysis

Key words: *dynamic analysis, multy body system simulation, 3D-printer, parallel kinematics, numeric optimization*

Some features of dynamic modeling in Autodesk Inventor CAD.´ The object of research into the dynamics of parallel structure mechanisms (PSS) [4] is a mathematical description of the forces and moments acting on it in the form of equations of motion, in other words, the creation and analysis of a dynamic model with a closed kinematic chain [3,7]. These equations are necessary for solving the direct and inverse problem of dynamics [2], synthesis of control laws, and evaluation of the quality of the mechanism design. In general, the preparation of the dynamic calculation model includes the following works: simplification of the composite model; creation of mechanical connections; determination of the loading scheme of model elements. To speed up the modeling process and make it more reliable, before adding connections and forces (in the extreme case after that), it is necessary to perform the following definitions: the features of the assembly of the node and the purpose of modeling; the most important (from the point of view of dynamic analysis) assembly components. In this way, the finished assembly should be simplified to a model for analysis [8]. In the Autodesk Inventor environment, this can easily be done using the Suppress command. Fig.1 shows the model after simplification, which is completely ready for dynamic modeling.

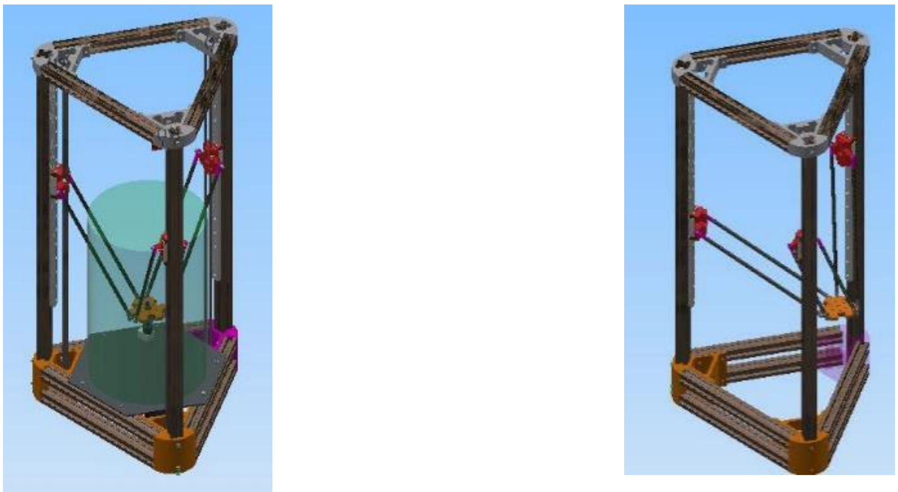


Fig. 1. **Triglide-based 3D printer model before and after simplification**

Using the "Automatically convert dependencies to standard connections" command, which is enabled by default in the "Dynamic Modeling Parameters" dialog box in the form of a checkbox with the caption "Automatically convert dependencies to standard connections" (The program automatically converts component structural connections into standard Connections made using automatic transformation can be edited later by changing the direction vectors of the main part and the attached one. Let's check the correctness of the automatic generation of connections in the model using the "Model state" command (Fig. 2)

