DEVICE AND METHOD FOR STUDYING THE DYNAMICS OF SYSTEMS SUBJECTED TO FRICTIONAL IMPACT. PART II: ROLLING FRICTION CASE

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Abstract: *The paper continues the analysis of dynamical systems behavior under impact. As shown in Part I, between the system's elements only sliding friction occurred on the contact planes of prismatic parts. Knowing the kinematical and inertial parameters of the involved elements, the percussions occurring within the system can be found by applying the dynamical equations. In the second case, one of the prismatic bodies is replaced by a cylindrical part, and therefore, after percussion is applied, the motion is characterized by the mass center linear velocity and angular velocity. As the angular velocity of the cylinder could not be found experimentally after applying an external percussion, the percussions cannot be found. To solve the problem, system dynamic simulation is requested and the MSCADAMS software, which is based on multibody method, is utilized. One of the thorniest problems encountered in this case is specifying the parameters required by contact modeling.*

Keywords: *System dynamic, impact, dry friction*

1. Introduction

The effect of percussion upon a system in the presence of sliding friction was studied in Part I of the paper. Applying the Newton-Euler equations, [1], the percussions emerged inside the system can be obtained. To attain this goal, specifying the kinematical parameters and the inertial characteristics of the system elements was required. The present paper presents the kinematical parameters specification for each of the system's bodies.

2. Experimental device and tests

If one of the prismatic elements of the system is replaced by a cylindrical one, as shown in Fig. 1, the case becomes more intricate because the kinematical state of the cylinder assumes knowing both the linear velocity and the angular velocity of the mass centre.

Acquiring a motion picture of the cylinder motion was a challenge and didn't allow for finding the cylinder's position, because it gets a rotation motion, Fig. 2.

Accurate determination of cylinder's kinematical parameters is affected by the systematic errors of the camera.

