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	JIF = 1.500	SJIF (Morocco) = 6.004	OAJI (USA) = 0.350

SOI: 1.1/TAS DOI: 10.15863/TAS

International Scientific Journal

Theoretical & Applied Science

p-ISSN: 2308-4944 (print) **e-ISSN:** 2409-0085 (online)

Year: 2026 **Issue:** 09 **Volume:** 161

Received: 09.09.2026
Accepted/Published: 30.09.2026

https://T-Science.org

Issue Article



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IMPROVING THE VIBRATION PROTECTION EFFICIENCY OF SHIP POWER EQUIPMENT USING CONSTANT-LOAD AND ZERO-STIFFNESS SUPPORTS

Abstract: *The paper presents a method for reducing vibration of ship diesel generators based on the creation of a constant bearing load and an effective stiffness approaching zero. The study is based on the single-axis dynamic model, weight-compensation method, and structural solution of a gravity balancer using a multi-disk dry-friction coupling developed in the dissertation. It is shown that conventional elastic and viscous elements may produce resonant regimes at certain frequencies and contribute to the transmission of structural noise. In the proposed model, the weight of the equipment is compensated by a force F(q) which, within the operating range, should not depend on the position of the equipment; consequently, the stiffness and viscosity of the compensator are targeted toward zero. The paper presents the equation of motion, friction-torque calculation, geometric optimization of the friction surface, and a new vibration-isolation evaluation criterion based on the difference between excitation and inertial forces. The results of the dissertation and related publications are consolidated; the additional graphs are analytical calculations derived from the same equations and do not represent new experimental measurements.*

Key words: *ship diesel generator; vibration isolation; zero stiffness; constant load; weight compensator; gravity balancer; dry friction; multi-disk coupling; structural noise.*

Language: *English*

Citation: Purtskhvanidze, G., Jijavadze, O., Chokharadze, T., Jijavadze, N., & Jijavadze, A. (2026). Improving the vibration protection efficiency of ship power equipment using constant-load and zero-stiffness supports. *ISJ Theoretical & Applied Science, 09 (161)*, 21-27.

Soi: https://s-o-i.org/1.1/TAS-09-161-3 ***Doi***: https://dx.doi.org/10.15863/TAS.2026.09.161.3

Scopus ASCC: *2200.*

1. INTRODUCTION

Vibration of ship power equipment, particularly diesel generators and internal-combustion engines, is one of the important sources of structural-noise transmission to the hull. The research materials indicate that engine vibration is associated with unbalanced masses, irregularities in operation, and moments of inertia forces of the connecting rod. The dissertation analysis also shows that, with conventional supports, resonant regimes and the transmission of high-frequency vibration cannot be completely eliminated. [1–3]

One limitation of a conventional elastic support is its stiffness: the force transmitted to the base depends on displacement. In a viscous element, the force depends on velocity. Therefore, an idealized vibration-isolation system should be sought in which the principal support reaction does not change within the operating range of the equipment. The zero-stiffness concept is intended to address this problem. [4, 5, 11]

In previous publications, the authors' group considered zero-stiffness supports and the constant-load principle for ship power equipment. The 2026 paper proposed a multi-disk dry-friction compensator and a gravity balancer, while vibration-isolation effectiveness was evaluated through the difference between excitation and inertial forces. [4, 6]

2. SOURCES OF VIBRATION IN SHIP POWER EQUIPMENT AND PROBLEMS OF EXISTING VIBRATION ISOLATION

According to the dissertation analysis, a significant part of ship power-equipment supports is represented by viscoelastic systems. For standard supports considered on a massive foundation, a resonant zone of approximately 15–28 Hz is reported; under ship conditions, the lower mass of the foundation structure may shift the natural-vibration range to approximately 18–32 Hz. [1, 10]

