

## VIBRATION-ISOLATION SYSTEM WITH GYROSCOPIC STABILIZER

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**Abstract.** *Research of vibration-isolation system with gyroscopic stabilizer is based on the research of vibration-isolation of lying human body during transportation. During the previous research there was designed the three-degree of freedom vibration-isolation system - one vertical translation and two rotations around transversal and longitudinal axes. Vertical translation is conducted by paralelogram mechanism or scissors mechanism; rotations are conducted by gimbals (Cardan suspension). We have suggested extending this system by two-axis gyroscopic stabilizer with power gyroscopes. The power gyroscopes with sufficient magnitude of angular momentum move the natural frequencies (without gyroscopic stabilizer approximately 1-5Hz) of gimbals frames into the higher values (nutation frequency) and precession motion of gyroscopes is slow or aperiodic. Thus it means that the natural frequencies of gimbals, which previously have been located into the band of excitation signals spectra, have been moved outside this band. The paper describes suggested solution of the lying human body vibration-isolation and its mathematical models. There is also described the optimization of the systems of radial corrections. There is introduced the new solution of the compensation drive – based on the control of pneumatic springs. And also there are discussed some aspects of hard nonlinearities effects on the behaviour of described mechanical (or mechatronic) system.*

## 1 INTRODUCTION

Our research of vibration-isolation system with gyroscopic stabilizer is based on the research of vibration-isolation of lying human body during transportation. In the course of previous research there was designed the three-degree of freedom vibration-isolation system; vertical translation and two rotations around transversal and longitudinal axes. Vertical translation is conducted by paralelogram mechanism or scissors mechanism; rotations are conducted by gimbals (Cardan suspension). We have suggested extending this system by two-axis gyroscopic stabilizer with power gyroscopes (control moment gyroscope) mounted on the inner gimbal frame. The power gyroscopes with the sufficient magnitude of angular momentum move the natural frequencies (without gyroscopic stabilizer approximately 1-5Hz) of gimbals frames into the higher values (nutation frequency), while precession motion of gyroscopes is slow or aperiodic. Thus it means that the natural frequencies of gimbals, which previously have been located into the band of excitation signals spectra, have been moved outside this band. This property of gyroscopic stabilizer could be very advantageous for vibration-isolation of e.g. laying patient.

## 2 DESCRIPTION OF TWO-AXIS PLATFORM

Mechanical system of vibration-isolation platform with three degrees of freedom and gyroscopic stabilizer (Figure 1) consists of the paralelogram or the scissors mechanism, which leads the vertical suspension. Air springs and damper could be mounted between upper and lower base of paralelogram (scissors mechanism) or between paralelogram (scissors mechanism) arms and upper or lower base, if certain transmission is required. The gimbals are mounted on the upper base of vertical suspension – outer gimbal axle is transverse oriented, inner gimbal axle is longitudinal oriented. Air springs are mounted between inner and outer gimbal for springing of inner gimbal tilt about longitudinal axle, and also mounted between outer gimbal and upper base of vertical suspension mechanism for springing of outer gimbal tilt about transverse axle. The two-axis gyroscopic stabilizer is mounted in the inner gimbal.

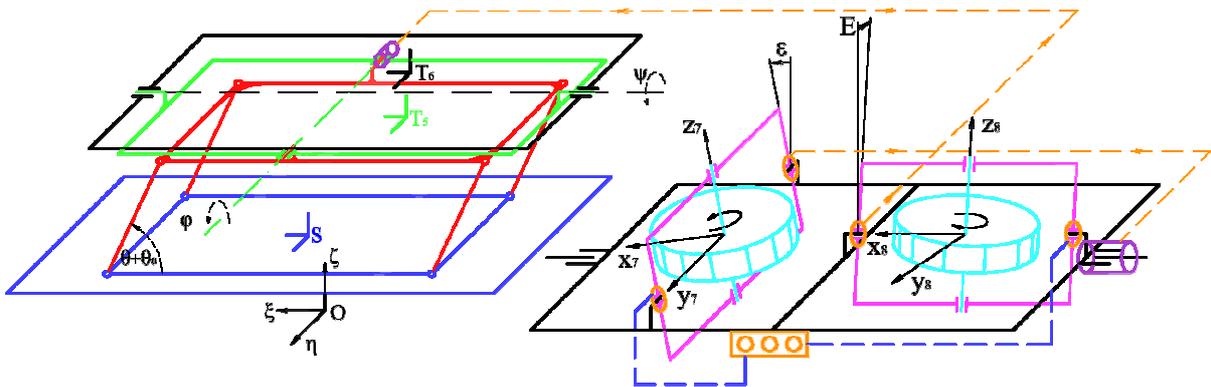


Figure 1: Vibration-isolation platform with 3DOF and two-axis gyroscopic stabilizer.

