GUIDANCE NOTES
GD15-2021

ISClass

GUIDELINES FOR SHIPBOARD VIBRATION CONTROL

2021

Effective from 1 July 2021

INTERNATIONAL SHIP CLASSIFICATION

Content

Chapter 1 GENERAL

1.1 Introduction

1.1.1 Purpose

Different vibration levels will be caused as ships navigate at sea, where the natural frequency of vibration system is consistent or approach to exciting frequency, resonance phenomenon may be induced. Although some units are not in resonance state, severe vibration may be arisen due to the increase of excitation. In addition to the adverse effects on the personnel, excessive vibration may bring fatigue damage to the local structural members or fault to the machinery equipment, even to influence the ship's normal operation.

Classification societies carry out the research working constantly to control the ship's vibration, however due to its complexity, except the shafting vibration, the ship's vibration has not been included as a class condition in their rules. With the increase of ship's dimension and speed, adverse vibration occurs to endanger the ship's safety and personnel's health severely. Therefore, classification societies have developed the class notations of habitability (for vibration), comfort (for vibration), comfort (for noise), structural vibration, mechanical vibration and noise as the criteria for shipowners to select.

ISO has also rewritten and issued ISO 6954-2000 Mechanical Vibration – Guidelines for the Measurement, Reporting and Evaluation of Vibration with regard to Habitability on Passenger and Merchant Ships, etc.

To improve the accommodation and working conditions for crew onboard ships, ILO has made provisions to "prevent the risk of exposure to hazardous levels of noise and vibration" and "taking the due consideration for the relevant international standards, review on an ongoing basis the problem of noise onboard ships with the objective of improving the protection of seafarers, in so far as practicable, from the adverse effects of exposure to noise" respectively in Standard A3.1 – Accommodation and recreational facilities, Guideline B3.1.12 – Prevention of noise and vibration and Guideline B4.3 – Health and safety protection and accident prevention of Maritime Labour Convention, 2006.

Therefore, to effectively control shipboard vibration is the essential measures for ensuring the habitability, safety and functionality onboard ships.

In order to effectively control the effects of adverse shipboard vibration and make the ship to meet the relevant requirements of vibration criteria, the Guidelines for Shipboard Vibration Control (hereinafter referred to as the Guidelines) has been developed by ISC.

1.1.2 Application

(1) The Guidelines applies to all the vibration controls for all sea-going ships, high-speed craft, light-type ships and surface naval ships.

(2) Unless otherwise specified, ships applying for ISC class notations of vibration are to comply with the related requirements in the Guidelines.

1.1.3 References

ISC Guidelines for Shipboard Vibration Control, 2000.

The Guidelines is developed on the basis of ISC experience and research results, including the requirements of the updated related international standards.

The Guidelines includes the related contents of ISC Rules for Classification of Sea-going Steel Ships and the related standards, and attention is to be given to the updated edition of the related rules and standards in application.

The Guidelines is mainly referred to the following international standards and GBs:

— ISO 6954-1984 Mechanical Vibration and Shock – Guidelines for the General Assessment of Vibration on Merchant Ships;
— ISO 6954-2000 Mechanical Vibration – Guidelines for the Measurement, Reporting and Evaluation of Vibration with regard to Habitability on Passenger and Merchant Ships;
— ISO 20283-2-2008 Mechanical vibration – Measurement of Vibration on ships Part 2: Measurement of Structural Vibration;
— ISO 20283-3-2008 Mechanical Vibration – Measurement of Vibration on Ships Part 3: Measurement of Pre-installation Vibration for Shipboard Equipment;
— GB/T 7452-2007/ISO 6954:2000 Criteria for the Measurement, Reporting and Evaluation of Vibration with regard to Habitability on Passenger and Merchant Ships.

1.1.4 Adverse vibration

(1) During the ship’s operation, common vibrations are as follows:

— vibration of hull girders, superstructures and stern;
— vibration of panels, grillages, masts and structures in engine room;
— vibration of propulsion shafting;
— vibration of engine frame and machinery equipment.

(2) Adverse vibrations occur with the following phenomenon:

— fatigue damage of hull structure and mechanical parts;
— effects on the normal operation of machinery and equipment;
— effects on the normal working and life for the personnel onboard ships.

1.1.5 Vibration control

(1) In order to predict ship’s vibration at various stages, the Guidelines provides a set of flow to minimize the vibration, which is for the purpose of prediction or elimination of shipboard adverse vibration at design and construction stages through statistic data, empirical formula or estimation of computer program and assessment of vibration characteristics.

For the vibration onboard in-service ships, necessary calculation analysis and full-scale ship measurement may be carried out in accordance with the existing drawings and information, where the level of vibration exceeds the allowable values, vibration damping measures are to be taken as appropriate to improve the ship’s vibration characteristics.

(2) Means for vibration control are:

— to minimize the excitation force and its transmission;
— to carry out frequency modulation so as to change the natural or exciting frequency of vibration system and preventing the resonance;
— to increase damping so as to deplete vibration energy and achieve the purpose of amplitude decrease;
— to increase structural strength so as to raise the anti-vibration ability;

— to install dampers for main equipment.

(3) The elements of vibration control are:

Within the maximum running speed range of main engine (85% to 100%), to control:

— main exciting force not exceeding the allowable values;

— the ratio between main exciting frequency and main frequency of vibrating body meeting the requirements of design criteria.

1.2 Effective Means for Controlling Shipboard Vibration to Achieve the Anticipated Habitability

1.2.1 The habitability of shipboard vibration (or comfort of vibration) is the general assessment of vibration conditions, the vibration phenomenon likely occurred is to be controlled case by case, to achieve the expected habitability vibration criteria,. Hence, the flow of vibration control in the Guidelines, including the control methods provided in each Chapter are the effective means to achieve the anticipated habitability of vibration.

1.2.2 The habitability of shipboard noise (or comfort of vibration) is the general assessment of noise conditions, the noise phenomenon likely occurred is to be controlled case by case, to achieve the expected habitability vibration criteria. Ship's noise includes mechanical noise, hydrodynamic noise and electromagnetic noise. Generally, ship's noise occurs accompanied with vibration. Hull and local structural vibrations induced by excitation of main, auxiliary engines and propellers are transmitted from foundations, hull structures to compartment periphery structures in an elastic wave mode and could radiate the secondary airborne noise. Hence, the flow of vibration control in the Guidelines, including the control methods provided in each Chapter are one of the effective means to achieve the anticipated habitability of noise.

1.3 Meaning of Vibration Criteria

1.3.1 In addition to the definite requirements in ISC Rules for Classification of Sea-going Steel Ships, the related parts of the Guidelines provide the corresponding vibration criteria.

1.3.2 The ship's vibration criteria are the standards to evaluate the vibration in locations where the personnel normally work and accommodate onboard, other than those to accept or inspect the machinery equipment.

1.3.3 Vibration criteria in the Guidelines mean the following respectively:

(1) Structure vibration criteria: related to structural fatigue cracks;

(2) Mechanical vibration criteria: related to fatigue damage or acceleration weardown of moving parts;

(3) Habitability vibration criteria in accommodation and working spaces: related to acceptable vibration levels by crew and passengers;

(4) Comfort vibration criteria in accommodation and working spaces: related to comfortable environment levels of related spaces as specified by classification societies;

(5) Restriction of forces and moments: related to the induced excitation or response.

1.4 Relationship Between Shipboard Vibration and Excitation

1.4.1 Due to the fact that a ship itself is a complex elastic system, vibration response of a certain local structure or equipment induced by excitation may be a secondary excitation of the other local structure or equipment and coupling relationship may exist between different kinds of vibrations.

Therefore, the vibration relationship of each part onboard is also complicated, which should be noted by the designers at the very beginning.

1.4.2 The excitation sources of ship vibration include primary and secondary ones, which may cause the local or global vibration. For ships, both the vibrations caused by primary excitation source and the vibrations caused by secondary excitation source are required to be acceptable.

(1) The primary excitation mainly includes:

— excitation of unbalanced moment for main engine;
— excitation of poor shafting alignment;
— excitation of auxiliary engine;
— excitation of pump;
— excitation of motor;
— excitation of propeller (fluctuation pressure and bearing force);
— excitation of rudder;
— excitation of turbulence.

(2) The secondary excitation mainly includes:

— shafting longitudinal vibration;
— shafting torsional vibration;
— shafting whirling vibration;
— transverse vibration of diesel engine frame;
— longitudinal vibration of diesel engine frame.

(3) Wave is also a primary exciting force, the ship's vibration caused by wave may lead to adverse vibration, however, it may be minimized by the change of ship's speed and course. With the increase of ship's size, for ships of more than 300 m in length, the natural frequency of vertical vibration for hull girders will be decreased with the minimum one of 0.5 Hz, then wave may lead to hull vertical vibration - wave induced vibration. Therefore, the effects of wave induced vibration on hull strength are to be taken into consideration in ship's design.

(4) The exciting frequency of various excitation sources may see the relevant Chapters in the Guidelines.

1.4.3 In order to be beneficial to analyse, judge and prevent the adverse shipboard vibration, the basic concepts of relationship between main excitation sources and vibration bodies are to be established.

The relationship between excitation source and vibration for main diesel engine onboard ships is shown in Figure 1.4.3-1 and the relationship between excitation source and vibration for turbine engine or main electric propulsion engine onboard ships is shown in Figure 1.4.3-2.

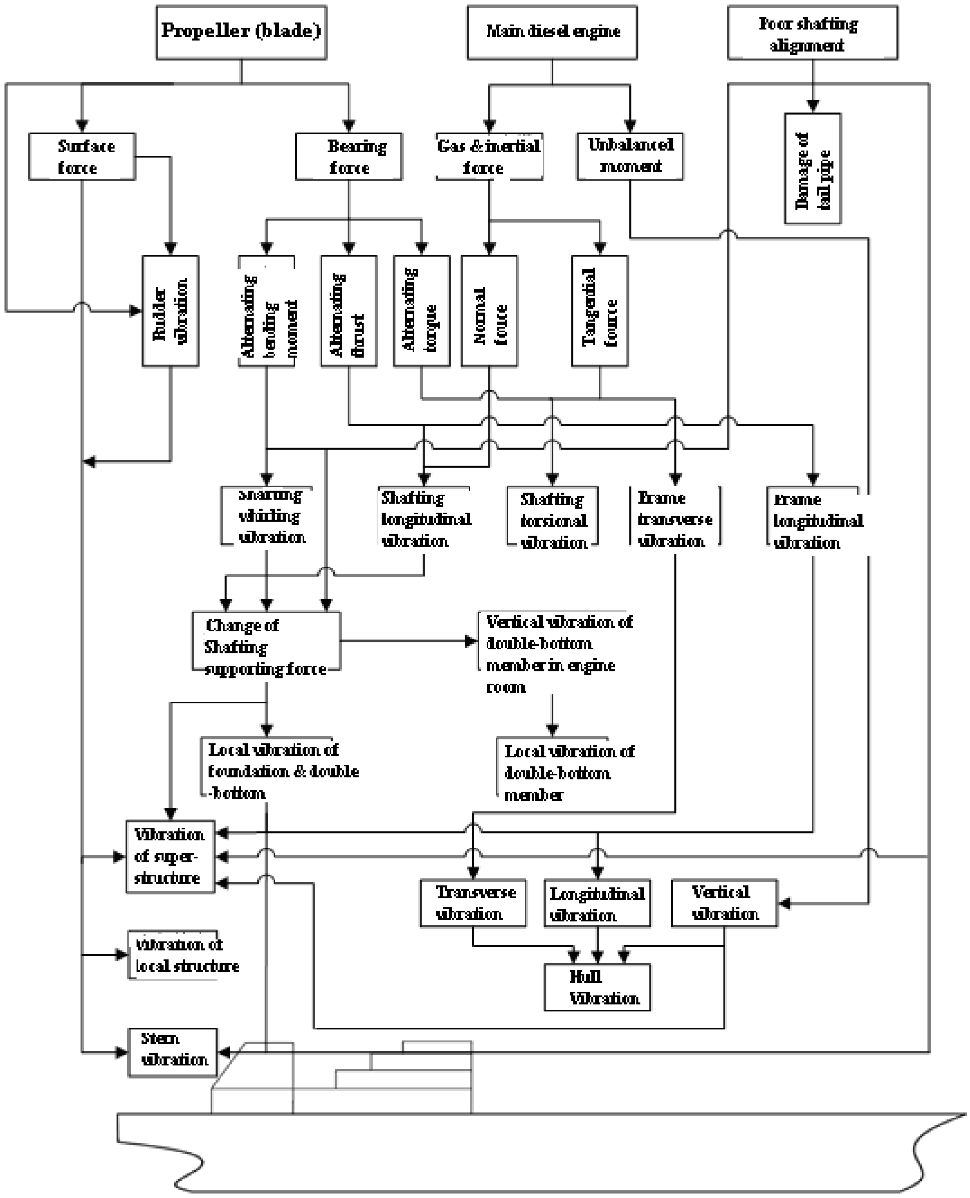


Figure 1.4.3-1 Relationship Between Excitation Source and Vibration for Main Diesel Engine

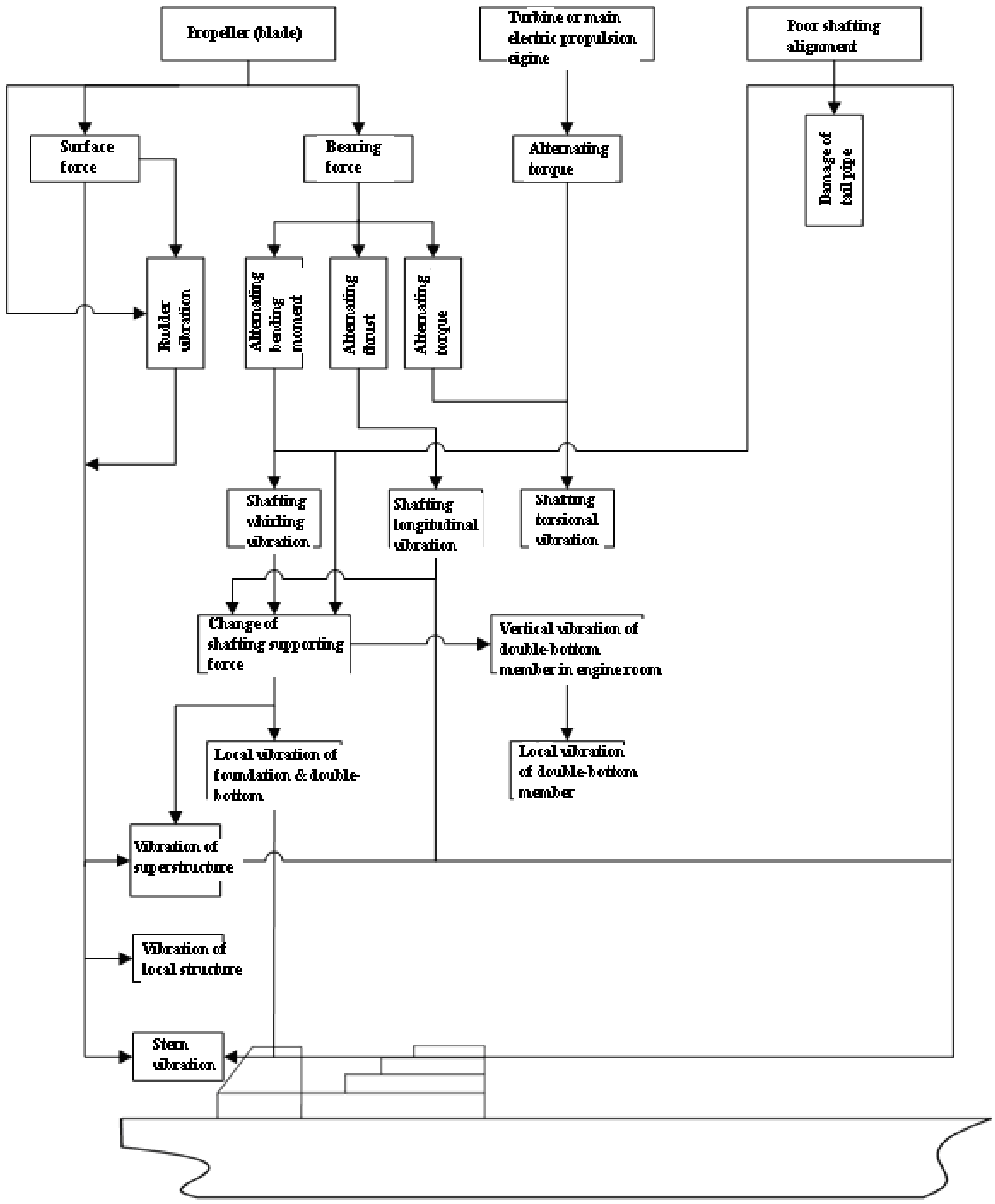


Figure 1.4.3-2 Relationship Between Excitation Source and Vibration for Main Turbine Engine or Electric Propulsion

1.5 Technical Routes to Research and Solve Shipboard Vibration

1.5.1 Where necessary vibration control is carried out in accordance with the requirements of the Guidelines, the adverse vibration may be prevented, and the anticipated vibration comfort of the ship may be achieved. However the shipboard vibration is to be researched and solved if adverse vibration is found by measurement or the anticipated vibration comfort is not achieved due to unexpected reasons.

1.5.2 The basic method to research and solve the shipboard vibration is first to anslyse the primary and secondary excitation sources, then to study the transmission path of sources, and then to estimate the possibility of resonance together with the vibration body. For example, if resonance or forced vibration occurs, the criteria level is to be decided whether or not being exceeded, and if it exceeds the criteria, the reasons are to be investigated and the economical and effective solutions are to be provided, as shown in Figure 1.5.2.

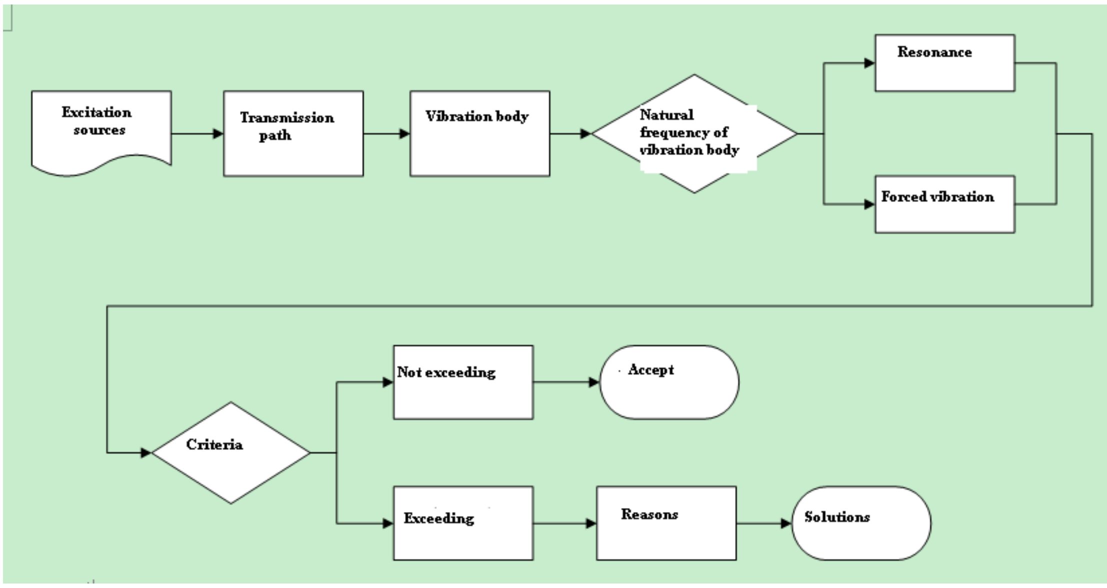


Figure 1.5.2 Technical Routes to Solve Shipboard Vibration

1.6 Basis of Vibration

1.6.1 Introduction

The actual vibration system is so complicated that full vibration analysis in accordance with its own conditions are not possible or necessary. For this purpose, a simplified mechanical model which represents an actual system is first to be established. Such model is to have the basic vibration characteristics of the original system and not so complicated. The simplification depends on the complexity, required analysis precision and calculation method of the original system.

