STUDY AND ANALYSIS OF VERTICAL AMPLITUDE FORCED VIBRATIONS OF ENGINE FIXED ON THE PRINCIPAL BEAM FRAME AT MACHINES TOOLS FOR WOODWORKING

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Abstract

Deformable material systems are characterized by conducting vibratory movement when operated from the outside. From elastic systems category are: installations for buildings, process equipment and stationary machinery and equipment. After a certain time, as was evidenced, that vibrations lead to degradation of the system: construction, machine or equipment.

Key words: vibrations, woodworking, engine, beam.

INTRODUCTION

Linear elastic systems with or without damping was observed that free vibration is the result of the initial conditions imposed displacement or velocity component of the system mass. Therefore at baseline the energy is transferred from the outside at the elastic system and it is set in motion. At later moments the energy transfer is canceled, so elastic system performs an free oscillatory motion.

This study corresponds to the case where on the elastic system is continuously present a dynamic or kinematic disturbing factor.

MATERIALS AND METHODS

On the horizontal beam of the frame deflection is considered as fixed the electric motor. The unbalanced motor is considered to be represented as a concentrated mass located at a fixed distance from the axis of rotation, It rotates around its axis with rpm n.

The problem is to determine the amplitude of the forced vertical vibration motor. Mass frame is considered to be negligible.

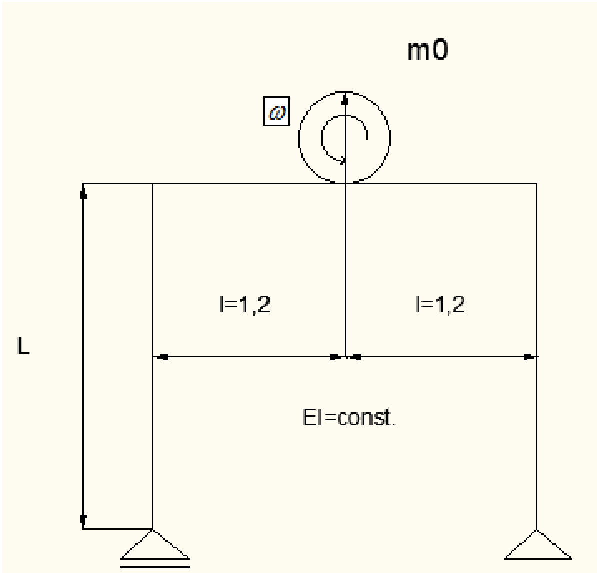


Fig. 1 The mechanical system.

The mechanical system is replaced by an equivalent system.

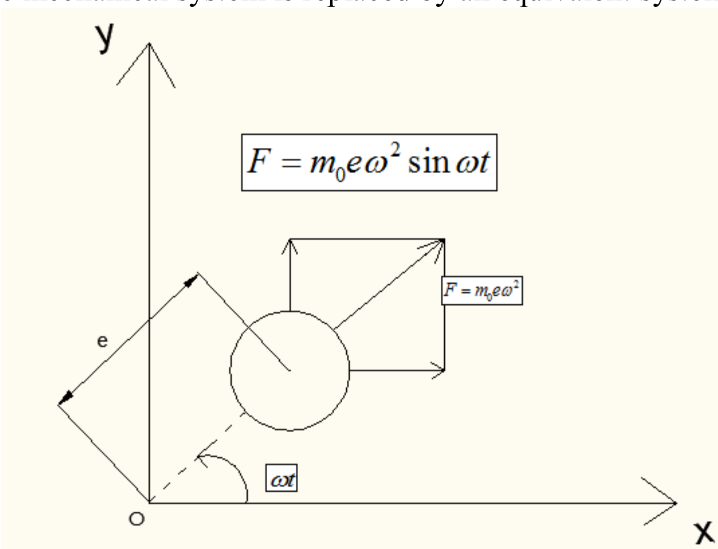


Fig. 2 The centrifugal force of inertia.

Determining forced vibration amplitude involves the following steps:

1. determine the angular velocity of the rotor motors (Bratu P, 1993)..

$$\omega = \frac{\pi n}{30} \tag{1}$$

2. calculate the vertical component of the centrifugal force of inertia due to eccentric rotating mass with angular velocity determined (Bratu P, 2000).

$$F = m_0 e \omega^2 \sin \omega t \tag{2}$$

3. Considering the middle of horizontal beam the point of application of disturbing force, is determined the stiffness coefficient of the frame on vertical direction of structure. For this initial, is calculate the static displacement given by a force applied to direction of centrifugal force of inertia (N) (Catarig A, 2003).