Gyroscopic stabilizer consists of two flywheels with vertical spin axis; each of them is suspended in precession frame with horizontal precession axis; longitudinal and transverse respectively. Correction and compensation feedbacks (in Russian literature also called “radial corrections”) are very important parts of gyroscopic stabilizer. Compensation feedback acts a torque on the corresponding gimbal frame with respect to the angular displacement of corresponding gyroscope precession frame (corresponding gyroscope has precession axle perpendicular to the corresponding gimbal frame). Correction feedback acts a torque on the

gyroscope precession frame with respect to signal from the apparent vertical sensor – tilt of inner gimbal about longitudinal axis relates to the gyroscope with transversal precession axis; tilt about transversal axis relates to the gyroscope with longitudinal axis. Apparent vertical sensor is mounted on the inner gimbal and its function could be achieved by electronic level or e.g. 3D piezo-resistive accelerometer, which can indicate the relative direction of the acceleration resultant. Correction feedbacks allow the system to respond to the longitudinal or transversal accelerations by tilt of corresponding gimbal frame so that the normal line of the inner gimbal plane achieves the apparent vertical orientation. This function cause that the resultant of acceleration, which acts on the e.g. laying patient, is oriented normally to the plane of the bed and it could be very beneficial for the patient.

## 2.1 Mathematic model of two-axis platform

Mathematic model of above mentioned mechanical system could be derived e.g. using Lagrange equations of second kind:

$$\frac{d}{dt} \frac{\partial T}{\partial \dot{q}_i} - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial q_i} = M_{pi} + M_{Ti} + M_{rk} + M_{pasi} \quad i = 1..5, \quad (1)$$

where indexes  $i$  correspond to the angular displacements of parallelogram arms, outer and inner gimbal frame and precession frames of the gyroscopes respectively. On the right hand sides of these equations there are the torques of air springs, dampers, feedbacks and passive resistances respectively. Equations of gyroscopes flywheels are subsequently:

$$\frac{d}{dt} \frac{\partial T}{\partial \dot{q}_i} - \frac{\partial T}{\partial q_i} + \frac{\partial U}{\partial q_i} = M_{hgi} + M_{pasi} \quad i = 6..7, \quad (2)$$

where on the right hand sides of equations are driving torques of gyroscopes and torques of passive resistances in flywheels bearings. Torques on the right hand sides of the last two equations (2) are balanced after activation of flywheel drive and achieve of its operation spin velocity. The kinetic energies of flywheels are not dependent on the rotation displacement of flywheel spin axis. Thus the angle of the flywheel displacement about its spin axis is the cyclic coordinate. If we suppose that the momentum of the flywheel  $\partial T / \partial \dot{q}_i = H = const.$  (for  $i = 6, 7$ ) we can leave out these equations and consider the constant spin velocity of flywheels in the equations for  $i = 1..5$ .

We have used above described mathematical model (linearized) for tuning of correction and compensation feedback parameters. We have used Hurwitz criteria of stability and also the procedure of optimization of correction and compensation feedbacks parameters which was suggested for these purposes and which was described in [1]. Subsequently we used the linearized model also for modal analysis of described mechanical system [2].

## 3 ONE-AXIS PLATFORM

Mathematical model of two-axis platform has 5 degrees of freedom and is complicated for optimization of feedbacks parameters settings. However the coupling between vertical suspension and gimbals is weak (in [3]), thus it is possible to perform decomposition of this system and to deal with the gyroscopic stabilizer separately. Gimbals can be represented as two subsystems of outer and inner gimbal. Knowledge gained by research of so called one-axis platform – one-axis vibration-isolation platform with the gyroscopic stabilizer (pictured in Figure 2) – can be applied on both gimbals with two-axis gyroscopic stabilizer. Thus it is possible to use a simpler model with less degree of freedom for simulations and analyses.

