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ASSESSMENTS ON UNIDIRECTIONAL MECHANICAL VIBRATION GENERATORS

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ABSTRACT

This paper deals with some practical aspects of the design and evaluation of operational capabilities provided by unidirectional mechanical vibration generators embedded in technological equipment with a specific dynamic working regime. The study was focused both on theoretical computational basics and on specific responses evaluated based on the technique of the finite element method. A theoretical model with two degrees of freedom was considered to simulate the influence of technological process on the real and practical response of the vibration generator during the exploitation practice. Modal characteristics of the main parts of the mechanical generator were evaluated using the FEA technique. The main results were presented in detail and discussed, taking into account the interface reaction of processed materials due to the variable and intensive dynamic action provided by the vibration generator.

KEYWORDS: unidirectional mechanical vibration generator, eigenfrequencies, resonance, FEA

1. INTRODUCTION

The unidirectional vibration generator is designed to produce controlled oscillations in a single direction, typically using rotating eccentric masses, electromagnetic actuators, or pneumatic systems. The technological design process involves several essential stages, including [1-18]:

- Defining the functional requirements (vibration frequency and amplitude, generated force, durability, and energy efficiency);
- Selecting the operating principle (type of motor, method of vibration generation, direction and sense of vibrations) [18-25];
- Choosing appropriate materials based on dynamic loads, fatigue resistance, and working conditions;
- Calculating and modelling the main components (shaft, bearings, housing).

Centrifugal vibration generators periodically produce varying inertial forces due to the rotation of unbalanced masses. The most commonly used are statistically unbalanced flywheels, where the mass center is offset from the rotation axis. Dynamically unbalanced flywheels are also employed, characterized by non-zero centrifugal inertia moments, as well as combinations of these two types [12-16,24-29].

The magnitude, spatial orientation, and time variation of the generated forces and moments depend both on the motion of the element and on the characteristics of the driving motor. Typically, the interaction between the forces and moments developed at the flywheel shafts and the driving motor is neglected, assuming constant angular velocity during operation [30].

2. THEORETICAL BASICS

The centrifugal force is calculated using the expression [27]:

$$Fcf = m_{e0} \times r \times \omega^2 \quad (1)$$

where: m_{e0} is the eccentric mass, r is the eccentricity, and ω is the angular velocity.

For the entire vibratory assembly, four eccentric masses will be considered (two for each wheel in the system).

Thus, the total eccentric mass is:

$$m_e = 4 \times m_{e0} = 4 \times 0{,}092 = 0{,}368\ kg \quad (2)$$

It follows that the minimum excitation force is:

$$F_{cfmin} = m_e \times r \times \omega^2 \cong 100 \quad (3)$$

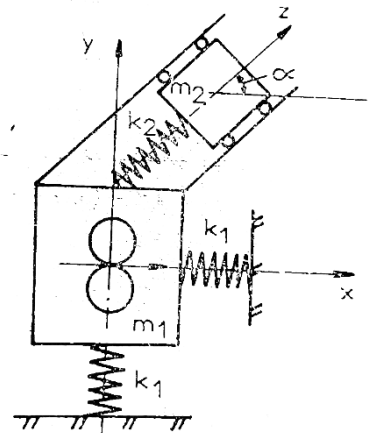


Fig. 1. Mathematical model with three degrees of freedom and negligible damping

The kinetic energy and the force function of the system are [27]:

$$E_c = \frac{(m_0+m_1)\dot{x}^2}{2} + \frac{(m_0+m_1)\dot{y}^2}{2} + \ldots + \frac{m_2(\dot{x}+\dot{z}\cos\alpha)^2}{2} + \frac{m_2(\dot{y}+\dot{z}\sin\alpha)^2}{2}$$
$$U = -\frac{k_1x^2}{2} - \frac{k_1y^2}{2} - \frac{k_2z^2}{2} \quad (4)$$

where z denotes the relative displacement of the working organ with respect to the balancing mass, and α denotes the direction of the relative displacement.

