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KINEMATIC DISCONTINUITIES EFFECT UPON GIMBALS WORKING

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***Abstract*–**The paper presents the dynamical analysis in the case when sudden kinematics parameters are applied upon one element of gimbals. Two cases are considered: first, when the rotor of the gimbals is dynamically balanced and the second one, when this condition is not satisfied. The dynamic simulation was made using specialized software based on multibody dynamics method. An accurate characterisation of impact force versus time using specialised software requires refined meshing of the sampling interval and numerical modelling of a shock phenomenon arises as a intricate project. To avoid this aspect, the shocks were applied to the system by step variation of kinematical driving parameters. The step function was modelled using a third degree polynomial. The balanced rotor has a smoother dynamical behaviour than the unbalanced one.

Keywords—gimbals dynamics, impact phenomena, multibody dynamics.

I. INTRODUCTION

A number of methods are mentioned in the literature concerning the dynamical analysis of a system obtained by connecting rigid elements. Among these manners, one can mention the methods from vector mechanics [1], using the Newton-Euler equations, the Lagrange equations method [2], the dynamic equations method proposed by Kane [3] etc. Applying the above methods, it results a system of second order ordinary differential equations whose difficulty depends straightforwardly on the complexity of the modelled mechanical system. The analytical solving of the obtained differential equations system is possible for only some systems which have, in general, a simple configuration. For most of the situations, this analytical solution cannot be found and numerical methods are necessary [4].

One of the modern numerical analysis methods is the multibody dynamics [5]. The principle of the method consists in writing the scalar differential equations characteristic to a running system, in matrix form; it results a matrix differential equation whose unknowns are a vector characteristic to the unknown kinematical parameters and a vector characteristic to the magnitudes of the reaction forces from the system's joints. All unknowns are found simultaneously by numerically solving this matrix equation.

A particular situation occurs when the system is forced by external shocks. Characteristic to this case is a considerable variation of some dynamical parameters during a very short time period. For the impact of two metallic bodies, the impact scale time is in the order of $10^{-4} \div 10^{-3}\,s$, [6]. Provided that for an accurate characterisation of impact force versus time a refined meshing of the sampling interval is required, numerical modelling of a shock phenomenon arises as a complicated task. Thus, the step number of the numerical analysis increases considerably and implicitly the analysis demands more powerful computer resources. A solution for overcoming this difficulty consists in writing in a convenient manner the matrix differential equation to describe the system's behaviour [7]-[9]. The dynamic model becomes more complicated when the friction from system's bearings is considered, because both its amplitude and direction depend directly on the relative motion from mechanism's joints [10]. The dynamic behaviour of gimbals when percussion is applied to one of its elements is analyzed in the present paper. Based on the method of multibody systems, the software MSC.ADAMS was used in the current dynamic analysis.

II. GIMBALS MODEL AND IMPACT PHENOMENON MODELLING

The analysed example refers to gimbals subjected to percussions. The mechanisms presented in Fig. 1., is a spherical mechanisms simulating the motion of rigid body about a fixed point, the fixed point coinciding with the mass centre. The spherical joint applied in the rotor's mass centre is replaced by three joints with concurrent axes in the rotor's mass centre. The three degrees of freedom from the spherical joint are replaced by three rotations (motions with one degree of freedom) around three concurrent axes.

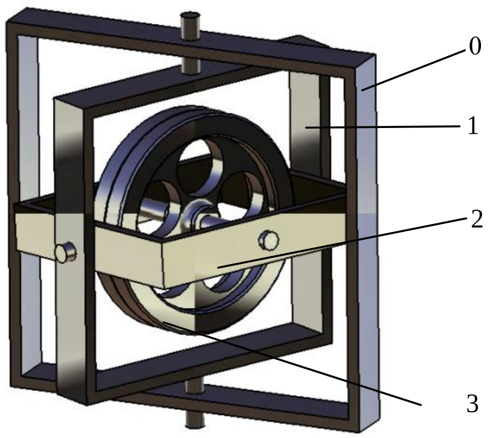

Fig. 1. Gimbals.

The three rotations around axes are practically obtained using two bodies, an external frame 1 and an internal frame 2. The revolute joints occur between: the ground 0 and the external frame 1; the external frame 1 and internal frame 2; internal frame 2 and rotor 3, respectively. One assumes that there is a preset initial angular velocity of the rotor. The aim of the paper is to emphasise the system's response to a sudden external forces action.