For rubber and rubber-metal supports, changes in material properties during operation are problematic, whereas for metal springs there is a compromise between achieving low stiffness and maintaining stability. The dissertation data show that, for ДПН–2 and ДПН–665 supports, dynamic stiffness exceeds static stiffness in several cases, indicating the dynamic character of the support system. [1, 12]

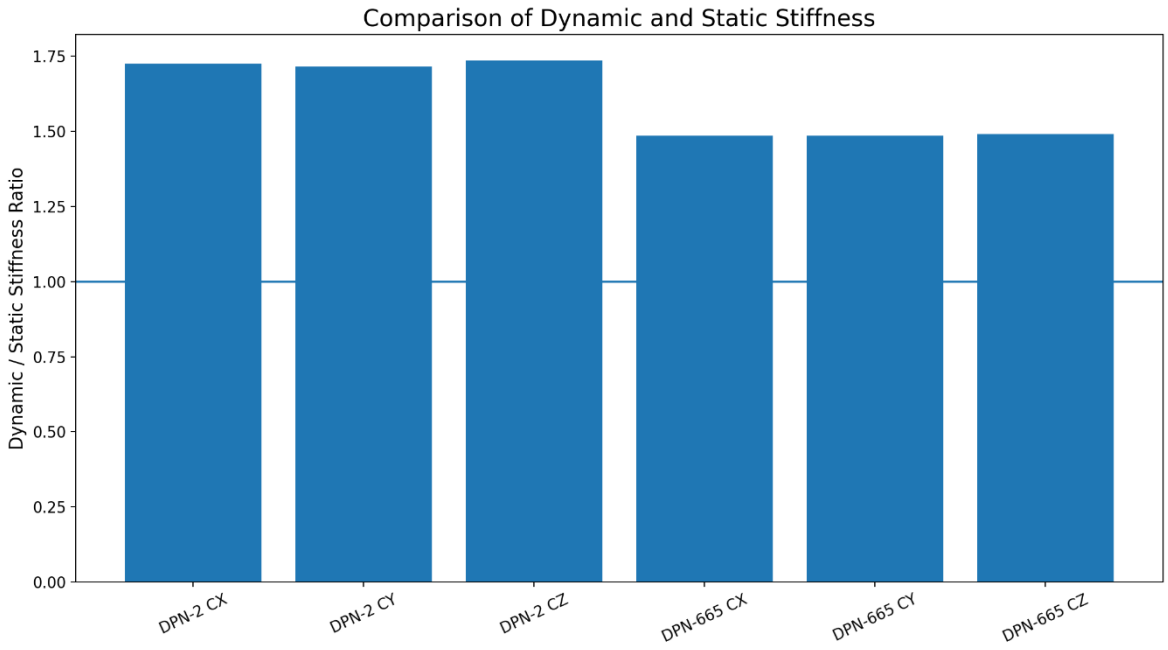


Fig. 1. Dynamic-to-static stiffness ratios of the ДПН-type supports reported in the dissertation (author's calculation from the data of Table 1.5).

The same materials report an increased vibration-acceleration level in the 40–50 Hz octave range, considered to be associated with the firing frequency or with imbalance of the moments of inertia forces of the connecting rod. These results show that changing only the elastic properties of the support is insufficient for protection over the entire vibration spectrum. [1]

3. THEORETICAL BASIS OF CONSTANT-LOAD SUPPORTS AND ZERO-STIFFNESS SYSTEMS

The idea of a zero-stiffness support is to create a force for the equipment that does not depend on its static position. A gravity balancer has two principal requirements: the generated force should not depend on the position of the equipment, and the generated pressure/force should not depend on vibration velocity. Accordingly, the target values are k≈0 and b≈0. [5, 6]

Several principles for generating a constant force are considered: sliding of a slider on a flat belt, sliding of a flat surface on a drum, and relative rotation of coaxial disks. A multi-disk dry-friction coupling is particularly attractive because the required constant force can be obtained in a compact structure through the parallel action of several friction pairs. [5, 6, 12].

