Potential use of program Dynamic Designer in spring modelling

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ABSTRACT: Program Dynamic Designer serves for kinematic and dynamical analysis of rigid body system. Models of springs and relations between force and spring deformation can be chosen and inserted into the system. In the article there is presented a model of flexibly fastened body. Result of dynamical analysis is dependence of selected point deflection on time (Figs. 5, 7, 8, 10). The results can be put to use in the design of spring-loaded parts of agricultural machines.

Keywords: software Dynamic Designer; rigid body system; dynamical analysis; springs; dampers

Dynamic Designer program is one of many upgrades of the design modelling programs such as AutoCAD, Mechanical Desktop and Autodesk Inventor. Similarly, it is specifically focused on a particular technical field, in which it allows to apply various advanced algorithms and calculations to the model system, which elementary modules are not capable of.

Dynamic Designer is suited for kinematic and dynamic analysis of a system of rigid bodies (User's Guide 1999). This program enables you to define the joints between model bodies, set forces and couples acting on bodies or directly enter the motion parameters of driving parts and monitor the movements of other system parts as well as the reactions at supports and connections. This article deals with models of springs and dampers, which may be included in the system of rigid bodies.

MATERIALS AND METHODS

Fig. 1 shows the Motion menu located in the left part of the window, which also becomes available once the Dynamic Designer upgrade is installed. In a clearly arranged manner, the menu is split up into five sections.

The first section (Assembly Components) displays a list of modelled bodies allowing to drag and drop them onto second section (Parts), so as to choose which of them would become moving parts and which would represent ground parts. The third section (Constraints) enables to restrict the movement through the various constraints, e.g. by defining the joints between the bodies. Acting forces, comprising spring and damper forces may be set within the fourth section (Forces). The fifth section (Results) makes it possible to select the required results: the trajectory of the particles (Cplr Curve), displacements, velocities and acceleration of the selected particles, reaction and inertial forces and couples; furthermore, enables to plot the function of the chosen parameter.

Clicking the right mouse button while in the Springs menu enables to choose whether a cylindrical or torsion spring is to be entered into a mechanical model (Fig. 2). The same applies to dampers. Fig. 3 shows the dialog window which appears for the cylindrical spring when bodies with flexible connection are selected. Here, basic spring parameters can be set, particularly the force deflection curve (Spring Expression), which does not necessarily have to be linear. First, the user either selects one of the default expressions or puts in his own one (Adams Function). Then the free length and spring stiffness need to be entered. Other parameters (Display Parameters) are only used in drawing the picture of the spring and have no impact on the force effect whatsoever. The parameters of torsion springs and cylindrical and torsion dampers are entered in a similar fashion.

The next paragraphs demonstrate how the Dynamic Designer program may be used in modelling a spring-suspended body. The dynamical analysis is required to produce a displacement-time curve as a result.

Fig. 1. Dynamic Designer menu

Fig. 2. Right mouse button Springs menu

Fig. 4 shows a body, suspended on four identical springs with the acting force linearly dependent on the deformation. The joint is restricting body movement to rectilinear translation. If added to the system, the dampers, with the force linearly proportional to the velocity, and acting impressed periodic force F (N), allow to study the characteristics of damped forced vibrations.

Let us denote the natural circular frequency of undamped free vibrations as Ω_o (rad/s) and compute it using the expression

$$\Omega_o = \sqrt{\frac{k}{m_b}}$$

where k (N/m) stands for the total constant of springs attached in parallel and m_b (kg) represents the weight of the vibrating body. The constant of damping δ (s^{-1}) is given by the expression

$$\delta = \frac{b}{2 \cdot m_b}$$

where b (N.s/m) is a total coefficient of viscous damping. The natural circular frequency of damped free vibrations Ω_d (rad/s) then follows from

$$\Omega_d = \sqrt{|\,\Omega_o^2 - \delta^2\,|}.$$

The body is subjected to a periodic force F (N) of magnitude

$$F\,(t) = F_a \sin\,(\omega \cdot t)$$

where F_a (N) is the amplitude, ω (rad/s) the circular forced frequency and t (s) stands for the time. Let us introduce the following dimensionless parameters: damping factor

