Limitation of dynamic power parameters in transitional processes

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Abstract

Introduction. The article is devoted to the analysis of the prospects of the complex use of the accumulated potential energy on the example of a mechanism for lifting, horizontal movement and loading of cargoes, for example, in transport container with limitation of dynamic loads.

Materials and methods. Synthesis of technological systems is aimed at overcoming the contradictions between the kinematic parameters of a highly productive machine, energy costs, and dynamic load.The study is theoretical, based on the laws and principles of mechanics, with the creation and analysis of appropriate mathematical formalizations that relate to energy and mechanical transformations with the final result of the combination of elevated kinematic parameters and limited force actions.

Results and discussion. On the basis of the analysis of the peculiarities of the shock loads in two-mass systems, the possibility of using the rigidity of the springy pendant as a variable factor to achieve the given relations between static and dynamic loads during their course in the gravitational field was determined. The idea of a sharp increase and fixation of the rigidity of the pendant system under maximum deformations and loads was used. Mechanical transformation of the springy system in the oscillatory process from low-frequency to high-frequency is carried out by parallel connection to springy elements of high rigidity in the form of a two-link hinge.

The fixed position of the deformed springy element allows to perform the role of "holder" of the potential energy of the field of springy forces, the use of which is carried out at the stage of inserting of loads to limit the speed of their contact with the supporting receiving planes. Exclusion of low-frequency oscillations from the system means limitation of energy losses on internal friction in springy elements.

The developed mathematical formalizations confirmed the validity of the idea of using a variable rigidity system, which, in addition to the positive result of limiting extreme dynamic loads, achieves positive energy and dynamic effects in the final stages.

Conclusions. The idea of using a system with variable rigidity of springy elements allows to limit the contradiction between the values of kinematic and dynamic parameters.

Introduction

Solving the problems of synthesis of technological machines is traditionally associated with the choice of structure, analysis of kinematics and dynamics with the need to limit the irregularities of the course of the leading links, loads of transition processes and energy costs. Such a set of characteristics of the parameters of systems has an inherent contradiction [1]. On the one hand, it is desirable to limit the mass characteristics of the equipment, dynamic loads and energy costs, on the other hand, solving the task of a given performance requires increasing kinematic parameters and inertial loads with them. Obviously, the latter requirements are necessarily related to transition processes in overclocking and running modes [3]. The dynamics of their implementation undergoes energy transformations and costs, since the work of the driving forces to overcome the forces of inertia corresponds to the increase of the kinetic energy of the moving masses of the system [1, 2].

Since the transition processes occur in the gravitational field, the number of force parameters in the system includes the forces of gravity, driving force and resistance factors (forces and moments of forces), forces and moments of forces of inertia and springy force factors [1]. The latter belong to the field of forces of springiness. So for the linear force of springiness which obeys the Hooke's law, due to which force action $\bar{F} = -c\bar{\Delta}$, where c is the stiffness coefficient; $\bar{\Delta}$ is the distance from the center of mass at which the force $\bar{F}$ acts to the position of its static equilibrium at $\bar{F} = 0$. In the case of linear deformation, the springy force is reflected by a partial derivative of the energy potential [3]:

$$\frac{\partial u}{\partial \Delta} = F = -c\Delta . \tag{1}$$

Integrating the latter condition allows to write:

$$u = -\frac{c\Delta^2}{2} + const . \tag{2}$$

Given that $\Delta^2 = x^2 + y^2 + z^2$, we finally have:

$$u = -\frac{c\Delta^2}{2} + const = -\frac{c}{2}\left(x^2 + y^2 + z^2\right) + const . \tag{3}$$

The characteristic of the potential force field is expediently supplemented by a function that determines the energy reserve at a given point of the field, that is, the potential energy of the system at a given position, which is equal to the work that the potential forces of the field can accomplish when moving the material point from the flowing point to the starting point of the system [ ]. From this position it follows that the potential energy P depends on the coordinates of the material point:

$$P = A; \quad P = P\left(x, y, z\right) . \tag{4}$$

The above considerations allow to conclude that the potential energy of deformation of the springy system can be determined by the dependence:

$$\begin{gathered} P = -u = Pz + const; \\ P = -u = \frac{c}{2}\left(x^2 + y^2 + z^2\right) + const \end{gathered}; \tag{5}$$

The presence of springy elements in the drives of the working bodies of technological machines is one of the directions of limiting the dynamic components of the loads, each of which is accompanied by the corresponding deformations [4, 5]. In a considerable number of cases, systems with springy elements are modeled by two-mass, and the most difficult conditions correspond to shock interactions between the leading and the driven masses [6, 7].

