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Ships and marine technology — Guidelines on vibration isolation design methods for shipboard auxiliary machinery

Navires et technologie maritime — Lignes directrices pour la conception de l'isolation antivibratoire des machines auxiliaires de bord

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Contents

Page

Foreword

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This document was prepared by Technical Committee ISO/TC 8, *Ships and marine technology*, Subcommittee SC 8, *Ship design*.

Introduction

This document has been developed in response to the demand, noted by the International Maritime Organization (IMO) and its Marine Environment Protection Committee (MEPC), for an International Standard on reduction of vibration and noise from machinery onboard ships, considering the negative effects of ship vibration and noise on the health of seafarers and the negative effects of underwater noise radiated from ships on marine life.

The reduction of ship vibration and noise may necessitate machinery vibration isolation measures. Although marine machinery vibration isolation design can be done by professional groups, such as consultant companies, with the development of modern shipbuilding technology, the integration of various technical fields for ship design is becoming a necessity.

The ship designers themselves should also understand the procedures, requirements and basic methodology of vibration isolation for shipboard machinery. The purpose of developing a basic methodology in this document is not limited to better vibration isolation measures, but to help ship designers take into account the need of machinery vibration isolation, planning for the space and weight for machinery vibration isolation in advance at the design stage and to consequently promote the application of vibration isolation design technology onboard.

Vibration isolation concerns the use of comparatively resilient elements called vibration isolators. Practical vibration isolators usually consist of springs, of elastomeric elements with damping or of a combination of these. The primary purpose of isolators is to attenuate the transmission of vibrations, whereas the main purpose of dampers is the dissipation of mechanical energy. Simple models based on systems with a single degree of freedom are useful for establishing some fundamental relations, such as single-stage isolation. Extensions of these models can account for the non-rigidity of supports and isolated items, as well as for reaction effects on vibration sources. More complex models apply to two-stage isolation that can provide greater attenuation than single-stage isolation systems. As common measures onboard a ship, single-stage and double-stage vibration isolation design methods are suggested in this document.

Ships and marine technology — Guidelines on vibration isolation design methods for shipboard auxiliary machinery

1 Scope

The purpose of this document is to provide general guidelines on the design of ship vibration isolation based on the basic methodology of vibration isolation for shipboard machinery, for example, auxiliary engine, compressor, fan, pump, etc. A well-designed vibration isolation system can significantly reduce the vibration transmission from shipboard machinery to ship structures lowering the noise level onboard the ship or the underwater noise radiated from the ship.

2 Normative reference

The following documents are referred to in the text in such a way that some or all of their content constitutes requirements of this document. For dated references, only the edition cited applies. For undated references, the latest edition of the referenced document (including any amendments) applies.

ISO 20283-3, *Mechanical vibration — Measurement of vibration on ships — Part 3: Pre-installation vibration measurement of shipboard equipment*

3 Terms and definitions

For the purposes of this document, the following terms and definitions apply.

ISO and IEC maintain terminological databases for use in standardization at the following addresses:

— IEC Electropedia: available at http://www.electropedia.org/

— ISO Online browsing platform: available at http://www.iso.org/obp

3.1
single-stage vibration isolation
vibration isolation where a single machinery or assembly is mounted on the ship foundation through a stage of elastic connection by a group of isolators

3.2
double-stage vibration isolation
vibration isolation where a single machinery or assembly is flexibly installed on an intermediate frame through a first stage isolator group, and an intermediate frame is mounted on the ship foundation through a second stage isolator group

3.3
displacement limiter
rigid element which is used in combination with the vibration isolator to avoid excessive displacement and consequent damages of the vibration isolation system

3.4
coast down test
test to determine the resonant frequencies of the machine isolation system for the machinery from a certain speed while it is shut down, simultaneously recording the vibration level until the machine stops

Note 1 to entry: Annex A gives information on the determination of the resonant frequencies of machine isolation system.