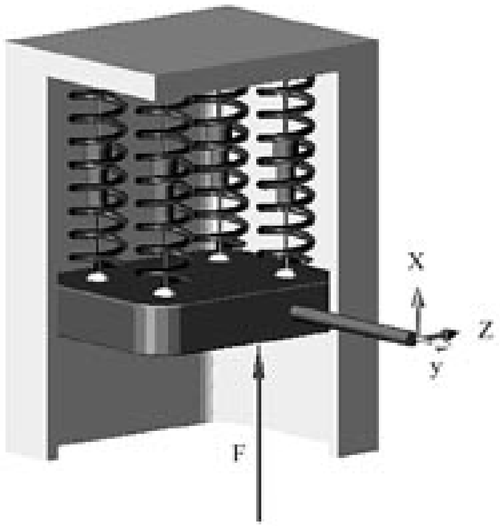

Fig. 4. A model of a spring-suspended body with rectilinear translation

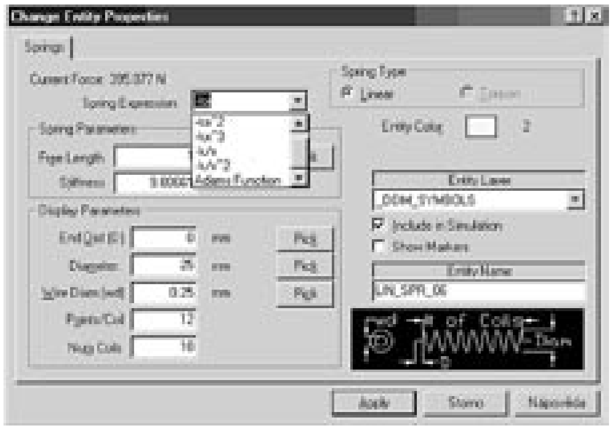

Fig. 3. Setting cylindrical spring parameters

$$\delta p = \frac{\delta}{\Omega_o}$$

and frequency ratio

$$\eta = \frac{\omega}{\Omega_o}$$

If an origin of the coordinate system is chosen in the static position, the damped forced linear vibrations in the direction of x-axis yield a constant coefficient non-homogeneous second order differential equation of the form $m_b\,\ddot{x} + b\,\dot{x} + k\,x = F$. Its solution represents the desired function of the body displacement x (m) versus time t (s) in the form

$$x\,(t) = e^{-\delta t}\,.\,G\,.\,\sin\,(\Omega_d\,.\,t + \kappa) + x_m\,.\sin\,(\omega\,.\,t + \Delta\phi) \qquad (1)$$

where: the amplitude x_m (m) of a steady-state vibration is given by

$$x_m = \frac{F_a}{m_b}\left[\sqrt{(\Omega_o^2 - \omega^2)^2 + (2\,.\,\delta\,.\,\omega)^2}\right]^{-1}$$

and the phase difference $\Delta\phi$ (rad) of a steady-state vibration is given by

$$\Delta\phi = -\mathrm{atan}\left(\frac{2\,.\,\delta\,.\,\omega}{\Omega_o^2 - \omega^2}\right)$$

(JULIŠ, BREPTA 1987).

Integration constants G (m) and κ (rad) may be derived from the given initial displacement x_o (m) and velocity v_o (m/s) e.g. in the following form:

$$G = \sqrt{(x_o - x_m\,.\,\sin\,(\Delta\phi))^2 + \left[\frac{v_o}{\Omega_o} - x_m\,.\,\eta\,.\,\cos\,(\Delta\phi) + \delta_p\,(x_o - x_m\,.\,\sin\,(\Delta\phi))\right]^2 \cdot \frac{1}{1 - \delta_p^2}}$$

$$\kappa = \mathrm{asin}\left[\frac{(x_o - x_m\,.\,\sin\,(\Delta\phi))}{G}\right]$$

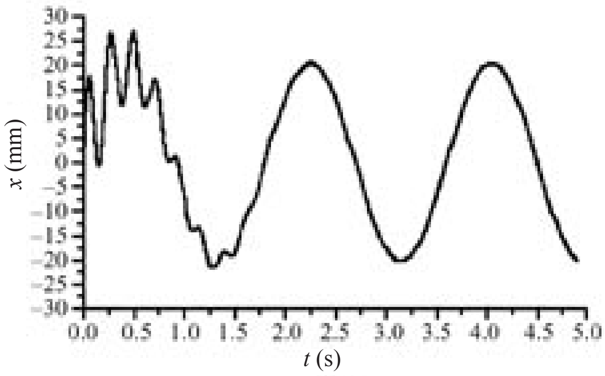

Fig. 5. The function of the displacement versus time computed by Dynamic Designer program