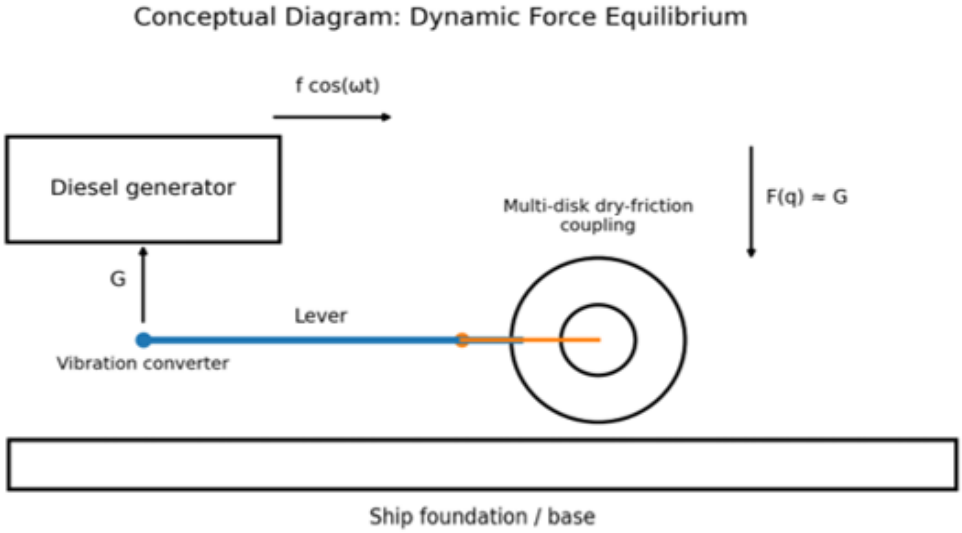


Fig. 2. Conceptual scheme of a constant-load and zero-stiffness support.

4. MATHEMATICAL MODEL OF THE DIESEL GENERATOR

For the single-axis model presented in the dissertation and the 2026 publication, the equation of motion of the equipment can be written in Newtonian form:

$$m\,\ddot{q} = f\cos(\omega t) - G + F(q) - b\,\dot{q} \quad (1)$$

Here, m is the equipment mass; q is displacement; $\dot{q}$ and $\ddot{q}$ are velocity and acceleration; f is the amplitude of the excitation force; ω is the angular frequency of the excitation force; G is the equipment weight; F(q) is the weight-compensator force; and b is the viscous coefficient. [5]

Under ideal operation of the compensator, its force is equal to the equipment weight within the operating range. In the numerical model used, this condition is expressed by a conditional operator:

$$F(q) = \text{if}(q^2 < v,\ G,\ 0) \quad (2)$$

where v is the threshold parameter for compensator activation, while q^2 in the programmatic description is related to the amplitude of vibration velocity. The source notes that insufficient compensator speed causes disk slip and loss of isolation, whereas excessive speed increases energy consumption. [5]

An important new criterion for the vibration-isolation system is obtained when the elastic force of the support is excluded. In this case, the inertial and excitation forces, under the appropriate phase relationship, balance each other, while their residual is considered the reaction transmitted to the foundation. [6]

$$R(t) \approx F_in(t) - F_exc(t) \quad (3)$$

Thus, the effectiveness problem is transformed from merely determining a transmissibility coefficient into minimizing the difference between excitation and inertial forces.

5. NUMERICAL MODELING AND ANALYTICAL PRESENTATION OF THE RESULTS

According to the dissertation and related publications, the numerical model allows the amplitude and vibration velocity to be varied. The dissertation considers initial data for a ship power unit in the 50 kW class. [1, 5]

One important conclusion of the model is that, in the zero-stiffness case, instead of reducing the natural frequency of the equipment motion, the force reaction

of the support is controlled. Consequently, the principal indicator becomes force balance rather than simply reducing the conventional natural frequency. [6]

The two graphs below do not represent a new series of experimental data. They are constructed directly from the geometric relationships given in the dissertation and are used to visualize the interaction of the design parameters.