4 Basic requirements of vibration isolation system design

4.1 Vibration isolation performance

The design of vibration isolation should meet the requirements of vibration isolation performance and avoid resonance according to the structural dynamic characteristics of the ship structure including machinery foundation. The choice of vibration isolation type depends on the vibration levels difference between shipboard machinery and ship structure, as well as the limit of weight and space onboard the ship. The design requirements on the limit of the ship structure vibration response in ISO 20283-5 and relevant ship specifications can be referred to.

In general, the effect of vibration reduction can reach more than 15 dB when the single-stage vibration isolation is adopted and reach 25 dB to 30 dB when the double-stage vibration isolation is adopted. Especially for the double-stage vibration isolation, transmissibility of the isolation system varies with different intermediate mass values. The heavier the intermediate mass block is, the better the vibration reduction effect is. Design of the weight of the intermediate mass block depends on the weight restriction and vibration reduction requirement. Meanwhile, the advantage of introducing the intermediate mass block is more obvious in the high-frequency range.

4.2 Vibration severity of machinery

The vibration response of mechanical machinery will be enhanced after elastic installation. When the value of the machinery vibration severity is large, it is easier to cause fatigue damage of machinery and to reduce the reliability of machinery operation. Therefore, machinery that is installed resiliently should meet the requirements of the vibration limits of machinery according to the product specification or ISO 10816.

4.3 Stability of machinery

Machinery installed resiliently should be ensured to work normally under conditions of machinery rocking by ship motion, such as yawing, rolling and pitching. Limiters can be mounted in combination with the vibration isolators to prevent the maximum displacement of any flexible joints isolating the system from exceeding the permissible value.

4.4 Environmental adaptability

Vibration isolation elements shall meet the requirements of marine products and be approved by the related administrator after inspection.

4.5 Ease of installation and maintenance

The vibration isolation system of shipboard machinery shall meet the requirements of space and weight for maintenance and machinery reinstallation of isolation elements. The specifications of vibration isolation elements should be universal for the convenience of maintenance.

5 Design procedure of single-stage vibration isolation

5.1 General

The design of single-stage vibration isolation depends on vibration characteristics, gravity centre, weight, size and form of machine mounting feet, etc., of ship machinery according to the general requirements. Through optimizing the parameters and layout of isolators, the vibration isolation analyses are then implemented to reach the desired vibration isolation.

5.2 Spectral analysis of disturbance

Test and plot (in accordance with ISO 20283-3) spectrum of disturbance from machinery can describe the excitation characteristics and analyse the main excitation frequencies.

5.3 Natural frequency

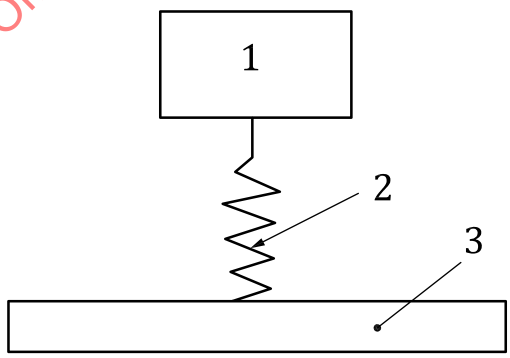
There are six degrees of freedom for the single-stage isolation system. The major exciting force can be projected to six directions of principal axes and there are corresponding major exciting frequencies. In the design of the vibration isolation system, the single-stage isolation system can be simplified as single-degree-of-freedom system for each direction of principal axes. For the considered directions, major exciting frequency, f, should be greater than $\sqrt{2}$ times of natural frequency of system, f_n, $f / f_n > \sqrt{2}$. The vibration transmissibility, T, from machinery to foundation of ship structure varies with the frequency ratio, f/f_n, and with the value of the loss factor, η, ($\eta = 2\xi(f/f_n)$, ξ is the damping ratio) as shown in Formula (1):

$$T = \sqrt{\frac{1+\eta^2}{[(f/f_n)^2 - 1]^2 + \eta^2}} \qquad (1)$$

For frequency ratios of more than $\sqrt{2}$, T value is less than 1. This reflects that better isolation (smaller transmissibility) is obtained at higher frequency ratios. Thus, in order to obtain good isolation in the presence of a disturbance at a given excitation frequency f, the natural frequency, f_n, shall be designed as much lower than f as possible.