Equations of motion which describe behaviour of the one-axis gyro platform could be derived analogously to the two-axis platform. In order to use the electronic level as the sensor of apparent vertical, we extended the system of equations of motion by equation of motion of the simple gravity pendulum with the pivot mounted on the stabilized gimbal frame (see Figure 3). We set its damping and natural frequency to reach the same behaviour as the electronic level.

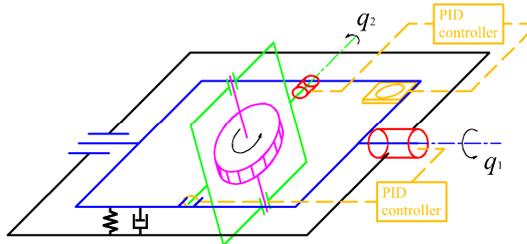


Figure 2: Scheme of one-axis vibration-isolation platform with gyroscopic stabilizer.

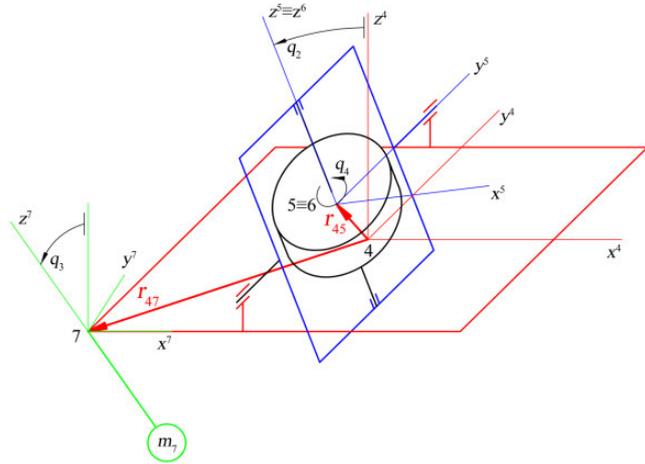


Figure 3: Scheme of connection of apparent vertical sensor model.

### 3.1 Substitution of compensation feedback drive

Compensation system drive applies the torque in the axis of the gimbal frame. The use of the torque motor as a compensation system drive takes several negatives. First should be mentioned the complication of gyro stabilizer design. If we use e.g. the pneumatic rotary motor, high friction torque due to sealing of moving part in the cavity (vane in the cavity or piston in the cylinder) is included into the system and affects its function. In case of the use of an electric motor with the reduction gearbox (usually with high gear ratio) the reduced inertia of the connected gimbal frame is increased. Also self-locking gearboxes avoid the function of the stabilizer absolutely.

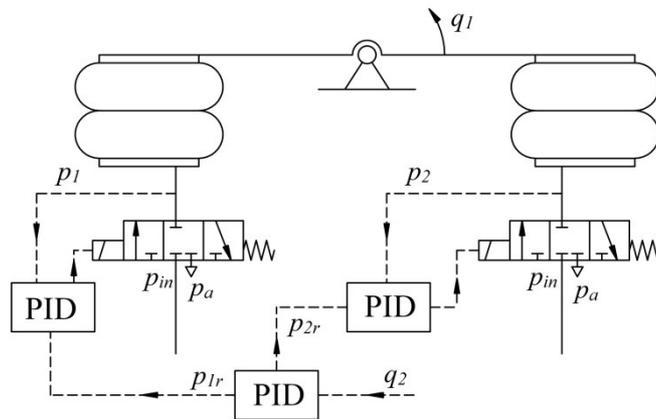


Figure 4: Scheme of the compensation system drive substitution by pressure control of the two air springs.