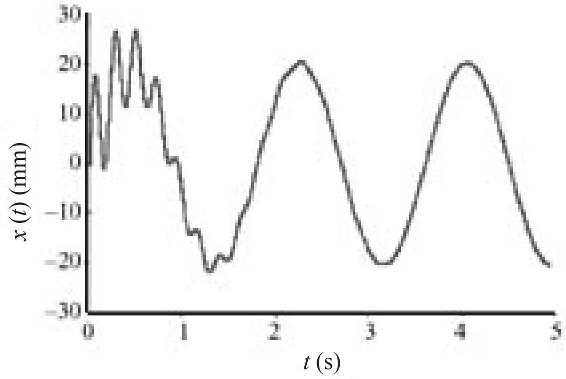

Fig. 6. The graph of the derived function x (t) (eq. 1)

RESULTS AND DISCUSSION

The resulting function of the body displacement x (mm) versus time t (s) computed by the Dynamic Designer program (Fig. 5) may subsequently be compared with a graph of the function x (t) (Fig. 6), which can be analytically derived provided the vibrations are damped and forced (equation 1). Should the same input parameters be entered, the results will coincide.

Dynamic Designer program furthermore enables to take into account the influence of the tangential component of the reaction T (N) in a real joint (Fig. 7).

The impressed force F (N) may be entered with a variable frequency and modelled over the resonance area (Fig. 8).

Fig. 9 again shows a body suspended on four springs and dampers, yet now its movement is not restricted by any joint thus allowing the body to engage in a general three-dimensional motion.

Fig. 10 depicts the time-dependent displacement, calculated by the program and decomposed into the components x, y and z of the displayed coordinate system (Fig. 9). The stiffnesses of the springs differ; as the motion is caused merely by the initial velocity, thus what we are dealing with are free damped vibrations.