Generally, the simplified model is divided into two categories, such as discrete system model and distribution system model, the discrete system model also includes lumped parameter model and finite element model.

The lumped parameter model consists of three basic elements, i.e. inertial element (concentrated mass), damping element (damper without mass) and elastic element (spring without mass) which are indicated as M, C and K respectively.

The distribution system model consists of distribution parameter element, which is normally a continuous elastomer with the mass and stiffness distributed homogeneously or regularly.

Obviously, for the same real system, various models according to the different needs may be simplified. In most circumstance, it may be multiple degree of freedom system which is similar to the discrete system, even may be simplified to single degree of freedom system.

In the vibration analysis, vibration of single degree of freedom system has the basic characteristics, by which may establish many basic concepts of vibration.

There are infinite natural frequencies for the distribution system theoretically, however only several lower-node/ lower-order natural frequencies and corresponding mode shape have practical significances.

1.6.2 Free vibration of single degree of freedom system

The single degree of freedom system is shown in Figure 1.6.2.1-1.

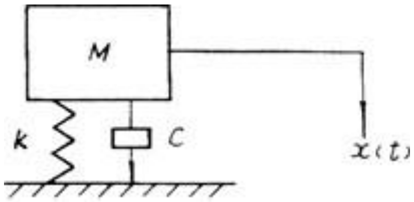


Figure 1.6.2-1 Single Degree of Freedom System

The equation of motion for single degree of freedom system is as follows:

$$M\ \ddot{x}+C\ \dot{x}+K\ x=0 \qquad (1.6.1)$$

where: M – mass, in kg;

C – coefficient of viscous damping, in Ns/m;

K – rigidity, in N/m,

x、$\dot{x}$、$\ddot{x}$ – displacement, in m, velocity, in m/s and acceleration, in m/s^2 of mass M distanced from static balance location.

Where the damping of system is ignored, the vibration displacement $x\ (t)$ with the time change may be shown as:

$$x(t)=A\sin(\omega_n t+\varphi_0) \qquad (1.6.2)$$

where: A – amplitude of displacement, to be determined by the initial condition:

ω_n – natural circular frequency of system without damping:

$$\omega_n=\sqrt{\frac{K}{M}} \qquad (1.6.3)$$

φ_0 – initial phase angle of free vibration without damping, $\varphi_0=tg^{-1}\dfrac{\omega_n x(0)}{\dot{x}(0)}$.

where: $x(0)$ — initial displacement as $t=0$;

$\dot{x}(0)$ — initial velocity as $t=0$.

This is a harmonic vibration with the circular frequency ω_n, depending on the mass and stiffness of the system, not related to the initial conditions. Where the damping is considered, the formula (1.6.1) is to be changed to:

$$\ddot{x}+2\xi\omega_n\dot{x}+\omega_n^2 x=0 \qquad (1.6.4)$$

where: ξ – damping ratio, calculated as:

$$\xi=\frac{C}{2M\omega_n}=\frac{C}{2\sqrt{MK}} \qquad (1.6.5)$$

In the most engineering issues, the damping ratio ξ is far less than 1, then the change of vibration displacement with the time may be shown as:

$$x(t)=A_1e^{-\xi\omega_n t}\sin(\omega_d t+\varphi_1) \qquad (1.6.6)$$

where: A_1 — amplitude, calculated as:

$$A_1 = \sqrt{[x(0)]^2 + \frac{1}{\omega_d^2}[\dot{x}(0) + \xi\omega_n x(0)]^2}$$

ω_d — for system with natural circular frequency of damping:

$$\omega_d = \omega_n\sqrt{1-\xi^2} \qquad (1.6.7)$$

φ_1 — for system with initial phase angle of free vibration of damping, to be calculated by:

$$\varphi_1 = tg^{-1}\left(\frac{\omega_d x(0)}{\dot{x}(0) + \xi\omega_n x(0)}\right)$$

Others are same as formula (1.6.2).

This is a vibration which the amplitude value decays with the time change based on the exponential rules, the time history is shown in Figure 1.6.2-2, circular frequency of vibration ω_d is less than ω_n . However, in practice, the minor difference may not be considered.

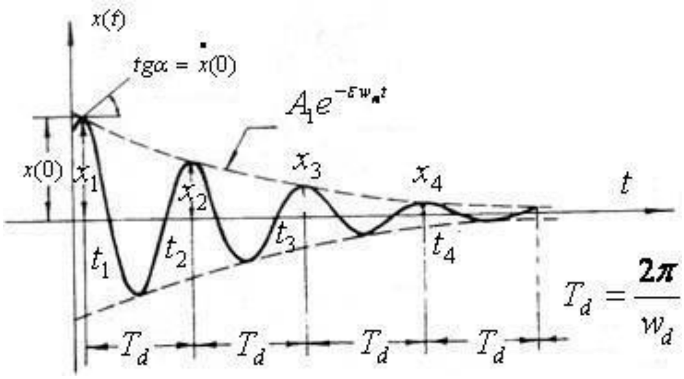


Figure 1.6.2-2 Free Vibration with Damping

1.6.3 Vibration response under harmonic excitation for single degree of freedom system

The single degree of freedom system with damping under the harmonic excitation is shown in Figure 1.7.3-1.

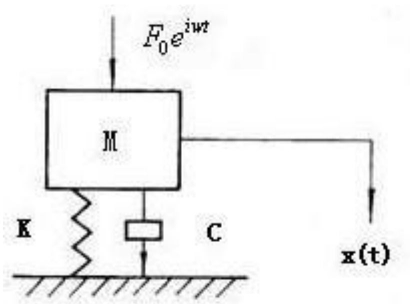


Figure 1.6.3-1 Single Degree of Freedom System with Damping under Harmonic Excitation

The equation of motion for single degree of freedom system under harmonic excitation is as follows:

$$M\ddot{x} + C\dot{x} + Kx = F_0 e^{i\omega t} \qquad (1.6.8)$$

where: F_0 — amplitude value of excitation;

ω — circular frequency of excitation;

Others are same as formula (1.6.1).

The steady-state vibration response is as:

$$x(t) = A_2 e^{i(\omega t-\varphi_2)} \tag{1.6.9}$$

where: A_2 — amplitude:

$$A_2 = \frac{A_{st}}{\sqrt{(1-\frac{\omega^2}{\omega_n^2})^2+(2\xi\frac{\omega}{\omega_n})^2}} \tag{1.6.10}$$

where: $A_{st} = \frac{F_0}{K}$ — static displacement;

φ_2 — initial phase angle of forced vibration, to be calculated as:

$$\varphi_2 = tg^{-1}\left(\frac{2\xi\ (\omega/\omega_n)}{1-(\omega/\omega_n)^2}\right) \tag{1.6.11}$$

The change of φ_2 with frequency ratio is indicated by the curve of phase frequency characteristic, as shown in Figure 1.6.3-2.

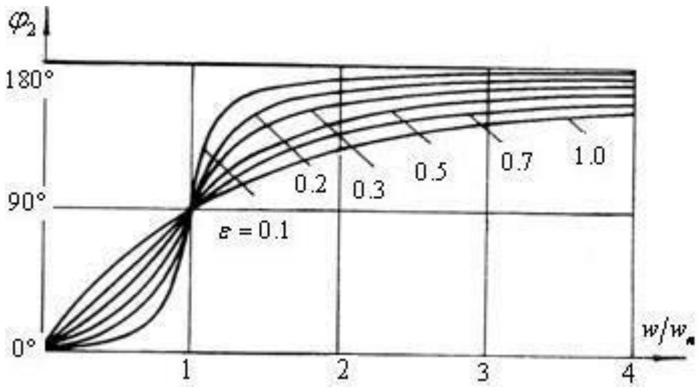


Figure 1.6.3-2 Curve of Phase Frequency Characteristic

The vibration response of system is to be shown by dynamic amplification factor T_m :

$$T_m = \frac{A_2}{A_{st}} = \frac{1}{\sqrt{(1-\frac{\omega^2}{{\omega_n}^2})^2+(2\xi\frac{\omega}{\omega_n})^2}} \tag{1.6.12}$$

The change for T_m with ratio of ω/ω_n is indicated by the curve of amplitude frequency characteristic, as shown in Figure 1.6.3-3.

Where ω/ω_n=1, the formula (1.7.12) may be as follows:

$$T_m = \frac{1}{2\xi} = Q \tag{1.6.13}$$

where: Q — quality factor to describe the damping characteristic of system.

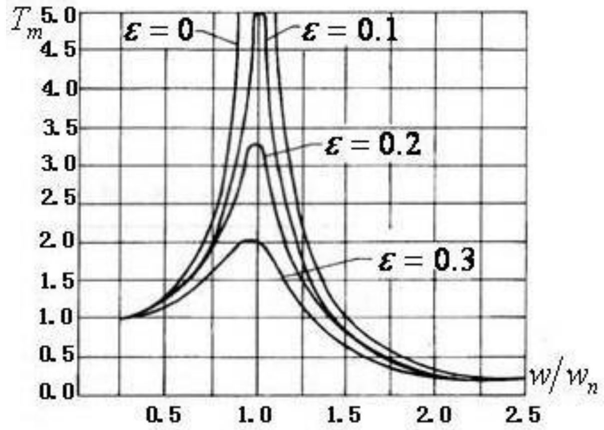


Figure 1.6.3-3 Curves of Amplitude Frequency Characteristic

1.7 Definitions and Vibration Parameters

1.7.1 For the purpose of the Guidelines, the following main terms are defined:

(1) Displacement: means vector for distance change of targeted objects or particles relative to a certain referred position, in mm.

(2) Velocity: means vector of targeted displacement relative to time coefficient, in m/s.

(3) Acceleration: means vector of targeted velocity relative to time coefficient, in m/s^2.

(4) System: means combination of the related parts to complete certain functions.

(5) Vibration: means a phenomenon that the magnitude to describe the motion or position of mechanical system alternatively changes with the time relative to a certain average value either large or small.

(6) Periodic vibration: means a period of the argument that reoccurs with the increase in a certain same time.

(7) Vibration system: means any system either with or without damping and including mass and rigidity.

(8) Steady-state vibration: means the continuous periodic vibration.

(9) Harmonic vibration: means vibration changes with the time *t* in accordance with harmonic function rule.

(10) Excitation: means external force or other inputs on the system.

(11) Harmonic order or number of vibration: means number of harmonic waves for each speed of shafting or the value of vibration frequency divided by the speed of shafting.

(12) Blade order: the harmonic order of exciting force caused by blades of propeller in uneven wake field is the integral multiple of number of blades may be called blade order, twice blade order, etc.

(13) Blade order frequency: the frequency of exciting force caused by blades of propeller in uneven wake field is the blade order multiplying with revolution frequency of propeller.

(14) Response: means the output of system which is effected by excitation.

(15) Free vibration: means vibration occurs after the excitation or restraining is removed.

(16) Forced vibration: means steady-state vibration caused by steady-state excitation.

(17) Frequency: means number of vibration at time unit, in Hz for number of vibration at each second, in *l*/min for number of vibration at each minute.

(18) Natural frequency: means vibration frequency determined by the mass and stiffness of system.

In general, n-degree of freedom system has n natural frequency and is arranged in the order from small to large, which is called 1st node/order natural frequency, 2nd node/order (node) natural frequency, n^{th} node/order (node) natural frequency, etc.

(19) Vibration mode: means a targeted vibration characteristic, which is determined by characteristic value and the corresponding characteristic vector of the system.

(20) Amplitude of vibration: means the maximum value of harmonic vibration.

(21) Relative amplitude: means the ratio between amplitude at a certain point and that at a referred point.

(22) Mode shape: means a certain vibration mode, which is a graph to show the relative amplitude of each point.

(23) Node: means the position which the relative amplitude is zero for a certain mode shape of the system.

(24) Wave loop: means the position which the relative amplitude is the maximum for a certain mode shape of the system.

(25) Resonance: means vibration state which any minor change of exciting frequency will decrease the response of system during the forced vibration.

(26) Resonance frequency: means frequency as the resonance occurs.

(27) Resonance speed: means the speed at which the resonance is excited, is also called critical speed.

(28) Circular frequency: means number of vibration at each 2π second, in rad/s or *l*/s, is also called circular frequency.

(29) Phase: means advanced cycle numerical of periodic function when measurement is taken as a certain value of argument is used as a basic one, in degree, is also called phase angle.

(30) Phase difference: means difference of phase angles with same frequency between two periodic functions, is called as phase angle difference.

1.7.2 Vibration parameters

(1) Main vibration parameter

In the Guidelines, ship's habitability (vibration) applies RMS velocity as the main parameter to evaluate the shipboard vibration.

Local structure and mechanical vibration apply displacement amplitude of vibration, amplitude of velocity or amplitude of acceleration as the main parameters to evaluate the shipboard vibration respectively. The displacement is applicable to the lower frequency, velocity is applicable to the higher frequency, and acceleration is applicable to even higher frequency.

The stress, torque, amplitude or frequency of vibration are applied as the main parameters to evaluate the shafting vibration.

(2) Relationship between vibration parameters

Displacement: $x = A\sin\omega_n t$

Velocity: $v = \omega_n A\sin\left(\omega_n t + \frac{\pi}{2}\right)$

Acceleration: $a = \omega_n^{\ 2} A\sin(\omega_n t + \pi)$

where: A— amplitude of vibration, in mm;

ω_n — vibration frequency, in rad/s;

t— time, in s.

(3) Relationship among amplitude of vibration and RMS, etc.

For the sinusoidal curve, the relationship among amplitude of vibration (peak value), peak to peak value and RMS parameters is shown in Figure 1.7.2.

Peak to peak value = $2A$;

Peak value or amplitude $= \frac{1}{2}$ (peak to peak value)= $\pm A$;

RMS = $\frac{1}{\sqrt{2}}A$.

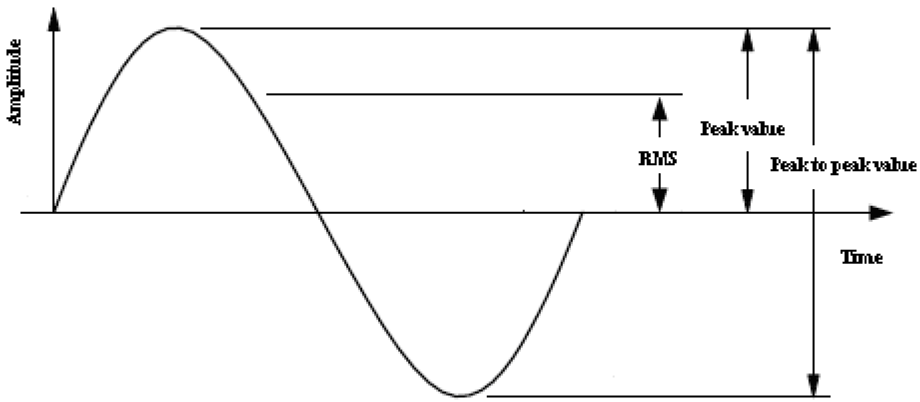


Figure 1.7.2 Relationship among Amplitude of Vibration and RMS, etc.

Chapter 2 FLOW OF VIBRATION CONTROL

2.1 Introduction

The Flow of Vibration Control is specially made for providing guidance for ship's designers and shipyards to effectively control the possible adverse vibrations (ship hull vibration, superstructure vibration, local vibration, mechanical equipment vibration and shafting vibration) for newbuildings.

Due to the fact that vibration control needs relevant ship's information, the vibration control is a progressive process. From ship's design to being put into service, the vibration control may be divided into the following four stages:

(1) initial design;

(2) detailed design;

(3) construction stage;

(4) stage after construction.

The corresponding vibration calculation is to be made according to the completed drawings and known data by reference of the partial or whole process of the flow at each stage on the basis of ship's conditions through estimation method or detail calculation method respectively by the designers to predict ship's vibration characteristics and then perform the evaluation in terms of the relevant criteria to provide the existing issues and necessary measures to be taken, as well as the further research direction.

In general, unless fairly consistent vibration information of parent ships, including vibration research report, detailed vibration calculation or vibration testing report of full-scale ships are available, the vibration characteristics of new design ships are nomally mastered deeply with the design processes gradually.

2.2 Flow of Vibration Evaluation

2.2.1 Flow of vibration evaluation

The flow of vibration evaluation provides the reference for vibration control analysis at each stage as shown in Figure 2.2.1.

Ship's vibration evaluation is actually a calculation and analysis of various exciting forces which may induce the structural vibration, and comparing with the natural frequency of vibration body, the occurrence possibility of resonance may be eliminated gradually.

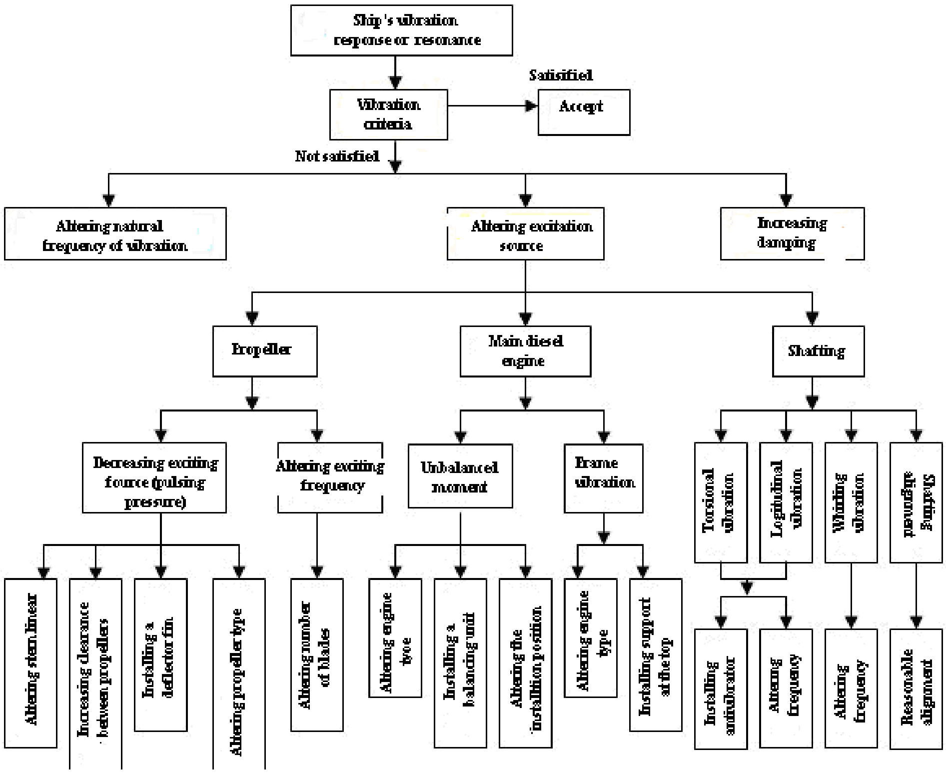


Figure 2.2.1 Flow of Vibration Evaluation

2.2.2 Contents of vibration evaluation

Currently, the accuracy of calculation of ship's vibration response has not yet been satisfied, thus particular attentions are to be given to the vibration evaluation so as to prevent from strongly vibrating as follows:

(1) within the range of maximum service speed of main engine (85% to 100%), analyzing and determining the main exciting force and its frequency, the exciting force is not to exceed the specified value;

(2) within the range of maximum service speed of main engine (85% to 100%), analyzing and determining the main vibration frequency of vibration body, and the ratio between vibration frequency of vibration body and main exciting frequency is to meet the requirements of design criteria.