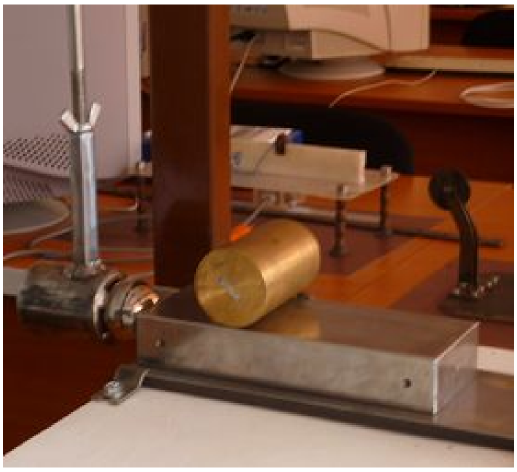

Figure 1: *Experimental setup*

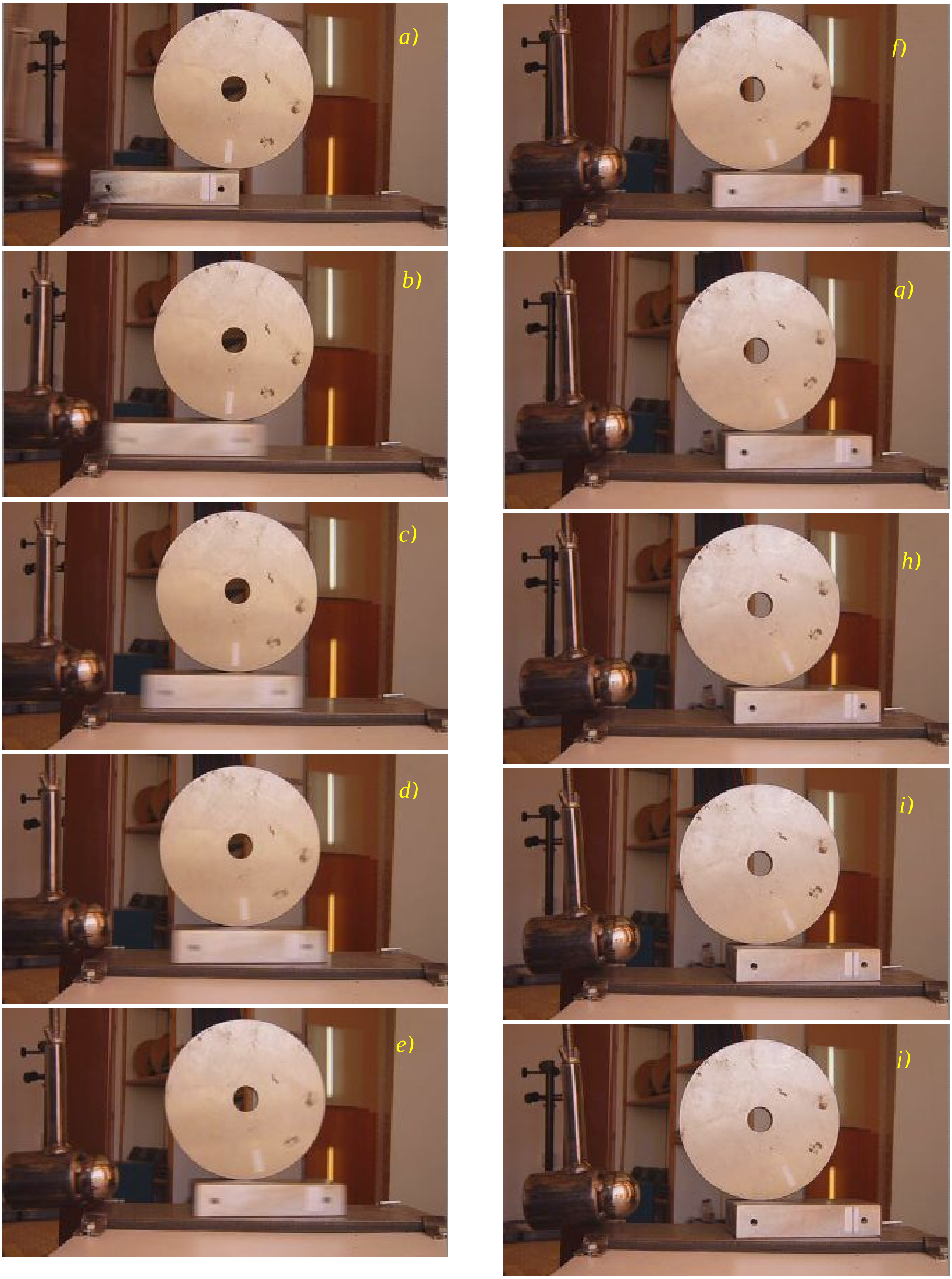

Figure 2: *Successive images of the dynamic system obtained from a film captured at 30 frames/sec.*

3. Dynamic modeling and results

A matter of maximum importance in obtaining accurate results is the correct parameters specification, required by the simulation software.

Consequently, dynamic modeling software, namely MSCADAMS was employed in simulating the system behaviour, as seen in Fig. 3.

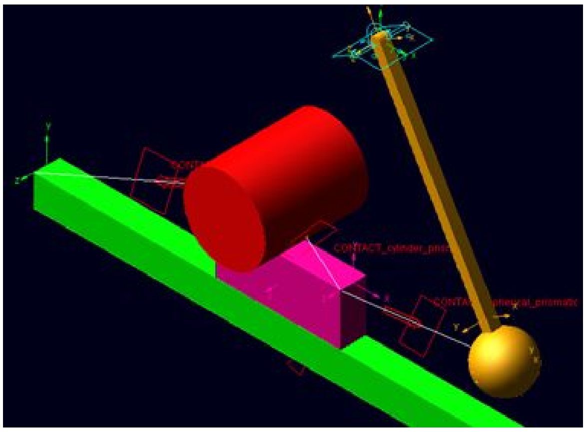

Figure 3: *System modeling using MSC ADAMS software*

The dialog box from Fig. 4 presents the window with the parameters, compulsory to be specified for the cylinder-prism contact modeling. All the considered parameters have a wide range of variation and therefore, this is an open problem, [2].