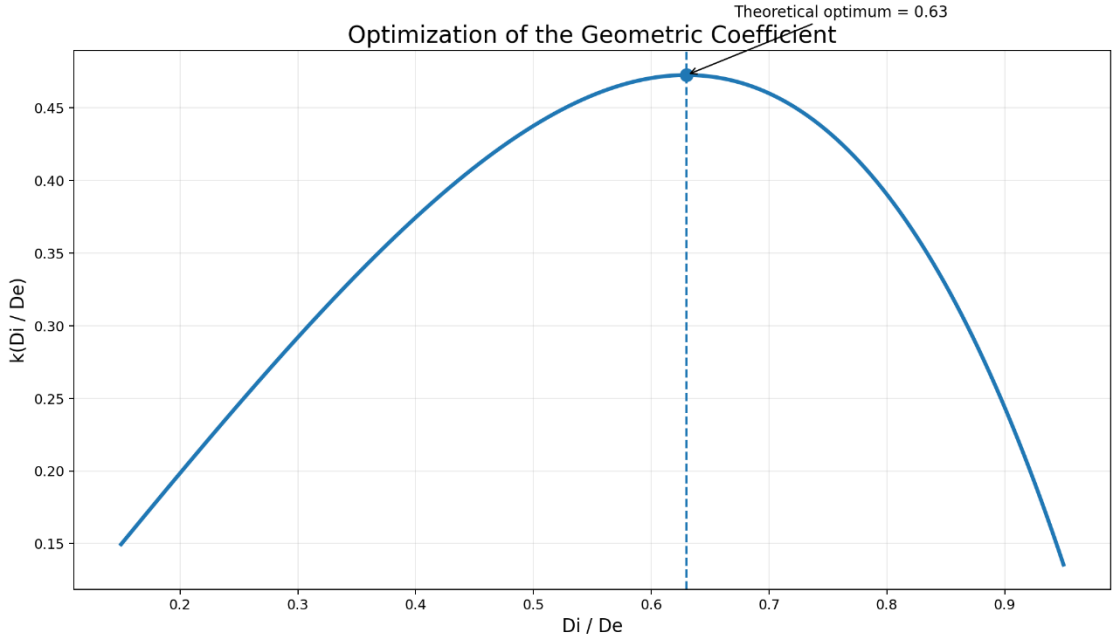


Fig. 3. Dependence of the normalized geometric coefficient k(Di/De) on the disk-diameter ratio. According to the source, the optimum is obtained at Di/De≈0.63. [1]