The exciting frequency is determined, while the exciting force may generally be controlled within a certain range. In the case of greater exciting force, the natural frequency of related vibration body is to be estimated as accurate as possible, and the exciting frequency and natural frequency of related vibration body are to meet the requirements of design criteria. Effect of the secondary exciting force on the related vibration body is also to be noted.

2.3 Antivibration Design

2.3.1 Introduction

To prevent adverse vibration, necessary structural vibration calculation in the design is required and the precautions are to be taken. However, many factors affecting the ship's vibration and exciting forces may involve the ship's general performances, hull structure, power plant, etc. The antivibration measures taken in the ship's design and vibration damping measures taken for ships in-service are the object difference and the difference for the view of processing, the basic principles with the most of methods are consistent. In general, the antivibration measures adopted in the ship's design and also applies to those for ships in-service. Nevertheless, some of the precautions in the design are difficult to implement for the constructed ships.

The precautions to minimize exciting forces or avoid resonance are given in the relevant chapters of the Guidelines respectively. This paragraph summarizes the related precautions to reduce the adverse shipboard vibrations.

2.3.2 Prevention of resonance

(1) Altering natural frequency of structures

In the ship's detailed design or being put into service, the alteration of natural frequency for its hull vibration is difficult, but for various local structures onboard, i.e. girders, plating, panels, grillages, pipes, etc., vibration damping to alter the natural frequency of structures and keep it far from resonance zones by altering the stiffness or mass of structures when the natural frequency is equivalent or similar to the exciting frequency are an effective measure. For example, supporting saddle may be added in the span of girder, stiffener may be added for plating along with the direction of its length, column, bulkhead or web frame may be provided for grillage, and the plate may be increased by mass.

(2) Altering exciting frequency

To alter exciting frequency is one of the effective means to prevent from structural vibration. However, the feasibility and economical analysis for the alteration of exciting frequency of excitation sources which may cause structural resonance are first to be made, and where possible, to be treated according to the methods as specified in Chapter 3 to Chapter 10.

(3) Altering function positions of excitation sources

When the i^{th} order resonance of hull girders occurs, the unbalanced force induced by diesel engine is removed to node or nodel line of the i^{th} order main mode shape while the unbalanced moment is removed to loop point of the i^{th} order main mode shape, thus, the resonance of this node is avoided.

2.3.3 Minimizing amplitude of excitation

Where hull or its local structural vibration response exceeds the assessment criteria, particularly for the high frequency vibration response, reduction of amplitude of excitation is the most effective means. The specific measures are to be taken according to the requirements of Chapter 3 to Chapter 10 based on the analysis of excitation source inducing the hull vibration, i.e. to choose the main diesel engine with balanced inertial force and moment of inertia, to alter the stern linear, change wake distribution, and increase clearance size between hull and propeller, etc.

2.3.4 Minimizing transmission of excitation

To minimize the transmission of excitation and consume the energy of excitation are also the effective means to reduce vibration response, i.e. vibration isolator is installed between medium, high-speed diesel engines and its pedestal to decrease the exciting force on the hull which is transmitted from diesel engine. In general, a single-order vibration isolator is to be used. Floating raft technology or double-layer vibration isolation may be applied as required higher, see Chapter 10 for details.

2.3.5 Reasonable matching of ship, engine, shafting and propeller

(1) The main diesel engine, propeller, shafting are excitation sources of shipboard vibration. Reasonable matching of them and matching with hull are the key factor to minimize the shipboard vibration.

(2) Particular attention is to be given to shafting alignment, hull deformation factor is to be considered in the design to minimize shaft frequency excitation induced by the shafting. Meanwhile, excessive shafting torsional, longitudinal and whirling vibration are to be prevented to minimize the secondary excitation to the hull.

(3) For the large low-speed diesel engine with long stroke or super long stroke, supporting is to be installed along the two directions of the frame. The natural frequency of frame is to be altered to prevent the longitudinal and transverse resonance and then to minimize the secondary excitation of frame vibration to the hull.

2.3.6 Rational design of hull structure

Rational design of hull structure and raising of structural stiffness are also effective means to minimize structural response. The following particular attentions are to be given during structural design:

(1) To ensure the continuity of longitudinal structural members, effectively connect the deck, side and bottom structure to constitute a whole with greater rigidity.

(2) To avoid the design of grillage with large area and girder with big span, special measures or unconventional structure type is to be taken where a design of deck plate without column is necessary.

(3) To increase torsional stiffness for ships with large openings, such as container ships, i.e. installation of torsional box, deck strip to prevent from the adverse torsional vibration.

(4) To design rationally the stiffeners of side, deck, platform, etc., in engine room, the connection between main engine pedestal and keel (or bottom girder) is to have a uniform transition, and longitudinal folding line is to be avoided within the engine room. The engine pedestals onboard small ships is to be extended to the fore and aft bulkheads of engine room as far as possible.

(5) For aft peak tank and aft suspension structure, attention is to be given to reduction of the spans of floor and stiffener, as well as the dimensions of plating. In order to increase the stiffness of aft peak tank and aft suspension structure, column, truss or bulkhead is to be installed to connect the deck plate with hull grillage. For aft suspension structure, longitudinal stiffness is also to be increased by installing of girder grillage, longitudinal bulkhead or reinforced frame.

(6) For ships with engine installed in the aft, within the whole length of engine room, the vertical and horizontal moment of inertia is reduced sharply, and the shear stiffness is also reduced with the thinning of shell plating and change of hull geometrical shape, attention is to be given to compensation of the lost stiffness in the structure design. The method to compensate vertical moment of inertia is to install the poop with large quantity of longitudinal plates from one side to the other; that to compensate horizontal moment of inertia is to install the platform and that to compensate shear stiffness is to install the side tanks along the whole length of engine room.

(7) Stress concentration is to be avoided or mitigated, particularly in the engine room and at stern.

Due to being close to the excitation sources, fatigue damage of vibration structures is likely to occur in the stress concentration areas of structural members such as continuous grooves, section ends, hard welding points, etc.

(8) To increase the shear stiffness and support stiffness of superstructure may raise its natural frequency of longitudinal vibration effectively, with the following means:

① the side walls of superstructure are arranged in a line;

② internal longitudinal walls are to be installed for the superstructure with more than four layers;

③ stiffened girders or other stiffened structural members are to be installed below the main deck where the superstructure is located;

④ the transverse bulkheads of main hull and the fore end bulkheads of superstructure are to be arranged in a line, if not, bracket is to be used to connect;

⑤ the aft end bulkheads of each layer of superstructure are to be arranged in a line as far as possible, and also arranged in line with the hull transverse bulkheads, if not, stiffened transverse girders are to be installed beneath the main deck.

2.3.7 Installing damping material and vibration damping device

Vibration damping coating is to cover (including by injecting, painting or bonding methods) the surfaces of shell plating, deck, bulkhead or other structures subjected to strong vibration to constitute the artificial damping which may consume vibration energy. Two structural types are free damping layer and constrained damping layer. The damping coating has obvious effect on minimizing the vibration transmission, reducing the vibration response, mitigating structural noise within a wider frequency domain.

Two kinds of vibration damping devices are active vibration damping device and passive damping device, which are widely used for various mechanical engineering, and for ships, mainly applied for local structures and some mechanical equipment, also applicable to reducing amplitude values within the range of a certain natural frequency. The vibration absorber is also used to minimize the ship hull vibration.

2.3.8 Attention is to be given to the antivibration design for non-calculation items, such as fixing and supporting, i.e. handrails, railings, pipes, windows and doors, funnels, exhaust pipes, etc.

2.4 Initial Design

2.4.1 Introduction

In the initial design, designers are to gather the related particulars of parent ships and the technical information of main equipment in accordance with the ship's navigating areas, performance indexes, requirements of use and main equipment and propulsion system may be applied onboard which are provided in the product design task.

When the ship's main dimensions are determined, type of main engine and number of propeller blade are selected, the natural frequency of vertical vibration of hull girders and natural frequency of superstructural vibration are to be estimated in accordance with ship's main dimensions or ship type and displacement to prevent vertical resonance of hull girders caused by unbalanced moment of two-stroke cycle diesel engine and superstructure resonance caused by excitation of propeller blade order frequency. Meanwhile, diesel engine with less unbalanced moment is to be selected as the main engine installed onboard as practicable as possible.

2.4.2 Research contents of vibration control design

(1) According to the formula given in Chapter 11 of the Guidelines, the natural frequency of 1st order to 3rd order vertical vibration of hull girders is to be estimated and to avoid resonance occurring with the 1st order, 2nd order and 4th order unbalanced moment which are greater within the normal speed range of main engine. Where severe vibration is predicted to occur, the program is to be adjusted timely and issues to be resolved at this stage as far as possible. Where the adverse state can not be improved due to the limitation of objective condition, the existing issue is to be pointed out for further research in the detailed design. Meanwhile, particular attention is to be given to the effect of fluctuation pressure of propeller blade order.

(2) According to the formula given in Chapter 12 of the Guidelines, the natural frequency of longitudinal vibration of superstructure is to be estimated and to avoid resonance with the frequency of propeller blade order within the range of normal speed of main engine.

(3) The related information of unbalanced moment for the latest two-stroke cycle diesel engine is to be required to provide by the manufacturers, and according to the formula given in Chapter 4 of the Guidelines, the PRU of unbalanced moment of diesel engine unit power is to be estimated, then based on PRU, the effect on ship hull vibration is to be evaluated and whether to take antivibration measures to be considered. Where balance compensation device is used, the type is to be determined preliminarily. Where installation on the main engine is required, a remark is to be indicated when ordering.

(4) The information of moment values for Type *H* and Type *X* exciting moment for two-stroke cycle diesel engines and the natural frequencies of Type *H*, Type *X* and Type *L* vibrations for the corresponding frames are required to be provided by the manufacturers, and an estimation is to be made in accordance with the requirements of Chapter 5 of the Guidelines.

In order to prevent or minimize the frame transverse vibration which will further induces the hull vibration, it is recommended to install a transverse supporting on the top of main diesel engine.

In order to prevent Type *L* vibration of frame caused by propellers which will further induces the hull vibration, the longitudinal supporting may be installed in the front end of the top of engine frame.

2.5 Detailed Design

2.5.1 Introduction

In the detail design, the basic drawings, including general arrangement, structural type, dimension of structural members and primary structural drawings, main mechanical equipment and arrangement of engine room, shafting arrangement, propeller drawings, etc., have been completed. Based on these drawings, more detailed calculation and research for each excitation which may cause ship’s vibration, ship hull vibration, typical local vibration, shafting torsional vibration, longitudinal vibration and whirling vibration, mechanical equipment vibration may be carried out. If the requirements are not satisfied with, improvement measures are to be taken.

2.5.2 Research contents of vibration control design

(1) Estimation of propeller fluctuation pressure

Based on the designed linear, the propeller fluctuation pressure is to be estimated according to the requirements of Chapter 3 of the Guidelines, and comparing with the criteria, the inhomogeneity of wake filed at stern is to be improved to an acceptable scope as far as possible. Particular attention is to be given that the number of propeller blades is to match with the stern shape, in general, Type *V* can not be used for the stern of single screw ship, but Type *U* is applicable. Meanwhile, the parameters of propeller are to be considered to amend, the clearance between blade tip and shell plating is to meet the recommended values provided in Chapter 3 of the Guidelines. If the clearance is smaller, the diameter of propeller is to be reduced or antivibration cave is to be provided.

(2) Unbalanced moment of diesel engine

For selected two-stroke cycle diesel engine, the PRU of unit power unbalanced moment is to be calculated and confirmed to be in the allowable range. Where the requirements are not complied with, a balance device is to be considered or a vibration isolation device is to be used.

(3) Calculation of diesel engine frame vibration

For the selected two-stroke cycle diesel engine, the transverse and longitudinal vibrations are to be calculated in accordance with the requirements of Chapter 5 of the Guidelines. To avoid and minimize the secondary excitation to hull caused by frame vibration, consideration may be given to longitudinal and/or transverse supporting to be provided on the top of the frame where necessary, and the hull structure in way of supporting is to have sufficient stiffness. Meanwhile, the structural configuration of frame intended for use is to be evaluated.

(4) Shafting vibration and alignment calculation

According to the requirements of Chapter 6, Chapter 7 and Chapter 8 of the Guidelines, the natural frequencies and responses of shafting torsional vibration and longitudinal vibration and the natural frequency of whirling vibration are to be calculated with the results meeting the rules requirements, otherwise, design is to be revised or vibration damping measures are to be taken. The secondary excitation to hull caused by shafting vibration is to be taken into consideration. Meanwhile, shafting alignment calculation is to be carried out in accordance with the requirements of Chapter 9.

For the low-speed two-stroke cycle diesel engine, longitudinal damper may be considered to provide for the main engine in order to avoid the frame longitudinal vibration or superstructure longitudinal vibration induced by the shafting longitudinal vibration.

(5) Calculation of mechanical vibration

According to the requirements of Chapter 10 of the Guidelines, the natural frequency of main mechanical equipment vibration is to be calculated and compared with the exciting frequency which likely occurs, to satisfy the requirements of design criteria.

Meanwhile, antivibration measures, arrangement and installation requirements for mechanical equipment and pipes are to be taken into consideration.

(6) Calculation of hull girder vibration

According to the requirements of Chapter 11 of the Guidelines, the calculation for natural frequency of hull girder vibration by finite element method is to meet the requirements of design criteria. Meanwhile, evaluation of vibration effects is to be carried out.

Due to the vertical vibration of hull girders being caused by 1st order, 2nd order and 4th order unbalanced moment of two-stroke cycle diesel engine, the conditions of vertical vibration of hull induced by unbalanced moment are as follows:

① certain unbalanced moment frequency is as same as certain order frequency of hull vertical vibration;
② the related unbalanced moment achieves to a certain value;
③ the main diesel engine is installed in way of node of hull vertical vibration.

The exciting frequency being equivalent or similar to ship hull vibration frequency within the normal speed range is to be avoided. Where it is not satisfied with the design criteria and difficult to alter the natural frequency of ship hull vibration, either alteration of exciting frequency or reduce of exciting force is to be taken into consideration respectively according to the excitation source.

(7) Natural frequency calculation for superstructure vibration

According to the requirements of Chapter 12 of the Guidelines, the natural frequency of superstructure vibration is to be calculated by finite element method and to meet the design criteria.

The resonance caused by blade order frequency excitation of propeller is to be prevented (its frequency is obtained by propeller speed multiplying number of blades), which is transmitted to the superstructure through hull structures.

The resonance caused by radial force on the crankshaft affected by the two-stroke cycle diesel engine to induce shafting longitudinal vibration is to be prevented, which is further transmitted to the superstructure through thrust bearing and hull. The main harmonic order of radial force of two-stroke cycle diesel engine is relative to the cylinder number of diesel engine, see Chapter 12 of the Guidelines for details. Where the diesel engine is provided with damper of shafting longitudinal vibration, consideration of the effects of shafting longitudinal vibration is not necessary.

(8) Calculation of local vibration

According to the requirements of Chapter 13 of the Guidelines, the natural frequencies of local vibrations for typical structures, such as engine room grillage, engine room floor, bottom shell plating above the propeller, blade, platform, mast, etc., are to be calculated. For large platform structure exceeding 40 m^2, calculation for several natural frequencies is to be carried out by finite element method and compared with the main exciting frequency with the results meeting the design criteria.

Within the normal speed range, if design criteria are not satisfied with, for local vibration, the natural frequency may be altered by changing of structure type or increasing of supporting structure or increasing mass, or the exciting frequency may be altered by changing of gear ratio or propeller blade number.

2.5.3 Consideration for vibration control

In the detailed design, the vibration control design is to be considered as follows:

(1) Vibration analysis is to be carried out for local structure, i.e. the structures in the areas where excitation of propeller and main engine directly affect, such as shell plating above the propeller, engine room floor, stern, etc., in order to find issues timely and eliminate the hidden perils.

(2) Propeller fluctuation pressure is one of the main excitations for stern vibration, superstructure vibration and local vibration, alteration of design (blade number, diameter, speed, etc.) and structural parameters of propeller is an effective means to minimize the shipboard vibration, particularly for the stern vibration.

(3) For the engine frame vibration and shafting vibration, particular attention is to be given to either the vibration response of itself or that it may become the secondary excitation source for shipboard vibration.
(4) Where the shipboard vibration is minimized by installing the vibration damping device, it is necessary to carry out the detailed vibration calculation according to the real structure and made the further full-scale ship testing program.
(5) Various vibration damping devices, such as vibration isolator, antivibration cave and vibration absorber are to be provided with process and installation technology documents, providing defined and specific process flow and quality requirements based on different characteristics and material natures, to ensure the performance of vibration damping devices, as well as listing the corresponding maintenance requirements.

2.5.4 General analysis of vibration prediction

Through the research and calculation for main excitation sources and the vibration characteristics of hull girder, superstructures and local structures onboard, general analysis of vibration prediction for the whole ship is to be carried out according to the requirements of 1.5 of Chapter 1 of the Guidelines to avoid the negligence which may lead to adverse vibration induced by original excitation sources or secondary sources.

Rational design of hull structure and raising of structural stiffness are also the effective means to minimize the structural vibration response. Therefore, the rational design for the hull structure and superstructure is to be made in accordance with the relevant requirements of 2.3 of this Chapter.

2.5.5 Antivibration design information

After completion of the research working of antivibration design, the following are to be fulfilled:
(1) Analysis report of shipboard vibration calculation, including:
① analysis for shipboard vibration characteristics;
② various vibration calculations;
③ filling of main result summary (see Tables 1 to 10 below).
(2) Shipboard vibration measuring items and the arrangement of measuring points.
2.5.6 Where the ship applies for ISC class notation of vibration, analysis report for shipboard vibration calculation and shipboard vibration measurement program are to be submitted to ISC for examination, see the requirements of Chapter 15 of the Guidelines for details.

2.6 Construction Stage

2.6.1 Introduction

The construction stage includes the ship’s construction stage and trial stage.

The detailed product drawings and construction processing documents for the whole ship have been completed at the ship’s construction stage. Due to the fact that the vibration characteristics of new-built ship depend on the construction factors in addition to the design, for the above-mentioned drawings and documents, due consideration is to be given for the vibration control requirements to make the relevant provisions, improve the construction quality and control the adverse factors of shipboard vibration induced by manufacturing process deviation.

2.6.2 Construction stage

(1) Each construction drawing and processing document are to meet the requirements of related rules and standards.

(2) Antivibration measures: construction drawings and processing documents for various antivibration measures are to be prepared and installation is to be carried out in accordance with the requirements of processing documents.

(3) Vibration damping devices: construction drawings and processing documents for various vibration damping devices, such as vibration isolator, antivibration cave, vibration absorber are to be prepared, providing defined and specific process flow and quality requirements based on different characteristics and material natures, to ensure the performance of vibration damping devices, as well as listing the corresponding maintenance requirements to the ship.

(4) Shipboard vibration measurement program is to be prepared. Where a ship applies for ISC class notation of vibration, the shipboard vibration measurement program is to be approved by ISC prior to the measurement. The vibration measuring organization is also to be approved by ISC.

2.6.3 Trial stage

(1) At the trial stage, various vibration measurements are to be carried out in accordance with the shipboard vibration measurements program, such measurement is to be in compliance with the relevant requirements of Chapter 14 of the Guidelines.