The present work doesn't insist on the significance of each parameter, this is very well explained elsewhere, [3], but, in order to emphasize how intricate the problem is, some aspects concerning static and dynamical friction are briefly presented. A plot of friction force variation with relative velocity is presented in Fig. 5. Providing between the bodies there is no relative velocity, the friction force values lay in the range $[-\mu_s N, \mu_s N]$, where N is the pressing normal force. At the instant when $v \neq 0$, the friction force takes a limiting value: $\mu_d N$.

In practical application is difficult to employ this dependency, [4], [5]. To eliminate this complexity, the dependence on velocity of the coefficient of friction was approximated by a function with continuous variation.

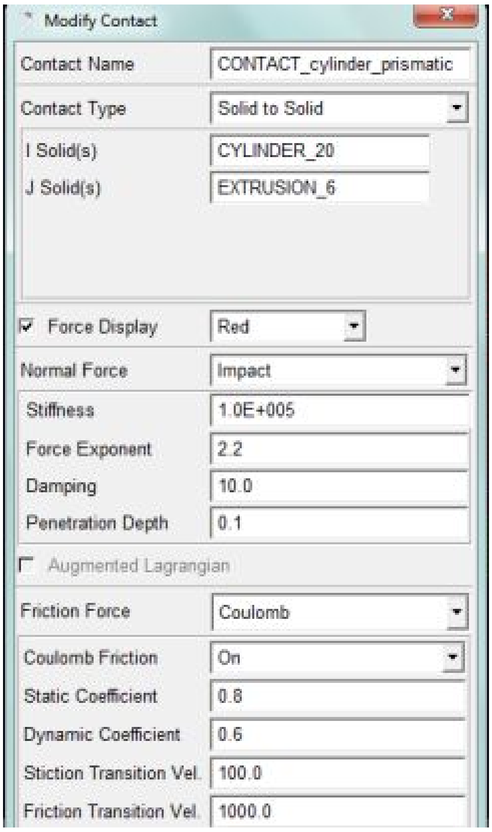

Figure 4: *The dialog box from simulation software for friction parameters setting*

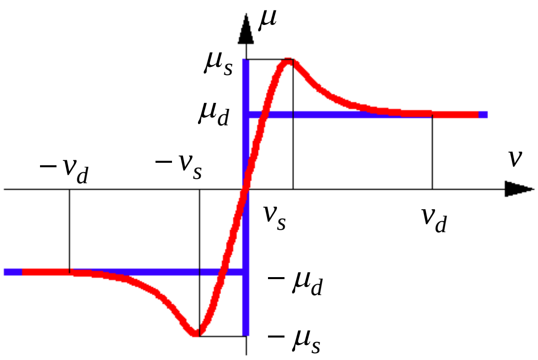

Figure 5: *Theoretical variation and approximation of friction coefficient versus relative velocity*

This approximation brings in a transition domain, within which the friction coefficient varies between μ_s and μ_d, a range that cannot be identified in actual cases.

From Fig. 2, it can be observed that instantly after impact, (Figs. 2.a, 2.b, 2.c) while the prism is in motion, the marker from the disc doesn't change its orientation and the conclusion that, in these figures, between the contacting points, sliding friction is present.

In Figs. 2d-2g, the prism continues the translation motion but the disc rotation is also perceptible.