It is known that in such cases, the velocity of the driving mass remains stable, and the load of the springy element is reflected by the sum of static and dynamic components (Figure 1) [1, 2]:

$$P_{sp} = V\sqrt{m_2 c}\sin\sqrt{\frac{c}{m_2}}\,t + P_{re} \tag{6}$$

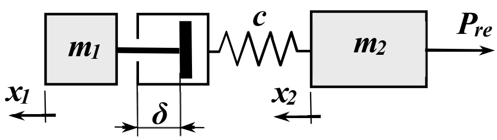

Figure 1. Diagram of a two-mass model

The presence of such dependence indicates the possibility to determine the maximum springy load and the corresponding maximum deformation of the springy element with rigidity c, as well as the potential energy of the deformed element [8–10].

The purpose of the study is to develop mathematical formalizations in search of opportunities to improve the performance of systems and machines with dynamic load limitations based on the use of the proposed springy pendant.

Materials and methods

The **object** of research is the synthesis of technological machines and features of transition kinematic and dynamic processes.

The **subject** of research are mechanisms for lifting, horizontal movement and stacking of goods, in particular a device for the insertion and removal of bottles from a transport container (patent of UA65929).

The research is theoretical, performed on the basis of the laws of physics, principles and methods of mechanics [3, 11] with the creation and analysis of analytical models [1].

Results and discussion

Henceforth we turn to the problem of the prospects for the integrated use of accumulated potential energy at the example of a mechanism for lifting, horizontal movement and stacking of goods, for example, in transport containers. In its composition there is a leading element, which moves the programmable route with vertical and horizontal sections with a definite sequence [3].

At a given productivity of the system, the velocity V cannot be considered as variable as the driven mass [1]. This means that the limitation of the dynamic component is associated with the rigidity of the system in the direction of its reduction. However, under this condition, the frequency of natural oscillations $\sqrt{c/m_2}$ also decreases and the system turns into a slow decay. The latter can be considered as a disadvantage, although it is counteracted by the addition of a parallel springy damper of oscillatory processes.

An alternative to such a partial solution of the problem is the proposal reflected in the patent of Ukraine 65929, which combines the possibility of limiting shock dynamic loads in systems of fixed movement of loads in vertical and horizontal combined movements. The invention relates to a device for the insertion and removal of bottles from a transport container, consisting of mounted with the possibility of horizontal reciprocating movement of the carriage mounted on it with the possibility of vertical movement from the drive gripper head with clamps, characterized in that the gripping head united with the drive of vertical movement. The gripping head of the device is connected to the drive of vertical movement by traverses 1 and 2, interconnected by springs 3 and two-hinged hinge 4, one of the traverses is fitted with a retainer 5 for two-hinged hinge, and the other is fitted by lever 6 and connected with it rod 7 for fixing a double-joint hinge (Figure 2).