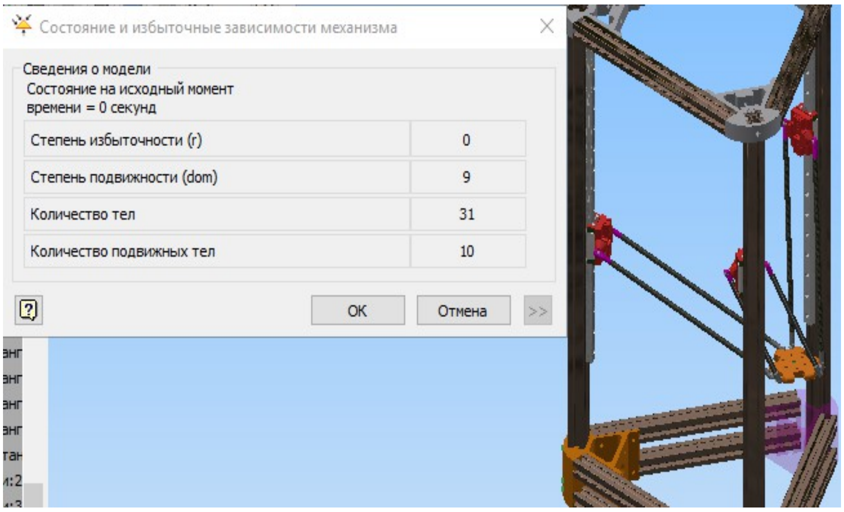
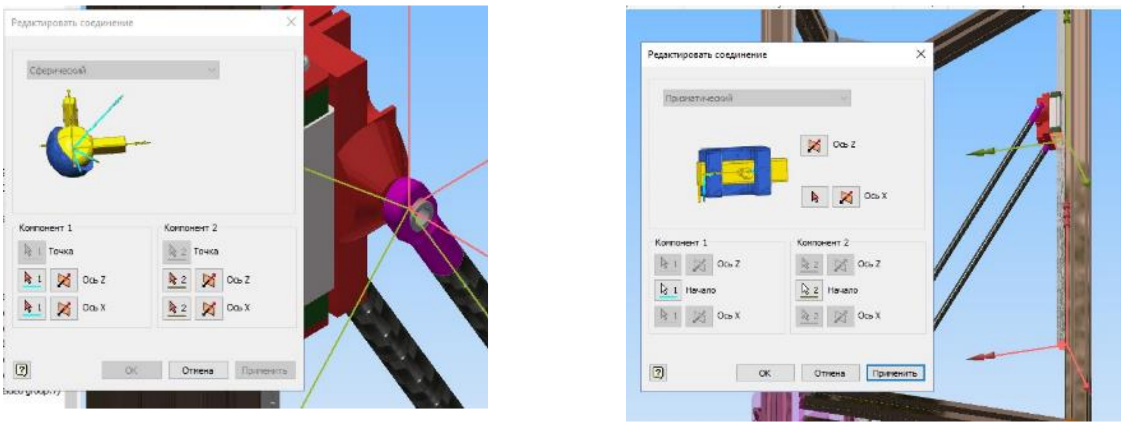


Fig. 2. **Verification and structural analysis of the mechanism in Autodesk Inventor**

As can be seen from Fig. 2, the number of degrees of mobility is 9. This is the correct value, because 3 of them are the movement of the platform and 6 are excess degrees of freedom in the rods around their own axis, due to the use of spherical joints, not cardan joints. Thus, after starting the dynamic simulation, 6 spherical joints were automatically created in the hinges, with the further possibility of their editing (Fig. 3, a) and 3 prismatic joints between rails and carriages. (Fig. 3, b). So in fig. 3, b. it can be seen that the corresponding lower planes of the parts are selected as the origin of the coordinate system of each element of the connection, the Z axis is directed upwards.



a) b)

Fig. 3. **Editing of automatically generated connections: a-spherical; b-prismatic**

In the dynamic modeling module, types of kinematic connections are available, implementing all existing classes of kinematic pairs: - standard connections ("Rotation", "Prismatic", "Cylindrical", "Spherical", "Flat", "Point - segment", "Segment - plane", "Point - plane", "Space", "Welded" "); - rolling joints ("Cylinder on plane", "Cylinder on cylinder", "Cylinder in cylinder", "Cylinder-curve", "Belt", "Cone on plane", "Cone on cone", "Cone in cone", "Screw", "Worm gear"); - sliding joints ("Cylinder on plane", "Cylinder on cylinder", "Cylinder in cylinder", "Cylinder-curve", "Point-curve"); - 2D contact connections ("2D contact"); power connections ("Spring / shock absorber / jack", "3D contact"). Next, configure the connection properties using the "Connection Properties" tab. For prismatic connections, we will accept the following restrictions and settings. To limit the mobility of carriages along the rail, we set the minimum and maximum position settings so that the position of the working body (carriage) in these extremes corresponded to the minimum and maximum at the points of the working space along the Z axis (Fig. 4). After creating a connection (by inserting or transforming structural connections), the correct operation of the mechanism can be checked in two ways. After using the "State and redundant mechanism dependencies" mode to eliminate redundant dependencies, you can perform a check of the new solution. The mechanism can also be activated without running the simulation. There are several such ways of putting the mechanism into action. One of them involves the use of the "Dynamic movement" function (Fig. 5).

