

Dynamic Contact Vibration Damping

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Abstract

In any contact between two or more solid bodies, a number of physical phenomena occur, some of which cause oscillatory movements. When taking into account the real conditions in the system, the resistance forces are taken into account, which lead to the damping of the oscillatory movements that have occurred. In addition to the vibration-exciting physical phenomena during the period of contact between the bodies, there are physical phenomena that dynamically extinguish the oscillatory movements that have occurred. The material considers three main physical phenomena of dynamic vibration excitation and dynamic vibration damping of oscillatory movements. These are self-centering, as a physical phenomenon as a result of impact, friction and the natural frequency of the bodies.

Keywords— vibration damping, friction, natural frequency, shaping

INTRODUCTION

Impact and Self-Centering Theory.

The theory of impact in Theoretical Mechanics is associated with a number of conventions and assumptions established during the numerous observations of this phenomenon [1]. Due to the fact that this phenomenon is very short-term, the changes in mechanical quantities are realized almost simultaneously [2,3]. It is not possible with the known Newtonian mechanics to describe the processes in phenomena that occur almost simultaneously and some quantities tend to infinity, thus blocking the mathematical apparatus.

The solution to the problem is to introduce a time interval of the impact τ , which tends to 0 and therefore the impact force $\vec{F}$, which acts during the infinitely small time interval τ is not an infinite quantity, but takes a finite value. In this way, the mechanical quantities for the phenomenon acquire finite values.

The impulse I^m of the impact force $\vec{F}$ for the impact time interval τ will be:

$$I^m = \int_{t_0}^{t_0+\tau} \vec{F} dt$$

- where: t_0 is the initial moment of the impact phenomenon;

- and $t_0+\tau$ is the final moment of the impact.

From the law of change of momentum, the basic equation of the theory of impact of a material point follows:

$$(2) \quad \begin{aligned} mv_x - mv_{0x} &= I_x^m \\ mv_y - mv_{0y} &= I_y^m \\ mv_z - mv_{0z} &= I_z^m \end{aligned}$$

After substitution we get:

$$(3) \quad mv - mv_0 = \int_{t_0}^{t_0+\tau} \vec{F} dt$$

Therefore, the difference in velocities before and after the impact phenomenon on a body is inversely proportional to its mass.

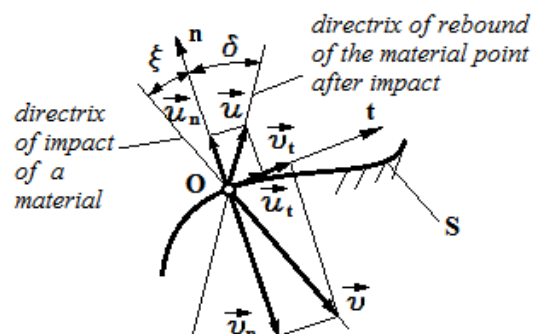


Fig. 1. Schematic diagram of the impact of a material point on a stationary surface.

If we hit a material point with mass m on a stationary surface $S(x,y,z,t) = 0$, at an angle ξ to the normal n of the surface S , with a speed v , then the reaction to the impact will depend on the physico-mechanical properties of the matter of the material point and the surface S , at the moment of impact.

static friction $\vec{F}$ is proportional to the pressure $\vec{N}$ between two bodies:

$$(9) \quad \vec{F} = \mu \cdot \vec{N}$$

where: μ is the coefficient of friction (proportionality).

The coefficient of friction depends on the type and condition of the surfaces of the two bodies at the point of contact, but not on the number of contact points, as well as on the type of surfaces around the contact area.

Static friction has its ultimate value and in order to bring two statically contacting bodies into a state of relative motion relative to each other, a certain force is required to overcome the force of static resistance. Usually, the coefficient of static friction is greater than the coefficient of sliding friction $\mu_{stat.} > \mu_{slid.}$ and with the same normal reaction $\vec{N}$ the force of static friction is greater than the force of sliding friction. This difference in the magnitudes of the coefficients of friction when transitioning from a state of static friction to a state of sliding causes a jump in the driving force.