(2) If the vibration level is found to exceed the rules provisions or the relevant standards at the trial stage, reasons are to be further sought by the shipyard in accordance with the methods in 1.6 of Chapter 1 and the flow of vibration evaluation in 2.2 of this Chapter, to take the economical and effective vibration damping measures or antivibration measures.

(3) Local vibration such as handrails, railings, antennas, masts, plates, pipes, etc., may occur as the new ship is on trial. The local vibration generally includes local structure resonance, which may be easily corrected by local strengthening.

(4) Measurement report of shipboard vibration is to be prepared. Where the ship applies for ISC class notation of vibration, the measurement report of shipboard vibration is to be approved by ISC.

2.7 Stage after Construction

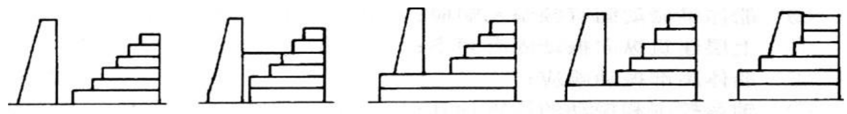
2.7.1 The hull vibration measurement at trial is to be carried out in accordance with the provisions. However, during the ship's operation, shipboard vibration may occur when the speed of main engine is different from that as the vibration measurement is taken. The structure resonance may be avoided and the vibration response may be minimized by means of regulating the speed of main engine.

2.7.2 Due to the constrain of arrangement of measuring points when the vibration measurement is carried out at trial, certain vibration phenomenon is not found, but some local vibrations may occur during the ship's operation, appropriate strengthening method may be applied to alter the vibration frequency and vibration response effectively.

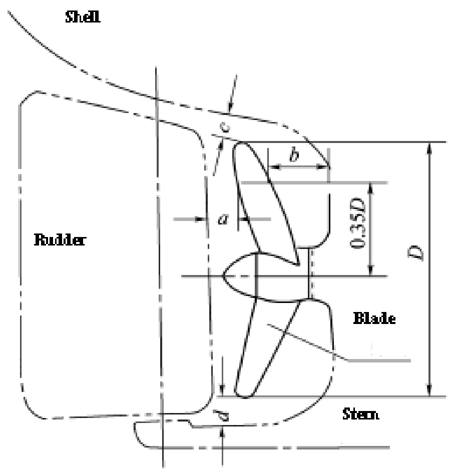
2.7.3 Due to the fact that the local structural vibration evaluation only involves 1st order natural frequency, however for lager platform structures, multiple order vibration may be neglected as the vibration measurement is constrained by the arrangement of measuring points in trial, which may cause greater vibration response in ship's operation. In this case, detailed vibration measurement and calculation are to be carried out so as to take effective solution.

2.7.4 Where complicated vibration phenomenon occurs in practice, reasons are to be further sought with necessary calculation and measurement in accordance with the methods in 1.6 of Chapter 1 and the flow of vibration evaluation in 2.2 of this Chapter, to take the economical and effective vibration damping measures or antivibration measures.

General Arrangement of Hull **Table 1**

Ship's type							
Principal dimension		Length between perpendiculars (m)		Molded breadth (m)	Molded depth (m)	Design draught (m)	Displacement (t)
Position of engine room		Middle/middle rear/aft		AP distanced from fore bulkhead (m)	AP distanced from aft bulkhead (m)	Length of engine room (m)	
Superstructure	Position	Middle front /middle /middle rear/after		AP distanced from fore end (m)	AP distanced from aft end (m)	Total height (m)	
	Size	Number of layer		Length for each layer (m)		Areas of each layer (m^2)	
	Type of structure	 					

Ship's Linear and Clearance of Propeller **Table 2**

Ship's linear	Characteristics			Wake		
Aft linear	Normal/aft	Characteristic value τ	UV Type	Asymmetry parameter W_{Δ}	Maximum wake	Average wake
Clearance of propeller (if satisfied with the rules)						
 						

Propeller **Table 3**

Propeller type	Conventional propeller/controllable pitch	Blade type	Conventional/skew

	propeller/ducted propeller/tandem propeller					
Number of propeller	Number of blade	Diameter of blade (m)	Pitch (m)	Speed (r/min)	Blade order frequency (Hz)	Cavitation
Existing force	Homogeneity of wake					
	Fluctuation pressure (N/mm^2)					
If satisfied with the criteria				Vibration damping measures		

Main Engine **Table 4**

Type	Model	Number	Number of cylinder	Rated speed (r/min)	Rated power (kW)	Unbalanced force (kNm)	Unbalanced moment (kNm)	Unbalanced moment per unit power (Nm/kW)
Two-stroke cycle diesel engine						—		
Four-stroke cycle diesel engine								—
Turbine			—	—		—	—	—
If satisfied with the criteria					Vibration damping measures			

Two-Stroke Cycle Diesel Engine Frame Vibration **Table 5**

Mode shape	Vibration mode	Natural frequency of vibration (Hz)	Frequency of exciting force (Hz)	Vibration damping measures
Transverse vibration	H			
	X			
Longitudinal vibration	L			

Shafting Vibration **Table 6**

Type of shafting vibration	Propulsion shafting of diesel engine/propulsion shafting of turbine/ gear-driven propulsion shafting system of diesel engine				
Torsional vibration	Natural frequency (Hz)	Additional torsional vibration stress for shafting (N/mm^2)			
		Crankshaft	Intermediate shaft	Thrust shaft	Screwshaft
	f_1 =				
	f_2 =				
	f_3 =				
	Gear vibration torque (kNm)		Torsional vibration torque of Elastic coupling (kNm)		
	If satisfied with the rules		Vibration damping measures		
Longitudin	Natural frequency of vibration (Hz)		Vibration acceleration of gear (m/s^2)		

al vibration	Amplitude of Longitudinal vibration (mm)		If satisfied with the rules	
	Secondary excitation (kN)		Vibration damping measures	
Whirling vibration	Natural frequency of vibration (Hz)			
	Critical speed of 1st order positive whirling (r/min)		Critical speed of positive whirling of blade order (r/min)	
	If satisfied with the rules		Vibration damping measures	

Vibration of Mechanical Equipment **Table 7**

Name of mechanical equipment	Natural frequency of vibration (Hz)	Frequency of exciting force (Hz)	If satisfied with design criteria	Vibration damping measures

Vibration of Hull Girders **Table 8**

Order	Natural frequency of vertical vibration (Hz)	Natural frequency of longitudinal vibration (Hz)	Natural frequency of transverse vibration (Hz)	Natural frequency of torsional vibration (Hz)	Frequency of exciting force (Hz)	If satisfied with the design criteria
1						
2						
3						
4						
5						
6						
Vibration damping measures						

Vibration of Superstructure **Table 9**

Natural frequency of longitudinal vibration (Hz)	Frequency of exciting force (Hz)	If satisfied with the design criteria	Vibration damping measures

Vibration of Local Structures **Table 10**

Name of structure	Natural frequency of vibration (Hz)	Frequency of exciting force (Hz)	Is satisfied with the design criteria	Vibration damping measures

Chapter 3 PROPELLER EXCITATION

3.1 Introduction

3.1.1 Exciting force induced by propeller

The exciting force induced by propeller is one of the main excitation sources for ship's vibration, which may be divided into two types, one is propeller-induced hull surface fluctuation pressure below the waterline adjacent to propeller when the propeller runs, this fluctuated water pressure is called fluctuation pressure or surface pressure, the integration of fluctuation pressure along the hull surface is called the surface force. In general, the propeller-induced hull surface fluctuation pressure will not cause the vertical vibration of hull girders other than the stern vibration, superstructure vibration and local vibration. Most of the shipboard vibrations are caused by the propeller-induced hull surface fluctuation pressure.

The other is that working conditions of each blade within a cycle is altered with the change of wake due to the inhomogeneity of wake when the propeller runs adjacent to hull, then to cause the force withstood by each blade is not equivalent and changes periodically, the formed resultant of forces and moments also change periodically and are transmitted to propulsion mechanical and hull through shaft and bearings, these forces are called bearing force, including alternating torque Mx, alternating (fluctuating) thrust Fx and alternating bending moment Mz, as shown in Figure 3.1.1. The bearing force for propeller will not induce hull girder vibration, however, alternating (fluctuating) thrust Fx may induce longitudinal vibration of superstructure through thrust bearing.

3.1.2 Effect of propeller blade cavitation on fluctuation pressure

In the uniform stream filed, due to the number of propeller blade is limited, pressure at each point of stream field will fluctuate when the propeller runs, sometimes, cavitation will be caused on the blade in the uniform stream field, the appearance of such cavitation is stable, it is called as steady cavitaiton, which has less effect to the fluctuation pressure. Where the shapes of blades are same and the angles between blades is fully equivalent, no other bearing force is caused than the steady thrust and torque.

Where the wake filed formed by hull is inhomogeneous, cavitation will be induced on blades when propeller blades enter the high wake areas, and the cavitation disappears when blades leave the high wake areas, such cavitation occurring from time to time is called unsteady cavitation. The change of cavitation volume on the blades will cause a huge pressure fluctuation on the hull surface to form a greater surface force. However, the full-scale ship measurement shows that the propeller blade cavitation has less effect on the bearing force.

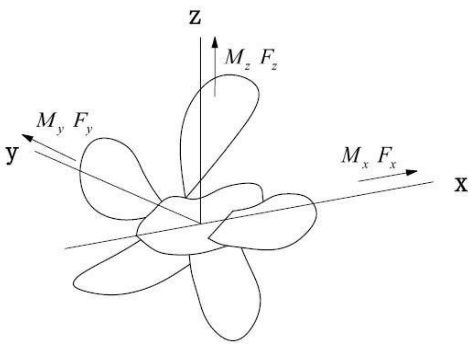


Figure 3.1.1 Propeller-Induced Bearing force

3.2 Wake Filed

3.2.1 Introduction

As mentioned above, the main reason of exciting force of propeller (either bearing force or surface force) is the inhomogeneity of wake field at stern. The wake is to be divided into three components such as axial wake, tangential wake and radial wake, which is normally shown by the ratio of wake velocity and ship speed, *W* (wake fraction). The axial wake has greater effect on the exciting force of propeller, and sometimes, the tangential wake also has greater effect. Currently, the wake distribution is to be obtained by model testing with three methods, the first is the pure meshing method which only stimulates the axial nominal wake field of propeller, the second is to apply the assumed stern plus meshing method to stimulate the axial nominal wake field of propeller, the tangential wake filed may partially stimulated due to the existence of assumed stern, the third is to apply the hull-appendage model to stimulate the three-dimensional effective wake field. It is recommended to apply the third one, particularly for ships with larger tangential wake. For conversional single-screw ships, estimation may be used to calculate the wake filed of ship model in the absence of testing information, then the full-scale ship's wake filed is obtained by dimensional correction.

3.2.2 Estimation of maximum axial wake fraction $W_{a\max}$

The axial wake fraction on the top of propeller disk area is the maximum. Where the estimated maximum axial wake fraction is less than 0.75 and less than block coefficient C_b, it is regarded that the design of stern is in compliance with the vibration damping requirements. The maximum axial wake fraction may be estimated as follows:

$$W_{a\max} = \frac{X_2 + X_3}{2} + 0.01(\phi_{1.0R} + \phi_{0.7R}) \qquad (3.2.1)$$

where: X_2— shape parameter of transverse section in way of 0.1L (L being the ship length between two perpendiculars) in front of aft perpendicular, calculated as:

$$X_2 = \frac{1}{2B}[3(y_4 - y_1) + (y_3 - y_2)]$$

B — molded breadth, in m;

R— radius of propeller, in m;

y_j— half-breadth value of each waterline corresponding to j, in m, the specific measurement is referred to Figure 3.2.2-1;

j— No. of waterline;

X_3— shape parameter of transverse section in way of 0.15 L in front of aft perpendicular, the calculation method is similar to that of X_2;

$\phi_{1.0R}$— angle of run at the end of waterline in way of 1.0 R above the propeller disk in (°), see Figure 3.2.2-2;

$\phi_{0.7R}$— angle of run at the end of waterline in way of *0.7R* above the propeller disk in (°), see Figure 3.2.2-2.

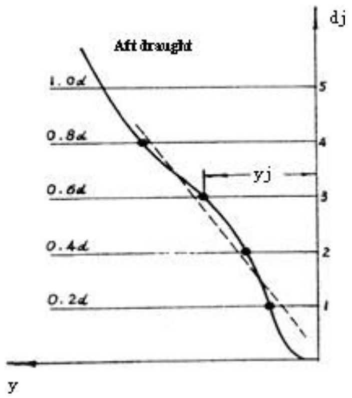


Figure 3.2.2-1 Shape Parameter of Transverse Section

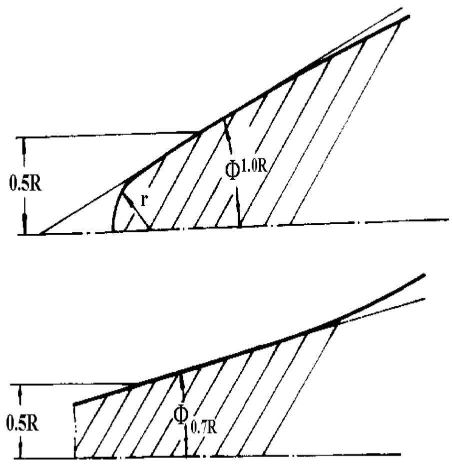


Figure 3.2.2-2 Angle of Run

3.2.3 Estimation of wake gradient W_Δ

According to some testing information of ship models, the wake gradient W_Δ is to gather statistics as one of the following formulae:

$$W_\Delta = 0.4074(X_2 + X_3) + 0.4074(b_1 + b_2) + 0.01573(\phi_{1.0R} + \phi_{0.7R}) - 0.6832 \quad (3.2.2)$$

$$W_\Delta = \frac{W_{a\max} - W_{a\min}}{1 - \overline{W}_a} \quad (3.2.3)$$

where: $\overline{W}_a$ — circumferential mean value of axial wake fraction on the radius of 1.0 R;

$W_{a\max}$— maximum value of axial wake fraction on the radius of 1.0 R;

$W_{a\min}$ — minimum value of axial wake fraction on the radius of 1.0 R;

X_2— shape parameter of transverse section in way of 0.1L in front of aft perpendicular, same as formula (3.2.1);

X_3— shape parameter of transverse section in way of 0.15 L in front of aft perpendicular, same as formula (3.2.1);

b_1 — characteristic parameter of rectifier cave in way of 0.05 L in front of aft perpendicular, calculated as formula (3.2.4):

b_2— characteristic parameter of rectifier cave in way of 0.1 L in front of aft perpendicular, calculated as formula (3.2.4):

$$b_i = \frac{l_i}{l_i + \Delta l_i} \quad (3.2.4)$$

where: i — No. of frame;

$l_i, \Delta l_i$ — see Figure 3.2.3.

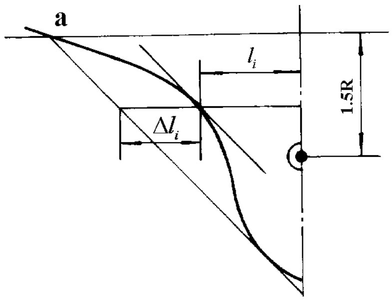


Figure 3.2.3 Depth of Rectifier Cave

3.2.4 Determination of full-scale ship nominal wake

The Reynolds number of ship model and that of full-scale ship are different, their wake and distribution are also different and there is scale effect between them. In the initial design, conversion is to be made according to Figure 3.2.4-1, *y, z* are the coordinate points of rectangular coordinate system within propeller disk area as the screwshaft for the origin. The relationship between full-scale ship wake distribution w_s and model wake distribution W_m is as follows:

$$W_s(y'z) = W_m(yz) \quad (3.2.5)$$

$$y' = \frac{c_{fs}}{c_{fm}} y \quad (3.2.6)$$

where: $c_{fs} = \dfrac{0.075}{(\lg R_{ns} - 2)^2}$ — full-scale ship's coefficient of frictional resistance;

$c_{fm} = \dfrac{0.075}{(\lg R_{nm} - 2)^2}$ — ship model's coefficient of frictional resistance;

R_{ns} — Reynolds number of full-scale ship;

R_{nm} — Reynolds number of ship model.

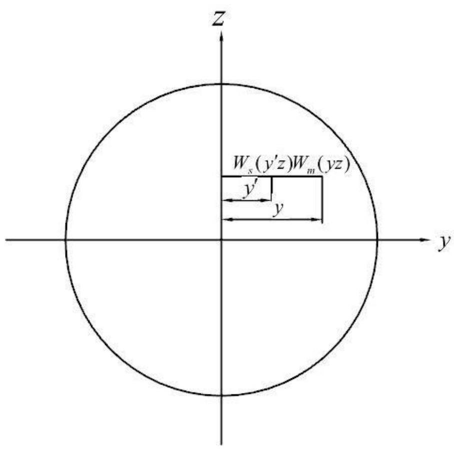


Figure 3.2.4-1 Wake Distribution of Full-Scale Ship and Ship Model

For a rational designed stern shape, in the absence of detailed original information, the maximum axial wake fraction of full-scale ship may be determined in accordance with Table 3.2.4, as the shaft bracket, shaft rack and bossing of major axis are shown in Figure 3.2.4-2.

Maximum Axial Wake Fraction of Full-Scale Ship **Table 3.2.4**

Number of propeller and shipern shap		$W_{a\max}$ (minimum value is taken for Type U stern while maximum value is taken for Type V stern)
Single-screw ship	Oil tanker, combined ship, bulk carrier, LPG carrier, LNG carrier	$W_{a\max} = 0.6 \sim 0.8$
	Dry cargo ship, container ship, ro-ro ship	$W_{a\max} = 0.5 \sim 0.7$
	Costal vessel, trawlboat	$W_{a\max} = 0.5 \sim 0.8$
Double-screw ship (see Figure 3.2.4-2)	Shaft bracket	$W_{a\max} = 0.2 \sim 0.35$
	Bossing of major axis A	$W_{a\max} = 0.3 \sim 0.5$
	Shaft rack B	$W_{a\max} = 0.4 \sim 0.7$

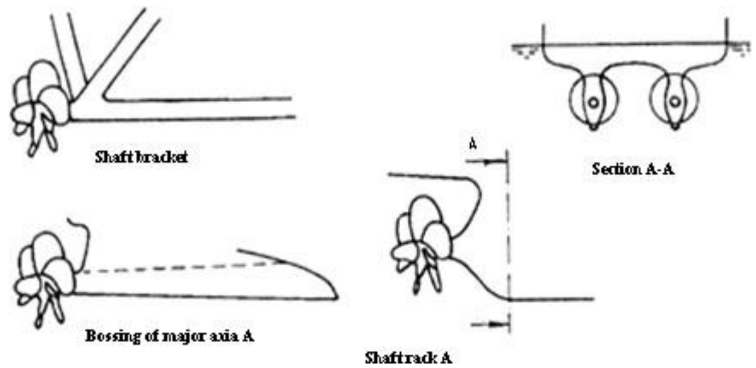


Figure 3.2.4-2 Shaft Bracket, Shaft Rack and Bossing of Major Axis

3.2.5 Cavitation of propeller

Number of cavitation on blade tip σ_n is:

$$\sigma_n = \frac{9.90 - 0.5D + h_a}{0.051(\pi n d_a)^2} \qquad (3.2.7)$$

where: D— diameter of propeller, in m;

h_a— submersed depth of screwshaft, in m;

d_a— aft draught of ship, in m;

n — speed of propeller per second, in r/s.