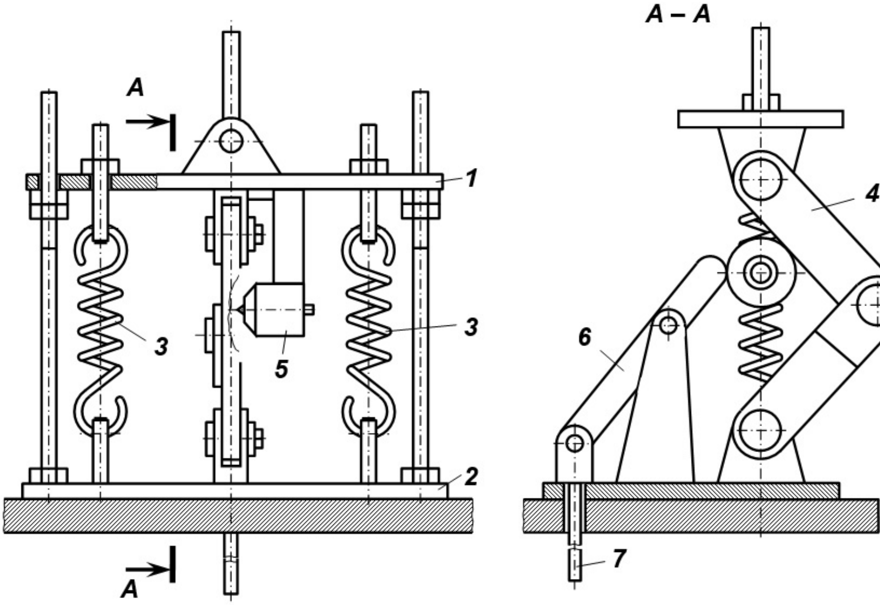

Figure 2. Scheme of springy pendant of cargo [patent UA 65929]

Moving loads by the gripping head vertically takes place in two stages. At the first stage from the beginning of the movement of the traverse 1 stretching and loading of the springs 3 takes place to a value corresponding to the weight of the gripping head with the load and the opening of the hinge 4.

At the second stage, the accelerated movement of the traverse 2 begins together with the driven mass with the subsequent stretching of the springs. At the moment of reaching the maximum load on the springs 3, which comes at the equality of speeds of the traverses 1 and 2, full disclosure of the two-link hinge 6 and fixing in this position by the retainer 5 take place.

The introduction of the springs 3 to the pendant sharply reduces the dynamic load of the shock interaction, and their stretching is accompanied by the accumulation of potential energy, which in the absence of fixation would cause low-frequency oscillations and would disrupt the normal operation of the device. The actuation of the retainer 5 leads to the fact that in the future, the traverses 1 and 2 with the gripping head and load moved as one. The two-link hinge remains open and compressed force, which is equal to the difference of maximum load and gravity of the gripping head.

Loading of goods on the receiving plane corresponds to the third stage. In this case, the rod 7 rests on the support plane and, interacting with the lever 6, leads to the hinge 4 from the dead point, releasing the springs 3. Since the force in the springs exceeds the weight of the gripping head, the traverse 2 with the latter receives a movement towards the traverse 1, reducing their absolute speed of the lowering movement by triggering the potential energy of the springs. Depending on the kinematic parameters of the system and the choice of the moment of unlocking the hinge, it is possible to limit the speed of contacting the loads with the supporting receiving plane.

Such a combination of conditions makes it possible to limit the contradiction stated above between the set of kinematic and dynamic parameters.

In accordance with these stages of moving goods, let's turn to the tasks of drawing up their analytical models. The movement of the leading mass of the system together with the flexible pendant and the traverse 1 with a constant speed V corresponds to the first of them. At this stage, the driven mass m_2 remains stationary until the moment when the springy load P_{sp} is not equal the resistance to movement $m_2 g$ (Figure 3). Equations $x_1 = Vt$ and $x_2 = 0$ and the initial load of the springyelement $P_{sp(i)} = 0$ correspond to this condition.

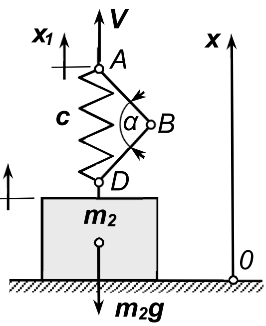

Figure 3. Calculation scheme of the first stage

The final value of the moving coordinate corresponds to the completion of the first stage:

$$x^{I}_{(f)} = \frac{m_2 g}{c} \tag{7}$$

where c is spring stiffness, N/m.