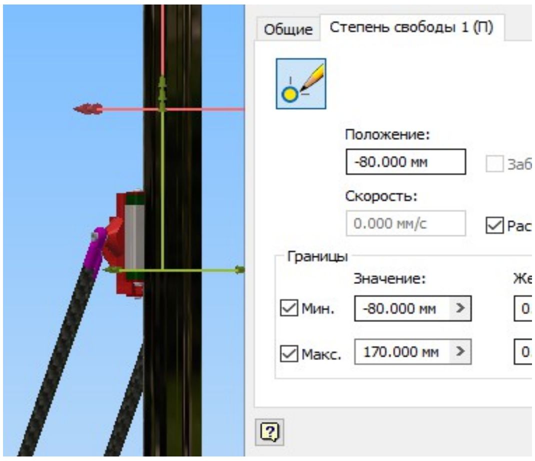


Fig. 4. **Determination of the minimum and maximum position of the carriage**

Other methods are described below. Dynamic movement of the part (simplified mode). To activate the mechanism, use the drag function, which is available both in the environment for working with assembly models and and in the dynamic modeling environment. This mode is similar to the Dynamic Motion mode. For example, it is not possible to control the damping of the overall motion. If the drag parts are connected to the joints, their behavior is limited by the degrees of freedom. However, you can drag and drop details that are not connected to connections.

The mechanism should be carefully studied to find out the observed type of movement. To set the mechanism in motion, it is necessary: - In the graphics window, select a point on the part that should move. - Press and hold the left mouse button to move the part. - To stop the movement of the part, simply release the mouse button. In the graphic window, the part will be displayed in the position in which it was at the moment when the mouse button was released. - Kinematic movement. The mechanism can be activated by changing the initial position of the degree of freedom of the standard connection. This process allows you to check the correctness of the kinematic function of the mechanism and determine the tolerances for correct operation. For the correctness of the further modeling, we set the internal forces in the connections, so there are certain frictional forces in the spherical joints and guides, which should be taken into account. Thus, the result of the work done in the dynamic modeling environment is a prepared model for further dynamic modeling.

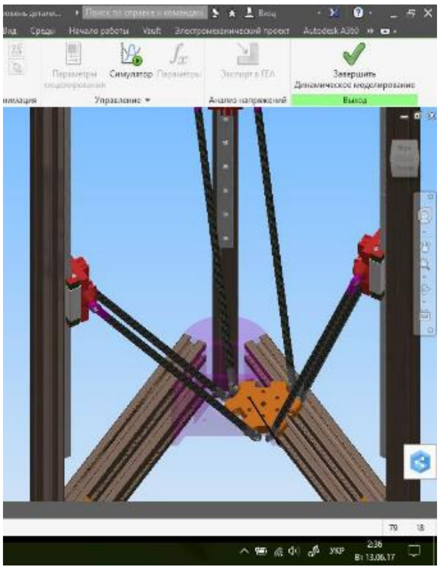


Fig. 5. **Checking the functionality of the model without using the function "Dynamic movement"**

Modeling of dynamic processes in ADAMS

ADAMS is a software product from MSC Corporation for dynamic and kinematic analysis of rigid body systems. Main features and advantages of ADAMS: • Development of calculation models of the researched products, taking into account the features of their design to the maximum extent, including the high identity of the appearance, which in many cases facilitates the construction of models, their adjustment and analysis of the obtained results; • Calculate product parameters that determine their performance and accuracy (movement, speed and acceleration of product components, operating loads, dimensions of the space required for moving parts of the machine, etc.); • Perform parameter optimization. One of the tasks that arose during the execution was the optimization of the 3D printer and the creation of a multi-purpose small-sized robotic platform based on it based on the modular principle. So, for example, the implementation of the possibility of replacing such a module as a tool presents us with the following task. Let there be two tools A nozzle with an extruder for printing and an adaptive gripper [1,6]. In this way, it is necessary to "tear off" the printed sample from the surface and transfer it with the help of a gripper to a given position, which may go beyond the working area of the robot's printing. In order to simulate the separation and gripping process, it was necessary to recreate a flexible grip (Fig. 6).

a) b)

Fig. 6. **Adaptive gripping device, a – working prototype, b – ADAMS model**

ADAMS allows modeling: multi-contact; dynamic friction; contact between three-dimensional solid geometry; contact between two-dimensional geometries The Adams solver (C++) has two geometry engines that it uses for 3D contacts. It uses Parasolid, a geometry toolkit with EDS / Unigraphics and RAPID. Currently, RAPID is the default, and Adams Solver (C++) supports version 2.01. Adamc Solver distinguishes some geometries as analytic. The following contact types are fully analytical. These two geometries are considered analytically: sphere by sphere; cylinder by cylinder. Numerical methods must be used to model the grip of an arbitrary object with an adaptive grip. Contact modeling can be conventionally divided into two stages. The first is the simulation of the normal component of the force and the second is the simulation of the tangential force. Fig. 7 shows the components of the contact modeling function in ADAMS.