3.2.6 Design criteria of wake homogeneity

The more wake inhomogeneity of propeller, the greater fluctuation pressure occurs. The wake inhomogeneity is related to ship's stern linear. In general, where the wake gradient W_Δ calculated as formula (3.2.3) and the number of cavitation σ_n calculated as formula (3.2.7) are in the acceptable area of Figure 3.2.6, the wake homogeneity and stern linear are considered as appropriate.

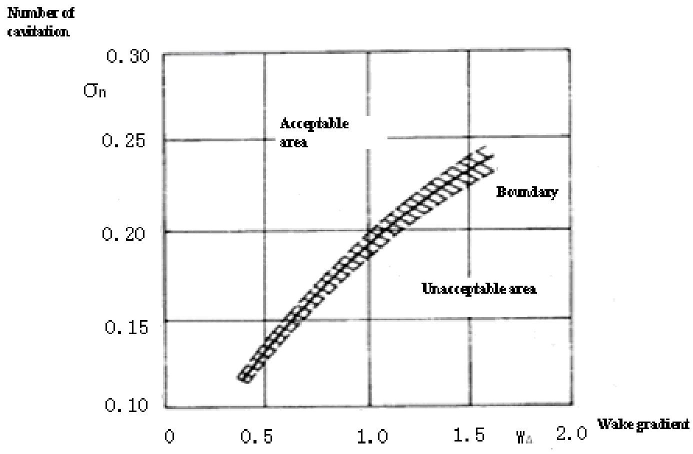


Figure 3.2.6 Design Criteria of Wake Homogeneity

3.3 Propeller Exciting Force and Frequency

3.3.1 Exciting force

The exciting force of propeller means the value at its rated speed, where it is less than the rated speed, the exciting force of propeller will be reduced with the square of its speed ratio.

3.3.2 Exciting frequency

When the propeller runs at the inhomogeneity field, the exciting frequency f_s is:

$$f_s = \frac{nZ_p v_p}{60} \quad \text{Hz} \qquad (3.4.1)$$

where: Z_p — number of propeller blade;
n — speed of propeller, in r/min;
v_p — number of exciting force for propeller, (v_p =1, 2, ……, called blade order, twice blade order……individually).

3.4 Bearing Force

3.4.1 Exciting function of bearing force

The bearing force for propeller will not induce hull girder vibration, but alternating (fluctuating) thrust *Fx* may induce longitudinal vibration of superstructure through thrust bearing. The alternating torque *Mx*, alternating thrust *Fx* and alternating bending moment *Mz* are also called amplitude of exciting force to calculate one of exciting forces for shafting torsional vibration, longitudinal vibration and whirling vibration respectively, see the relevant requirements in Chapter 6, Chapter 7 and Chapter 8 of the Guidelines.

In general, the fluctuating thrust of propeller with the blades in odd number is less than that in even number, but the component of moment for bearing force will be reduced with the increase of number of blades.

3.5 Fluctuation Pressure

3.5.1 Exciting function of fluctuation pressure

In general, the fluctuation pressure of propellerwill not induce the vertical vibration of hull girder, but may induce the stern vibration, superstructure vibration and local vibration. 90% shipboard vibration is induced by the fluctuation pressure of propeller, particularly in the areas above the propeller, local structural resonance or forced vibration will be induced by fluctuation pressure of propeller, and further to cause structure fatigue damage or reduce the habitability.

The fluctuation pressure of propeller without cavitation is mainly for the component of blade order, the fluctuation pressure of multiply blade order is always reduced monotonously and rapidly with the increase of harmonic order. As a general principle, the component of twice blade order is half of once, and the proportion for component of three times blade order is less. For the most merchant ships, cavitation of propeller will occur within the range of operation speed. For the propeller with serious cavitation conditions, sometimes, the component of twice blade order may also be larger.

3.5.2 Distribution of fluctuation pressure

Where the ship bottom above the propeller is flat, and the propeller is without cavitation or the cavitation is not serious, the maximum fluctuation pressure is located at 0.1 D in front of propeller disk area along the direction of propeller axis (D – diameter of propeller), and the distribution range is $D \times D$. If the ship bottom is in V-shape, the distribution of fluctuation pressure is irregular.

3.5.3 Fluctuation pressure testing and theoretical prediction

(1) Fluctuation pressure testing

The propeller-induced fluctuation pressure may be assessed through theoretical prediction and model testing in the design. Due to the complexity of physical phenomenon for cavitation, generally, the assessment for performances of cavitation and fluctuation pressure are to be carried out through the observation of cavitation and measurement of fluctuation pressure by the propeller model testing. Therefore, observation of propeller cavitation and model testing of fluctuation pressure are one of essential links in the design process, and how to stimulate the wake filed of propeller is the most important factor in model testing technology.

(2) Theoretical prediction

Currently, the theoretical prediction method of fluctuation pressure is grouped into two categories, one is the unsteady lifting surface theory or wortex lattice method, and the other is CFD method. Comparing the theoretical prediction method with empirical formula, more details may be taken into account, but the process is complicated with the following factors taken for consideration:

① effect of wake scale;
② distribution of solid boundary factor;
③ cavitation-induced fluctuation pressure;
④ self-induced fluctuation pressure by hull vibration.

Therefore, the effects of above-mentioned factors are needed to be considered when the theoretical prediction results are processed and applied.

3.5.4 Estimation of fluctuation pressure

(1) Fluctuation pressure induced by propeller without cavitation on the hull surface P_0 is:

$$P_0 = \frac{n^2 D^2}{70} \frac{1}{Z_p^{1.5}} \left(\frac{1}{d_s/R}\right)^{K_0} \quad \text{N/m}^2 \qquad (3.5.1)$$

where: n— speed of propeller, in r/min;

D— diameter of propeller, in m;

Z_p— number of propeller blade;

R— radius of propeller, in m;

$K_0 = 1.8 + (\frac{d_s}{R})$ for $d_s / R \le 2$;

$K_0 = 2.8$ for $d_s / R > 2$;

d_s— distance from the position in way of 0.9R to the calculated surface which is immersed, if the blade is at the top position, see Figure 3.5.3-1.

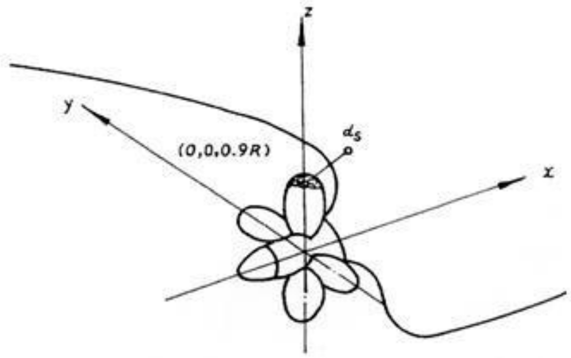


Figure 3.5.3-1 d_s Value

(2) Fluctuation pressure of blade order induced by propeller with cavitation P_c is:

$$P_c = \frac{n^2 D^2}{160} \frac{V_s(W_{a\max} - W_e)}{\sqrt{h+10.4}} \left(\frac{1}{d_s/R}\right)^{K_c} \quad \text{N/m}^2 \qquad (3.5.2)$$

where: V_s — ship's velocity, in m/s;

h_a — submersed depth of screwshaft, in m;

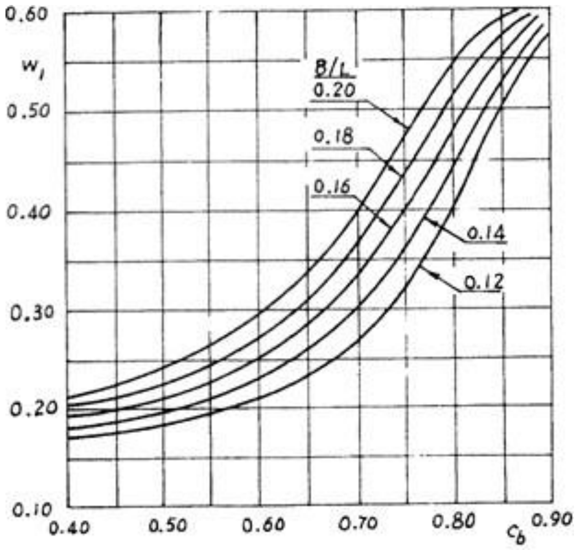
$K_c = 1.7 - 0.7(d_s / R)$; K_c=1 for $d_s / R \ge 1$;

$W_{a\max}$ — peak value of maximum wake; if no measuring information, data in Table 3.2.4 may be used;

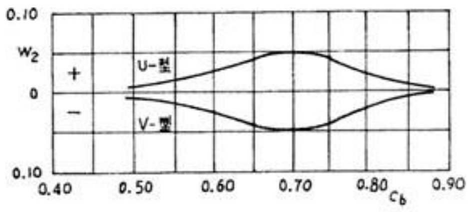
W_e — effective wake;

$W_e = 0.7(W_1 + W_2 + W_3)$;

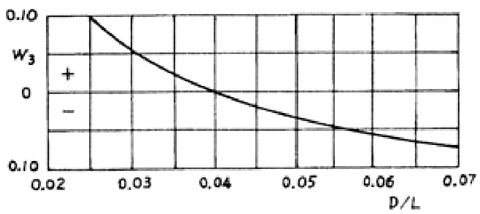
W_1, W_2, W_3 are to be determined according to Figure 3.5.3-2.



L - length in way of waterline; B - breadth in way of waterline;
(1) Corrective factor of block coefficient



Cb - block coefficient;
(2) Correction of shape coefficient



D - diameter of propeller;
(3) Corrective factor of propeller diameter

Figure 3.5.3-2 Effective Wake

(3) The total fluctuation pressure of propeller P is:

$$P = \sqrt{P_0^2 + P_c^2} \qquad \text{N/m}^2 \qquad (3.5.3)$$

(4) The maximum fluctuation pressure on the ship's flat bottom by propeller P_{max} may also be calculated by the following formula:

$$P_{max} = 5.773 K_1 K_2 \frac{N_p}{nD^3} \times 10^3 \qquad \text{N/m}^2 \qquad (3.5.4)$$

where: N_p— power of screwshaft, in kW;

n— speed of propeller, in r/min;

D— diameter of propeller, in m;

K_1— factors related to number of blade, clearance ratio of blade tip, i.e.:

$$K_1 = \left\{ \frac{c}{D}[a_1(\frac{c}{D})^2 + a_2(\frac{c}{D}) + a_3] \right\}^{-1} \qquad (3.5.5)$$

c— clearance of blade tip, in m;

a_i— to be determined by Table 3.5.1;

Z_p— number of propeller blade;

K_2— influence coefficient of angle between screwshaft and shell plate α is taken into account, to be determined by Figure 3.5.3-3; α is shown in Figure 3.5.3-4.

Coefficient Values a_i **Table 3.5.1**

a_i \ Z_p	3	4	5
a_1	27.97	26.61	23.81
a_2	-6.54	-3.78	0.00
a_3	1.92	2.07	2.62

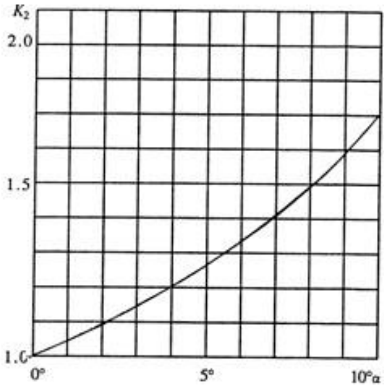


Figure 3.5.3-3 Value of K_2

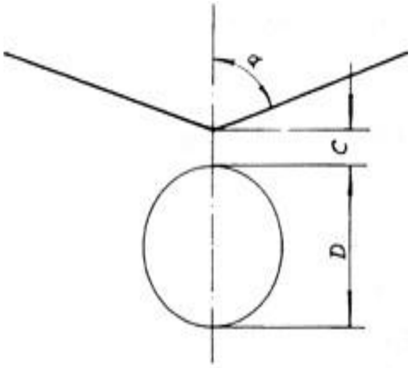


Figure 3.5.3-4 Diagram of α

3.5.5 Calculation for resultant of forces of propeller vertical fluctuation pressure

The resultant of forces of propeller vertical fluctuation pressure or surface force acting on the areas above the propeller F_z may be calculated as:

$$F_z = PD^2 \qquad \text{N} \qquad (3.5.6)$$

where: P— total fluctuation pressure of propeller, in N/m^2; to be calculated according to formula (3.5.3);

D — diameter of propeller, in m.

3.5.6 Calculation of fluctuation pressure of propeller for a certain tanker

Severe vibration occurred in the stern and engine room onboard a tanker at sea trial and in service. The clearance value of propeller was, by means of calculation and analysis, confirmed to be within the rules recommended range, and the fluctuation pressure of propeller blade order was also within the range of criteria, as shown in Table 3.5.6.

Through the calculation and analysis, the severe vibration induced around the rated speed of the tanker was found that a strong resonance phenomenon at stern was due to the exciting frequency of blade order fluctuation pressure being similar to the 1st order vibration frequency of stern vibration; and that a strong local resonance phenomenon on helicopter platform was due to the exciting frequency of four times the blade order fluctuation pressure being similar to the 4th order and 5th order local vibration frequency of helicopter platform.

Calculating Results of Propeller Blade Order Fluctuation Pressure Table 3.5.6

Blade order fluctuation pressure	Full-loaded condition	Ballasting condition	Criteria
Blade order fluctuation pressure induced by propeller without cavitation P_0 (kN/m^2)	2.107	2.107	–
Blade order fluctuation pressure induced by propeller with cavitation P_C (kN/m^2)	4.387	4.720	–
Total blade order fluctuation pressure P (kN/m^2)	4.867	5.170	8
Resultant of forces of vertical propeller blade order fluctuation pressure F_Z (kN)	376.9	400.3	

3.6 Criteria

3.6.1 Clearance dimension of propeller as the rules recommended

From point of view of reducing effect of wake, the propeller is to be kept away as far away from the hull as possible, and the clearance between the blade tip and hull becomes bigger. In gerneral, the clearance dimension of propeller is to be in compliance with the recommended values as specified in Chapter 2, PART TWO of ISC Rules for Classification of Sea-going Steel Ships, as shown in Figure 3.6.1. The Guidelines recommends c = 0.25D, then the recommended clearance dimension is as follows:

$a = 0.12D$ in m;

$b = 0.20D$ in m;

$c = 0.25D$ in m;

$d = 0.04D$ in m;

where: D— diameter of propeller, in m.

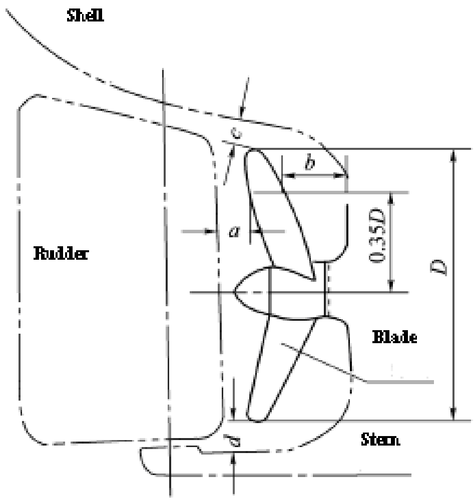


Figure 3.6.1 Clearance Dimension of Propeller

3.6.2 Propeller-induced fluctuation pressure (maximum amplitude)

The propeller-induced fluctuation pressure (maximum amplitude) is generally not to exceed the values in Table 3.6.2.

Criteria of Fluctuation Pressure **Table 3.6.2**

Type of ship	Fluctuation pressure (kN/m^2)
Pleasure-boat, passenger ships	2
Ro-ro ships	4
Container ship and high-speed cargo craft	6
Other cargo ships (including bulk carriers and oil tankers)	8

3.7 Precautions

3.7.1 Introduction

From point of view of reducing ship's vibration, the effective means is to minimize the excitation. For minimizing propeller excitation, a general consideration is to be given to stern linear, propeller parameter and their appropriate matching, as well as the matching of vibration characteristics of hull structure.

3.7.2 Improvement of wake field

When the stern linear is determined in the ship's design, general consideration is to be given to the propeller excitation, resistances and propulsion performances, for ships with higher vibration requirements, wake distribution is to be improved.

The stern linear of single-screw ship is different from that of double-screw ship, the single-screw ship is to adopt U-shaped stern while the double-screw ship is to adopt V-shaped stern. Normally, it is distinguished by the shape factor of transverse section $\tau_0 = a/b$ in way of 0.1 L in front of aft perpendicular (see Figure 3.7.2-1), and in general $\tau_0 > 0.5$ being for V-shape. The shape factor of transverse section τ_0, in general is to be $\tau_0 < 0.5$. For full-formed ships, $\tau_0 < 0.3$ is preferable.

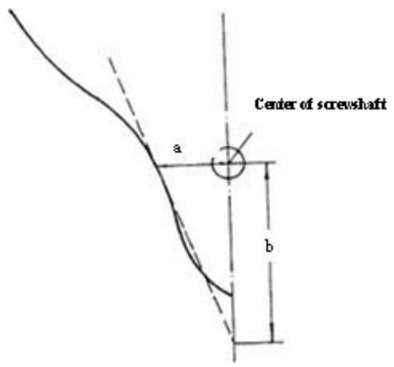


Figure 3.7.2-1 Definition of τ_0

It has been proven by a large quantity of practices that U-shaped or ball-shaped sterns for single-screw ships may improve the inhomogeneity of wake fields obviously and is beneficial to minimize the propeller excitation.

To modify the shape of transverse section for the fore part of propeller to cause the appropriate bilge vortex. Based on the positions of peak values of wake distribution within the propeller disk area, different modifications are to be carried out for the transverse section in order to control the center position of bilge vortex. Where wake peak is on the top of propeller disk, it is required that the maximum breadth is to be located at the center of screwshaft, where the wake peak is above the screwshaft, the breadth of lowest part of the transverse section is to be increased.

Figure 3.7.2-2 shows the transverse section of a certain stern in way of 0.1 L in front of aft perpendicular, the full line being a prototype, the corresponding wake distribution is shown in Figure 3.7.2-3. There is a wake peak more than 1.0 above the screwshaft, and the wake distribution in upper part is also unacceptable. Therefore, if a bilge vortex is introduced in a lower position on the disk area ratio, the wake field may be improved. Fine modification to the prototype is done as the dotted line shown in Figure 3.7.2-3, and the wake distribution will be greatly improved, as shown in Figure 3.7.2-4.

The angle of run is the angle between longitudinal centerplane of hull and afterbody waterline. The enlarging of angle of run will raise the mean wake and ship's resistance, then to increase the loads on the propeller. However, it will not greatly effect on the wake values provided that the angle or run is maintained within 30°, and ideal wake distribution may be obtained. It is important that if the maximum angle of run φ_m shown in Figure 3.7.2-5 is located in the front position and the value is big, serious interlayer separation will occur to cause the above-mentioned adverse consequence, W_s in the figure is the half-breadth of stern post.

