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ZERO-RIGIDITY SYSTEMS IN SHIP EQUIPMENT SUPPORTS

Abstract: *The purpose of the article is to study the periodic characteristics of vibration excitation created by ship equipment: engines and diesel generators, as well as to analyze the design of vibration-protective supports for power equipment.*

Key words: *engine, vibration, vibration-isolating devices, single-axis model of vibration-isolating mount.*

Language: *English*

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Introduction

Our research has shown that the main vibration system of ship equipment with periodic excitation characteristics is the engine and diesel generator [1, 2, 3, 4, 5, 6]. The vibration behavior of the power unit in the range of sound frequencies mainly depends on disturbing factors arising during the operation of the engine. The main cause of vibration is the imbalance of the translational masses of the power plant mechanism and the uneven working process. In this case, the acoustic radiation of the engine surface at low frequencies occurs as a result of its vibration on an elastic suspension as an ideal solid body.

Research Findings

The analysis of the design of vibration-suppressing supports of power equipment revealed the following disadvantages: 1) Rubber and rubber-metal supports have a high rubber density, which does not create acoustic resistance. This creates the prerequisites for vibration transmission to the foundation shelf. Such vibration isolators do not ensure the safe operation of ship power equipment; 2) Metal supports are made in the form of cylindrical springs. This form is the best in terms of energy capacity, since the coil material works in shear. Cylindrical springs are effective at low frequencies and a large number of turns, but are not stable enough. This disadvantage is eliminated by increasing the diameter of the turn, but the mass of the spring increases sharply; 3) The use of pneumatic supports does not guarantee protection from high-frequency noise and sufficiently fast restructuring. This type of support is difficult to manufacture and operate, does not provide transverse vibration protection from spatial vibrations of power equipment.

The development of zero-rigidity supports takes a period of sixty years [7]. The reason that researchers ignored the existence of zero and especially negative rigidity is that the equations of the dynamics of such systems give an unstable solution.

In zero-rigidity systems, the elastic element is deformed along a given trajectory without friction. Geometry makes the force nonlinear [8]. If we approach the concept of rigidity formally, we can get into the area of negative values [9]. Such a device is called a force compensator or a rigidity corrector.

During the synthesis of the corrector system with intermediate bodies, optimal solutions were obtained. The body with sharp ends turned out to be unsuitable for large loads and was replaced by a finite radius calculated by contact stresses (Fig. 1.).

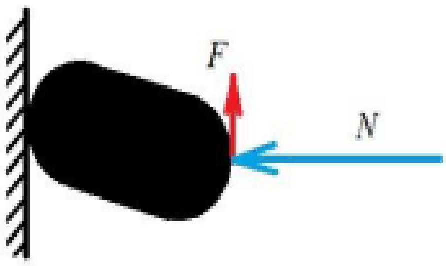

Fig. 1. The best form of intermediate element in the stiffness corrector:
N – Compression force of intermediate element,
F – Quasi-elastic force.

By connecting such a rigidity compensator in parallel with a conventional spring, it is possible to obtain a section of the power characteristic with zero rigidity. The main difficulty is that the transition to a new load was accompanied by a sharp deterioration in vibration insulation. This was due to the impacts on the slider travel limiters. The best characteristics were shown by systems with the stiffness corrector turned off during the restructuring process. Since many technical objects are constantly in a variable mode, the use of zero-stiffness supports is not widespread.

Another solution is to supply energy equal to work of spring deformations within the vibration limits. This problem has never been fully solved. The technical solution applied on the navigator's seat, provides for switching off the compensator during the transition to another load, which allowed to cut the rearrangement effort in half. The stiffness corrector consisted of three needle rollers with a diameter of 2.5 mm, installed side by side, enclosed in a separator and pressed by a spring. When the rollers approached the friction angle, the spring sat on the limiters and the rollers slipped.