At such low-frequency ratios, the items with isolation system are subjected to greater motions than they would experience without any isolation. For auxiliary machinery onboard, design of an isolation system is restricted by revolution times and the number of resonant resolutions is difficult to be designed under the revolution times. Therefore, low-speed auxiliary machinery isolation systems generally should be carefully designed.

The natural frequency is the frequency of free vibration in which a system vibrates to dissipate its energy. The natural frequency, f_n, expressed in radian per second, is a function of its stiffness, K, and its mass, M, as shown in Formula (2). See Figure 1.



Key

1 mass, M

2 stiffness of spring, constant, K

3 foundation

Figure 1 — Schematic diagram of single-stage vibration isolation

$$f_n = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \tag{2}$$

The calculated results from Formula (2) shall meet the requirements listed in Clause 4 and 5.3. Otherwise, isolators should be rechosen or redistributed and the calculation should be performed repeatedly.

5.4 Select type of vibration isolator

The selection procedure for the type of vibration isolator is as follows.

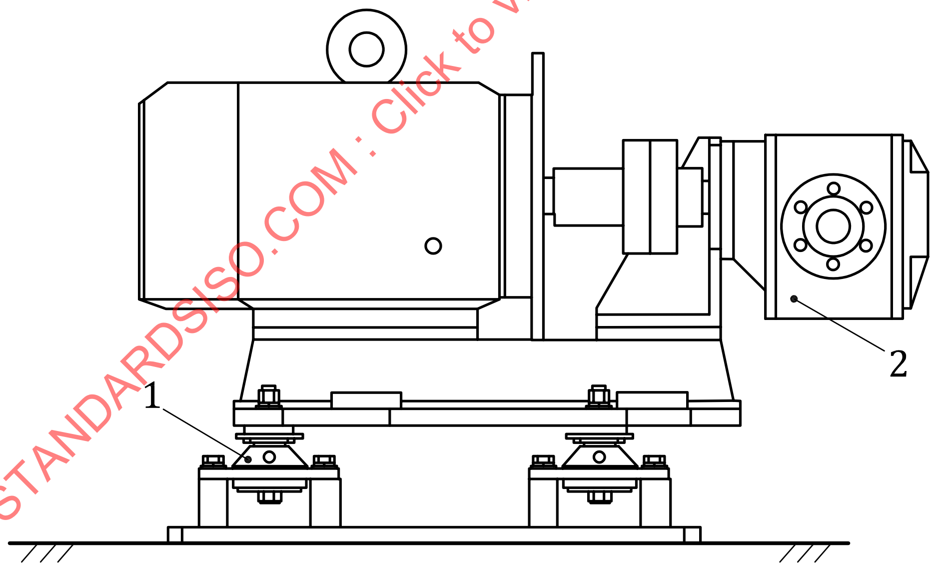
Firstly, determine the natural frequency based on the desired degree of isolation. Secondly, select isolators with the correct dynamic stiffness with which the isolation system will achieve the natural frequency and determine the static loads with the weight of the machine. Thirdly, check the amplitude at resonance so that the displacement of the isolator is below the allowable maximum.

5.5 Layout of vibration isolators

The layout of vibration isolators is designed with consideration of the directions of the disturbance force, machinery weight, gravity centre of mass, connection between the feet of the machine and hull structure and the location of machinery in the ship. Isolators should be arranged symmetrically about principal axes of inertia in order to reduce the vibration coupling of different directions.

