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ANALYSIS OF A GRAVITY BALANCER FOR A MARINE POWER PLANT

Abstract: *The paper addresses the design and analysis of a gravity balancer for a marine power plant, aimed at increasing vibro-isolation effectiveness and reducing structural noise. The research is based on the application of zero-stiffness systems in the supports of marine power units. A gravity balancer design based on a multi-disk friction coupling is proposed, ensuring the creation of constant pressure and compensation of the vibration source weight. A balancer's design algorithm, mathematical model, and new effectiveness evaluation criteria have been developed based on the difference between excitation and inertial forces. The results obtained confirm that the proposed approach is promising in terms of improving vibro-protection systems for marine power plants and reducing vibration levels.*

Key words: *marine power plant; vibro-isolation, gravity balancer; zero-stiffness system, multi-disk friction coupling; vibro-protection; structural noise; constant pressure system; mathematical modeling; vibration reduction.*

Language: *English*

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Introduction

The main sources of vibration on a ship are: the piston internal combustion engine, the diesel generator, and the deck machinery [1, 2]. Ship units operate at a frequency of 1500 rpm, therefore, they are always considered sources of structural noise on a ship. Noise has a significant impact on the living quarters where crew members are constantly present, some at their workplaces and others at rest. Other sources of vibration are less harmful, both in terms of spectrum and structural noise. The main cause of vibration is the unbalanced masses executing the translational motion of the power plant and the non-uniformity of the working process [3, 4, 5]. In addition, acoustic radiation from the engine surface occurs at low frequencies on its oscillatory elastic suspension, as an absolutely rigid body.

Object of research

We conducted a study on the application of zero-stiffness systems in the supports of ship equipment [6]. The study was conducted in the research laboratory of the Engineering Faculty of Batumi State Maritime Academy. Figure 1 illustrates a photograph of the research object [7].

Fig. 1. Object of research

Prerequisites

Purtskhvanidze, G. et al. (2025) [6] demonstrated that the vibration isolation methods currently in use are insufficient, which necessitates further research. Therefore, the goal of subsequent work is to develop effective criteria for designing a vibration isolation method for ship power plants that contains no viscous or elastic elements.

We conducted an analytical study of vibration-protective structures for the supports of ship power plants [8]. The analytical review revealed the shortcomings of the many vibration isolation methods currently in use; therefore, our goal is to reduce vibration by improving the effectiveness of vibro-isolation mounting.

We developed a vibration protection system for ship power plants based on the principle of constant pressure, as presented in [9]. In that work, a successful attempt was made to reduce the natural frequencies to zero by using constant-pressure systems whose operation was based on the independence of dry-friction forces from sliding velocity.

Basic Part

To stabilize the engine position, a special device is required, referred to as a gravity balancer. The gravity balancer must generate a force that is independent of the engine position; therefore, its stiffness must be zero. Secondly, the pressure generated by the balancer must be independent of the engine's vibration velocity; therefore, its viscosity must be zero.

Preliminary analysis identified three relatively simple methods for generating a constant force: (1) sliding of a slider on a flat belt; (2) sliding of a flat surface on a drum; and (3) relative rotation of coaxial disks. In the first case, the main drawbacks are low reliability and large overall dimensions. In the second case, the drawbacks are insufficient friction and excessive wear of the contacting surfaces. The third approach is more promising for several reasons, including its modular design, which allows the friction-contact area to be increased to the required level while maintaining low contact pressure. Another advantage is that the required force can be obtained by varying the lever arm while keeping the friction force constant. It should also be noted that the friction surfaces are enclosed and protected by the balancer body.

The gravity balancer is based on a multi-disk friction coupling design. The stationary half-coupling balances the vibration source with a handle mounted on its body. The three-dimensional model is shown in Figure 2.

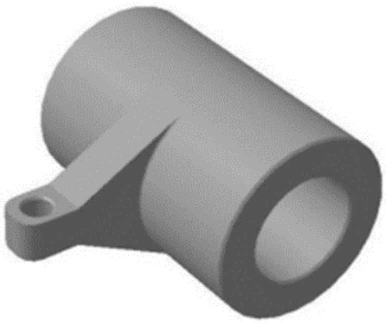

Fig. 2. Gravity balancer with a stationary half-coupling handle

Unlike the drive coupling, the balancer is additionally loaded by a force applied at a distance from the axis. This force compensates for the weight of the device and is calculated as the ratio of the moment to the distance to the point of force application.

Theoretically, the force can be arbitrary for a given moment of the friction force. Still, in practice, this is not possible, since the nonlinearity of forces as a function of the rotation angle manifests itself when the handle length is small. There is a problem associated with the occurrence of transverse oscillations due to the excessive length of the handle. In this case, we must rely on the balancer's primary design parameter, such as the mass of the device. The load acting on each support is determined based on the number of supports. The pressure on the balancer's bearings equals the nominal load on the support. Accordingly, the torque on the shaft equals the product of the weight and the moment arm. This torque twists the shaft and yields the drive power when multiplied by the rotational speed. However, the sliding speed of the friction disks is bounded from below by the vibration speed of the oscillation source. The expression for the amplitude of friction power in the force generator is as follows:

$$M_f = G \cdot V,$$

where G – is the weight of the device per support (N); V - amplitude of the source vibration velocity (m/s).