The second stage is displayed by a system of two equations:

$$\begin{aligned} x_1 &= Vt; \\ m_2\ddot{x}_2 &= c\left(x_1 - x_2\right) - m_2 g, \end{aligned} \tag{8}$$

the transformation of which leads to a condition:

$$\ddot{x}_2 + \frac{c}{m_2}x = \frac{Vc}{m_2}t - g \ , \tag{9}$$

which matches the original data:

$$t_{(i)} = 0; \quad x_{2(i)} = -\frac{m_2 g}{c}; \quad \dot{x}_{2(i)} = 0 \tag{10}$$

The solution of equation (9) taking into account (10) is:

$$x_2 = Vt - V\sqrt{\frac{m_2}{c}}\sin\sqrt{\frac{c}{m_2}}t - \frac{m_2 g}{c}; \tag{11}$$

$$\dot{x}_2 = V - V\cos\sqrt{\frac{c}{m_2}}t\ ; \tag{12}$$

$$\ddot{x}_2 = V\sqrt{\frac{c}{m_2}}\sin\sqrt{\frac{c}{m_2}}t\ . \tag{13}$$

The load of springy elements is reflected by the dependence:

$$\begin{aligned} P_{sp} &= c\left(x_1 - x_2\right) = c\left(Vt - Vt + V\sqrt{\frac{m_2}{c}}\sin\sqrt{\frac{c}{m_2}}t + \frac{m_2 g}{c}\right) = \\ &= m_2 g + V\sqrt{m_2 c}\sin\sqrt{\frac{c}{m_2}}t \end{aligned} \tag{14}$$

The latter condition implies the presence in the system of static and dynamic components of the load, the ratio of which can be guided in the choice of parameters. For example, if you agree on their equality, we can write down:

$$m_2 g = V\sqrt{m_2 c}\ , \tag{15}$$

where the extremum of the dynamic component is represented on the right side of the equation.

Since the output data of the system is represented by the weight of the load and the productivity in the form of velocity V of the driving mass, then the stiffness of the pendant should act as a variable parameter, so:

$$c = \frac{m_2 g^2}{V^2}\ , \text{ N/m.} \tag{16}$$

c = 153977 N/m corresponds to the parameters $m_2 = 100$ kg and V = 0.25 m/s.

Changing the velocity parameter to the value of V = 1 m/s provides a transition to the stiffness of the springs c = 9623.6 N/m. It is obvious that the calculated value of rigidity corresponds to each of the ratios of parameters m_2 and V. However, the condition of static load and extreme dynamic conditions is not necessary, which creates additional possibilities in the structural variations of the system.

Given that the interaction between the driving and driven masses of the system is estimated as a shock load, we conclude that it is advisable to complete the second stage at the time of equalization of firces $\dot{x}_1$ and $\dot{x}_2$.

A graphical interpretation of this situation is shown in Figure 4, which, considering equation (12), corresponds to the value:

$$V\cos\sqrt{\frac{c}{m_2}}t = 0. \quad (17)$$

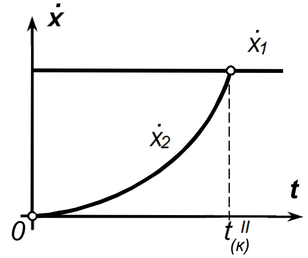

Figure 4. Graphs of the speeds of the driving and driven masses at the second stage

This means that the final time of the second stage is:

$$t^{II}_{(f)} = \frac{\pi}{2}\sqrt{\frac{m_2}{c}}. \quad (18)$$

For values $m_2 = 100$ kg and c = 153978 N/m, we get $t^{II}_{(f)} = 0.04$ s. During this time, the leading mass will perform the movement $x^{II}_{1(f)} = Vt^{II}_{(f)} = 0.25\cdot 0.04 = 0.01$ m, and the displacement of the driven mass is determined by the condition (11):

$$x^{II}_{2(f)} = 0.25\cdot 0.04 - 0.25\sqrt{\frac{100}{153978}}\sin\sqrt{\frac{153978}{100}}0.04 - 0.00637 = -0.00274 \text{ m} \quad (19)$$

The difference of the coordinates of the movements of the driving and driven masses is:

$$\Delta x = x^{II}_{1(f)} - x^{II}_{2(f)} = 0.01 + 0.00274 = 0.01274 \text{ m} \quad (20)$$

Checking the value of the springy load leads to a value

$$P_{sp} = c\left(x^{II}_{1(f)} - x^{II}_{2(f)}\right) = 153978(0.01 + 0.00274) = 1962 \text{ N} \quad (21)$$

which meets the previously accepted condition (15).