The common arrangement of vibration isolators is that through isolators, machinery would be mounted horizontally or inclined on the foundation of the ship hull.

a) A horizontally installed type is the most common arrangement and has the feature of easy installation for simple structures (see Figure 2).



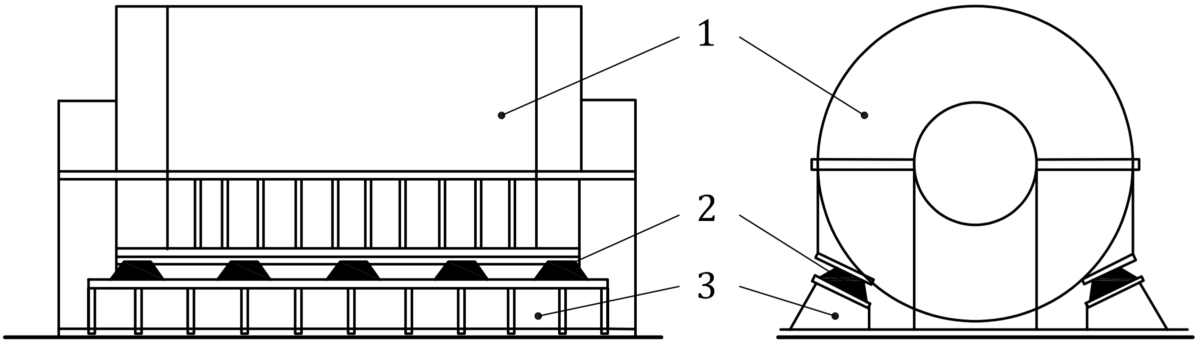
Key

1 isolator

2 oil pump

Figure 2 — Horizontal arrangement of isolators

b) An inclined installed type is adopted for the case of large machinery with a high centre of gravity (see Figure 3). The stiffness of three principal directions of the isolation system is determined by the projection of stiffness of the isolator reference axes in the principal directions.



Key

1 driving motor

2 isolator

3 foundation

Figure 3 — Inclined arrangement of isolators

5.6 Design of foundation

The design of the foundation should be carried out according to the mass, volume and layout of the machinery. The foundation of machinery shall be firmly fixed to the strong frame of ship structure. In addition to meeting the strength requirements, the stiffness of the foundation should be designed as high as possible. According to the practical engineering experience, the input mechanical impedance of the foundation should be at least 6 to 10 times higher than that of the vibration isolator.

5.7 Single-stage vibration isolation system calculation

Single-stage isolation systems shall meet the proposed requirements, such as coincidence of gravity and stiffness centre, relation of first order natural frequency to main disturbance frequency and stability of machinery over a resilient mount. The calculation process is listed below, which might need to be iterated to achieve the best degree of isolation:

— calculation of the natural frequencies of the isolation system;

— revised calculation on coincidence of gravity and stiffness centre;

— calculation on deviation of systemic natural frequency from main disturbance frequencies;

— calculation on stability of machinery over resilient mount.

6 Design procedure of double-stage vibration isolation

6.1 General

Double-stage vibration isolation can be commonly used for auxiliary equipment, etc., to improve the effect of vibration isolation effectively on the basis of the isolation requirements.

6.2 Vibration character analysis of machinery

Similarly for single-stage vibration isolation, vibration spectrum analysis of the machinery as an exciting source could be carried out additionally in order to determine major excitation frequencies, if necessary, at each considered directions along the principal axes, vibration spectrum of the machinery is ordinarily obtained by test.

6.3 Design of intermediate mass and foundation

a) Intermediate mass should have enough stiffness in order to avoid large deflection under static loads of the machinery. Structurally, the fundamental natural frequency of intermediate mass should be as high as possible, which is higher than excitation frequency of machinery. For design of intermediate mass, modal analysis and transfer function calculation should be carried out and optimal design is required for increasing the natural frequency and reducing the transmission ratio over the main excitation frequencies.

b) For achieving good vibro-isolation performance, high input impedance is necessary for design of foundation through increasing plate thickness or alterating the supporting structure. If necessary, a high damping material could be laid on the foundation to decrease resonant response.

c) Increasing the weight of intermediate mass would improve the performance of vibration isolation. The determination of weight depends on the weight restriction and vibration reduction requirement.