Two models are available in Adams Solver (C++) to calculate the normal force: IMPACT function model;

Coefficient of restitution or POISSON model. Both models result from the regularization of normal contact constraints. Penalty regularization is a modeling technique in mechanics in which a constraint is applied mathematically by applying forces along the gradient of the constraint. The magnitude is a function of constraint violation.

Contact between solid bodies theoretically requires that the two bodies do not penetrate each other. This can be expressed as a one-sided (inequality) constraint. The contact force is the force associated with the application of this constraint. Processing these auxiliary constraints is usually carried out in one of two ways - either by introducing Lagrange multipliers or by regularizing fines. For contact problems, the latter technique has the advantage of simplicity; no additional equations or variables are introduced. This is particularly useful for intermittent contact and algorithmic control of active and inactive conditions associated with unilateral constraints. In addition, the concept is easily interpreted from a physical point of view. For example, the magnitude of the reaction contact force is equal to the product of the stiffness of the material and the penetration between the contacting bodies, similar to the spring force. For these reasons, the Adams (C++) solver uses penalty regularization to enforce all contact constraints.

Adams Solver (C++) uses a relatively simple model based on friction velocity for contacts (Fig. 9). Indication of friction behavior is optional. The figure below shows how the coefficient of friction varies with sliding speed.

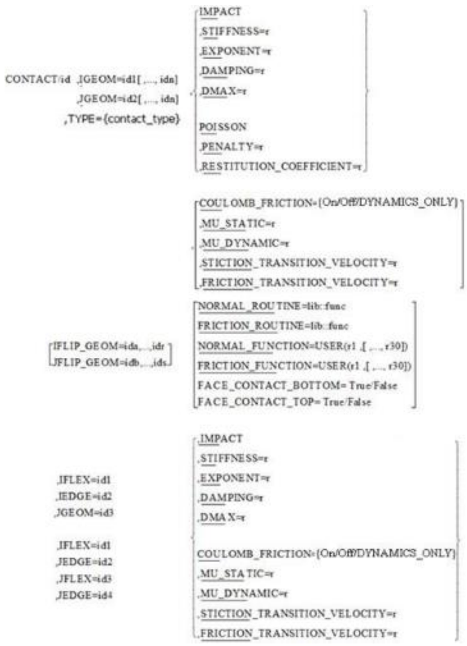
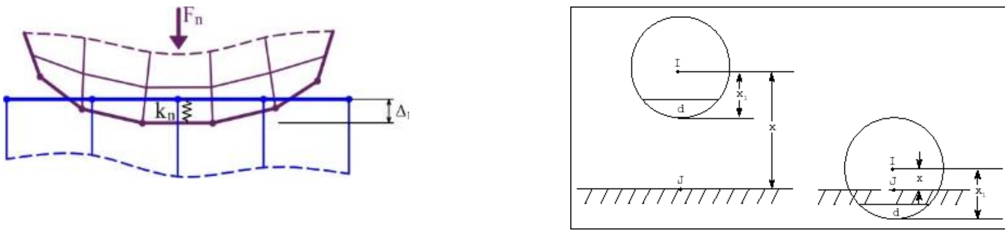


Fig. 7. **Elements of the contact function IMPACT in ADAMS**



a) b)

Fig. 8. **Numerical simulation of contact (a) and use of the IMPACT function for contact simulation (b)**