In the future, the calculation is related to the determination of the lengths of the links of the two-link hinge. Preliminary calculations made it possible to determine the movement of the driving and driven masses in two stages. The initial difference of coordinates is defined as $\Delta x_{(i)}$ taking into account the coordinates of the hinges D and A. $\Delta x_{(i)}$ takes into account the structural size $\Delta x_{(0)}$ and the value of deformation of the component of static load $x^{I}_{1(f)}$ (Figure 5).

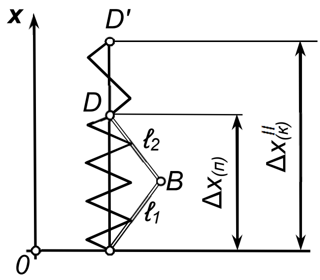

Figure 5. Scheme to determine the difference in the coordinates of the positions of the driving and driven masses

Then the condition of the hinges A and D corresponds to the completion of the second stage:

$$\Delta x^{II}_{(f)} = \Delta x_0 + \Delta x^{I}_{(f)} + \left(x^{II}_{1(f)} - x^{II}_{2(f)} \right). \tag{22}$$

The obtained value $\Delta x^{II}_{(f)}$ corresponds to the sum of the lengths of the links ℓ_1 and ℓ_2 of the double-link.

The fixed relative position of the hinges AB-D' on one vertical means the arrangement of the energy "trap". The potential energy of the deformed springy element is:

$$E = \frac{c}{2}\left(x^{II}_{1(f)} - x^{II}_{2(f)} \right)^2. \tag{23}$$

Subsequent movements of the system along horizontal and vertical lowering movements continue at a stabilized velocity V up to the moment of unlocking of the double-link, at which the force action of the deformed springy element will exert additional influence (Figure 6).

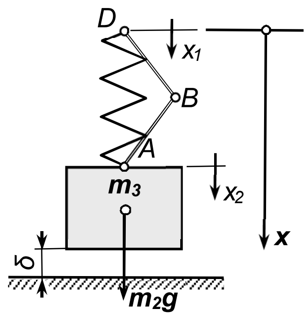

Figure 6. Scheme to the transitional process of loading goods

In this case, the driven mass will go into the transition mode with the deceleration of absolute speed according to the equations of the system:

$$\begin{aligned} x_1 &= Vt; \\ m_2 \ddot{x}_2 &= m_2 g - c\left(x_2 - x_1 \right). \end{aligned} \tag{24}$$

However, the latter requires confirmation in relation to the magnitude of the fixed springy force in the deformed element. The role of the latter is present in the meaning of the initial condition $x_{2(i)}$ at the third stage. We will show this in a further analysis. Condition (24) implies:

$$\ddot{x}_2 + \frac{c}{m_2} x_2 = g + \frac{cV}{m_2} t. \tag{25}$$

Under initial conditions

$$t_{(i)} = 0; \quad x_{2(i)} = \frac{m_2 g}{c}; \quad \dot{x}_{2(i)} = V \tag{26}$$

we get:

$$x_2 = A_1 \sin\sqrt{\frac{c}{m_2}} t + B_1 \cos\sqrt{\frac{c}{m_2}} t + \frac{m_2 g}{c} + Vt \tag{27}$$

and hence integration constants $A_1 = 0$ i $B_1 = 0$.

The corresponding substitution allows to write:

$$x_2 = \frac{m_2 g}{c} + Vt; \quad \dot{x}_2 V, \tag{28}$$

which means that there is no oscillation process at the third stage.

The transition to the new initial conditions

$$t_{(i)} = 0; \quad x_{2(i)} = \frac{2m_2 g}{c}; \quad \dot{x}_{2(i)} = 0 \tag{29}$$

leads to changes integration constants:

$$A_1 = 0 \text{ i } B_1 = \frac{m_2 g}{c}. \tag{30}$$

Then we have:

$$x_2 = \frac{m_2 g}{c} + Vt + \frac{m_2 g}{c} \cos\sqrt{\frac{c}{m_2}} t; \tag{31}$$

$$\dot{x}_2 = V - \frac{m_2 g}{c} \sqrt{\frac{c}{m_2}} \sin\sqrt{\frac{c}{m_2}} t; \tag{32}$$

$$P_{sp} = m_2 g + m_2 g \cos\sqrt{\frac{c}{m_2}} t; \tag{33}$$

$$P_{sp.\max} = 2m_2 g. \tag{34}$$

Obviously, the value $P_{sp.\max}$ corresponds to the initial load, and the presence of the oscillatory process at the third stage corresponds to the graphical interpretation in Figure 7.