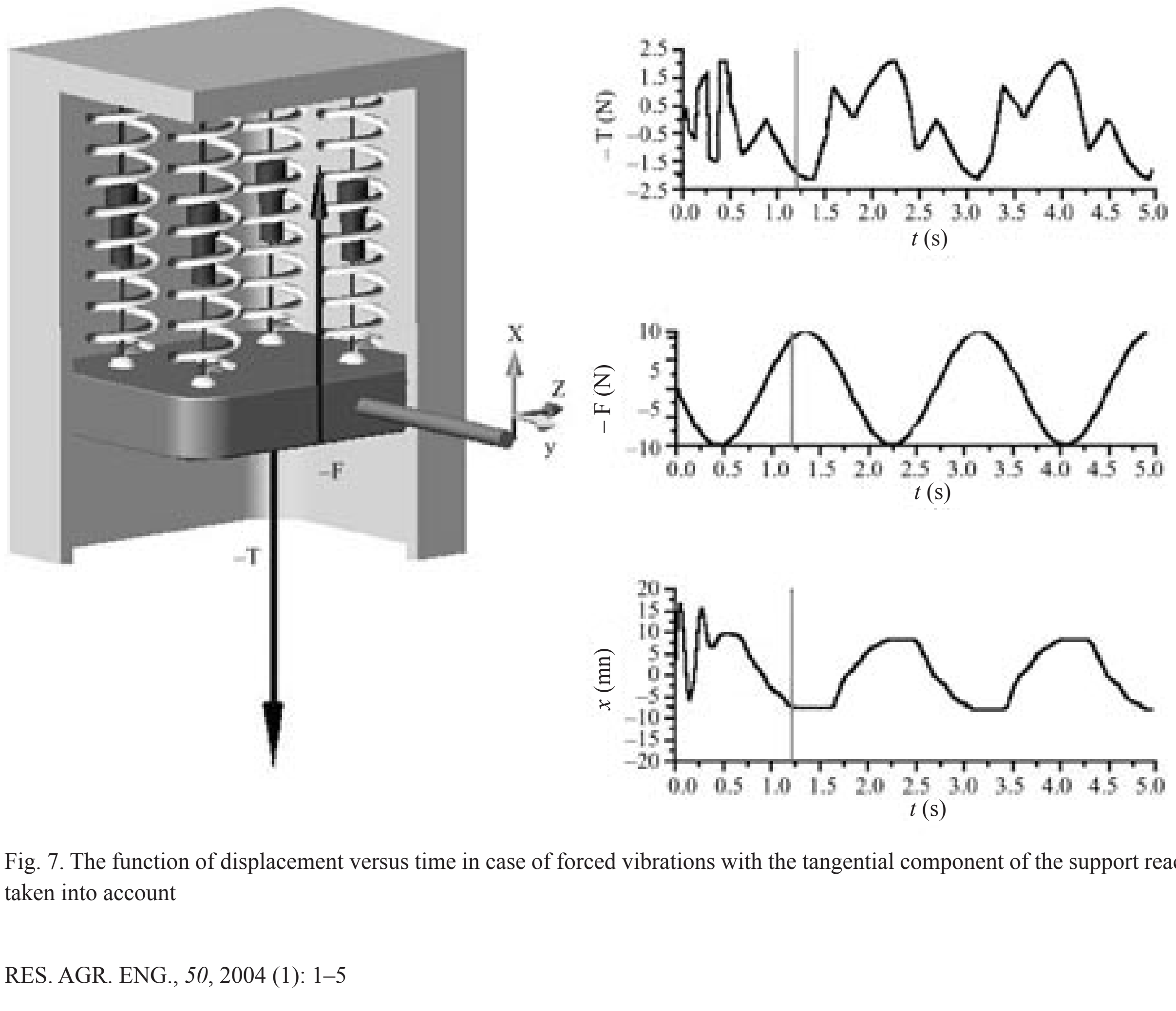

Fig. 7. The function of displacement versus time in case of forced vibrations with the tangential component of the support reaction taken into account

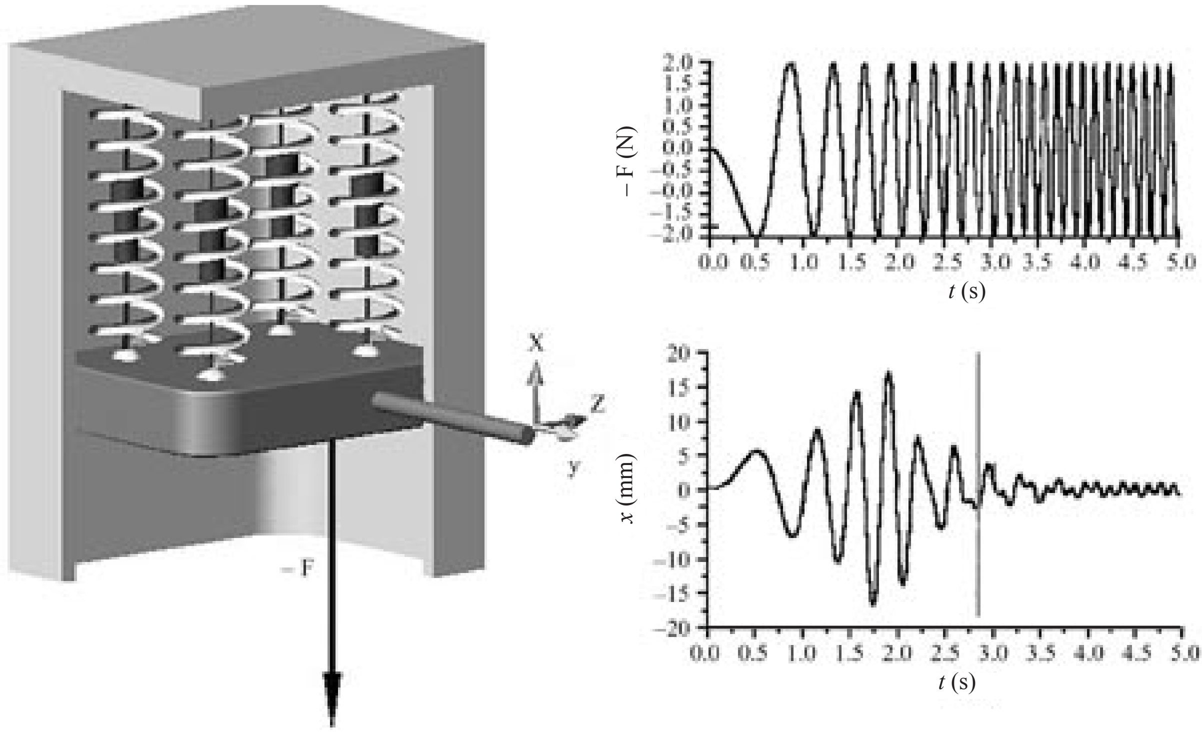

Fig. 8. Function of the displacement versus time in case of forced vibrations with a variable frequency of the impressed force