Model development for vibration and modal analysis

Signal analysis is the process of determining the system's response to an unknown, in general, disturbance and presenting them in a form that is easy to understand. System analysis is a method of determining the characteristic properties of systems. It can be conducted by disrupting the system by measuring forces and determining the response/force ratio (sensitivity). For linear systems, this ratio is an independent parameter characteristic of these systems. This parameter remains constant regardless of whether the system is in an excited or quiescent state. The frequency characteristic of a given output channel is the dependence of the amplitude of the output signal of the device or transmission, amplification or signal processing system on the frequency of the input signal of constant amplitude. To calculate the frequency response, the linearized model is represented as:

$$s\mathbf{x}(s) = \mathbf{A}\,\mathbf{x}(s) + \mathbf{B}\,\mathbf{u}(s) \qquad (1)$$
$$\mathbf{y}(s) = \mathbf{C}\,\mathbf{x}(s) + \mathbf{D}\,\mathbf{u}(s),$$

where: s - Laplace variable; A, B, C, D - state matrices for a linearized model; x (s) - the Laplace transform of the linearization states; u (s) and y (s) - Laplace transformations of input and output channels.

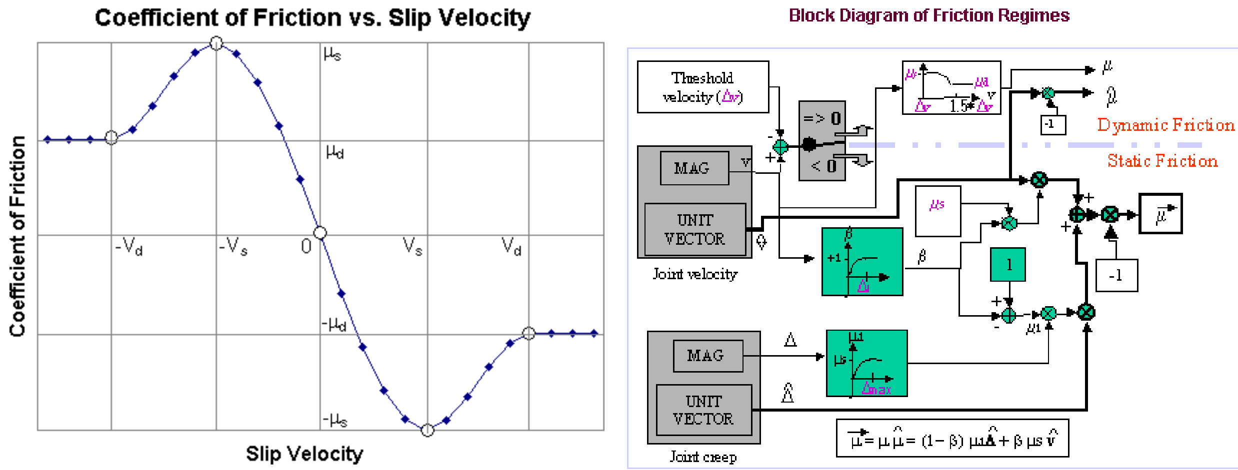
Frequency response is calculated by:

$$\mathbf{H}(s) = \frac{\mathbf{y}(s)}{\mathbf{u}(s)} = \mathbf{C}(s\mathbf{I} - \mathbf{A})^{-1}\mathbf{B} + \mathbf{D} \qquad (2)$$

The main problem in vibration analysis is the construction of an appropriate mathematical model.

It should reflect all important moving masses and elastic elements. Concepts and models shown in fig. 10.

Fig. 11 shows the frequency response in the low-frequency range for a 3D - printer with a ball-screw pair (BSP) instead of belt transmission [5].



a) b)

Fig. 9. **Modeling contact friction in ADAMS. Friction coefficient vs slip velocity chart (a), block diagram (b)**

The model with modal parameters is shown in Fig. 12. It is built using two parameters that can be obtained from the results of measuring frequency characteristics. In Fig. 12, the function H(ω) is determined by the coordinates of the pole (p) and subtraction (R) and their complex related values (p* and R*). The pole and offset coordinate, in turn, is determined through spatial parameters. The pole coordinate is a complex quantity. The numerical value of its real part (σ) represents the damping rate of oscillations. This is shown on the graph of the dependence of the impulse response on time. In the frequency domain, σ is half the bandwidth (-3 dB) of the peak of the frequency response. The imaginary part of the pole coordinate represents the modal frequency - the natural frequency of freely damped oscillations (ωd). The subtraction in the case of a system with one degree of freedom is a dummy value that reflects the intensity of the oscillation mode. Fig. 12 shows all modes of oscillations of a triglide-based 3D - printer.