The perturbing force is directed along the Ox axis and is equal to $F = F_0\cos\omega t$, with magnitude $F_0 = m_0 r\omega^2$. The differential equations of motion are [27]:

$$\begin{aligned}(m_0+m_1+m_2)\ddot{x} + m_2\ddot{z}\cos\alpha + k_1x &= F_0\cos\omega t\\ (m_0+m_1+m_2)\ddot{y} + m_2\ddot{z}\sin\alpha + k_1y &= 0\\ m_2\ddot{z}\cos\alpha + m_2\ddot{y}\sin\alpha + m_2\ddot{z} + k_2z &= 0\end{aligned} \quad (5)$$

The notations made are [27]:

$$p_0 = \sqrt{\frac{k_1}{m_0+m_1}};\ \varepsilon = \frac{m_2}{m_0+m_1};\ \varepsilon_1 = -\frac{m_0}{m_0+m_1};$$
$$\gamma = \frac{\omega}{p};\ u = \frac{k_2}{k_1}. \quad (6)$$

The equation of the natural frequencies is

$$\begin{vmatrix}[k_1-(m_0+m_1+m_2,)p^2]\cdots & \\ \cdots[k_1-(m_0+m_1+m_2)p^2]\cdots & \\ \cdots(k_2-m_2p^2-m_2^2p^4) & \end{vmatrix} = 0$$

from which the expressions of the dimensionless natural angular frequencies, denoted in order of their magnitudes, result [27-29]:

$$\left(\frac{p}{p_0}\right)_{I,II,III} = \gamma_{I,II,III},$$
$$\gamma^2_{I,III} = \frac{1}{2}\left[1+\left(1+\frac{1}{\varepsilon}\right)u \pm \sqrt{\left[\left(1+\frac{1}{\varepsilon}\right)u\right]^2 + 1 - 4\frac{u}{\varepsilon}}\right]$$
$$\gamma^2_{II} = \frac{1}{1+\varepsilon}. \quad (7)$$

The stationary solutions of the system (5) are [27-29]:

$$x = x_0\cos\omega t\ ;\quad y = y_0\cos\omega t\ ;\quad z = z_0\cos\omega t\ . \quad (8)$$

The expressions of the amplitudes can be written in the form [27-29]:

$$x_0 = \varepsilon_1 r\xi_{ca}\ ;\ y_0 = \varepsilon_1\eta_{ca}\ ;\ z_0 = \varepsilon_1\zeta_{ca} \quad (9)$$

The amplification factors are given by the relations [27]:

$$\xi_{ca} = \gamma^2\frac{[1-(1+\varepsilon)\gamma^2]\left[\frac{u}{\varepsilon}-\gamma^2\right]-\varepsilon\gamma^2\cos^2\alpha}{[1-(1+\varepsilon)\gamma^2]\left\{[1-(1+\varepsilon)\gamma^2]\left(\frac{u}{\varepsilon}-\gamma^2\right)-\varepsilon\gamma^4\right\}},$$
$$\eta_{ca} = \frac{\varepsilon\gamma^6\sin\alpha\cos\alpha}{[1-(1+\varepsilon)\gamma^2]\left\{[1-(1+\varepsilon)\gamma^2]\left(\frac{u}{\varepsilon}-\gamma^2\right)-\varepsilon\gamma^4\right\}}, \quad (10)$$
$$\zeta_{ca} = \frac{\gamma^4\cos\alpha}{[1-(1+\varepsilon)\gamma^2]\left(\frac{u}{\varepsilon}-\gamma^2\right)-\varepsilon\gamma^4}.$$

The relative amplitude z_0 has a minimum value at the dimensionless angular frequency [27-29]

$$\gamma' = \sqrt[4]{\frac{u}{\varepsilon}}. \quad (11)$$

The amplitude z_0 vanishes for [27-29]:

$$\gamma''_{12} = \sqrt{\frac{1+\varepsilon+\frac{u}{\varepsilon}\mp\sqrt{\left(1+\varepsilon-\frac{u}{\varepsilon}\right)^2-4\frac{u}{\varepsilon}(1+\varepsilon\cos^2\alpha)}}{2(1+\varepsilon\cos^2\alpha)}} \quad (12)$$

Relative to the resonance angular frequencies, the angular frequencies $\gamma', \gamma'_1, \gamma'_2$, are located as follows [27-29]:

$$\gamma_{II<\gamma'<\gamma'_{III}};\ \gamma_{I<\gamma'_1<\gamma_{II}};\ \gamma_{II} < \gamma''_2 < \gamma_{III} \quad (13)$$

From expressions (8), it follows that the trajectory of the balancing mass is a straight line with the equation [27-29]:

$$y = \frac{\eta_{ca}}{\zeta_{ca}} x \tag{14}$$

inclined to the horizontal by the angle β,

$$\tan\beta = \frac{\eta_{ca}}{\xi_{ca}} = \frac{\varepsilon^2\gamma^4 \sin\alpha\cos\alpha}{[1-(1+\varepsilon)\gamma^2](u-\varepsilon\gamma^2)-\varepsilon^2\gamma\cos^2\alpha}. \tag{15}$$

If the two-degree-of-freedom model is used, the elastic constant k_1 is calculated as shown, along the direction of the angle β.