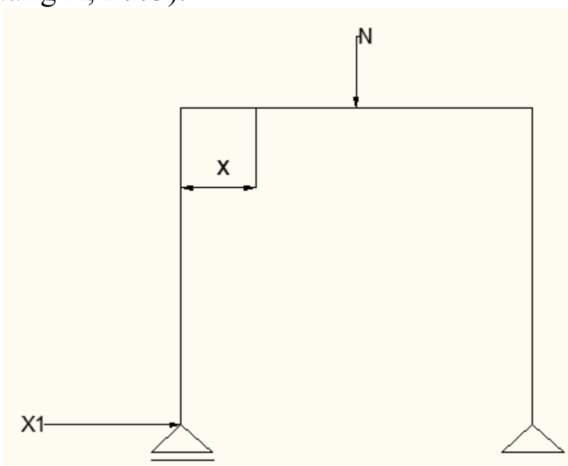


Fig. 3 The basic mechanical system.

By applying the efforts method is determined the equation of condition on the following form (Bia C, 1988):

$$x_{11}\delta_{11} + \Delta_{1P} = 0$$

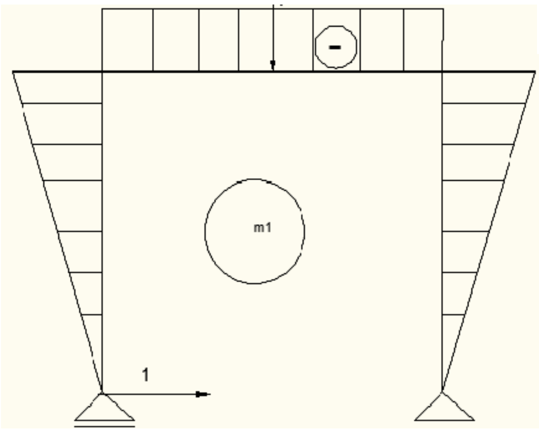
(3)

Where:

δ_{11}, Δ_{1P} – represents the point displacements.

Using Maxwell-Mohr formula and rule Veresceaghin integration, we obtain expressions for point displacements (Bratu P., 2000):

$$\delta_{11} = \sum \int \frac{m_1^2 ds}{EI}$$
$$\Delta_{1P} = \sum \int \frac{M_0 m_1}{EI} ds \tag{4}$$



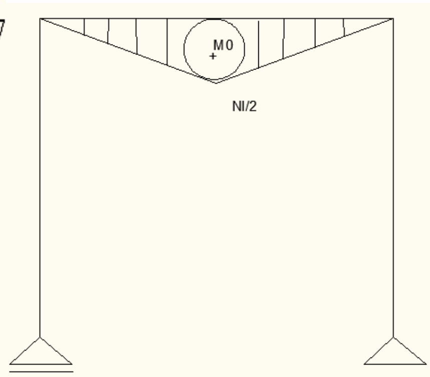


Fig. 4 Efforts sectional diagram m1 Fig. 5 Efforts sectional diagram M0

Using relation:

$$X_1 = \frac{\Delta_{1P}}{\delta_{11}}, \tag{5}$$

is determined the horizontal reaction of simply supported steel frame. By applying unit force on direction of disturbing force is obtained the static displacement.

$$f_k = \sum \int \frac{M m_2 dx}{EI} = \frac{2}{EI} \int_0^l \left(-x_1 l + \frac{N}{2} x \right) \frac{x}{2} dx \tag{6}$$

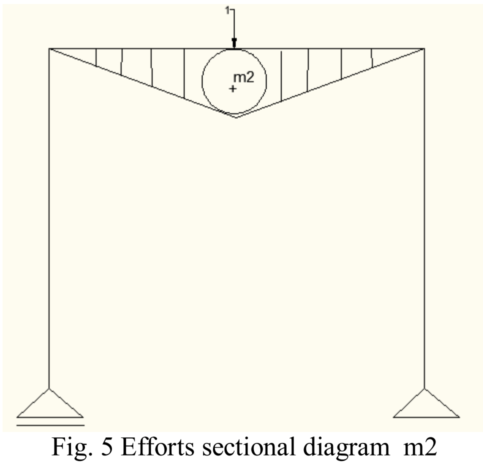


Fig. 5 Efforts sectional diagram m2

Using static displacement is determined stiffness coefficient of the frame with mathematical relation (Pantel E., 2002):

$$k = \frac{N}{f_k} \tag{7}$$

4. Using kinetic and potential energy expressions of the equivalent mechanical system is determined the inertia and stiffness coefficients (Barsan G., 1993).

$$E = \frac{1}{2} m \dot{y}^2 \rightarrow a = m$$
$$V = \frac{1}{2} k y^2 \rightarrow c = k \tag{8}$$

Where:

$a -$ represent the inertia coefficient.