The velocity on the friction coupling disk varies from 0 (at the center) to maximum at the outer edge of the disk. It is recommended to carry out the design of the gravity balancer in the sequence shown in Figure 3.

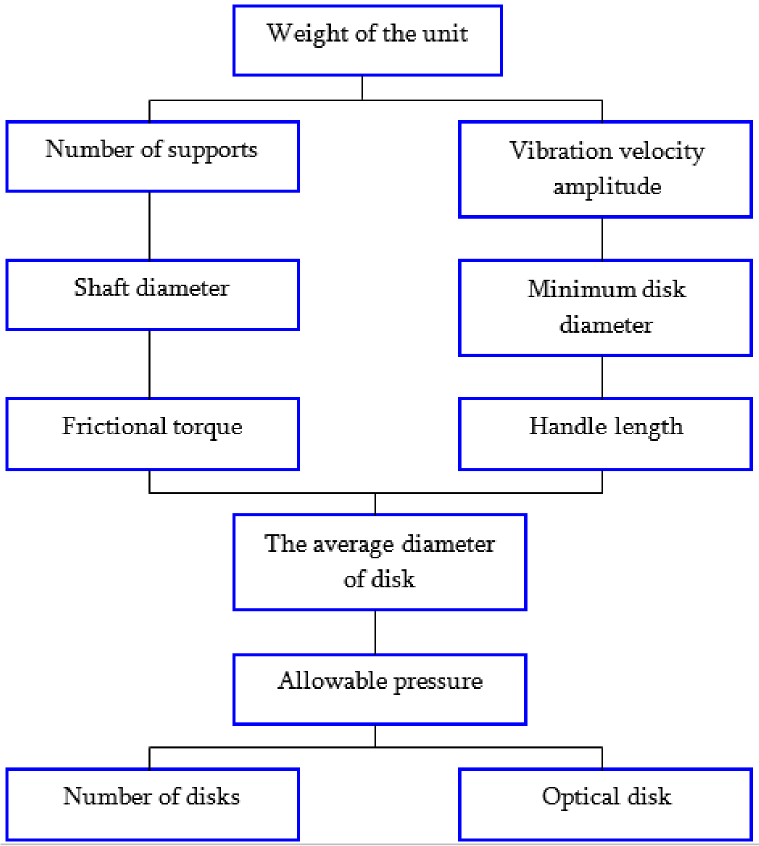

Fig. 3. Design algorithm for a gravity balancer

Let us take a look at one of the options for the structural embodiment of a constant-pressure element based on the dry friction of flat surfaces (Fig. 4).

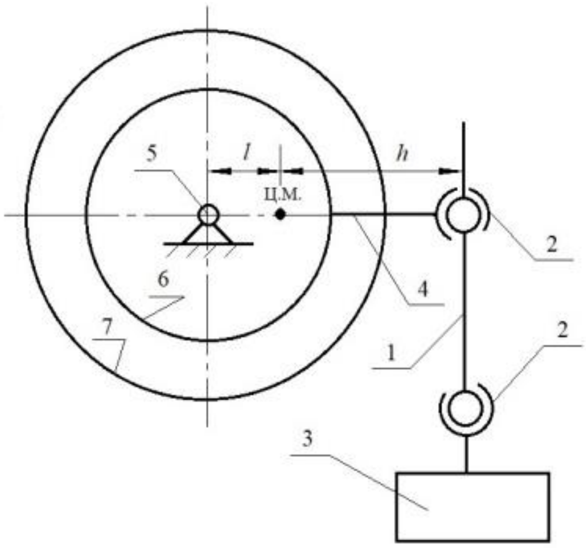

Fig. 4. Schematic diagram of a gravity balancer

Figure 4 illustrates a diagram containing the following combination of existing elements:

1. Spatial vibration converters, made in the form of a vertical rod (1), which is attached by a lower three-axis joint (2) to the body of the mechanical vibration source (3), and at the top to a horizontal axis (4), so that the center of percussion of the rod (1) coincides with the upper joint (2);

2. Planar vibration converters, made in the form of a horizontal rod (4), attached by a flat joint at the center of percussion to the protected object (5);

3. A friction element, made of two circular surface axes (6 and 7), one of which is attached to the engine shaft, and the other is rigidly connected to the axis (4) of the mechanical vibration converter;

4. A balancer, based on a multi-disk friction belt, with essentially small dimensions and mass.

Further studies were done on the latter, i.e., the balancer.

Let us choose a polar coordinate system with a massive disk at the center. The differential of the torque is equal to

$$dM = 2\pi \cdot r^2 \cdot f \cdot \rho \cdot dr,$$

where r – the current radius;

f – the friction coefficient;

ρ - contact pressure;

dr - the differential of the radius.