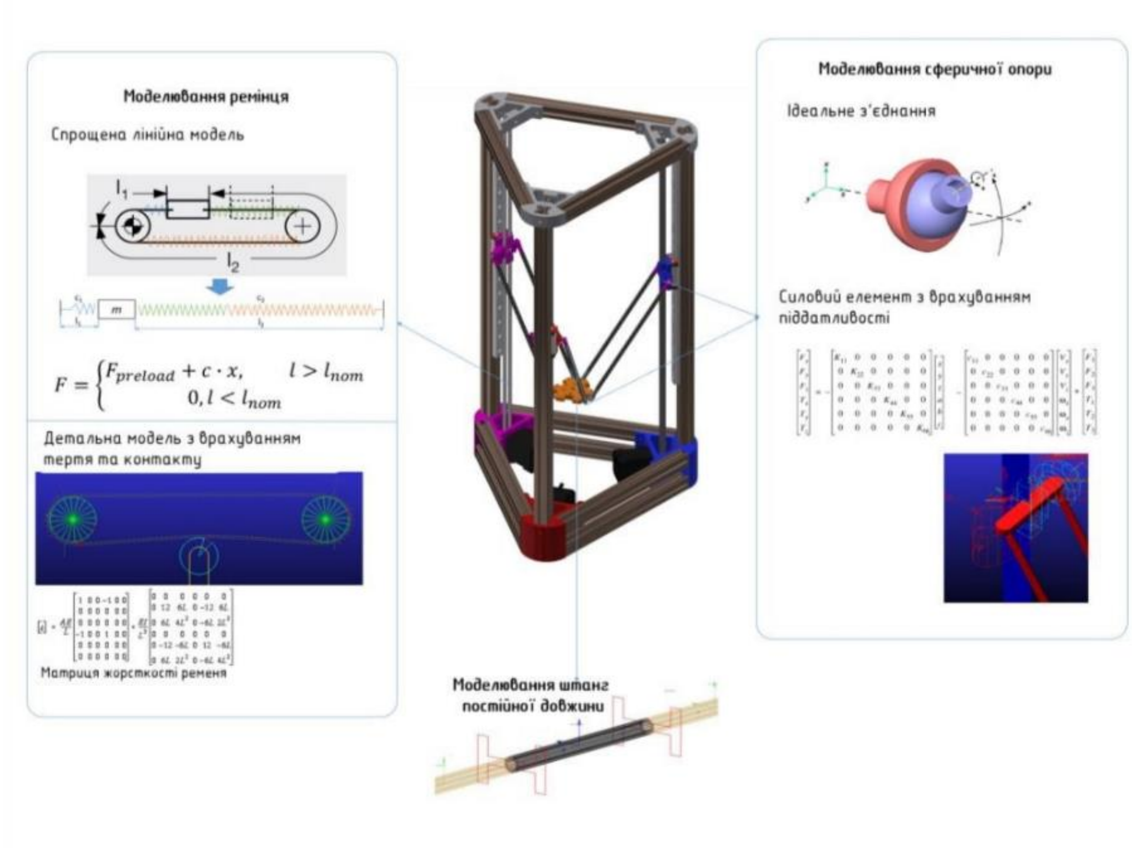


Fig. 10. **Some concepts and submodels (submodels) for the implementation of dynamic analysis**