A further development of the zero-rigidity method was the use of active supports using the energy of vibrations for readjustment to a new load. This solution was based on a self-braking slider [10]. The vibration mode was chosen so that negative feedback on the displacement of the protected object would occur. The test results were encouraging with a slow change in the position of the vibration source, as high speeds led to the abuse of feedback. The speed of readjustment was insufficient for use in ship power plants.

Another difficulty in improving the quality of vibration isolation is related to the transmission of energy through the elastic element [11, 12]. If the mass of the elements of the corrector mechanism can be made non-essential, then the mass of the main elastic element cannot be reduced due to the strength condition and it becomes a good waveguide.

The second difficulty that must be overcome is due to the design traditions. The contradiction between high rigidity accepted under the condition of system stability and low rigidity under the condition of effective vibration isolation is resolved in favor of high rigidity. The thrust, reactive moment, pitching and other factors require high rigidity, which is the reason behind the poor performance of vibration isolation [13].

The efficiency of the considered methods, devices and systems is limited for various reasons [14]. The linear approach to the problem allows considering each factor separately and then summing up the protection against vibration of the ship's hull [15, 16]. Figure 2 illustrates a single-axis model of the vibration-insulating mount.

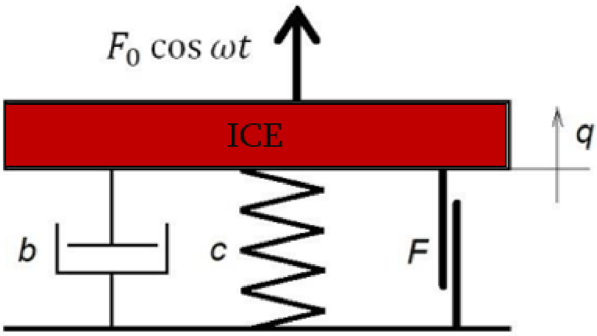

Fig. 2. Standard vibration protection model

The system contains a mass installed through an elastic linear element and a clutch of dry friction.
The harmonic force is applied to the mass of the unit

Write an equation of motion of mass in Newton form.

$m\ddot{q} = F_0 \cos \omega t - -b\dot{q} - cq - Fsign(\dot{q}),$

where m – a mass;

$\ddot{q}$ – mass acceleration;

F_0 - amplitude of the driving force;

ω – driving force angular frequency;

t – time;

b – a viscosity index of damper;

$\dot{q}$ – velocity of a mass;

c – rigidity;

q – the location of the mass;

F – dry friction force;

$sign(\dot{q})$ – velocity signature.

Three forces are transmitted to the protected base: viscous friction, elastic force, dry friction force. Thus, ideal vibration protection is possible when all three forces are equal to zero or constant. The mathematical model contains a dynamics equation in which the elastic force was replaced by the dry friction force. The conditional operator controlled the difference in vibration velocity and friction slip velocity. The model showed the existence of a critical speed, above which the force is not transmitted to the base [17, 18].

There are studies of the transfer of dry friction force to the protected base with kinematic excitation of the source [17]. In this case, instead of solving the equation of dynamics, an expression for the dry friction force was composed. The idea was that the method is fundamentally more effective than vibration isolation of elastic forces and viscous friction forces. Viscosity and elastic forces have some common basis through the time derivative. The viscous force according to Newton depends on the speed or on the first derivative of the displacement with respect to time. The inertial force depends on the second derivative of the displacement with respect to time. In accordance with this approach, devices based on elastic elements have been developed. For elastic force, a shift from resonance to the region of high frequency ratios is used. Protection from viscous force is practically impossible and it is used to reduce resonance peaks. The inertial force can be damped only with the help of dynamic vibration dampers or antivibrators, but these devices have a narrow range of operating frequencies or low efficiency.

The summary

Thus, the study revealed the shortcomings of many currently used vibration isolation methods. Therefore, the goal of further work is to develop an efficiency criterion for the vibration isolation system of ship power plants that do not contain elastic and viscous elements.

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