Natural frequency of the bodies.

Free oscillations are called *undamped oscillations* when the system is closed and conservative. Such oscillations are possible only in systems where friction and resistance forces are neglected, that is, these are methodological or idealized, oscillating systems. They are characterized by an amplitude that remains constant with time, and can continue indefinitely [6,7].

Free oscillations that occur in real systems are called *damped oscillations* (Fig. 3). Due to the acting friction forces, the amplitude of these oscillations decreases over time and they gradually damp out [8].

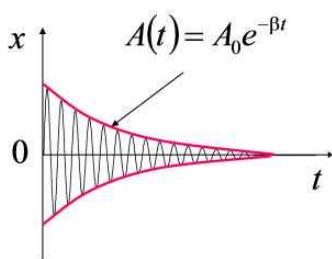


Fig. 3. Damping oscillations

If two forces act on a body performing harmonic oscillations: a driving force $\vec{F}_1$, proportional to its displacement from the equilibrium position, and a friction force $\vec{F}_2$, proportional to its velocity v :

$$(10) \quad \vec{F}_1 = -k \frac{d^2 x}{dt^2}; \quad \vec{F}_2 = -r \frac{dx}{dt}$$

The "-" sign in the friction force indicates that its direction is always opposite to the direction of the velocity of motion. The coefficient r is the coefficient of friction. From which it follows:

$$(11) \quad m \frac{d^2 x}{dt^2} = -k \cdot x - r \cdot v;$$

$$\frac{d^2 x}{dt^2} + \frac{r}{m} \frac{dx}{dt} + \frac{k}{m} x = 0$$

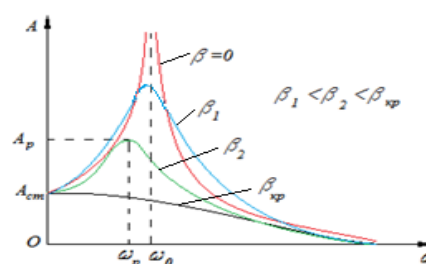
The above expression represents a differential equation of a damped oscillation.

Substituting:

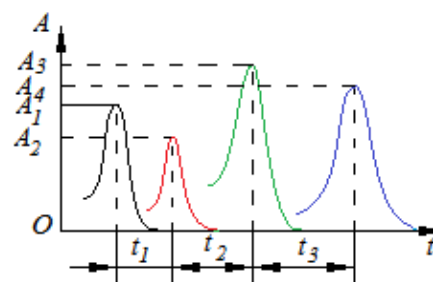
$$(12) \quad \frac{k}{m} = \omega_0^2; \quad \frac{r}{m} = 2\beta$$

Където: ω_0 - е собствената честота на трептящото тяло.

β - е коефициент на затихване.



a)



b)

Fig.4. Diagram of natural frequency and resonance of a solid body; a) amplitude-frequency characteristic of a body at different damping coefficients; b) shock pulses during friction between two bodies.

The natural frequency of a body is the frequency ω_0 (Fig. 4), which it would have in the absence of resistive forces, that is, friction forces. In such a case, the differential equation of damped oscillation. takes the form:

$$(13) \quad \frac{d^2 x}{dt^2} + 2\beta \frac{dx}{dt} + \omega^2 x = 0$$

The solution has the form:

$$(14) \quad x(t) = A_0 e^{-\beta t} \sin(\omega t + \varphi_0)$$

The damping coefficient at $\beta_k = \omega_0$ is called the *critical damping coefficient*.

If energy is introduced into the system from the outside to compensate for losses due to friction, resistance, etc., the oscillations can turn from damped to undamped, called *forced oscillations*. Forced oscillations are caused by the action of external periodically varying forces on the system and are also called *non-free oscillations*.

If, in an equilibrium position, the body receives a single, short shock, it will begin to perform harmonic oscillations with its own frequency ω_0 (the action of a tuning fork can serve as an example).

At a negligibly small damping coefficient ($\beta \approx 0$), the amplitude of the forced oscillations increases with increasing ω . When $\omega = \omega_0$ then $A \rightarrow \infty$, this phenomenon is called *resonance*.