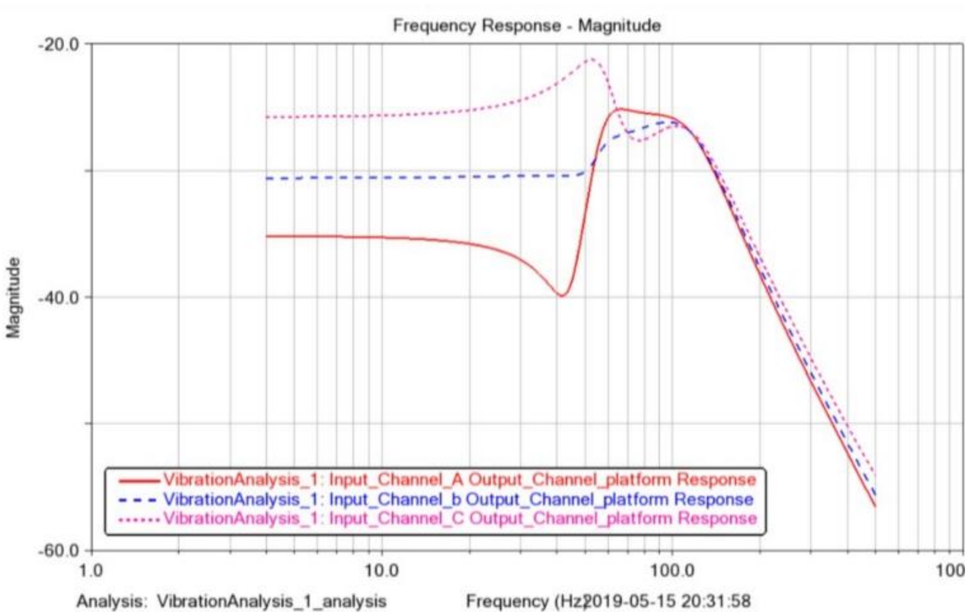


Fig. 11. **Frequency response for a triglide-based 3D - printer with KGP in low frequency range**

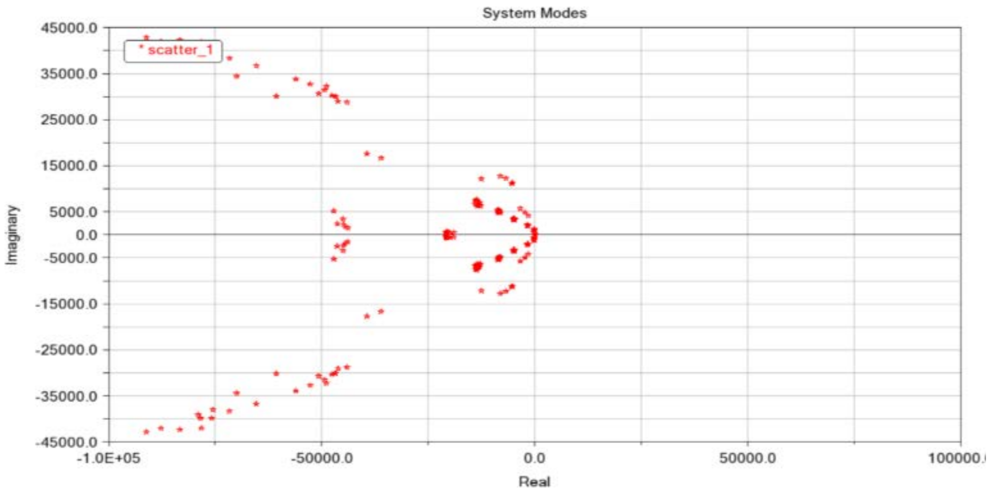
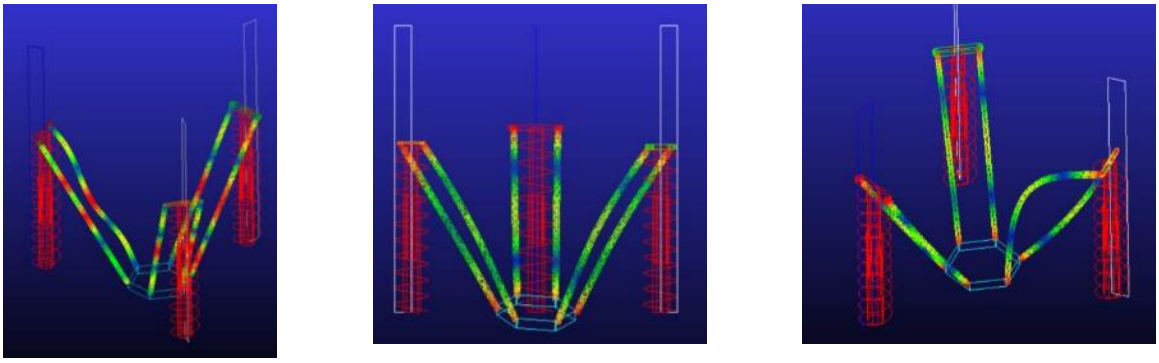


Fig. 12. **All modes of oscillations of a triglide-based 3D - printer**

Fig 13 shows some forms of modes from the working frequency range.



ω_d = 561 Hz; ε = 40%	ω_d = 3300 Hz; ε = 80%	ω_d = 1175 Hz; ε = 14%

Fig 13. **Some 3D printer vibration modes with corresponding frequencies and relative damping**

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