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DEVELOPMENT OF A WEIGHT COMPENSATION METHOD FOR THE SHIP DIESEL GENERATOR SUPPORT

***Abstract**: Vibration, generated by the ship's diesel generators, is one of the primary sources of structural noise, which negatively affects the living and working conditions of the crew. Traditional sound insulation systems are characterized by limited efficiency due to resonance phenomena across a wide frequency range, dependence on stiffness, and sensitivity to changes in operating conditions. The research presents a methodology for weight compensation in the support of a ship’s diesel generators, based on the concept of zero stiffness and implemented using a multi-disk dry friction compensator. A dynamic mathematical model of the system has been developed, and numerical simulations have been performed. The results obtained confirm that the compensation provides force balance and slightly reduces the dynamic reaction, transmitted to the base.*

Key words: *diesel generator, ship vibration, vibration isolation, zero stiffness system, dry friction compensator.*
Language: *English*
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Introduction

As our research has shown, the leading sources of vibration on board ships are internal combustion engines, diesel generators and deck machinery [1.2]. The ship's aggregates operate at a frequency of 1500 rpm; they are always considered a source of the structural noise on board ships. Noise has a significant impact on the living quarters, where crew members are constantly present—some working, some resting. Other sources of vibration are less harmful, both in terms of frequency spectrum and structural noise. The main cause of vibration is unbalanced mass in the power unit and irregularities in its operation [3,4,5]. In addition, low-frequency acoustic radiation from the engine occurs due to its oscillations while vertically suspended, behaving almost like a rigid body.

Over the past seventy years, scientists M.A. Fedoseeva, A.M. Baranovsky, and A.K. Zuev have studied methods to reduce transmitted forces, but complete success has not been achieved for several reasons. Firstly, zero stiffness implies restructuring the vibration isolation under a different load, when the operating mode of the vibration source changes, since the vibration source is in an undefined position, relative to the base. In this case, only power supply systems are effective, which, for a number of reasons, such as loss of stability, are not used in the shipbuilding industry. Thus, the problem of restoring the vibration isolator arises. Secondly, the elastic element in the support transmits vibrations, high-frequency noise and creates resonant modes. Thirdly, dry friction systems do not allow the creation of continuous bearing forces over a long period of operation.

Problem statement and research objective

We conducted a study of the course of vibration process in the supports of the ship equipment and the use of zero stiffness systems to relieve them [1,10]. The study was conducted in the research laboratory of the Faculty of Engineering of the Batumi State Maritime Academy. The object of the study was a diesel generator. The results of the study showed that it is necessary to develop a vibration isolation system that will allow us to reduce the vibrations and noise [11].

The scientific basis of the vibration isolation of the ship's power equipment is discussed in the article [12]. Comparison of analytical methods for calculating viscous systems and numerical modelling, using the Runge-Kutta method in the Mathcad program, shows practically identical results, so this method can be used to successfully integrate any, including nonlinear equations, in the work. An attempt was made to reduce the frequencies to zero by using constant pressure systems; the operation, sliding velocity of oscillations, is less than the sliding velocity, which does not violate the accepted concepts.

We also investigated the structures of the vibration-proof supports of the ship's power equipment and studied the periodic characteristics of vibration isolation methods used today [13]. So, the goal of our research is to reduce vibration by improving the efficiency of vibration isolation mounting.

Weight compensator concept

In order to balance the position of the motor, it is necessary to eliminate the resonances and compensate for the force of gravity. For this purpose, a special device is needed, which can be called a weight compensator. The main requirements for such a device are unique, as it must be competitive with traditional viscous systems in the entire frequency range.

Firstly, the weight compensator must create a bearing that does not depend on the position of the engine, so its stiffness must be equal to 0. Secondly, the compensator's bearing must not depend on the engine's oscillation speed, so its viscosity must be equal to 0.

Preliminary analysis has shown three relatively simple methods for creating a constant bearing: 1) Sliding of a crawler on a flat belt; 2) Sliding of a flat surface on a drum; 3)Relative rotation of the coaxial disks. In the first case, the design is characterised by low reliability and large dimensions. In the second case, the disadvantages are insufficient friction and wear of the contacting surface. The third case is more promising for several reasons. First, the modular construction between the two allows the friction surface area to be increased to a desired value, resulting in lower contact pressure. Another advantage is the ability to achieve the required force by adjusting its side at a given moment. It is also important to note that the compensator housing protects the friction surfaces.