PRESENTATION

Technological synthesis of a controlled contact

The creation of the technological process begins with the creation of the scheme of the contact itself. In general, there are three types of contact, these are *point* contact, contact along a *spatial line*, in particular straight lines, and contact along a *spatial surface*. There are also mixed contacts that start with one type and continue in the second and third types, that is, they start as a point contact and continue as a spatial line.

Another type of classification of the contact is according to the existence of an independent attractive force or force component between the bodies. In this case, the contact is divided into a support, a grip or the mixed variant of a hybrid support.

The classification of the contact suggests what technology should be applied in order to realize a reliable kinematic connection or mechanism for its implementation. The technology itself leads to the formation of the details and units in the contacting bodies.

The next stage of the development of the technological process is the type of load. In the most general form, it is spatial, which can be decomposed as projections along the six axes of a given coordinate system.

The interesting moment is when the external load is not constant, but is oscillatory or shock-periodic. In these cases, it is a question of vibration excitation of oscillatory processes in the system by the contacting bodies after or during contact. In these cases, the system is necessarily designed for vibration loading. One of the options for design under variable loading is shown in Fig. 5.

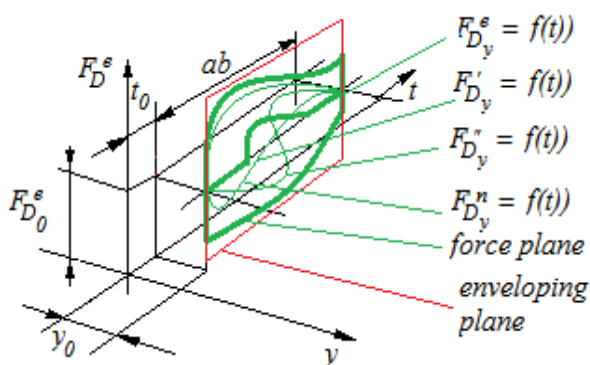


Fig. 5 The force plane formed by the set of possible forces $F_{D_y}^e = F(t)$ from the given technological implementation schemes.

The shown scheme of an exemplary force plane is described by the component $F_{D_y}^e = f(t)$, in its possible variations from the different given technologies to which the contacting bodies must respond, $F_{D_y}^s = f(t)$, $F_{D_y}^r = f(t)$, ..., $F_{D_y}^n = f(t)$. For some technologies, the maximum amplitude may be smaller, and for others it may be higher. This shows that the force plane is not of regular shape, but it is formed by enveloping all possible values of the given force components. Due to the fact that the force plane has an irregular shape, its mathematical description leads to large and complex expressions. The force plane thus obtained is replaced by an enveloping plane with regular geometric shapes (the enveloping plane shown in "Fig. 5" is a rectangle). According to this methodology, the work from spatial curves of the force load over time is transferred to the processing of stationary planes, characterizing the maximum load of the contact mechanism in a given direction. The projections of the spatial curves on the given plane slightly correspond to the projection shaping of the details.

In a similar way, the remaining components of the external load are also processed. The resulting enveloping planes in space create a complex spatial figure, which is replaced for the second time with an enveloping spatial figure. This enveloping figure gives the overload margin coefficient during sizing, as well as the selection of the contact points in space for specific bodies.

The approach is extremely elementary, fast and illustrative. This is actually a stage of the comfortable sizing method used for complex kinematic structures.

The creation of technological schemes for contact mechanisms and the synthesis of long kinematic chains to implement these technological schemes are constructed in such a way that they can implement multiple technological schemes similar to each other with extended capabilities of the contact mechanisms. This in turn leads to a mathematical description of planes and volumes that describe the force curves as a function of time for all given technologies.

Change in natural frequency

The natural frequency ω_0 of a given body is its invariable characteristic. It depends on the following factors:

1. The type of material from which the body is made. Its stiffness k , composition and distribution of the material in the volume of the body.
2. The type of shape of the body. It is important to note that the natural frequency of a body is not the same in different directions along the coordinates of the body. With an equal distribution of matter in the volume of the body, the natural frequency is a function of the shape.
3. The natural frequency depends on the mass of the body. More massive bodies have lower natural frequencies, with the same stiffness.