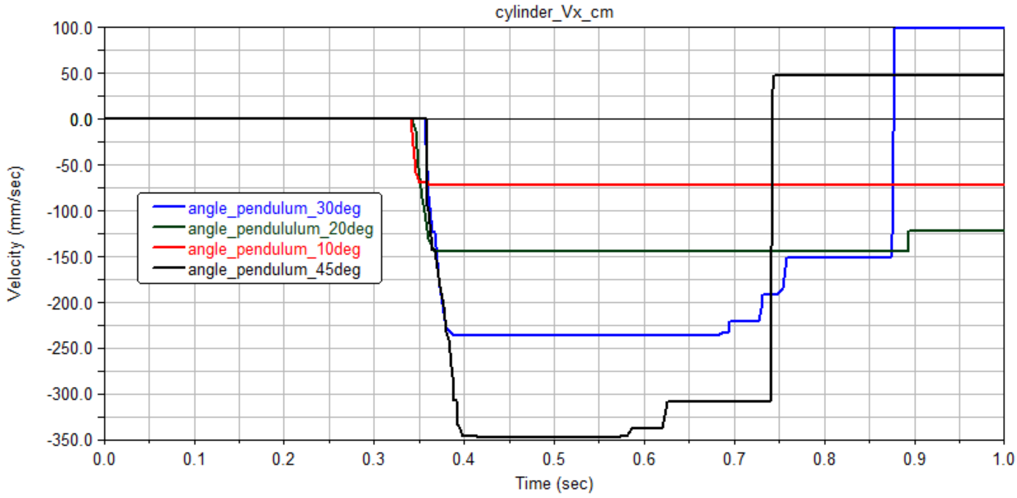

Figure 6: *Variation of centre disc velocity for different pendulum launching angles*

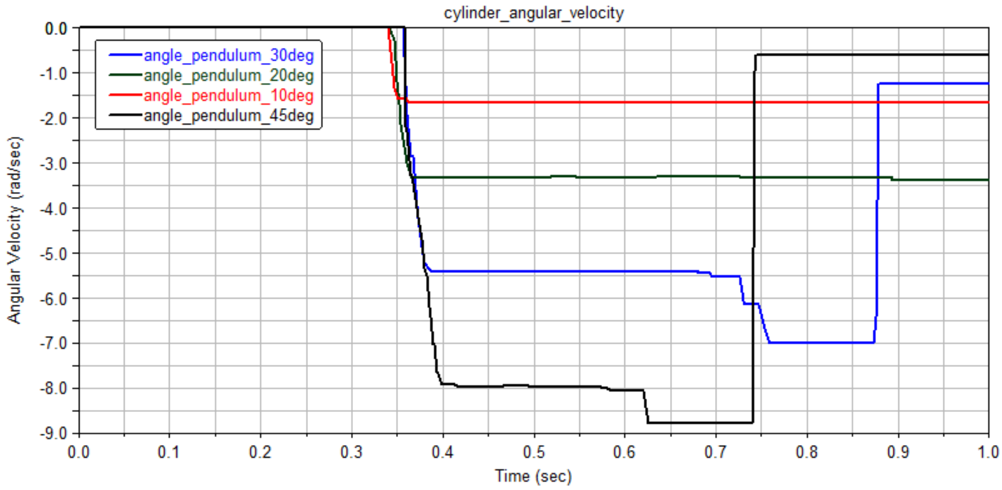

Figure 7: *Variation of disc angular velocity for different pendulum launching angles*

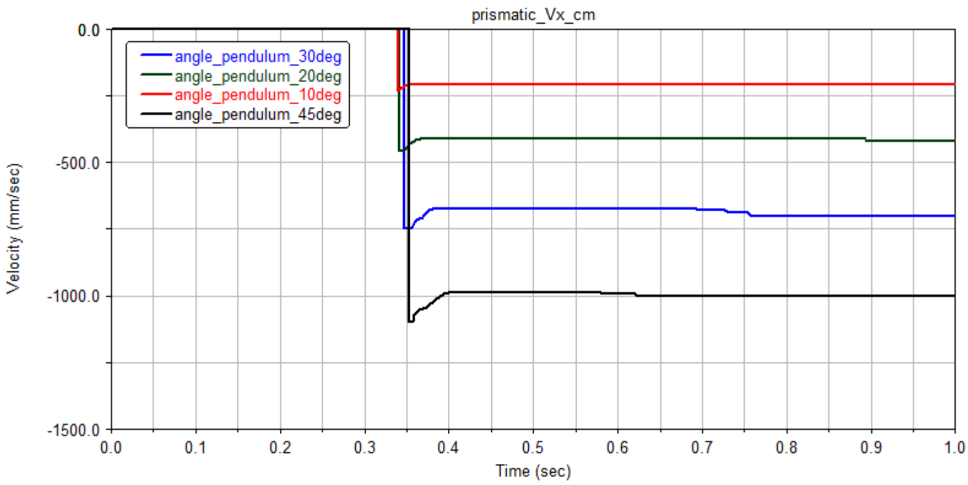

Figure 8: *Variation prism velocity versus time for different pendulum launching angles*