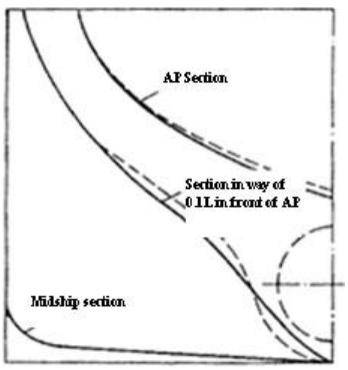


Figure 3.7.2-2 Transverse Section

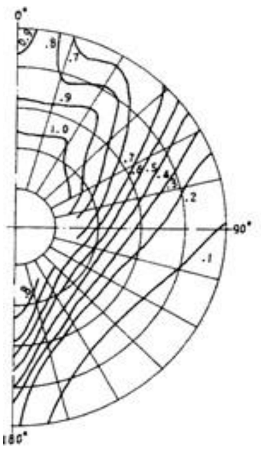


Figure 3.7.2-3 Prototype Wake Distribution

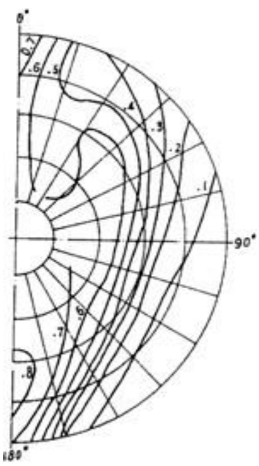


Figure 3.7.2-4 Modified Wake Distribution

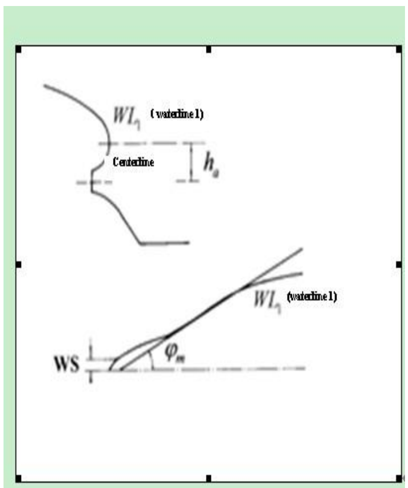


Figure 3.7.2-5 Maximum Angle of Run

3.7.3 Elements of rational selection for propeller

(1) Number of propeller blade

① Under the same circumstance, with the increase of number of blades, the diameter of propeller is generally reduced, the clearance of blade tip is increased and the propeller-induced fluctuation pressure trends to decrease.

② The bearing force is closely related to the number of blades, therefore, number of blades is to be considered according to the detail conditions of the given wake filed.

③ The number of blades is to be properly chosen to prevent the resonance of hull stern, superstructure and local structure.

④ To avoid the resonance of shafting whirling vibration and frame longitudinal vibration is also the element to choose the number of blades.

(2) Skew

Highly skewed propeller may reduce bearing force and surface force. The degree and distribution of skew are to be taken into consideration together with the each harmonic component of wake field. Where they are matched improperly, the effect is not obvious, even an opposite effect may be caused. The skew is to be chosen for the purpose of minimizing the bearing force and the effect of minimizing of surface force is to be inspected.

3.7.4 Unloading for blade tip

In the conventional design of propeller, radial loading distribution, i.e. radial circulation distribution is to be chosen the best shape in order to obtain the highest effective. However, due to a larger loading adjacent to the tip, the cavitation of tip bilge is likely to occur, thus influence the cavitation in way of external radius of propeller, cause the increase of the excitation of propeller. A proper reduction of the loading on the blade tip, i.e. use of the unloaded propeller may effectively reduce the excitation of propeller.

3.7.5 Means to reduce excitation induced by inhomogeneity of wake field

(1) Installing tail fin: installation of a tail fin may increase the velocity of flow in high wake area and homogenize the wake field. Figure 3.7.5-1, Figure 3.7.5-2 and Figure 3.7.5-3 show the arrangements for three different cases.

(2) Installing wake flow tunnel: installation of a wake flow tunnel may reduce the high wake peak value above the propeller disk, with the same effect of tail fin. Figure 3.7.2-2 shows the wake flow tunnel used by a single-screw container ship.

(3) Vortex generator: Sometimes, the adverse wake field is caused by the flow separation at tail. In addition to the above-mentioned means, vortex generator is to be installed at the side of stern to increase flow energy within interlayer and prevent separation, so as to improve the wake field. Figure 3.7.5-5 shows the shape of vortex generator used by a single-screw ship.

(4) Extending deadwood: if the double-screw ship is provided with a flat bottom, deadwood is to be extended and the flat-bottomed stern is converted to a similar double-screw stern.

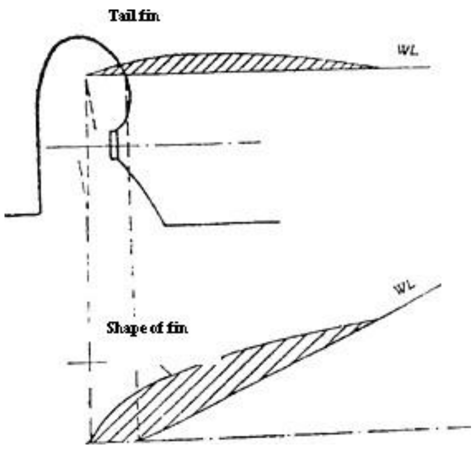


Figure 3.7.5-1 Tail Fin to Minimize Wake Peak Value above Blade Disk Area

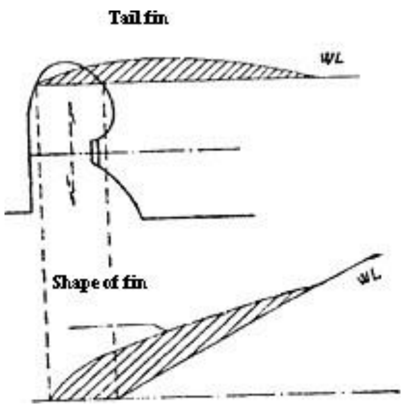


Figure 3.7.5-2 Tail Fin to Minimize Wake Peak Value above Blade Disk Area (2)

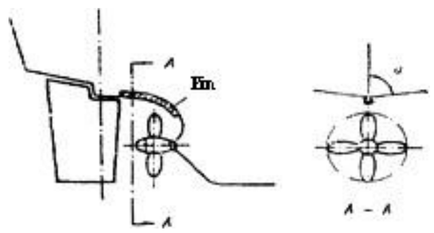
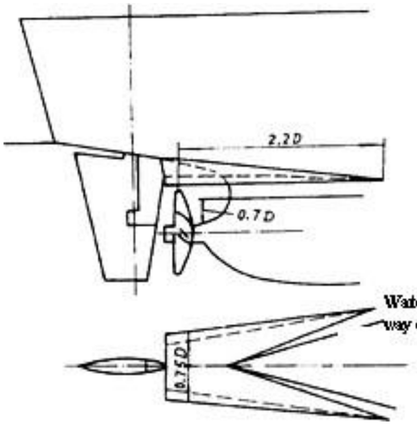


Figure 3.7.5-3 Vertical Fin to Reduce Vortex Cavitation of Propeller and Hull



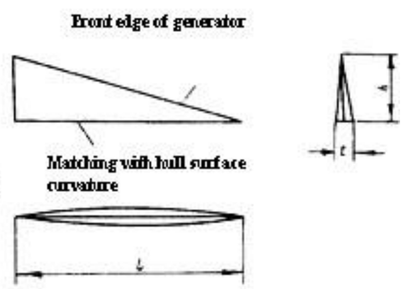


Figure 3.7.5-4 Arrangement of Wake Flow Tunnel **Figure 3.7.5-5 Shape of Vortex Generator**

3.7.6 Device to reduce energy transmission of propeller excitation

To reduce the transmission of excitation and consume the excitation energy is also an effective means to minimize the vibration responses, e.g. antivibration cave opened on bottom plate above the propeller (Figure 3.7.6) may reduce the propeller surface force transmitted on the hull so as to minimize the response of stern vibration. Among them, the damping antivibration cave is only to be opened with a small hole on the bottom plate and get rid of rubber elastic element, the manufacturing technology is simple and the maintenance working is greatly reduced.

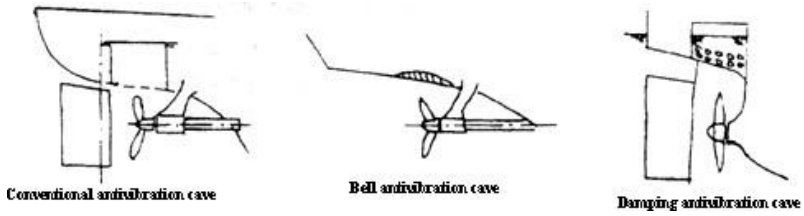


Figure 3.7.6 Diagram of Antivibration Caves

3.7.7 In the design process, comprehensive consideration is to be given to the propeller efficiency, cavitation and cavitation corrosion, vibration and noise performance. The wake design methods, blade tip unloading and high skew technology have been widely used in the design of modern ships propellers, which may effectively reduce fluctuation pressure induced by propellers.

Chapter 4 UNBALANCED MOMENT OF DIESEL ENGINE

4.1 Introduction

4.1.1 Introduction

When the diesel engine is running, the combustion gas will cause alternating gas pressure, alternating reciprocating inertial force and centrifugal force, these forces and moments may induce the following vibration phenomenon:

(1) tangential force acting on crankshaft may cause shafting torsional vibration so as to induce hull vibration;

(2) radial force acting on crankshaft may cause shafting longitudinal vibration so as to induce hull vibration;

(3) lateral force acting on cross head may cause frame transverse vibration so as to induce hull vibration;

(4) unbalanced moment acting on crankshaft may induce ship hull vibration.

This Chapter is to mainly discuss the external unbalanced moment of diesel engine and analyze the effect on ship hull vibration.

4.1.2 Excitation of diesel engine

(1) Gas pressure of diesel engine

The decomposition of gas pressure for each cylinder is shown in Figure 4.1.2. The gas pressure P_g for each cylinder may be decomposed to two components: connecting rod force P_c and side thrust force P_H. In way of crank pin, the connecting rod force P_c may be also divided into tangential force P_T and radial force P_R:

$$\vec{P}_g = \vec{P}_c + \vec{P}_H$$

$$\vec{P}_c = \vec{P}_T + \vec{P}_R$$

These forces on the unit piston area may be calculated as:

$$P_c = P/\cos\beta \quad \text{N/mm}^2 \qquad (4.1.1)$$

$$P_H = Ptg\beta \quad \text{N/mm}^2 \qquad (4.1.2)$$

$$P_T = P\sin(a+\beta)/\cos\beta \quad \text{N/mm}^2 \qquad (4.1.3)$$

$$P_R = P\cos(\alpha+\beta)/\cos\beta \quad \text{N/mm}^2 \qquad (4.1.4)$$

Tangential force P_T acting on crankshaft may cause shafting torsional vibration; radial force P_R acting on crankshaft may cause shafting longitudinal vibration; lateral force P_H acting on cross head may cause frame transverse vibration, and may further cause hull vibration.

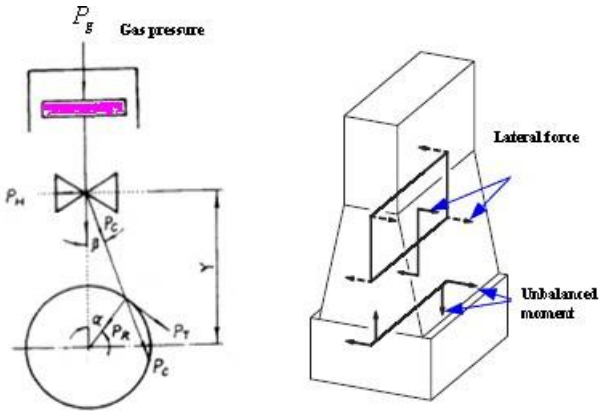


Figure 4.1.2 Decomposition of Diesel Engine Gas Pressure and Its Effect

(2) Reciprocating inertial force of diesel engine

For standard piston mechanism, the system type of reciprocating inertial force is similar to that of gas pressure. The reciprocating inertial force of diesel engine is the inertial force induced by the mass of reciprocating moving components and reciprocating moving parts of connecting rod, the line of action is coincided with the centerline of respective cylinder.

The reciprocating inertial force P_j of single cylinder for diesel engine is shown as:

$$P_j = -m_j R\omega^2 \cos\alpha - \frac{\lambda}{4} m_j R(2\omega)^2 \cos 2\alpha - \frac{\lambda^3}{64} m_j R(4\omega)^2 \cos 4\alpha \qquad (4.1.5)$$

where: m_j — mass of reciprocating parts;

R — radius of crank;

ω — angular velocity;

α — rotating angle.

$\cos\alpha$, $\cos 2\alpha$ and $\cos 4\alpha$ are called 1st order reciprocating inertial force, 2nd order reciprocating inertial force and 4th order reciprocating inertial force respectively, the others are less and not taken into account.

Tangential inertial force in the reciprocating force system is one of the exciting forces to induce shafting torsional vibration while radial inertial force is one of the exciting forces to induce shafting longitudinal vibration.

The reciprocating unbalanced resultant of forces (external forces) caused by reciprocating inertial force of multi-cylinder diesel engine, straight-type two-stroke cycle diesel engine (excluding 2nd order of two cylinders) and straight-type four-stroke cycle diesel engine (excluding 2nd order of two cylinders and 2nd order of four cylinders) are balanced, i.e. the reciprocating unbalanced force is equal to 0, consideration of the influence of reciprocating unbalanced force is not necessary.

(3) Whirling inertial force of diesel engine

The whirling inertial force of diesel engine is 1st order centrifugal force of crank, which acts on the main bearing. The value of whirling inertial force is not changed with the rotating angle of crankshaft and is outward along the centre of crank.

The 1st order whirling unbalanced resultant of forces (external forces) caused by reciprocating inertial force of multi-cylinder diesel engine is to be balanced, i.e. the whirling unbalanced force is equal to 0, consideration of the influence of whirling unbalanced force is not necessary.

4.2 Unbalanced Moment

4.2.1 Meaning of unbalanced moment

The meaning of unbalanced moment is shown in Figure 4.2.1-1.

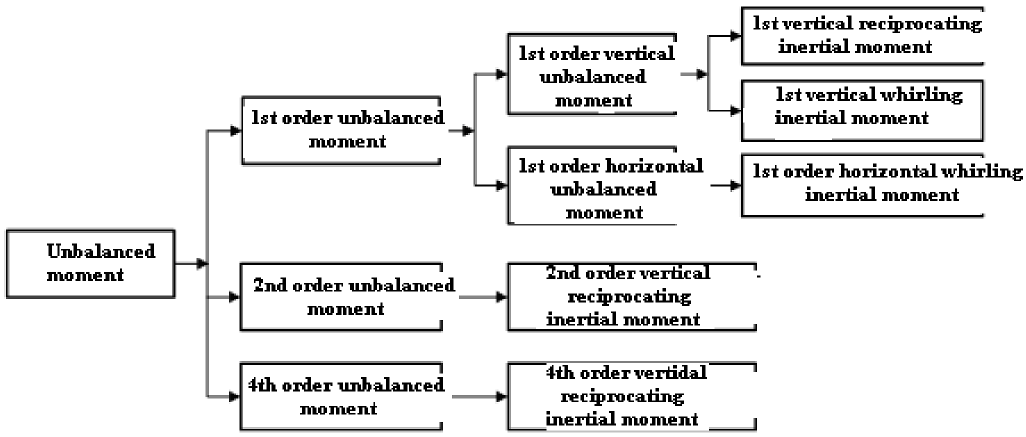


Figure 4.2.1-1 Meaning of Unbalanced Moment

The 1st order, 2nd order and 4th order vertical reciprocating unbalanced moments caused by 1st order, 2nd order and 4th order reciprocating forces of multi-cylinder diesel engine, acting on the centerline of crankshaft will induce vertical buckling of crankshaft.

The 1st order whirling inertial moment caused by multi-cylinder diesel engine may be divided into 1st order vertical whirling inertial moment and 1st order horizontal whirling inertial moment. The 1st order vertical whirling inertial moment acting on the centerline of crankshaft will induce vertical buckling of crankshaft. The 1st order horizontal whirling inertial moment acting on the centerline of crankshaft will induce torsional of crankshaft.

The sum of 1st order vertical reciprocating inertial moment and 1st order vertical whirling inertial moment is called 1st order vertical unbalanced moment of diesel engine.

The 1st order horizontal whirling inertial moment is called 1st order horizontal unbalanced moment of diesel engine.

The 1st order vertical unbalanced moment and 1st order horizontal unbalanced moment may be called 1st order unbalanced moment.

The 2nd order vertical unbalanced moment and 4th vertical unbalanced moment are also be called 2nd order unbalanced moment and 4th order unbalanced moment.

The above-mentioned unbalanced moments may generally be called as unbalanced moments of diesel engine.

Unless specially specified, the unbalanced moments in this Chapter are to be in the vertical direction. The unbalanced moment values provided by the manufacturers are the 1st order, 2nd order and 4th order unbalanced moment values. Particular attention is to be given to ask for the latest information of diesel engine type intended to choose in the design of vibration control.

Function of unbalanced moment for two-stroke cycle diesel engine is shown in Figure 4.2.1-2. M_{1H} and M_{1V} are 1st order horizontal and 1st order vertical unbalanced moments respectively.

M_{2V} and M_{4V} are 2nd order and 4th order balanced unbalanced moments respectively.

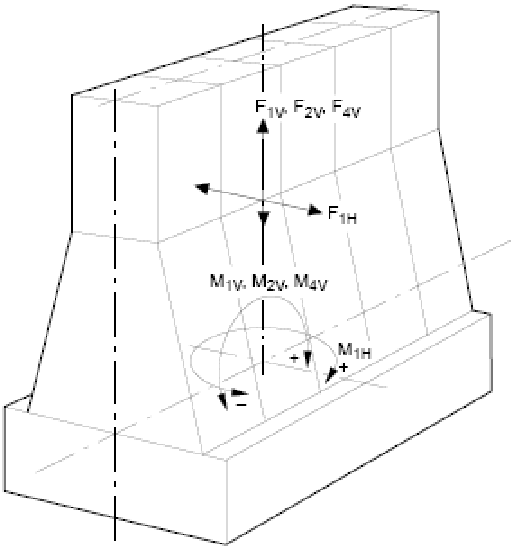


Figure 4.2.1-2 Unbalanced Moment of Two-Stroke Cycle Diesel Engine

4.2.2 Unbalanced moment value

(1) Unbalanced moment value provided by manufacturer

After the design of diesel engine is finished, the unbalanced moment value may be calculated by manufacturer, and together with the Type *H* exciting moment and Type *X* exciting moment are to be referred for the vibration control design. Explanation of two types is as follows.

The unbalanced moment M_e provided by the manufacturer means the value under the rated working condition of the diesel engine. The unbalanced moment M_r under other speed is:

$$M_r = M_e\left(\frac{n_r}{n_e}\right)^2 \tag{4.2.1}$$

where: n_r — other speed of diesel engine, in r/min;

n_e — rated speed of diesel engine, in r/min.

(2) Example 1 – unbalanced forces and unbalanced moments of Type K90MC two-stroke cycle diesel engine.

The unbalanced forces and unbalanced moments of Type K90MC two-stroke cycle diesel engine produced by MAN B&W are shown in Table 4.2.2-1, as follows:

① unbalanced forces of diesel engine with all cylinders are equal to 0;

② the unbalanced moments of diesel engines with various numbers of cylinder are different, the 1st order and 2nd order unbalanced moments of diesel engine with four cylinders are larger, the 2nd order unbalanced moment of diesel engine with five or six cylinders are large, but the 1st order and 2nd order unbalanced moments of diesel engine with more than seven cylinders are not large;

③ the 4th order unbalanced moment of diesel engine with all cylinders is smaller.