Air springs, which should be mounted between gimbals or gimbals and base frame, should provide the springing of gimbals rotations about longitudinal or transverse axes. These

springs should also, by applying the active control of air pressure, provide the achievement of horizontal position of the loading area, in case of non symmetric placement of the load (e.g. human body). We have suggested utilizing the active control of the pressure in the air springs instead of the correction feedback drive (see Figure 4). In order to prove the feasibility of this solution, we have derived the nonlinear mathematic models of the active controlled air spring and the control valve, which covers subsonic and choked flow through the control valve. We verified these models by comparison of simulated and real behaviour of the single mass mechanical system with the controlled air spring, during step changes of desired position of the load [4]. Subsequently we have extended the mathematical model of one-axis platform by above mentioned model of controlled air spring accompanied by newly designed control algorithm to produce appropriate correction torque. We have proved the applicability of the suggested solution of compensation feedback drive by comparing the behaviour of the model extended by controlled air springs and of the model where the hard source of correction torque was supposed (in [5]). The delays during pressurizing the air springs cause the difference between desired compensation torque and the torque generated by air springs. This difference is compensated by gyroscopic torque, which causes increase of the gyroscope angular displacement about precession axle (see Figure 5 –  $q_2(t)$ ). Behaviour of the stabilized frame – Figure 5, while  $q_1(t)$  is almost identical for both models.

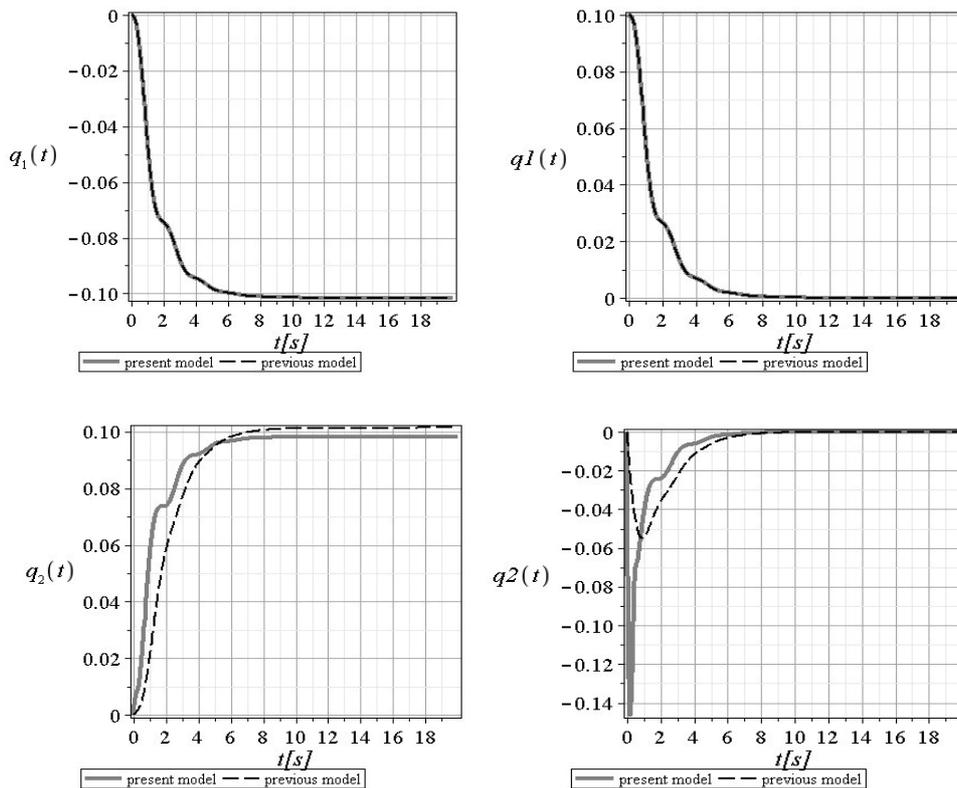


Figure 5: Comparison of responses of model extended by controlled air springs and model with hard source of compensation torque –  $q_1$  is for angular displacement of stabilized frame [rad];  $q_2$  is for angular displacement of gyroscope about precession axis [rad].