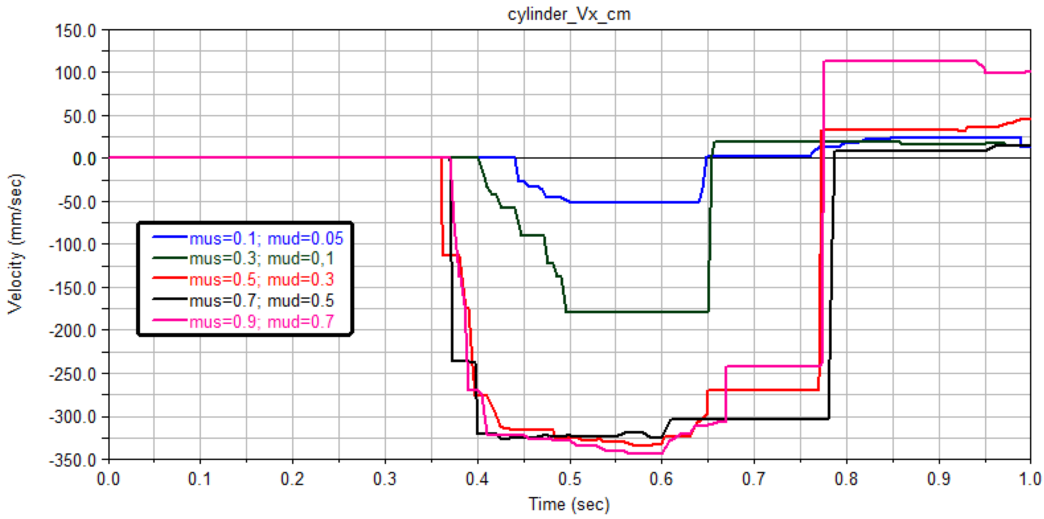

Figure 9: *Variation of centre disc velocity versus time for different values of friction coefficient*

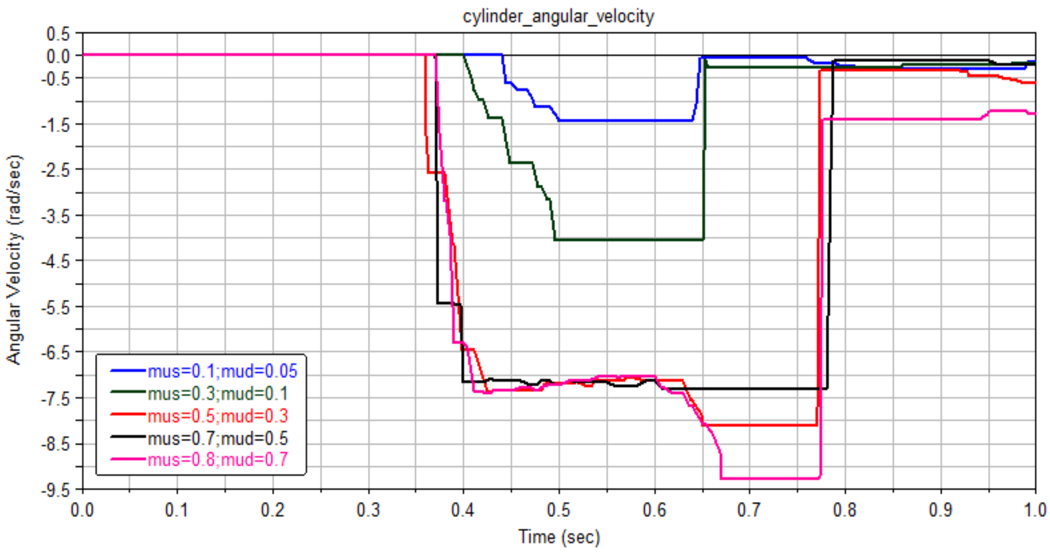

Figure 10: *Variation of disc angular velocity versus time for different values of friction coefficient*