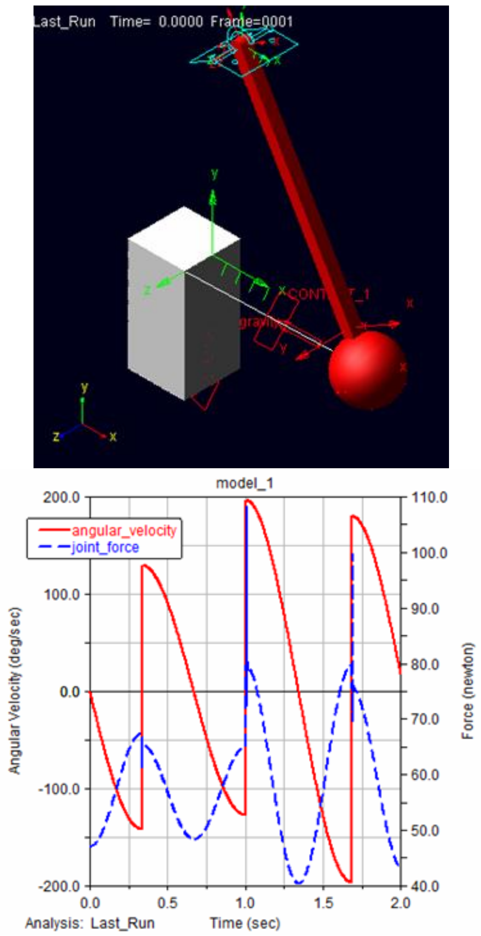

Fig. 2. Angular velocity and force variation with time in a revolute joint – impacting pendulum case.

The simplest case when external percussion appears is when impacting an external body with one of the system's parts. But applying this solution for modelling in MSC.ADAMS software leads to doubtful solutions. To prove this claim the simulated impact between spherical end of a pendulum and a fixed wall is presented. It was assumed that the coefficient of restitution (COR) for the chosen model was $COR = 0.9$.

The pendulum model is shown in Fig. 2., together with the angular velocity and force from the revolute joint variations. It is observed that, because the actual impact time, $\cong 10^{-5}\,s$, is much smaller than the mesh element, $\Delta t = 0.002$ s, the variations of the two above parameters happen instantaneously and in addition, a paradoxical situation occurs, that the angular velocity after impact takes greater value than the initial one. The gimbals model, modelled using MSC.ADAMS software is presented in Fig. 3.

The rotor revolves at an imposed angular velocity and in the revolute joint between the two frames no kinematical constraint is imposed. The shock performed against the system is applied through the revolute joint between the ground and external frame. More precise, the internal ring is subjected to a translated step (Heaviside $H(x - x_0))$ motion. This signal, characteristic to impact phenomena, [11], has a derivative in the variation point - the Dirac function $\delta(x - x_0)$, which should be viewed from distribution theory perspective[12].

$$\delta(\mathbf{x}) = \begin{cases} \infty, \ \mathbf{x} = \mathbf{x_0} \\ \mathbf{0, otherwise} \end{cases} \qquad (1)$$

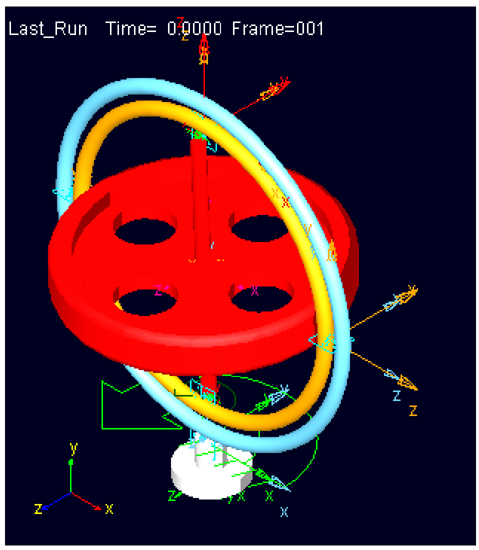

Fig. 3. Gimbals model.

The program can approximate the step function in two ways: by approximation of the step from x_0 point with a 3-rd degree polynomial, as shown in Fig. 4., and both the function derivability and the continuity of the derivative are feasible, or by a 5-th degree polynomial, ensuring the derivability of both the function and of its first derivative.