Unbalanced Moment Values of Type K90MC Two-Stroke Cycle Diesel Engine

Table 4.2.2-1

Number of cylinder	4	5	6	7	8	9	10	11	12
Order of firing	1-3-2-	1-4-	1-5-3-4-	1-7-2-5-	1-8-3-4-	1-6-7-3-			1-8-12-4-

		4	3-2-5	2-6	4-3-6	7-2-5-6	5-8-2-4-9			2-9-10-5-3-7-11-6
Unbalanced force (kN)		0	0	0	0	0	0	0	0	0
Unbalanced moment (kNm)	1st order (a)	2505 (b)	794	0	473	207	1630	291	202	0
	2nd order	5322 (c)	6635 (c)	4069 (c)	1338	0	1540	34	203	0
	4th order	0	21	163	463	188	234	334	427	326

Notes: (a) For all the numbers of cylinder, as a standard, 1st order vertical and horizontal unbalanced moments are to be balanced and equivalent.

(b) For four-cylinder diesel engine, if necessary, 70% of the 1st order vertical unbalanced moment may be moved to 1st order horizontal unbalanced moment through the adjustment of balance weight, vice versa.

(c) For four, five and six-cylinder diesel engine, a compensator for 2nd order unbalanced moment may be installed at both fore and aft ends of crankshaft so as to remove the 2nd order unbalanced moment.

(3) Example 2 — unbalanced forces and unbalanced moments of Type RT-flex 96C two-stroke cycle diesel engine.

The unbalanced forces and unbalanced moments of Type RT-flex 96C two-stroke cycle diesel engine produced by MAN B&W are shown in Table 4.2.2-2, as follows:

① unbalanced forces of diesel engine with all cylinders are almost equal to 0;

② for diesel engines with various numbers of cylinders, the 1st order vertical unbalanced moment is basically same as 1st order horizontal unbalanced moment, the 1st order unbalanced moment of diesel engine with nine and eleven cylinders are larger;

③ except the diesel engine with eight and twelve cylinders, the 2nd order unbalanced moment of other diesel engines are larger;

④ the 4th order unbalanced moment of diesel engine with six cylinders is large, and measures are to be taken. It is recommended by WÄRTSILÄ that an electronic balance compensator is to be installed on the deck.

Unbalanced Moment Values of Type RT-Flex 96C Two-Stroke Cycle Diesel Engine Table 4.2.2-2

Number of cylinder		6	7	8	9	10	11	12	14
Power (kW)		34320	40040	45760	51480	57200	62920	68640	80080
Unbalanced force (kN)	1st order vertical	0	0	0	0	43	0	0	15
	1st order horizontal	0	0	0	0	45	0	0	15
	2nd order vertical	0	0	0	0	26	0	0	21
	4th order vertical	0	0	0	0	22	0	0	86
Unbalanced moment (kNm)	1st order vertical	0	562	628	1941	51	1347	0	45
	1st order horizontal	0	580	628	1997	42	1388	0	45
	2nd order vertical	6753	1960	0	2204	1612	1769	0	11
	4th order vertical	3450	981	399	497	489	188	690	598

4.2.3 Exciting frequency of unbalanced moment

The unbalanced moment of diesel engine is induced by its inertial force, the relationship between exciting frequency and engine speed:

(1) exciting frequency of 1st order unbalanced moment is equivalent to the engine speed, in numerical;

(2) exciting frequency of 2nd order unbalanced moment is equivalent to twice of the engine speed, in numerical;

(3) exciting frequency of 4th order unbalanced moment is equivalent to four times the engine

speed, in numerical.

4.3 Assessment of Unbalance Characteristics for Diesel Engine

4.3.1 Introduction

Whether the unbalanced moment of diesel engine will induce ship hull vibration depends on the estimation precision of natural frequency for ship hull vibration, scale of unbalanced moment and the installation position of main diesel engine.

The unbalanced moment of diesel engine is not the only cause for hull vibration, and is not a kind of excitations being not eliminated or minimized. The effect of unbalanced moment of diesel engine on the hull vibration may be controlled in an acceptable range through the rational design. Currently, The estimation of effect of unbalanced moment on the ship hull vibration has not achieved the precision required by engineering application. In general, the unbalanced moment is restrained in an acceptable and relative safe range so that an excitation source of adverse shipboard vibration can be controlled.

For this purpose, the latest unbalanced moment values of corresponding engine types are to be obtained from the manufacturers for vibration design in the ship contracting design.

4.3.2 PRU

(1) PRU of unbalanced moment per unit power for diesel engine

This Chapter provides the PRU of 1st order and 2nd order unbalanced moments given by MAN B&W and WÄRTSILÄ and may be referred for evaluation.

PRU of unbalanced moment per unit power for diesel engine is as follows:

$$\text{PRU} = \frac{M_r}{n_e} \quad \text{Nm/kW} \tag{4.3.1}$$

where: M_r— unbalanced moment value, in Nm;

n_e— rated power of diesel engine, in kW.

(2) PRU of unbalanced moment per unit power for Type MC diesel engine

PRU of unbalanced moment per unit power for Type MC diesel engine produced by MAN B&W is shown in Figure 4.3.2-1, where:

— PRU≤ 60: compensator not relevant;

— 60< PRU< 120: compensator unlikely;

— 120< PRU< 220: compensator likely depending on further evaluated;

— PRU> 220: compensator most likely.

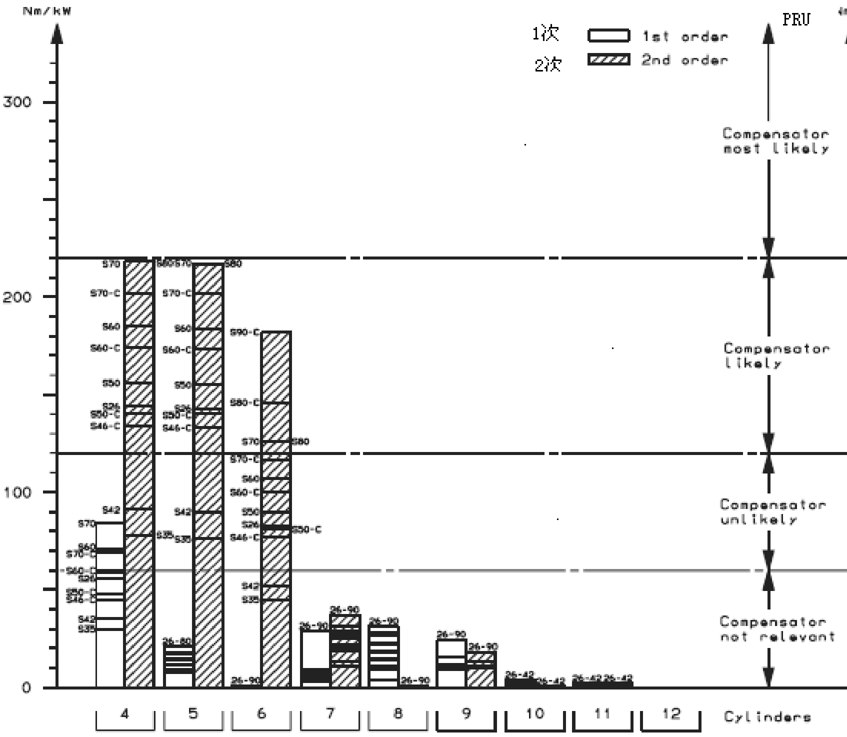


Figure 4.3.2-1 PRU for Type MC Diesel Engine

(3) PRU of unbalanced moment per unit power for Type RT-flex 96C diesel engine

PRU of unbalanced moment per unit power for Type RT-flex 96C diesel engine produced by WÄRTSILÄ is shown in Figure 4.3.2-2, where:

— C (PRU< 60＝: compensator not relevant;

— B (60< PRU< 120＝: compensator unlikely;

— A (120< PRU＝: compensator likely.

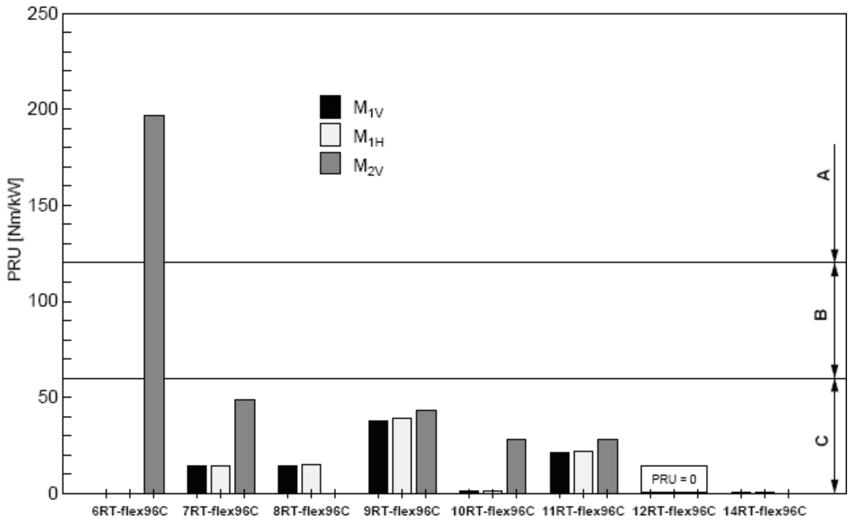


Figure 4.3.2-2 PRU for Type RT-flex 96C Diesel Engine

4.3.3 Analysis for influences of unbalanced moment of diesel engine

(1) Influence factor of unbalanced moment

The scale of unbalanced moment of diesel engine is related to its inertial force, number of cylinder, order of firing and speed.

(2) Two-stroke cycle diesel engine with four cylinders

For two-stroke cycle diesel engine with four cylinders, 1st order and 2nd order unbalanced moments are large, balance compensator must be used to control the effects of unbalanced moments. For example, where the 1st order unbalanced moment for Type K90MC four-cylinder diesel engine exceeds 2000 Nw, the PRU is about 135; where the 2nd order unbalanced moment is approximate to 4500 Nm, the PRU is about 290.

(3) Two-stroke cycle diesel engine with five and six cylinders

For two-stroke cycle diesel engine with five and six cylinders, 1st order unbalanced moment has no effect on the ship hull vibration; however 2nd order unbalanced moment is larger and may cause ship hull vibration, precautions are to be considered. Where the diameter of cylinder is larger, the 2nd order unbalanced moment is also larger, and the PRU is approximate to or exceed 120, an appropriate balance compensator is to be installed.

(4) Two-stroke cycle diesel engine with more than seven cylinders

For two-stroke cycle diesel engine with more than seven cylinders, the effects of 1st order and 2nd order unbalanced moments on the ship hull vibration may not be taken into consideration.

(5) 4th order unbalanced moment of two-stroke cycle diesel engine

The 4th order unbalanced moment of two-stroke cycle diesel engine is normally smaller and has no effect on the ship hull vibration. However, for certain diesel engine with larger diameter, e.g. Sulzer RT-flex 96C, the 4th order unbalanced moment of diesel engine with six cylinders is large, and/or hull vibration frequency is lower, the effect of 4th order unbalanced moment on the ship hull vibration is to be taken into account.

(6) Four-stroke cycle diesel engine with six cylinders and above

For four-stroke cycle diesel engine with six cylinders and above, the unbalanced moment is smaller and is not to be an excitation source of adverse shipboard vibration.

4.4 Effects of Unbalanced Moment on Ship Hull Vibration

4.4.1 Conditions of hull vertical vibration induced by unbalanced moments

The conditions of hull vertical vibration induced by unbalanced moments are as follows:

(1) a certain order unbalanced moment frequency is same as certain order of hull vertical vibration;

(2) the relevant unbalanced moment achieves a certain value;

(3) the main diesel engine is installed in way of nodes of hull vertical vibration.

4.4.2 Evaluation for influences of unbalanced moment excitation

In the design, an evaluation for hull excitation may be conducted through calculation and analysis of unbalanced moments.

(1) Evaluation of unbalanced moment frequency

A diagram of engine speed – frequency may be drawn according to the estimation method of frequency for ship hull vibration given in Chapter 11 of the Guidelines in order to evaluate the engine speed for 1st order unbalanced moment, twice of engine speed for 2nd order unbalanced moment, four times the engine speed for 4th order unbalanced moment and whether the frequency of engine speed with larger unbalanced moment is kept away from the natural frequency of hull vertical vibration.

(2) PRU of unit power unbalanced moment

When a certain unbalanced moment frequency is same as certain order overall vibration frequency of hull, PRU may be calculated in accordance with the unbalanced moment provided by the manufacturer, and based on the PRU, the effects on ship hull vibration are to be evaluated and antivibration measures are to be taken.

(3) Evaluation of engine installation position

As shown in Figure 4.4.2-1, the work W of unbalanced moment on the hull is directly proportional to the related modal angle θ, which is the maximum in way of node of vibration mode. Where the main engine is located in way of node of mode shape curve for the relative hull vertical vibration, the maximum energy input by unbalanced moment at resonance will induce a strong hull vibration. With the engine position being far away from the node of mode shape curve, excitation will be gradually reduced.

Therefore, when the natural frequency of hull vertical vibration is equivalent (or similar) to the excitation frequency, the engine position is to be kept far away from the node of relative mode shape curve in order to minimize the vibration energy to the ship.

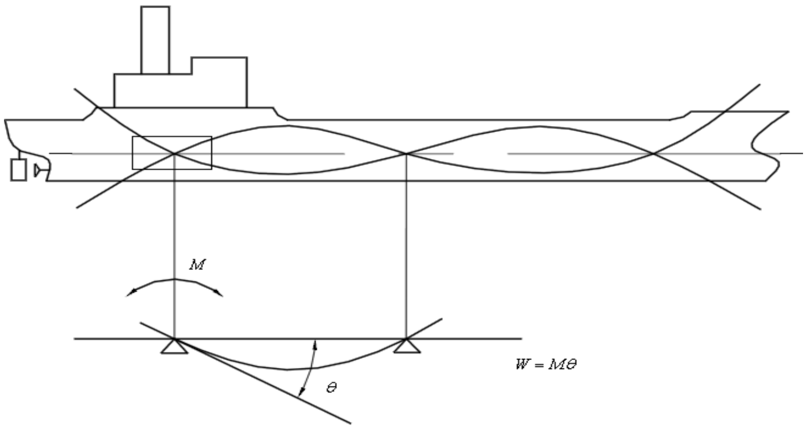


Figure 4.4.2-1 Work Principle of Unbalanced Force and Moment

In the design, the frequency and mode of hull vertical vibration are to be estimated according to the methods provided in Chapter 11, and the evaluation is to be carried out for the engine installation position to determine whether and how to install a balance compensator.

The installation of balance compensator is associated with the position of main engine corresponding to node of vibration mode relative to hull vertical vibration. Where the main engine is located at the node of hull vertical vibration, balance weight is to be installed at both fore and aft ends of main engine so as to balance the 2nd unbalanced moment M_{2V}, as shown in Figure 4.4.2-2(1); where the main engine is located after the node, balance weight is only to be installed at the aft end of engine, and the 2nd unbalanced moment M_{2V} may be balanced, as shown in Figure 4.4.2-2(2).

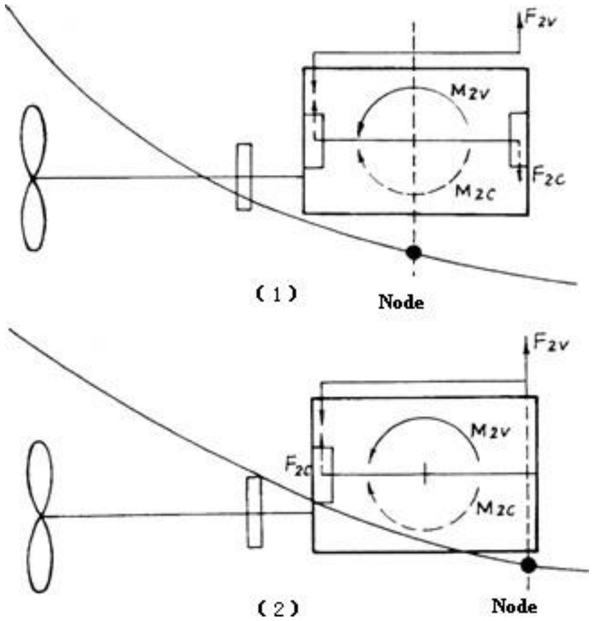


Figure 4.4.2-2　Effect of Installation Position for Main Diesel Engine

4.5　Precautions

4.5.1　Altering of frequency

Where the natural frequency of overall vibration is equivalent or close to engine excitation frequency, both of the frequencies are to be kept away as far as possible. For this purpose, the operating speed of main diesel engine is to be re-defined so as to alter the exciting frequency of main diesel engine or the ship's general design is to be remade to change the ship's mass, stiffness so as to alter the natural frequency of vertical vibration.

However, for alteration of exciting frequency of main diesel engine or natural frequency of hull vertical vibration, the adjustment range available is restrained with less function and is not applied widely.

4.5.2　Balance compensator for diesel engine

Where it is considered by the PRU evaluation that the unbalanced moment value is large and may induce the adverse vibration, a usual measure is to install a balanced compensator.

Most of the balance compensators are installed on the diesel engine and may neutralize or minimize the unbalanced moment through the balance weight installed on diesel engine. Some engines produced by the manufacturers are provided with standard balance weight, which may adjust the scale and position on the diesel engine for balance weight in order to get proper balance compensating moment to meet the requirements of vibration control.

The balance compensator of diesel engine is mainly as follows:

(1) 1st order balance compensator

The 1st order balance compensator is a pair of balance weights arranged at the fore and aft ends of crankshaft, as shown in Figure 4.5.2-1. The fore and aft balance weights are installed at the fore end of flange of crankshaft and flywheel of output end of diesel engine, respectively.

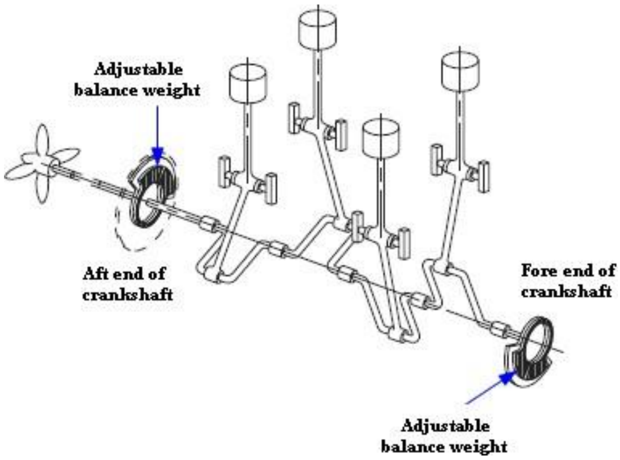


Figure 4.5.2-1 1st Order Balance Compensator

(2) 2nd order balance compensator

The 2nd order balance compensator of diesel engine is two pairs of balance weights arranged at the fore and aft ends of diesel engine, as shown in Figure 4.5.2-2. However, they are to be installed at both fore and aft ends only when the main engine is just located in or close to the node of hull vibration mode. Where the main engine is located in front of or after the node, one pair of balance weights is installed at the fore and aft ends of the diesel engine.

(3) Combined-type balance compensator

It is installed to compensate for the 1st order and 2nd order unbalanced moments together, as shown in Figure 4.5.2-3. This compensator is generally used for diesel engine with four cylinders.