The parametric equations of the absolute motion of the working organ are:

$$\begin{aligned} x_a &= (x_0 + z_0\cos\alpha)\cos\omega t, \\ y_a &= (y_0 + z_0\sin\alpha)\cos\omega t. \end{aligned} \tag{16}$$

The trajectory of the working organ is also a straight line with the equation:

$$y_a = \frac{\eta_{ca}+\zeta_{ca}\sin\alpha}{\xi_{ca}+\zeta_{ca}\cos\alpha} x_a \tag{17}$$

Inclined to the horizontal by the angle β_a,

$$\tan\beta_a = \frac{\eta_{ca}+\zeta_{ca}\sin\alpha}{\xi_{ca}+\zeta_{ca}\cos\alpha}. \tag{18}$$

As shown, for reasons of reduced vibration transmissibility, a small constant k_1 is recommended $k_1 < k_2$. The three-degree-of-freedom model allows for a more complete highlighting of the operating ranges, in relation to the trajectories of the masses and stresses on the leaf springs.

The introduction of damping into the systems does not fundamentally change the method of analyzing the problem.

3. COMPUTER-BASED MODELLING AND SIMULATION

In the case of generators equipped with statically unbalanced flywheels arranged in a plane, the resulting inertial forces are coplanar with the motion plane, while the vectors of the corresponding moments, if present, are perpendicular to this plane.

The 3D modelling of the unidirectional vibration generator was carried out using Solid E4dge CAD environment.

Solid Edge is an advanced 3D CAD software developed by Siemens Digital Industries Software, used for the design of mechanical products and equipment. It provides a comprehensive solution for product development by integrating functionalities for 3D modeling, simulation, manufacturing, technical drawing, and product data management.

The main components of the software are:

a) Part Design (3D part modeling)
b) Assembly (assembly design)
c) Drafting (2D drawings)
d) Sheet Metal (sheet metal parts design)
e) Simulation (CAE analysis)
f) Manufacturing (CAM processes)