### 3.2 Description of suggested compensation feedback drive

Experimental model is equipped by bellows type air springs. Two air springs are mounted symmetrically left and right from the gimbal frame pivot; between gimbal and base frame. For their control we have suggested the electronic controlled pressure regulator (electromagnetic

controlled solenoid valve with integrated electronic regulator), each air spring is connected to the separate control valve. Input signal for each valve defines the desired pressure in the related air spring. Desired pressures in the air springs are dependent on the requested compensation torque, considering the constant arm of torque acted by the air spring and the force caused by the  $i$ -th air spring is

$$F_{Si} = S_{efi}(l_i) \cdot p_i, \quad i = 1, \dots, 2, \quad (3)$$

where  $S_{efi}$  stands for effective area of  $i$ -th air spring, which is dependent on the air spring stroke  $l_i$ ;  $p_i$  is for pressure in the  $i$ -th spring. Expression (3) is valid only for the bellows type air springs, generally the effective area of the air spring is function of its height and pressure of the air inside. Effective area  $S_{efi}$  is known function – given by manufacturer or it is necessary to measure it. Air pressures inside the springs are another state variables defined by another two equations, which must be added to the equations of motion of the mechanical system. These equations are derived from ideal gas law, when isothermal process is considered; respecting the time variability of pressure and dependency of the effective area on the height of spring we obtain equation for the pressure in the  $i$ -th spring

$$p_i \cdot S_{ef}(l_i(q_1)) \frac{dl_i}{dq_1} \dot{q}_1 + \dot{p}_i \cdot V(l_i(q_1)) = RTG_i, \quad (4)$$

where  $q_1$  is for gimbal frame tilt;  $V$  is for volume of the air inside the spring dependent on its stroke;  $G_i$  is mass flow of the air through the control valve in the direction in or out the spring, which is described by subsequent expressions

$$G_{PAi} = \begin{cases} p_{in} \cdot C \cdot \rho_a \sqrt{1 - \left( \frac{p_i/p_{in} - b_{PA}}{1 - b_{PA}} \right)^2}, & \frac{p_i}{p_{in}} > b_{PA}, \\ p_{in} \cdot C \cdot \rho_a, & \frac{p_i}{p_{in}} \leq b_{PA}, \end{cases} \quad (5)$$

$$G_{ARi} = \begin{cases} -p_i \cdot C \cdot \rho_a \sqrt{1 - \left( \frac{p_a/p_i - b_{AR}}{1 - b_{AR}} \right)^2}, & \frac{p_a}{p_i} > b_{AR}, \\ -p_i \cdot C \cdot \rho_a, & \frac{p_a}{p_i} \leq b_{AR}. \end{cases}$$

where  $G_{PAi}$  stands for the flow from source of pressurized air into the  $i$ -th air spring;  $G_{ARi}$  is for the air flow from  $i$ -th air spring into the atmosphere. Upper part of both is valid for subsonic flow and second part is valid for choked flow, when  $b_{PA}$  and  $b_{AR}$  are critical pressure ratios for corresponding flow directions through the valve. Quantity marked as  $C$  describes so called pneumatic conductivity of the valve. It is dependent on the valve opening and should be measured for current valve or can be given by manufacturer. For this model, the pneumatic conductivity of control valve is function of difference of required pressure and current pressure in the output port of the valve, thus  $C = C(p_{ri} - p_i)$ . This function is given by valve design and by integrated regulator parameters, which should be also identified or obtained from the valve manufacturer. We have derived these parameters by modeling of the deflating and inflating of the pressure vessel of given volume and comparing with same characteristics given by manufacturer.

#### 4 EXPERIMENTAL MODEL OF ONE-AXIS PLATFORM

In purpose of verification of theoretic and analytic knowledge we have designed and composed the experimental model of the one-axis vibration isolation platform with gyroscopic stabilizer – see Figure 6. The experimental model is composed of modular aluminium alloy profiles. As a compensation system drive we have used the suggested solution of air springs control, correction system drive is solved by pneumatic rotary motor with swivel vane. On the stabilized frame, we have mounted so called gyroscopic tandem – two identical flywheels with vertical spin axis, opposite spin direction (same magnitude) and perpendicular precession axes. Their precession movements are coupled by gearing – see Figure 6. Controllers of compensation and correction feedbacks are provided by the computer NI PXI, as well as recording of measured data.