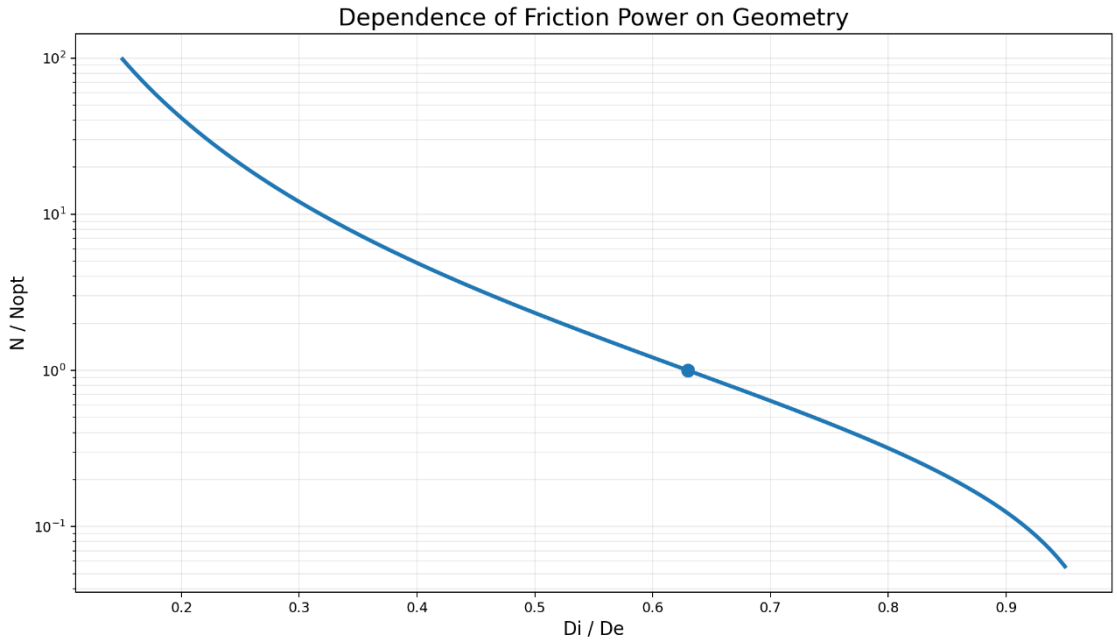


Fig. 4. Dependence of normalized friction power on Di/De according to the N/Nopt relationship given in the source. [1, 6]

6. STRUCTURAL SOLUTION OF THE WEIGHT COMPENSATOR AND GRAVITY BALANCER

A multi-disk friction coupling can be used to create a constant pressing force between flat surfaces. The friction torque obtained by integrating the friction force over the disk surface is given in the source as:

$$M = f \cdot \rho \cdot \pi \cdot D^3 / 12 \quad (4)$$

For the complete multi-disk friction surface, taking the inner and outer diameters into account:

$$M_f = f \cdot \pi \cdot \rho \cdot (D_e^3 - D_i^3) / 12 \quad (5)$$

Here f is the coefficient of friction, ρ is the parameter corresponding to the allowable/contact pressure, and De and Di are the outer and inner diameters. In the dissertation, optimization of the friction surface is considered as a compromise between energy losses and the dimensions of the structure. [1, 6]

The compensator design sequence includes determination of the equipment weight, vibration-velocity amplitude, minimum diameter, handle length, number of supports, shaft diameter, and friction torque, followed by selection of the number of disks and the optimum geometry. [1]

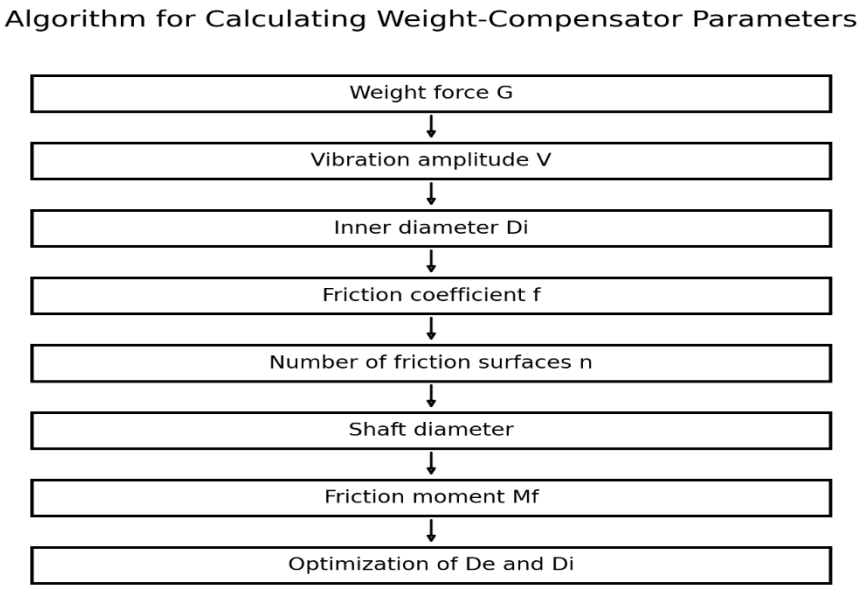


Fig. 5. Simplified design algorithm for the weight compensator, according to the sequence given in the dissertation. [1]