Structural Design, Mathematical Modelling, and Analysis of Results

The weight compensator is based on a multi-disc friction clutch design. The stationary half-clutch compensates for the vibration source by means of a handle fixed to its body. The three-dimensional model is presented in Fig .1;

Unlike a drive shaft, a compensator is additionally loaded with a force that is applied by removing it from the axis. This force compensates for the weight of the device and is calculated as the ratio of the moment to the distance from the point of Application of the Force

To ensure stable operation, the central zone—where the sliding speed is below the critical value—is excluded from the friction surface.

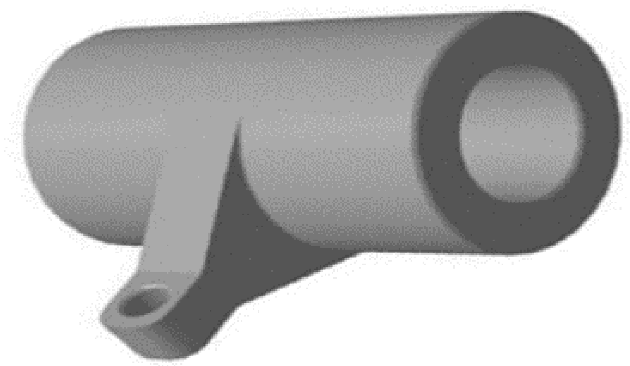

Fig. 1. fixed half – circle of the weight compensator with a handle

The speed on the friction clutch disc varies from zero (in the Centre) to a maximum on the outer side of the disc. It is recommended to design the weight compensator in the sequence shown in fig. 2.

Let us consider one of the options for the design of a permanent element, based on dry friction of flat surfaces (Fig 3).

The drawing shows a scheme containing the following set of existing signs.

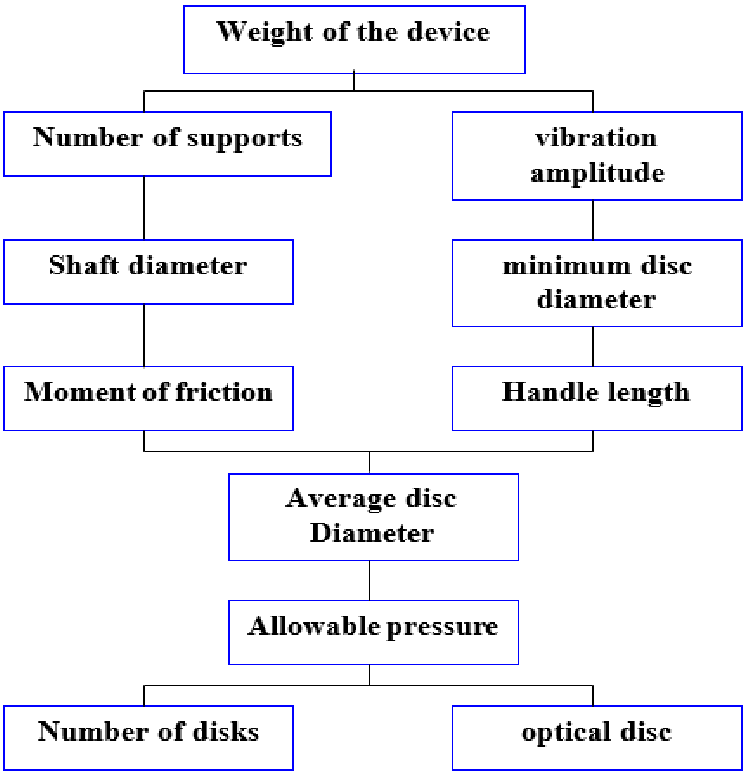

Fig 2. Design Algorithm of a Weight Compensator

1. Transducers of the spatial vibrations, made in the form of a vertical rod (1), Which is attached by the lower three – point joint (2) to the body (3) of the mechanical vibration source, and by the upper one to the horizontal axis (4), so that the Centre of impact of the rod (1) coincides with the upper joint (2).