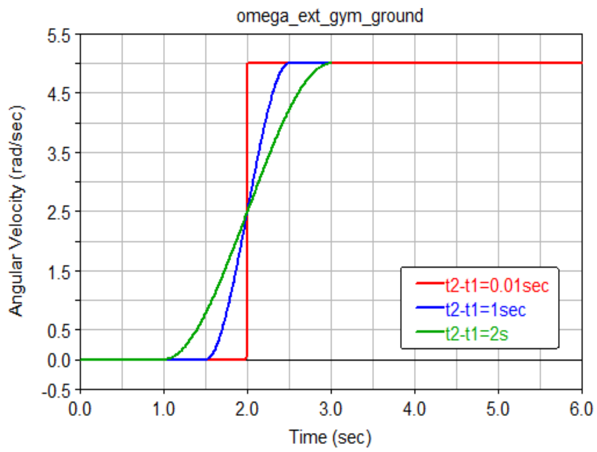

Fig. 4. Step function and approximation function.

The first approximation modality is presented in Fig. 4., for three specified values of the step duration. It can be observed that decreasing the variation time period the plot of the angular velocity is closer to the step function.

III. DYNAMIC GIMBALS SIMULATION. RESULTS.

The dynamic simulation of the gimbals for thee input signals as in Fig. 5., is presented next. The first variation is a sine function, the second is a step function of the same amplitude as the sine wave and the third corresponds to a step function that is twice the amplitude of the sine function.

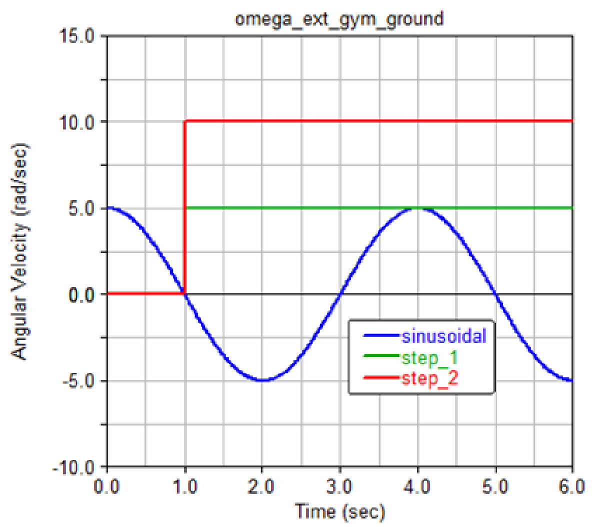

Fig. 5. Angular velocity imposed on external ring.

For the step functions, the approximation was performed using a variation achieved in $\Delta t = 0.01s$. It was assumed that Coulombian friction occurs in all the joints of the mechanism, characterised by static friction coefficient $\mu_{st} = 0.5$ and dynamic friction coefficient $\mu_d = 0.3$. The rotor's initial angular velocity is the same $\omega_{rotor} = 10\pi\, rad/s$. Two situations were considered: the case of dynamically unbalanced rotor and balanced rotor, Fig. 6.

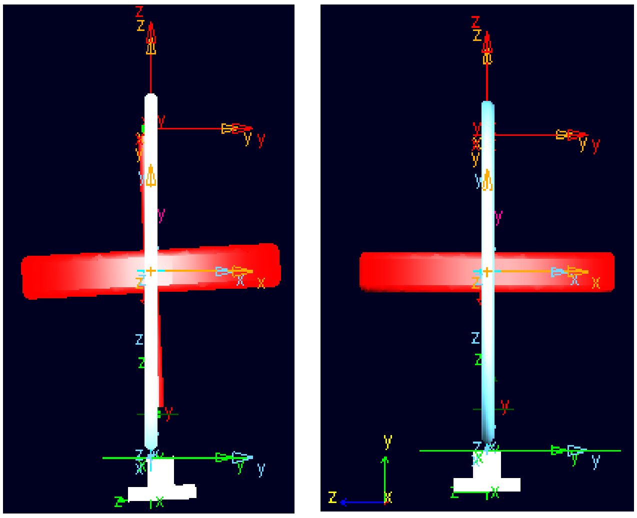

Fig. 6. Gymbals with dynamic unbalanced rotor a) and balanced rotor b).

The study period was of $6s$ and the discretization algorithm considers 1000 subintervals.

From Figs. 9.-10., it can be observed that, as expected, the dynamic balanced rotor gimbals has a dynamical behaviour much better than the one corresponding to Figs. 7.-8. The angular velocity also presents a step while in the joint between the two frames the angular velocity has a weakly variation after step input motion, Fig. 10. It is interesting to remark for the balanced rotor case, the rotor's velocity varies rapidly in the moment of applying the step velocity to the input element, remains constant for a while and after that, regardless of the amplitude of the step, acquires a periodical variation.