6.4 Selection of upper and lower isolators

For the two-stage isolation system, there are 12 degrees of freedom. For the linear orthogonal system, the two-stage isolation system can be simplified as a two-degree-of-freedom system for each direction of principal axes. Proper selection of stiffness of upper and lower isolators should make the natural frequencies of the system lie in a narrow frequency band and the natural frequencies should be away from the main exciting frequency. For each of the considered directions, natural frequency of system should be less than $1/\sqrt{2}$ of the main exciting frequency.

Good vibro-isolation performance occurs only at frequencies that are above the higher of the two natural frequencies of the system, so that it is desirable to make the natural frequency of the vibro-isolation system as low as possible because the higher natural frequency in practice often depends on the resonant frequency of the intermediate mass. And this frequency can be reduced by increasing the intermediate mass. In this case, displacement limiters should be considered to be adopted for preventing excessive displacement of machinery. Generally, reducing the stiffness of the lower isolators can decrease the high frequencies of the system and the vibration transmission and make the effect of the vibration isolation more effective.

6.5 Arrangement of isolators

The arrangement of isolators can be horizontal or inclined, which can refer to the layout of the single-stage isolation system.

6.6 Machinery items isolation

Sometimes, several machinery items need to be isolated, and it is often advantageous to support them on a common massive platform (sometimes called a “subbase” or “raft”), to isolate each item from the platform and to isolate the platform from the supporting structure. In such an arrangement, the platform acts as a relatively large intermediate mass for each of the machinery items isolation, resulting in more efficient double-stage isolation performance with a comparatively small total weight penalty. The platform should be designed so that it exhibits no resonances of its own in the frequency range of concern. If such resonances cannot be avoided, the platform structure should be highly damped.

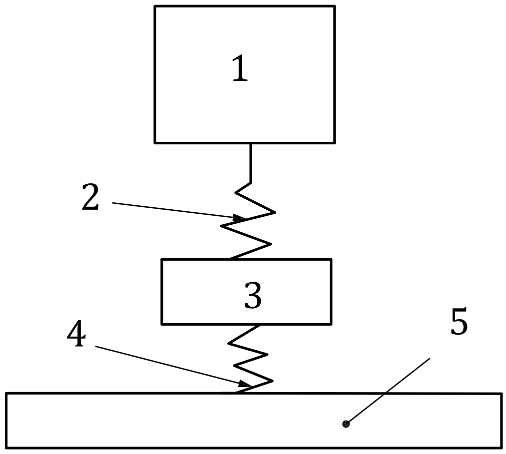
6.7 Calculation of double-stage vibration isolation system

The double-stage vibration isolation system calculations should include:

— natural frequency of system and transfer function calculation;

— modal analysis and structural strength calculation of intermediate mass;

— stability calculation of machinery over double-stage vibration isolation.



Key

1 mass, M_1
2 stiffness of spring, constant, K_1
3 mass, M_2
4 stiffness of spring, constant, K_2
5 foundation

Figure 4 — Schematic diagram of double-stage vibration isolation

Double-stage isolation systems, as shown schematically in Figure 4, can provide considerably greater high-frequency isolation. In a two-stage system, the mass, M_1, is connected to the supporting structure via two isolators (indicated as springs K_1, K_2 in Figure 4) and an intermediate mass, M_2.