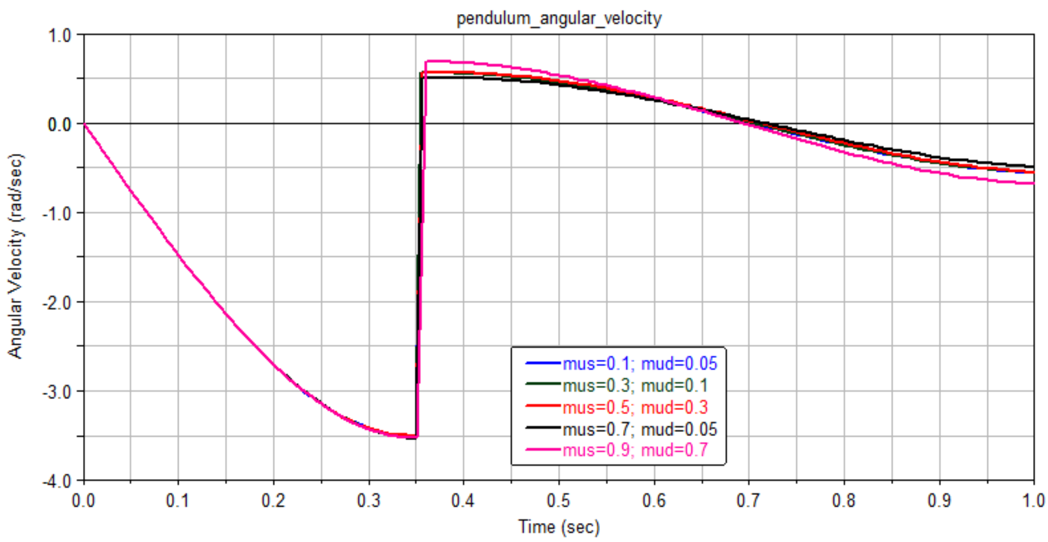

Figure 11: *Variation of pendulum angular velocity versus time for different values of friction coefficient*

The impossibility of precise measurement of kinematical parameters makes unattainable the validation of pure rolling condition:

$$v_P - v_O = \omega_d R \qquad (1)$$

where v_p, v_O are the velocities of the prism and of the disc's mass centre, respectively, ω is the angular velocity and R is the radius of the disc.

Subsequently, the effect of the impact velocity between pendulum and prism, characterized on one side by angular launching magnitude and by friction coefficients from superior pair, on the other side, upon kinematical parameters of the system: linear velocity of the prism, linear velocity of the mass of the disc and angular velocity of the disc, is presented.

For different values of friction coefficient, the variation of pendulum angular velocity before and after impact is presented, Figs. 6-11. The notation *mus* and *mud* in Figs. 9-11 refers to static and dynamic friction coefficient, respectively. Due to lack of accurate required input parameters, the graphs have only a perceptive character. From the figures presented (Figs. 6-11) one can observe that for small pendulum launching amplitudes, the velocity of the mass centre and the angular velocity of the pendulum have the same shape, the proportionality confirming the pure rolling presence. Together with augmenting the launching angle, the differences between the above mentioned plots become visible and the sliding velocity occurrence is expected.

In the case when the coefficient of friction varies, it is difficult to formulate a conclusion upon the manner this variation of coefficient of friction influences the kinematical parameters of the prism. As expected, the friction coefficient has an effect on the pendulum angular velocity. It was observed that the friction coefficient variation was not affecting the impact behavior of the pendulum and therefore, the hypothesis that during the impact phenomenon all forces can be neglected, excluding the percussions, is reasonable.

Conclusions

The paper presents the dynamical model of a system used in impact behaviour study, in the presence of both friction and rolling friction.

For a qualitative corroboration model, the motion of an actual system was filmed and analyzed by frames, the sliding friction and rolling friction occurrence being recognizable and evidenced.

Next, the major impediments that happen in modeling a dynamic system with dry friction forces were highlighted, claiming the necessity of precise stipulation of tribological parameters and possibility of measuring the kinematical parameters.

Once this drawback surmounted, the quantitative results obtained on the theoretical model could be compared to the experimental results, probably followed by a model refining process. Finally, the resultant model could be used in quantitative dynamical simulation for different input parameters.

References

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