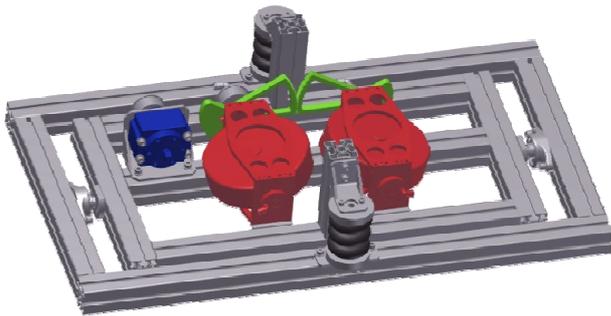


Figure 6: Experimental model.

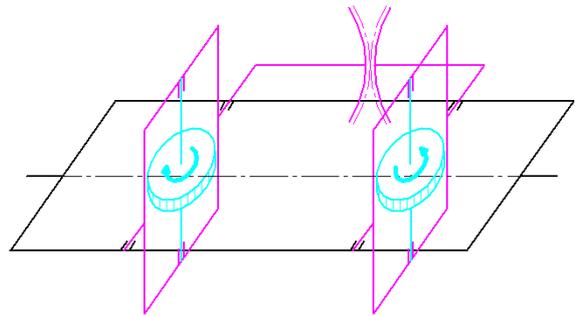


Figure 7: Scheme of the gyroscopic tandem.

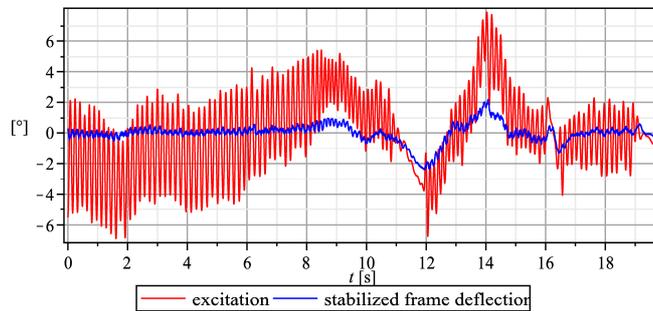


Figure 8: Absolute deflection of stabilized frame compared with absolute deflection of stabilized frame.

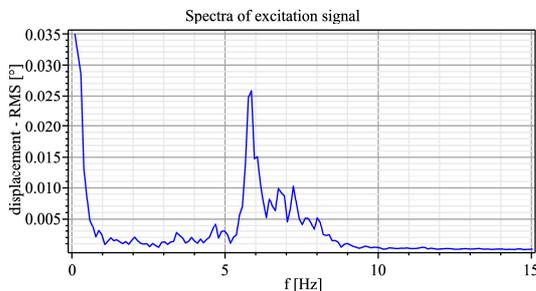


Figure 9: Example of spectra of excitation signal.

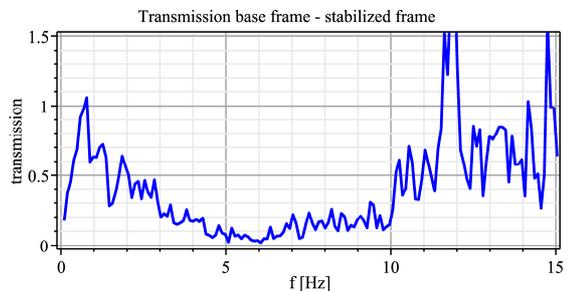


Figure 10: Transmission between base frame and stabilized frame

In order to verify the operation of experimental model we have provided series of measurements under several different excitation signals, usually stochastic signals. Example of

such measurement is pictured in Figure 8 – comparison of kinematic excitation applied on the base frame (rotation about stabilized axis) and response of the stabilized frame (its rotation). Transmission between stabilized frame and kinematical excited base frame is pictured in Figure 10, it is reliable in the band between 5 and 9 Hz, due to limited excitation signal spectra – see Figure 9. Within the mentioned band the transmission is very low. These results show the satisfactory operation of the experimental model.

## 5 MATHEMATICAL MODEL IN MSC.ADAMS

Previous mathematical models were derived in software Maple. This mathematical software supports symbolic calculus and it is very useful for derivation of nonlinear equations of motion. It also supports a lot of tools for mathematical analyses of analytic equations and numeric solving. However there are problems with interpretation of step function type of nonlinearities e.g. Coulomb friction. Another disadvantage would be very difficult derivation of equations which comprise the elasticity of stabilizer structure. For these reasons we have composed another mathematical model in MSC.Adams software – see Figure 11.