The first negative value of the extremum of the dynamic component of the speed of the driven mass and the zero value of the resultant comes when using the condition:

$$t_{(f)} = \pi\sqrt{\frac{m_2}{c}} \tag{35}$$

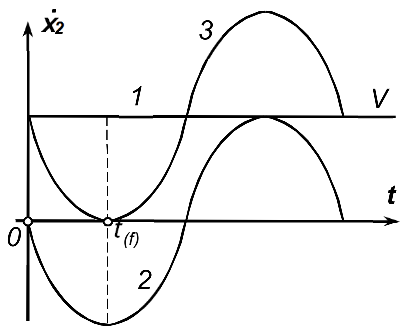

Figure 7. Schedule of change of speed of the driven mass:
1 – static component; 2 – dynamic component; 3 – the resulting of speed

The value of the displacement of the mass m_2 at the time $t_{(f)}$ is:

$$x_{2(f)} = \frac{m_2 g}{c} + Vt_{(f)} + m_2 g \cos\sqrt{\frac{c}{m_2}}t . \tag{36}$$

If the condition $x_{2(f)} = \delta$ is carried out, where δ is the gap between the mass m_2 and the receiving plane at the beginning of the transition process at stage III, we conclude that they contact at zero speed. This result is an expected positive based on the energy potential of the deformed springy element [1]. Thus, the initial conditions of the third stage, defined as the final conditions of the second, are variational factors in the third. From this point of view it is worth to note the importance of the ratio of the dynamic and static components of the load of the springy element [1]. If we take the value $x_{2(i)} = \frac{3m_2 g}{c}$, then we obtain $P_{sp.\max} = 3m_2 g$, and provided that their multiplicity is k, we obtain:

$$P_{sp.\max} = km_2 g . \tag{37}$$

The level of activation of the energy potential is estimated under the final conditions corresponding to equations (32) and (33). The value $\dot{x}_{2(f)} = 0$ leads to the conclusion that the kinetic energy of the driven mass $E_{kin.(f)} = 0$ is the same as the potential energy of deformation of the springy element at $P_{sp.(f)} = 0$.

Separation of the driven mass from the system of its movement at the moment of achievement $t_{(f)}$ means achievement of the mode of non-shocking loading of cargo on the receiving plane [3, 7]. The interaction between them also depends on the ratio of the masses and the receiving plane and the rigidity of the elements of its installation. It is known [3] that due to neglect of the mass of the receiving plane and at zero contact speed, the dynamic load of the springy elements is equal to static $m_2 g$, which in sum with the latter corresponds to the value of two static ones.

Conclusion

The proposed system of arrangement of the device for moving loads on the vertical and horizontal sections with their subsequent attachment to the receiving horizontal planes was intended to limit the force interactions between the driving and driven masses and energy losses associated with oscillatory processes and springy deformations. Performed theoretical studies allow to note the following:

1. The task of achieving high productivity of technological machines leads to the requirements of increasing kinematic parameters with the simultaneous intention of maintaining stable speed of the driving masses. This means the presence of modes of shock loads with dynamic amplitudes $P_{dyn} = V\sqrt{m_2 c}$.
2. The ratio of the stiffnesses to the masses determines the frequency of natural oscillations, whereby such systems are conventionally divided into slow and fast decay. In oscillatory processes, internal friction results in energy losses, the limitation of which is appropriate and achievable due to the use of elements of fixation of springyelements in the deformed state with the corresponding energy potential.
3. At the stage of loading of loads, the energy potential of springy elements provides for the restriction of the speeds of contacting the loads with the supporting receiving planes with restrictions of dynamic loads.
4. The rigidity of the springy element is the regulatory parameter of influence under limited shock loads during lifting. The latter simultaneously allows to maintain speeds that satisfy a given productivity. This is an important compromise in mechanical systems for dynamic power and kinematic parameters.

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