2. Transducers of plane vibrations, made in the form of a horizontal rod (4), which is attached to the protective object (5) at the impact Centre of a Flat Joint

3. Friction elements, made of surface axes (6; 7), one of which is attached to the motor shaft, and the other is rigidly connected to the axis (4) of the mechanical vibration transducers.

4. Compensator, based on a multi – disc friction clutch with substantially smaller dimensions and weight, has the form.

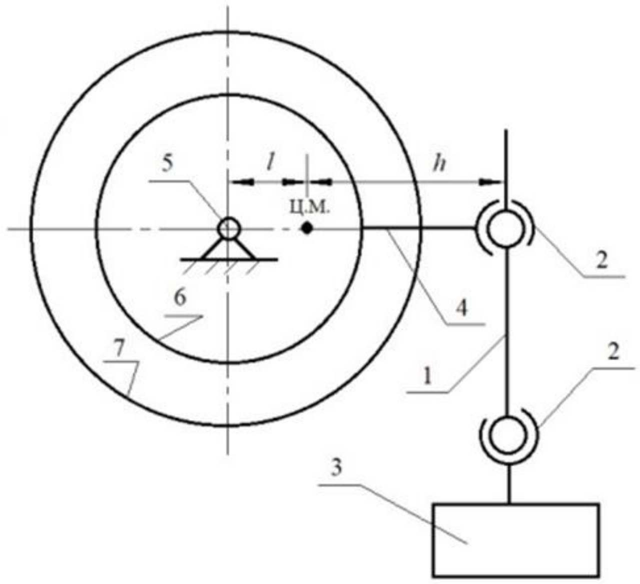

Fig 3. Circuit diagram of the weight compensator

The study was carried out on a model that included the weight, the compensator force, the drivingforce, and the viscous friction force. For this system, the governing equation has the form:

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

Where: $\frac{d^2q}{dt^2}$ alt 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)* - force of the weight compensator; b - coefficient of viscosity of the damper, and acceleration of the mass - $\frac{dq}{dt}$.

The damper viscosity is determined from the decay rate of the damping force during oscillatory motion of the vibration source. Its damping depends on the design of the electrical cables, the cooling system, the fuel supply system and the control and measuring equipment. Accounting for friction during numerical modelling reduces the transient process without reducing the effectiveness of the vibration isolation.

The force generated by the compensator $F(q)$ is equal to the weight of the vibration source (G), when the amplitude of the vibration velocity (q_2) is below a given value (v). In the Mathcad programming language, the following conditional operator is defined:

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

The mathematical model with the conditional operator is presented in Fig.4.

For a viscoelastic system, unlike the numerical integration model of the dynamic equation, this model neglects stiffness and instead incorporates oscillatory motion, the velocity V, and the compensator force $F(q)$, which depends on the amplitude of the oscillation velocity q_2, In addition, a third-order term corresponding to the integral of acceleration is included.

Let’s enhance the initial data based on the vibration speed norm for the electrical units. Insufficient speed of the compensator causes disk slippage and loss of vibration isolation. High speed of the compensator leads to an increase in power. The mass of the vibration source was set equal to the average mass of the ship's power plant at an output of 50 kW.

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

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

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

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

Fig 4. Numerical integration model of the dynamic equation with a mass compensator

In linear systems, the effectiveness of damping using elastic elements—either metal or rubber-metal—is determined by the vibration isolation coefficient, which is equal to the ratio of the transmitted force, or by the dynamic coefficient. This method cannot be used in this case, since the elastic force would require multiplying the amplitude by the stiffness, which is zero. Since the support with zero stiffness does not change the dynamics of the mass of the aggregate, the oscillations of the source will be harmonic, the support with the zero stiffness does not change the dynamics of the mass of the aggregate, the oscillations of the source will be harmonic, with the frequency of the driving force. This method allows us to calculate effectiveness using a new approach, by taking the difference between the inertial force and the driving force. Since the sum of the inertial and driving forces is close to zero, the residual corresponds to the reaction of the base.

Conclusion

The method for weight compensation in ship diesel Generator Mounts has been developed based on the concept of zero stiffness of the Multi-Disk Dry Friction Compensator:

- Eliminates resonance caused by the stiffness.
- Provides a constant support force, regardless of movement and speed.
- Significantly reduces the dynamic response, transmitted to the base.

The obtained results confirm the further development of the presented prospects, creating a basis for experimental approach research and practical implementation.

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