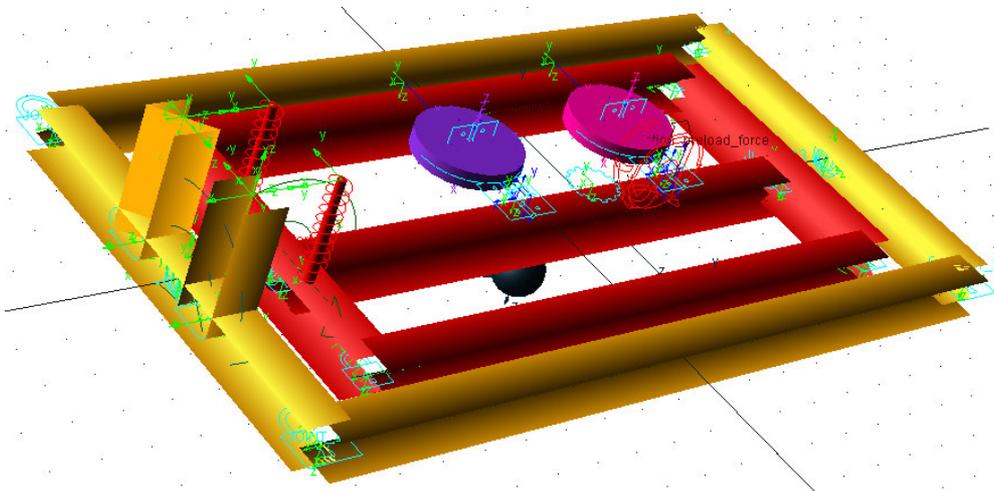


Figure 11: Model of one-axis platform with gyroscopic tandem in MSC.Adams

Base frame and stabilizer frame parts are considered and modelled as elastic beams with properties corresponding with aluminium profiles used to build the experimental model; these beams are connected by the fixed joints. For further work we suppose to replace these fixed joints by at least bushings (linear), better by nonlinear functions of force. Assembly of gyroscopes and precession frames are considered as rigid. Joints (base frame – stabilized frame) are considered as spherical joints which correspond with spherical roller bearings mounted on the experimental model.

### 5.1 Modelling of friction effects using MSC.Adams

Pneumatic rotary motor which is used as correction feedback drive, during its motion, must overcome a friction between vane and cavity. This motor is coupled to the precession frames of gyroscopes, which are coupled together by gearing with unitary gear ratio, also by unitary transmission ratio of belt transmission. In the literature [6] and [7], there is usually required very small magnitude of passive resistances on the gyroscopes precession axes bearings. Thus we have supposed friction in the correction system drive as a possible issue. In order of this supposition we have analysed affects of different types and magnitudes of friction passive resistance applied on the precession axes of the gyroscopes. We have used the MSC.Adams

model for this purpose. MSC.Adams allows applying the Coulomb friction as well as “stiction and sliding” type of friction. We have found out that the one-axis stabilizer is able to operate even when the friction on the precession axis of gyroscope (correction feedback drive) is applied [8]. Despite this friction does not cause serious problems, we recommend, in order of correct operation of the gyroscopic stabilizer, to minimize the friction on the gyroscope precession axle, e.g. by using an electric torque motor without gear transmission.

## 6 CONCLUSIONS

We have described the suggested solution of vibration isolation platform improvement. The gyroscopic stabilization by control moment gyroscopes can be very advantageous especially for lying human patient during transportation. Gyroscopic effects bring on the increase of the natural frequencies of stabilized gimbal frames – outside of the band of excitation frequencies caused by transportation. Experiments proved the functionality of suggested design of the gyro-stabilized vibration-isolation platform – low transmission of vibrations between base frame and stabilized gimbal frame in the case of one-axis platform. However the knowledge of one-axis platform can be extended for the two axis stabilizer. We also proved the usability of suggested design of compensation feedback drive, which simplifies the design of the stabilizer. On the end we have described the analyses of effects of friction in the precession axes bearing, which brings the recommendation of lowering the friction, but not necessarily close to the zero.

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