$c -$ represent the stiffness coefficient.

5. The natural pulsation expression of oscillatory equivalent mechanical system is (Muntean M., 2007):

$$p = \sqrt{\frac{c}{a}} \tag{9}$$

For the initial phase of harmonic disturbing force in vertical direction considered as zero ($\delta = 0$) and with the amplitude given by

$$P = m_0 e \omega^2, \tag{10}$$

is determined parameter h, as the ratio of the the amplitude disturbing force and inertia coefficient a (Bors I, 2003).

$$h = \frac{P}{a} \tag{11}$$

5. Using the data obtainedis determined the amplitude of forced vibration.

$$A = \frac{h}{p^2 - \omega^2} \tag{12}$$

RESULTS AND DISSCUSIONS

In developing work have considered the following calculation dates:

- electric motor mass fixed in the middle of the frame at $l = 1.2[m]$ of woodworking tools

$$m = 105[Kg],$$

- the unbalanced electric motor is represented as a concentrated

mass

$$m_0 = 7[Kg],$$

being at a distance of 7 centimeters from the axis of rotation.

$$e = 7[cm],$$

with

$$n = 1700[rpm].$$

- the axial moment of the horizontal beam and columns of wood processing

machine frame is considered

$$I = 500\left[cm^4\right].$$

- Young's modulus

$$E = 2 \cdot 10^6 \left[daN / cm^2\right]$$

- mass frame is considered to be negligible.

Using previously established relationships, given mechanical system is replaced by a mechanical system equivalent form

The angular velocity is:

$$\omega = \frac{\pi n}{30} = 177{,}93\left[s^{-1}\right].$$

Using Maxwell-Mohr formula and rule Veresceaghin integration, we obtain the point displacements:

$$\delta_{11} = \sum \int \frac{m_1^2 ds}{EI} = -\frac{8 \cdot l^3}{3EI}$$

$$\Delta_{1P} = \sum \int \frac{M_0 m_1}{EI} ds = -\frac{N \cdot l^3}{2EI}$$

The horizontal reaction of simply supported steel frame expressed with the ratio of displacements point

$$X_1 = \frac{\Delta_{1P}}{\delta_{11}} = \frac{3 \cdot N}{16}$$

By applying uniform force $X_2 = 1$ on the basic system in the direction of static force N, is determined static displacement

$$f_k = \sum \int \frac{M m_2 dx}{EI} = \frac{2}{EI} \int_0^l \left(-x_1 l + \frac{P}{2} x \right) \frac{x}{2} dx = \frac{7Nl^3}{96EI}$$

Stiffness coefficient is

$$k = \frac{N}{f_k} = \frac{96EI}{7l^3} = 8{,}333 \cdot 10^6 \left[N / m\right].$$

The inertia and stiffness coefficients are:

$$E = \frac{1}{2} m \dot{y}^2 \rightarrow a = m = 105\left[Kg\right]$$

$$V = \frac{1}{2} k y^2 \rightarrow c = k = 8{,}333 \cdot 10^6 \left[N / m\right]$$

The natural pulsation of oscillatory equivalent mechanical system computed is

$$p = \sqrt{\frac{c}{a}} - 281{,}71\left[s^{-1}\right].$$

The amplitude of harmonic disturbing force in vertical direction is

$$P = m_0 e\omega^2 = 15513\left[N\right].$$

Parameter h, as the ratio of the the amplitude disturbing force and inertia coefficient a is:

$$h = \frac{P}{a} = 147.74 \cdot 10^2 \left[cms^{-2}\right].$$

Accordingly, forced vibration amplitude is

$$A = \frac{h}{p^2 - \omega^2} = 0.0309\left[cm\right].$$

CONCLUSIONS

In the case linear elastic systems with or without damping is known that the free vibrations are the result of the initial conditions imposed by displacement or velocity component of the system mass. In the event that the elastic system in this case the machine woodworking tools, is operated continuously by a disturbing force having a dynamic nature, the system is dynamic perturbed.

In this paper the problem is to determine the amplitude of forced vibration system, whose value does not exceed certain values well established, which would lead to the entrance of the machine mechanical system in resonance, followed by its destruction by increasing the value of the vibration to the infinite.

Resonance problem appears in cases where the natural pulsation of wood processing machine system is equal with the harmonic pulsation of disturbing force applied to the considered system.

In this case pulsations determined values are different:

$$p \neq \omega,$$

therefore not appears the possibility of entry into resonance of woodworking machine.

Also the paper presents method of calculation of the forced vibration amplitude, which can be compared with values set or determined for each type of woodworking machine. Amplitudes must not exceed certain values to ensure a normal operation of the machine parameters.

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