The system in Figure 4 has two vertical natural frequencies, f_1 and f_2, given by the following:

$$f_1 = f_1' \left(C + \sqrt{C^2 - R^2} \right)^{\frac{1}{2}} \tag{3}$$

$$f_2 = f_1' \left(C - \sqrt{C^2 - R^2} \right)^{\frac{1}{2}} \tag{4}$$

where

$$2C = R^2 + 1 + K_1 / K_2 \tag{5}$$

$$R = f_2' / f_1' \tag{6}$$

where

$$f_1' = \frac{1}{2\pi \sqrt{M_1 \left(\frac{1}{K_1} + \frac{1}{K_2} \right)}} \tag{7}$$

and

$$f_2' = \frac{1}{2\pi} \sqrt{\frac{K_1 + K_2}{M_2}} \tag{8}$$

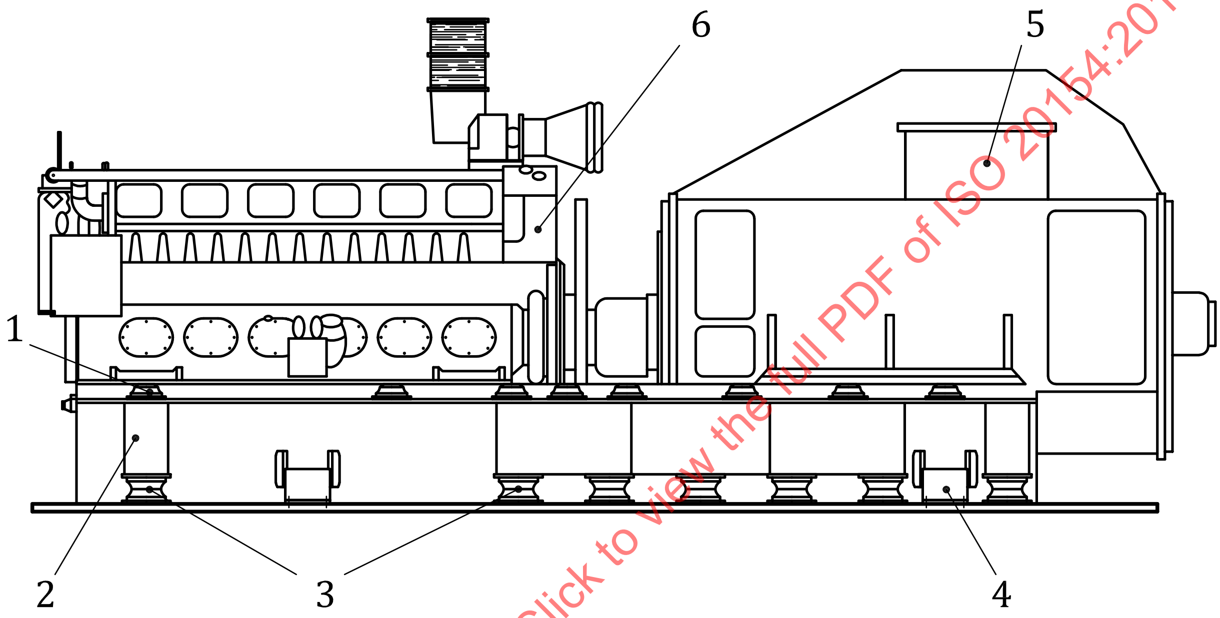
M_1, M_2, K_1 and K_2 represent the masses and stiffness as shown in Figure 4.

Formula (7) denotes the natural frequency of the system consisting of the mass, M_1, supported on the springs in mechanical series in the absence of the intermediate mass.

Formula (8) represents the natural frequency of the mass, M_2, between the two springs for the situation where mass M_1 is held completely immobile. The natural frequency f_1, as obtained with Formula (3), will always be greater than the larger of f_1' and f_2'. The natural frequency f_2, as obtained with Formula (4), will always be smaller than the smaller of f_1' and f_2'.

If any of the above results cannot meet the requirements in 6.4, the design procedure should be repeated.

Figure 5 is an example for a double-stage vibration isolation system.



Key

1 upper isolator
2 intermediate mass
3 lower isolator
4 limiter
5 generator
6 diesel

Figure 5 — Example for a double-stage vibration isolation system