The frictional torque of a massive disk is calculated using the following formula:

$$M = f \cdot \rho \cdot \frac{\pi D^3}{12},$$

where D – disk diameter.

In the central part of the disk, the sliding velocity becomes lower than the critical value. Consequently, the friction surface should be designed without a central section (Fig. 5).

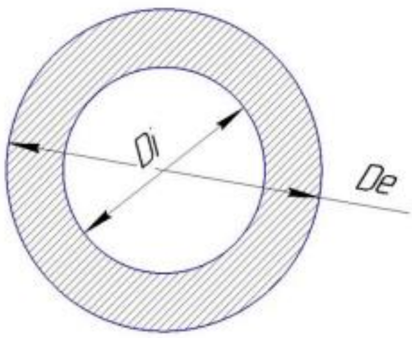

Fig. 5. Friction surface with outer D$_ℓ$ and inner D$_i$ diameters

The friction torque in this case is equal to

$$M = f \cdot \pi \cdot \rho \cdot \frac{D_\ell^3 - D_i^3}{12}$$

The torque can be determined through force and lever arm. If we choose the lever arm, it is possible to obtain the required force that compensates the weight of the device. At constant velocity, dry friction does not depend on the sliding velocity.

The study was carried out on a model containing weight, balancer's force, driving force, and viscous friction force. For this system, the equation of dynamics has the form as follows:

$$\frac{d^2q}{dt^2} \cdot m = f \cdot \cos(\omega t) - G + F(q) - b\frac{dq}{dt},$$

where $\frac{d^2q}{dt^2}$ - is the acceleration of the mass; m - mass; t - time; G - weight; f - amplitude of the driving force; ω - angular frequency of the driving force; $F(q)$ – gravity balancer's force; b - viscosity coefficient of the damper; $\frac{dq}{dt}$ - the acceleration of the mass.

The damper viscosity is determined from the logarithmic decrement during the vibration source's oscillations. Its damping characteristics depend on the design of the electrical cables, cooling system, fuel supply system, and control and measurement equipment. Incorporating friction into the numerical modeling shortens the transient response without reducing the vibration isolation effectiveness.

The balancer's force $F(q)$ equals the weight of the vibration source (G) when the amplitude of the oscillation velocity (q_2) is below a given value (v). In the Mathcad programming language, we obtain the following conditional operator

$$F(q) = if(q_2 < v, G, 0)$$

For a viscoelastic system, unlike the numerical integration model of the dynamics equation, the present model excludes stiffness and adds the vibration velocity V and the gravity balancer's force $F(q)$, which depends on the amplitude q_2 of the vibration velocity as well as on the third-order integral of acceleration.

Let us optimize the initial data based on the vibration velocity standard for electrical units. Insufficient balancer velocity causes disk grip and loss of vibro-isolation. Excessive balancer velocity, in turn, increases power consumption. The mass of the vibration source was set equal to the average mass of a marine power plant operating at 50 kW.

$$v := .02 \qquad f := 1000 \qquad m := 1000 \qquad \omega := 100$$

$$G := 9.81 \cdot m \qquad F(q) := \mathrm{if}(q_2 < v, G, 0) \qquad b := 700$$

$$init\_vals := \begin{pmatrix} 0 \\ 0 \\ 0 \end{pmatrix} \qquad init\_t := 0 \qquad final\_t := 6 \qquad Nsteps := 5000$$

$$D(t,q) := \begin{bmatrix} q_1 \\ q_2 \\ \frac{1}{m}\left(f \cdot \cos(\omega \cdot t) - G + F(q) - b \cdot q_2\right) \end{bmatrix}$$

Fig. 6. Numerical integration model of the dynamic equation with a gravity balancer

In linear systems with metal and rubber-metal elastic elements, the effectiveness of vibration-damping mountings is determined by means of the vibro-isolation coefficient, which is equal to the ratio of the forces acting on the base to the driving force, or by means of the dynamic coefficient. In the given case, this method is inapplicable, since to obtain the elastic force, the amplitude must be multiplied by the stiffness, which is equal to zero. Since a support with zero stiffness does not change the dynamics of the unit mass, the source vibrations will be harmonic at the driving force's frequency. This allows the effectiveness to be calculated using a new method, as the difference between the inertial and driving forces. Since the sum of the inertial force and the driving force is close to zero, the remainder will be the base reaction.

The analysis of the results revealed that the system is in a balanced state when the difference of forces is close to zero.

Conclusions

A design of a gravity balancer for a power unit is proposed, based on a multi-disk friction coupling operating in braking mode. A two-link handle mechanism was used to convert pure torque into force. To determine the effectiveness in the zero-stiffness model, new criteria were proposed based on the difference between the excitation force and the inertial force; an approach to the design of friction surfaces was developed as a compromise solution between energy losses due to friction and the dimensions of the coupling disk.

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