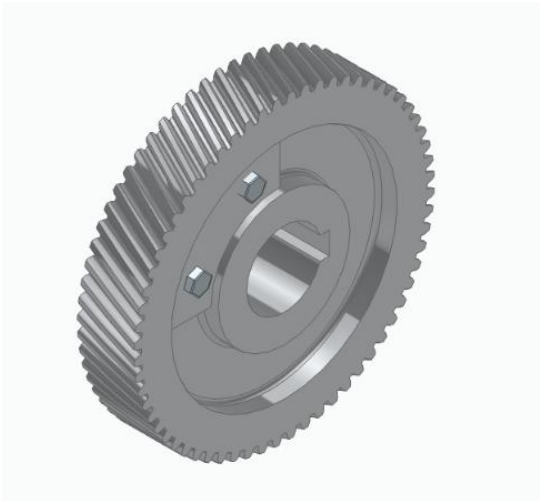


Fig. 2. The gear wheel with eccentric masses assembly



Fig. 3 The shaft

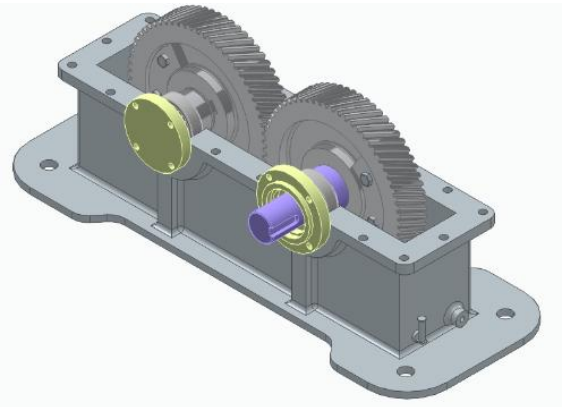


Fig. 4 The assembly

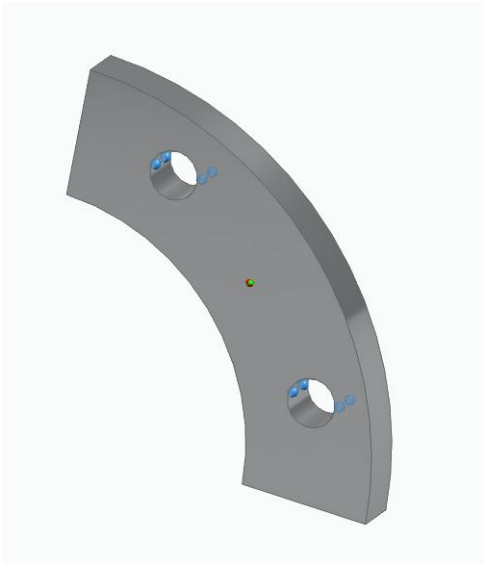
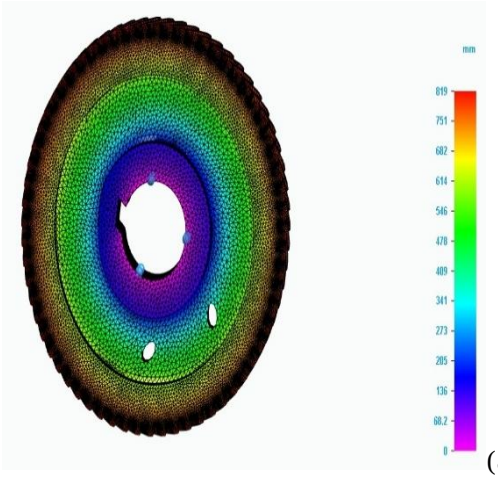


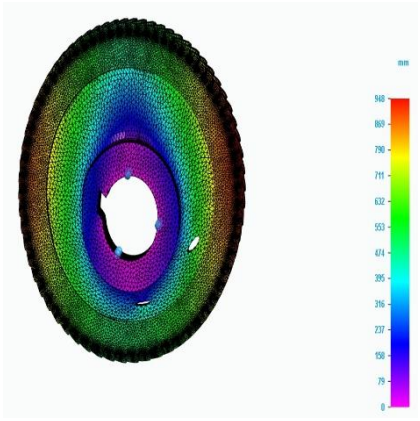
Fig. 5 Eccentric mass

4. RESULTS & DISCUSSIONS

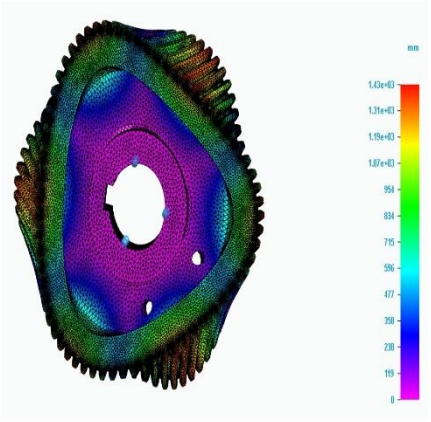
In the case of the eccentric mass, the support is provided on the inner surface of the two bores, being subjected to a centrifugal force-type load. For the gear wheel, the support was applied on the inner surface of the hub, while the load was applied to only one of the two bores used for mounting the eccentric masses on the wheel.



(a)

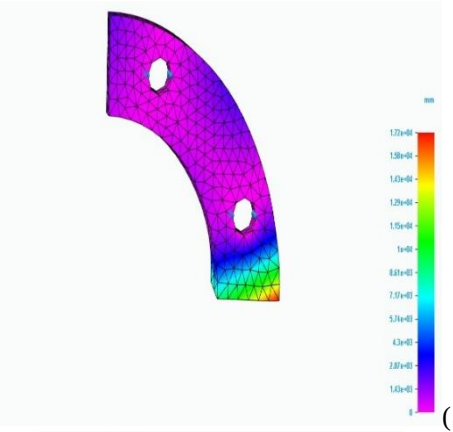


(b)

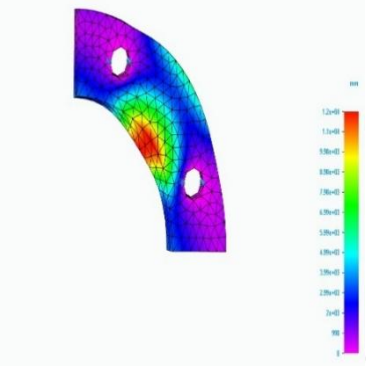


(c)