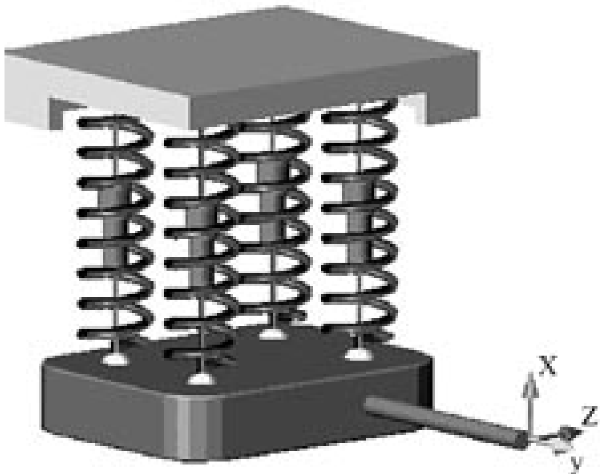

Fig. 9. A model of a spring-suspended body free to engage in a general three-dimensional motion

The modelling results can be utilized in the design of spring-loaded parts of agricultural machines (seats, cabs).

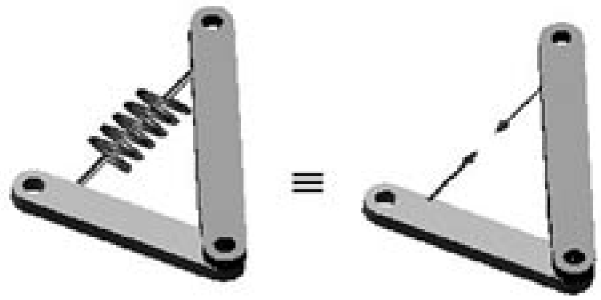

Fig. 11. A spring model in Dynamic Designer program

In closing it should be noted that the possibilities of Dynamic Designer program are restricted by its ability to work with rigid bodies only. The spring model does not exhibit the same behaviour as does a real material spring made of flexible steel wire (Fig. 11).

Placing a spring between two bodies only creates two forces of the same magnitude, the same line of action (the line connecting selected points on the bodies) and

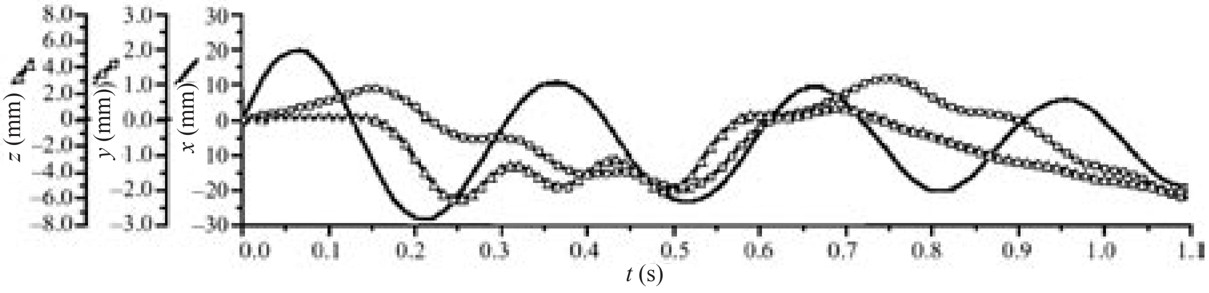

Fig. 10. The displacement in case of the body engaged in a general spatial movement

opposite sense. While in motion, their magnitude changes according to the selected function depending on the distance of the bodies (or more precisely on the distance of the given points thereon).

References

Dynamic Designer Motion and Motion Lite User's Guide, 1999. Ann Arbor, Mechanical Dynamics: 328.

JULIŠ K., BREPTA R.,1987. Mechanika, II.díl Dynamika. Praha, SNTL: 685.

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Možnosti využití programu Dynamic Designer při modelování pružin

ABSTRAKT: Program Dynamic Designer slouží ke kinematické a dynamické analýze soustavy tuhých těles. Do soustavy je možné vkládat modely pružin a volit vztah mezi působící silou a deformací pružiny. V článku je sestaven model pružně uloženého tělesa. Výsledkem dynamické analýzy je závislost výchylky zvoleného bodu na čase. Výsledky jsou využitelné při návrhu odpružených částí zemědělských strojů.

Klíčová slova: program Dynamic Designer; soustava tuhých těles; dynamická analýza; pružiny; tlumiče

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