Changing one of these three characteristics changes the natural frequency of the given body, and hence the behavior of the body during contact with other bodies.

Of great interest for the synthesis of details for mechanisms is the type and shape of bodies upon contact. The type and shape of bodies change the frequency characteristic of the bodies, and hence their behavior upon contact with each other.



Fig.6. Tuning fork with adjustable masses of the vibrating elements.

As an example of what has been said so far, the one presented in Fig.6. Design of a tuning fork (this is a musical instrument generating damped oscillations, based on the natural frequency, with a very low damping coefficient $\beta \approx 0$) with a variable position of the masses on the vibrating elements. This design has the ability to be readjusted, as the shape of the body of the tuning fork and the distribution of its mass over its volume can be changed, which leads to a change in the natural frequency.

Changing the position of only one movable mass leads to different frequencies of oscillation of the two vibrating elements, which leads to superposition of frequencies on the different branches of the tuning fork. This leads to a change in the sinusoidal shape of oscillation, to the physical effect of beating or to dynamic damping of vibrations.

Dynamic regulation of friction

As it was noted in point 1.2 that any resistance force is friction, here in this point, another property of friction will be shown, namely the dynamic excitation and damping of vibrations in contact between two solid bodies. The physical phenomenon on which this vibration excitation is based is the differences in the magnitudes of the coefficients of static friction and the coefficient of sliding friction. Usually the coefficients of static friction are greater than the coefficient of sliding friction, that is $\mu_{stat.} > \mu_{slid.}$ and with an equal normal reaction $N \rightarrow$ the force of static friction is greater than the force of sliding friction. This difference depends on the materials from which the bodies are made and it varies within wide limits.

The physical phenomenon is characterized by the creation of impulses that vibrate the system.

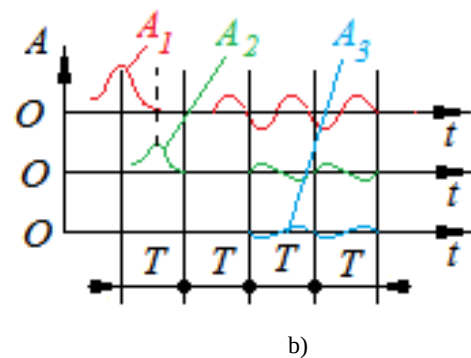
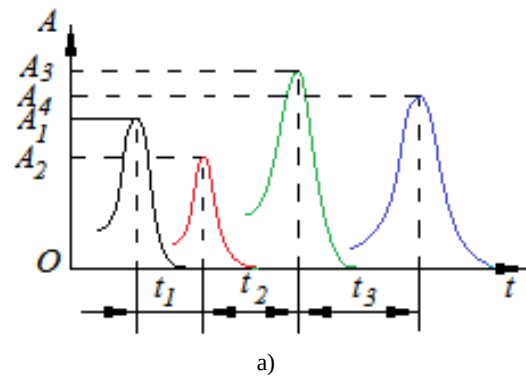


Fig.7. Vibrio Exciting Frictional Impulses

Fig.7. shows thrusts during the movement of sliding parts. It is clearly seen that the thrusts have different amplitudes and different periods. As a result of these thrusts, which act as impacts in certain directions, oscillations are excited in the system depending on the natural frequency of the bodies.

The dynamic damping of the excited oscillations can be adjusted by introducing a certain elasticity into one of the contacting bodies, thus adjusting the distances between the thrusts so that each subsequent thrust extinguishes the oscillations from the previous thrust (Fig. 7. b)).

Conclusion.

The dynamic damping of vibrations of vibroexcitation in contact of two or more bodies is especially important when there is periodic contact.

Dynamic damping of vibrations allows for optimal loading over the entire volume of the bodies, as well as optimal sizing of the parts. Forming is one of the main processes in the design of kinematic chains. The shown connection variant of the dynamic vibration damping makes it possible to select the appropriate contact technology, as well as the appropriate shape of the details when designing a given mechanism.

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