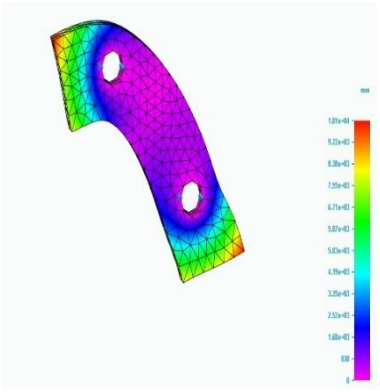
Fig. 6 Modal analyses of the gear wheel: first mode (a), 6th mode (b), and 9th mode



(a)

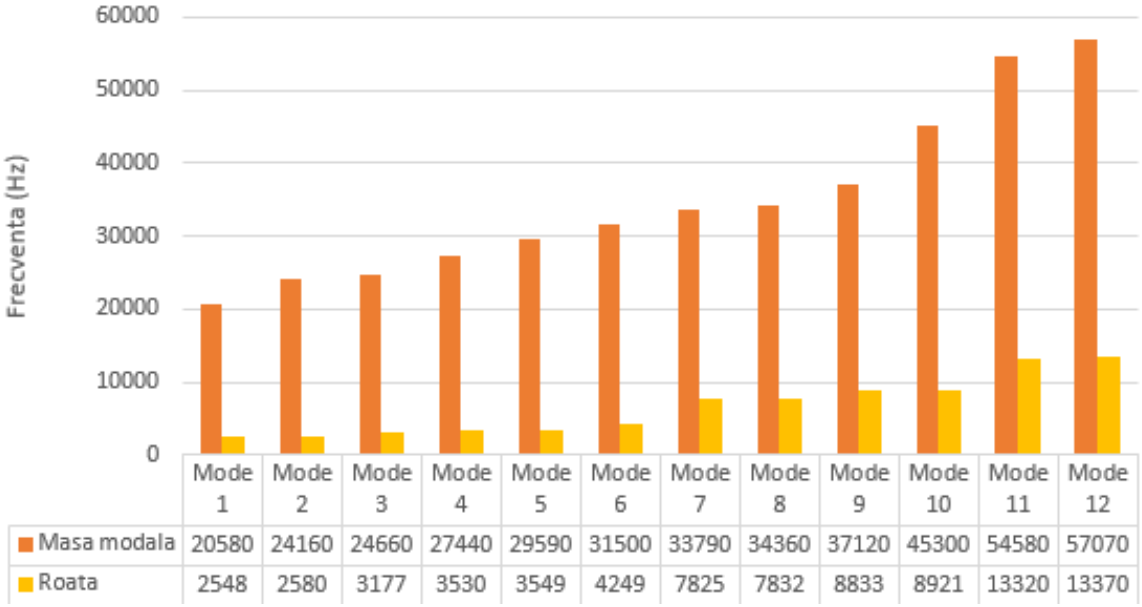


(b)



(c)

Fig. 7 Modal analyses of the eccentric mass: third mode (a), 6th mode (b), and 9th mode.



	Mode 1	Mode 2	Mode 3	Mode 4	Mode 5	Mode 6	Mode 7	Mode 8	Mode 9	Mode 10	Mode 11	Mode 12
Masa modala	20580	24160	24660	27440	29590	31500	33790	34360	37120	45300	54580	57070
Roata	2548	2580	3177	3530	3549	4249	7825	7832	8833	8921	13320	13370

Fig. 8 The modal parameters of the eccentric mass and the gear wheel respectively, in terms of first 12 natural frequencies for each part.

5. CONCLUSIONS

In this work, a complete assembly was designed, and after integrating all components, a 3D model of the unidirectional vibration generator was created.

The assembly was modeled using Solid Edge design software, version 2025. This software was chosen due to its advantages, such as ease of 3D modeling and the short time required to develop models. Additionally, it ensures compatibility with 2D and 3D files from AutoCAD, SolidWorks, and the DWG format. 3D models provide greater design flexibility, allowing designers to focus on functional aspects rather than geometric parameters.

The technological design of a unidirectional vibration generator involves a complex approach that combines knowledge from mechanics, materials science, manufacturing processes, and the dynamics of vibrating systems.

Key aspects include:

- operating principle: the use of a rotor with eccentric masses that generate centrifugal forces, producing unidirectional vibrations in the desired plane;
- material selection: a heat-treated alloy steel shaft, a cast iron housing, high-load-capacity radial bearings, and solid steel eccentric masses;
- maintenance: periodic lubrication of the bearings, inspection of the eccentric masses for wear or loosening, and continuous monitoring of vibrations over time.

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