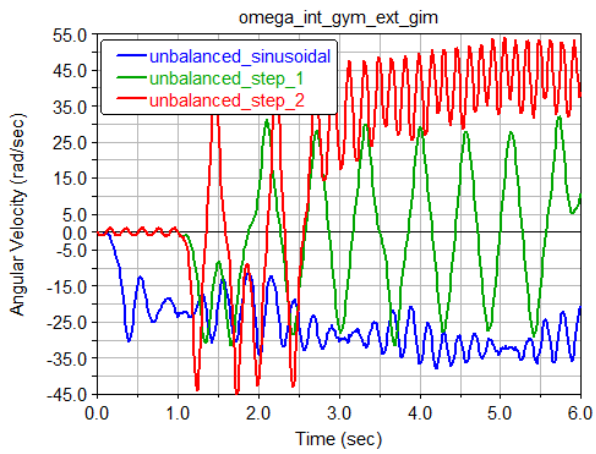

Fig. 7. Angular velocity in revolute joint between external end internal frames (unbalanced rotor).

It is impossible to state that the phenomenon is authentic, caused by the friction from the system's joints or by the decreasing convergence of the numeric process of the simulation software. A response to this question, as Giesbers [10] states referring to contact phenomena modelling, and in particular to the impact phenomena, assumes accomplishing an actual test-rig.

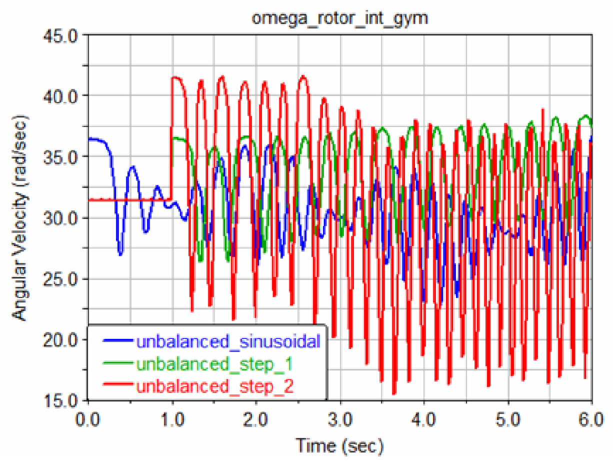

Fig. 8. Angular velocity in revolute joint between internal frame and rotor (unbalanced rotor).

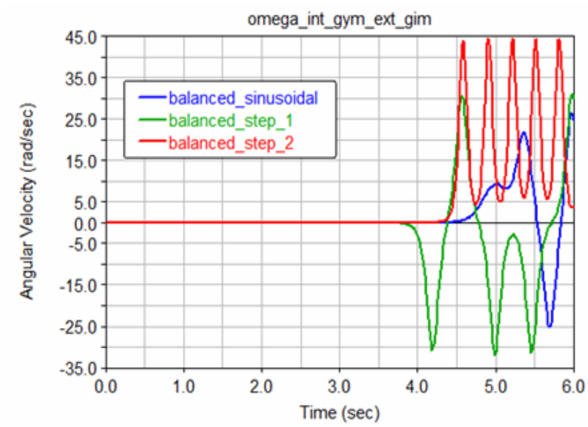

Fig. 9. Angular velocity in revolute joint between external end internal frames (balanced rotor).

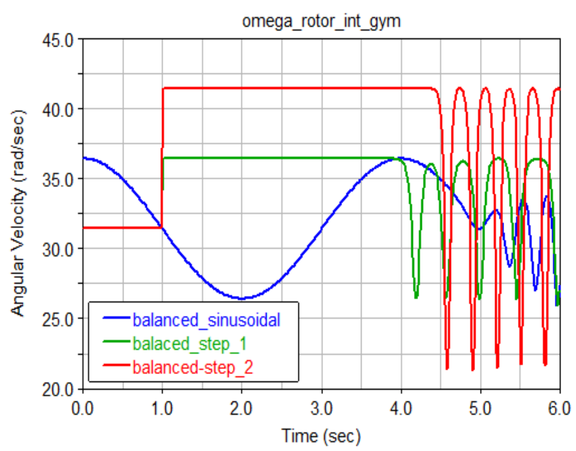

Fig. 10. Angular velocity in revolute joint between internal frame and rotor (balanced rotor).

Knowing the actual system behaviour one can respond to the issues if and how the parameters of the virtual model can be set as to obtain a simulation identical to the real model. Then, the results may be, cautiously extrapolated, in simulating the behaviour of other models.

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