The amplitude of friction power in the force generator is given in the source as:

$$M_f = G \cdot V \quad (6)$$

where G is the load per support/equipment weight in the corresponding dimensional form, and V is the amplitude of vibration velocity. This relationship shows that the energy requirement of the compensator is directly related to both the load and vibration velocity. [6]

The handle length of the gravity balancer is also an optimization parameter: an excessively short handle may increase angular nonlinearity, while an excessively long handle may produce transverse oscillations. Therefore, its selection should jointly account for equipment mass, number of supports, and friction torque. [6]

7. EVALUATION OF VIBRATION-ISOLATION EFFECTIVENESS

In classical systems, vibration-isolation performance is often related to the transmissibility coefficient, dynamic coefficient, or the ratio of the force transmitted to the base. In a zero-stiffness system, calculating the elastic force as kq loses its defining meaning because the target value is k≈0. Therefore, the paper uses a different approach—evaluation of the difference between excitation and inertial forces. [6]

If F_exc(t) is the excitation force and F_in(t)=−m q̈ is the inertial force, then in the balanced state their algebraic sum should be close to zero. The remaining component determines the reaction transmitted to the foundation.

$$E_vib \propto |F_exc(t) + F_in(t)| \quad (7)$$

Thus, improving effectiveness means minimizing E_vib over the operating frequency and amplitude range. This formulation directly corresponds to the force-balance principle described in the source.

8. DISCUSSION OF RESULTS

The combined consideration of the research materials shows that the constant-load and zero-stiffness approach is a concept different from conventional supports. The principal objective is not only to reduce the natural frequency of the support, but also to compensate the equipment weight and reduce the dynamic reaction transmitted to the foundation.

The 2025 paper formulates the problem of a single-axis model and force transmission for zero-stiffness systems. The 2026 paper continues this approach through the weight compensator and gravity balancer; it proposes a multi-disk friction coupling, a lever-type converter, and a force-difference-based evaluation criterion. [4, 6]

The dissertation gives Di/De=0.6939 as the ratio of the average diameters obtained through friction-surface optimization, whereas the theoretical optimization expression gives approximately 0.63. The dissertation notes that these values are close and that the practical construction can accept the value within approximately 10% accuracy. [1]

The practical significance of this result is that the compensator design can simultaneously account for friction-surface size, energy losses, and force-generation capability. It should also be emphasized that the present article consolidates the results of completed studies and does not claim new experimental measurements.

9. CONCLUSIONS

1. The use of conventional elastic and viscous supports to reduce vibration of ship diesel generators is accompanied by resonant regimes and transmission of structural noise.

2. The zero-stiffness concept allows the vibration-isolation problem to be considered not only in terms of reducing the natural frequency, but also in terms of controlling the force transmitted to the foundation.

3. The principal requirement for a weight compensator is to generate a constant force that is substantially independent of equipment position.

4. A multi-disk dry-friction coupling provides a practical structural basis for generating constant load.

5. The equation of motion allows the excitation force, weight, compensator force, and viscous resistance to be considered within a unified model.

6. A criterion based on the difference between forces is appropriate for zero-stiffness systems in which the classical kq-type indicator is no longer the governing parameter.

7. The friction-surface optimization presented in the dissertation indicates a theoretical optimum of Di/De≈0.63, while the average ratio obtained from the existing constructional data is 0.6939.

8. Selection of the compensator handle, disk diameters, and number of disks should be performed jointly, taking into account energy losses and stability requirements.

9. For further research, it is advisable to extend the physical model and add multi-axis dynamics and experimental vibro-acoustic measurements on an actual ship.

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