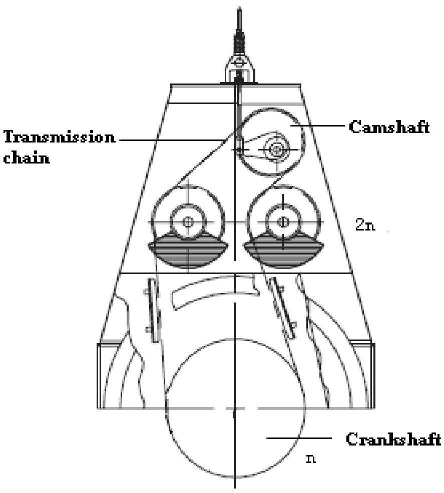


Figure 4.5.2-2
2nd Order Balance Compensator

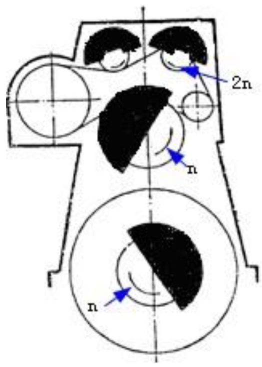


Figure 4.5.2-3
Combined-type Compensator

4.5.3 Electronic balance compensator

The electronic balance compensator is generally installed on deck. The response of ship hull vibration induced by electronic balance compensator is just neutralized the response caused by unbalanced moment of diesel engine (with the same frequency, opposite phase position) which may balance and compensate the 1st order and/or 2nd order unbalanced moment for diesel engine. Diagram of installation for electronic 2nd order balance compensator on deck is shown in Figure 4.5.3.

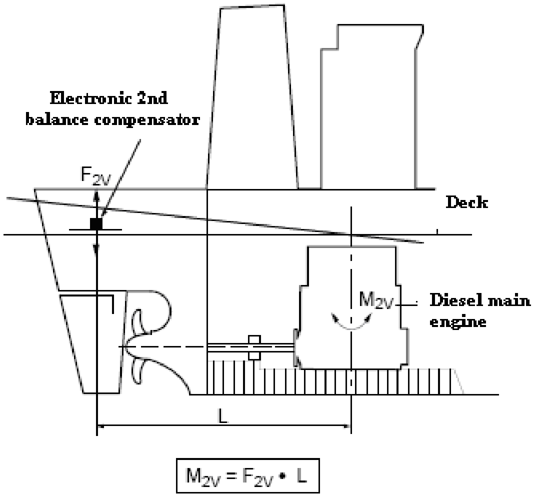


Figure 4.5.3 Diagram of Installation for Electronic 2nd Order Balance Compensator

4.5.4 Antivibration effect

The rational use of balance compensator may effectively control the adverse vibration induced by unbalanced moment of diesel engine to achieve the expected antivibration effect.

The measured vertical amplitude curve of the sixth cylinder cover of two-stroke cycle diesel engine is shown in Figure 4.5.4. The full line and dotted line in the figure shows the vertical amplitude curve before and after the installation of 2nd order balance compensator for the diesel engine respectively. As seen from the figure, the effect of balance compensator is obvious.

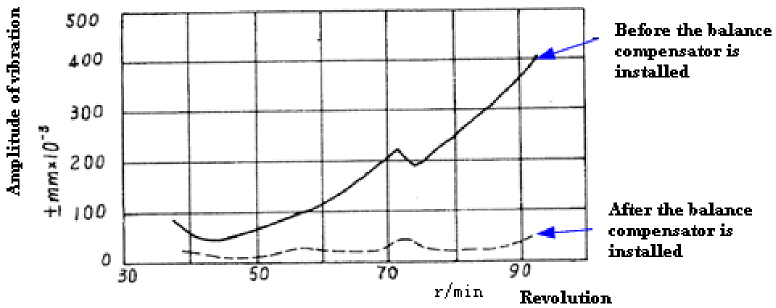


Figure 4.5.4 Measurement Results Before and After Installation of 2nd Order Balance Compensator

Chapter 5 VIBRATION OF DIESEL ENGINE FRAME

5.1 Introduction

5.1.1 Introduction

As mentioned in 4.1.2 of Chapter 4, the lateral moment acting on the diesel engine may induce engine frame transverse vibration. With the application of long stroke and super long stroke for marine two-stroke cycle diesel engine, the natural frequency of frame longitudinal vibration is minimized, frame longitudinal vibration is induced by the influences of shafting longitudinal vibration or alternating thrust of propeller. Furthermore, the frame vibration may become one of excitations of hull vibration. Currently, the frame vibration of two-stroke cycle diesel engine is still an innegligible issue to control the shipboard vibration.

Frame vibration may lead to the damage of diesel engine accessories, such as turbocharger support, and may also cause local vibration of engine room and double-bottom tank, etc. For a certain container ship with two-stroke cycle main diesel engine, due to strong longitudinal and transverse vibrations of engine frame, the value of vibrating acceleration occurred approximate to the resonance speed is so large that the vibrating acceleration of superstructure exceeds the allowable one.

The severe frame vibration may cause the tie rod bolts loosen and broken, turbocharger shaking in an unacceptable degree, vibration of superstructure being strongly, persons feeling uncomfortable, ship's operation being difficult, even the instruments in navigation bridge damaged. Therefore, the analysis and evaluation are required, and where necessary, effective measures are to be taken to control the frame vibration.

5.1.2 Types of frame vibration

(1) Types of frame vibration

Four types of frame vibration are: Type *H* transverse vibration, Type *X* transverse vibration, Type x transverse vibration and type *L* longitudinal vibration. Due to the fact that the natural frequency of Type *x* transverse vibration is higher, in general, it will not form the adverse Type x transverse vibration within the normal speed range, Type *x* vibration may not be further discussed. Type *H*, Type *X* and Type *L* vibration modes are shown in Figure 5.1.2.

(2) Type *H* transverse vibration

Type *H* transverse vibration (also called 0 node) is a vibration mode which the top of each cylinder moves left-and-right in-phase, shortly called Type *H* vibration.

(3) Type *X* transverse vibration

Type *X* transverse vibration (also called 1st node) is a vibration mode which the frame reverses around the centre of diesel engine, shortly called Type *X* vibration.

(4) Type *L* longitudinal vibration

Type *L* longitudinal vibration is a vibration mode which the frame moves around fore-and-aft direction, shortly called Type *L* vibration.

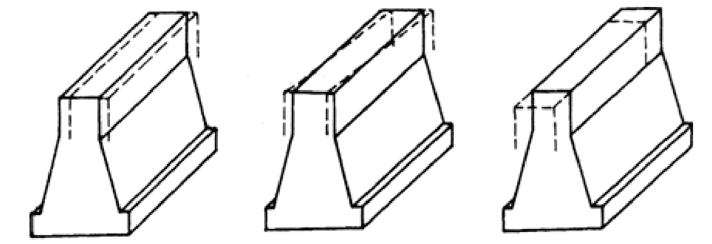


Figure 5.1.2 Vibration Modes of Engine Frame

5.2 Excitation Source

5.2.1 Excitation source of Type H vibration

(1) Exciting force of Type *H* vibration

The gas pressure and reciprocating inertial force have been analyzed according to 4.1.2, the horizontal force, i.e. lateral force P_H' decomposed by resultant of force P of gas pressure P_g and reciprocating inertial force on the cross head part, is shown in Figure 4.1.2, as follows:

$$\left.\begin{aligned} P_C' &= (P_g + P_j)/\cos\beta \\ P_T' &= P_C \sin(\omega t + \beta) = (P_g + P_j)\sin(\omega t + \beta)/\cos\beta \\ P_H' &= P tg\beta = (P_g + P_j) tg\beta \end{aligned}\right\} \tag{5.2.1}$$

Obviously, $P_H' Y = P_T R$

where: Y — distance between the center of crankshaft and center of cross head;

R — radius of crankshaft.

The exciting force inducing Type H vibration is the countertorque of output torque for diesel engine $P^*{}_H$:

$$P^*{}_H = P_T \frac{R}{Y} \tag{5.2.2}$$

In order to facilitate calculation, substituting $Y \approx L$ (L is the length of connecting rod) for formula (5.2.2):

$$P^*{}_H = \lambda P_T \tag{5.2.3}$$

The value of P_H' may be calculated according to formula (5.2.1) and the value of P_H^* is to be calculated according to formula (5.2.3), the former is the first method while the latter is the second method. The Fourier analysis is to be carried out for formula (5.2.1) is to be carried out so as to obtain the amplitude of lateral excitation C_{hv} (or called as harmonic coefficient of lateral force) and its initial phase angle θ_v at each harmonic order v:

$$P_H = P_{H0} + \sum_{V=1}^{\infty} C_{hv} \sin(\omega t + \theta_v) \tag{5.2.4}$$

where: P_{Ho} — mean value of P_H.

$$P_{Hv} = C_{hv} \sin(v\omega t + \theta_v)$$

P^*_{Hv} for multi-cylinder diesel engine is:

$$P_H^* = \sum_{i=1}^{Z} C_{hv} \sin(v\omega t - v\xi i + \theta_v) \qquad (5.2.5)$$

where: ξ_i — firing delay angle of i^{th} cylinder relative to the first cylinder;

Z — number of cylinders.

For homogeneous firing diesel engine, when $\nu = kZ$ (k =1, 2, 3……):

$$\sum_{i=1}^{Z} \sin(v\omega t - v\xi_i + \theta_v) = Z$$

when $\nu \neq kZ$:

$$\sum_{i=1}^{Z} \sin(v\omega t - v\delta_i + \theta_v) = 0$$

Hence, harmonic order of Type H vibration exciting force is only the integer multiples of number of cylinder (kZ), with the amplitude C^*_{hv} of:

$$C^*_{hv} = ZC_{hv} \qquad (5.2.6)$$

Similarly, Fourier analysis may be carried out for P_t according to formula (5.2.3), then the approximate value C_{hv} may be obtained directly by harmonic coefficient of tangential force in shafting torsional vibration calculation c_v multiplying λ:

$$P^*_{Hv} = \lambda c_v \sin(v\omega t + \theta_v) \qquad (5.2.7)$$

$$C^*_{hv} = Z\lambda c_v \qquad (5.2.8)$$

The amplitude of Type H vibration excitation C^*_{hv} may be obtained by formula (5.2.6) or formula (5.2.8) respectively.

Therefore, the scale of Type H exciting moment is related to firing order, harmonic order, power and speed.

(2) Exciting frequency of Type H vibration

The exciting frequency of Type H vibration is to be obtained by speed of main diesel engine multiplying harmonic order, e.g. when the rated speed of main diesel engine is 100 r/min, the 6th order Type H exciting frequency for diesel engine with six cylinders at the rated speed is 600 l/min.

5.2.2 Excitation source of Type X vibration

(1) Exciting force of Type X vibration

The ν th order exciting moment M_{Xvi} for the i th cylinder of diesel engine is to be obtained by lateral force multiplying the distance between engine body center to the centerline of various cylinders, as shown in Figure 5.2.2, i.e.:

$$M_{Xvi} = \frac{\pi D^2}{4} P_{Hvi} L_i \qquad (5.2.9)$$

where: P_{Hvi} — harmonic value of the ν th order lateral force for the i^{th} cylinder;

L_i — distance between centerline of the i^{th} cylinder and the center of engine body.

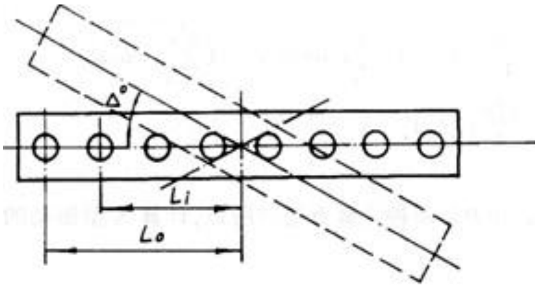


Figure 5.2.2 Exciting Moment of Type *X* Vibration

The formula (5.2.9) may also expressed as:

$$M_{Xvi} = \frac{\pi D^2}{4} L_o x_i C_{hv} \sin(v\omega t - v\delta_i + \theta_v) \qquad (5.2.10)$$

where: $x_i = L_i / L_0$;

L_0 — distance between the centerline of 1st cylinder (or z^{th}) and the engine body center.

The change of Type X torsional angle $\triangle$ is:

$$\Delta = \Delta_0 \sin(v\omega t + \theta_v - \xi) \qquad (5.2.11)$$

where: Δ_0 — amplitude of torsional angle;

ξ — lag angle of response.

Work of the v^{th} order exciting moment for all cylinders within a vibration period, E is:

$$E = \sum_{i=1}^{Z} \int_0^{2\pi} M_{xvi} \frac{d\Delta}{dt} dt$$

$$= \frac{\pi D^2}{4} C_{hv} L_v \Delta_0 \sqrt{(\sum_{i=1}^{Z} x_i \sin v\delta_i)^2 + (\sum_{i=1}^{Z} x_i \cos v\delta_i)^2} \sin(\varphi - \xi) \qquad (5.2.12)$$

$$\varphi = tg^{-1}\left\{\frac{\sum_{i=1}^{Z} x_i \sin v\delta_i}{\sum_{i=1}^{Z} x_i \cos v\delta_i}\right\}$$

When resonance occurs, the input power of exciting moment to the system will achieve the maximum value, E_{max}, and $\sin(\varphi - \xi) = 1$,

$$E_{max} = \frac{\pi D^2}{4} C_{hv} L_0 \Delta_0 \sqrt{(\sum_{i=1}^{Z} x_i \sin v\delta_i)^2 + (\sum_{i=1}^{Z} x_i \cos v\delta_i)^2} \qquad (5.2.13)$$

The equivalent value of exciting moment M_{xv} when the resonance occurs may be obtained by formula (5.2.13):

$$M_{xv} = \frac{\pi D^2}{4} C_{hv} L_0 \sqrt{(\sum_{i=1}^{Z} x_i \sin v\delta_i)^2 + (\sum_{i=1}^{Z} x_i \sin v\delta_i)^2}$$

$$= \frac{\pi D^2}{4} C_{hv} L_0 V_s \qquad (5.2.14)$$

Due to two methods to calculate exciting force of Type H vibration, P_{Hv} and P_{Hv}^*, two formulae to calculate the exciting moment for Type X vibration are as follows:

$$M_{xv} = \frac{\pi D^2}{4} C_{hv} L_0 V_s \quad (5.2.15)$$

$$M_{xv} = \frac{\pi D^2}{4} \lambda C_v L_0 V_s \quad (5.2.16)$$

$$V_s = \sqrt{(\sum_{i=1}^{Z} x_i \sin \nu \delta_i)^2 + (\sum_{i=1}^{Z} x_i \cos \nu \delta_i)^2} \quad (5.2.17)$$

The V_s mentioned above is a vector sum of exciting moment for Type *X* vibration.

For example: the firing order of two-stroke cycle diesel engine with eight cylinders is 1—8—3—4—7—2—5—6 and 1—6—5—2—8—3—4—7, respectively, the value of vector sum, V_s is shown in Table 5.2.2. As seen from the Table, when the harmonic order ν =3, ν =5 and ν =11, the vector sums of Type *X* exciting moments are the maximum values, i.e. the 3rd order, 5th order and 11th order Type *X* exciting moments are the maximum, which may induce severe Type *X* vibration.

Therefore, the scale of Type *X* exciting moment is related to firing order, harmonic order, power and speed of diesel engine.

Vector Sum of Exciting Moment of Type *X* Vibration for Certain Diesel Engine

Table 5.2.2

Harmonic order	Firing order		Harmonic order	Firing order	
	1-8-3-4-7-2-5-6	1-6-5-2-8-3-4-7		1-8-3-4-7-2-5-6	1-6-5-2-8-3-4-7
1	0.2562	0.4016	7	0.2562	0.4016
2	0	0	8	0	0
3	3.6049	3.6814	9	0.2562	0.4016
4	1.1429	0	10	0	0
5	3.6049	3.6814	11	3.6049	3.6814
6	0	0	12	1.1429	0

(2) Exciting frequency of Type *X* vibration

The exciting frequency of Type *X* vibration is to be obtained by speed of main diesel engine multiplying harmonic order, e.g. when the rated speed of main diesel engine is 100 r/min, the 3rd order Type *X* exciting frequency for the diesel engine with eight cylinders at the rated speed is 300 *l*/min.

5.2.3 Excitation source of Type L vibration

(1) Exciting force of Type *L* vibration

The secondary exciting caused by shafting longitudinal vibration and inhomogeneous axial thrust caused by propeller may induce Type *L* vibration through thrust bearing housing, and larger local longitudinal vibration may be caused at the fore and aft ends of engine frame. The exciting force may be calculated in accordance with formula (7.4.5) and formula (7.4.10) respectively.

(2) Exciting frequency of Type *L* vibration

If it is induced by propeller, the exciting frequency of Type *L* vibration is to be obtained by the speed multiplying number of blade, e.g. where the rated speed of main diesel engine is 100 r/min, and 4 propeller blades, the exciting frequency of Type *L* vibration at the rated speed is 400 *l*/min. If it is induced by shafting longitudinal vibration, the exciting frequency is to be obtained by resonance speed multiplying harmonic order of shafting longitudinal resonance, e.g., where the shafting longitudinal resonance speed of two-stroke cycle diesel engine with six cylinders is 70 r/min, the exciting frequency of Type *L* vibration is 420 *l*/min.

5.2.4 Relationship between exciting moment of frame transverse vibration and firing order

The scale of exciting force of frame transverse vibration and the induced frame transverse mode shape is closely related to the firing order of diesel engine, which reflects by vector sum of relative amplitude of engine C_{hv}^{*} and V_s.

As seen from Figure 5.2.4, the main harmonic order induces the Type *H* transverse vibration while the secondary harmonic order induces the Type *X* transverse vibration.

For the diesel engine with 10 cylinders, the original firing order is 1—9—5—6—2—10—4—3—8—7, which induces 5th order exciting moment within the normal speed range and causes the strong Type *X* resonance of main engine and the superstructure vibrates severely. As the firing order is changed to 1—8—7—3—5—9—4—2—10—6, the vector sum of 5th order exciting moment V_s calculated according to formula (5.2.17) is reduced from 3.49 to 0.33, this is obvious to improve the frame transverse vibration and hull vibration.

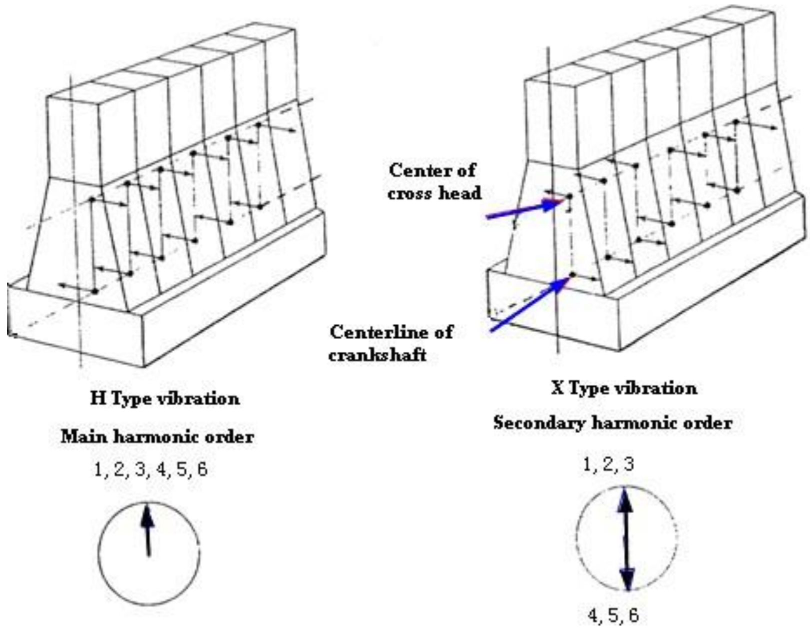


Figure 5.2.4 Relationship Between Transverse Excitation and Mode Shape of Diesel Engine

5.2.5 Values of Type H and Type X exciting moments

After the design of diesel engine is finished, the values of Type *H* and Type *X* exciting moments for related engine type provided by manufacturers, together with the unbalanced moments are to be referred for the vibration control design. Two types of engines are shown as follows:

(1) Example 1 – Type *H* and Type *X* exciting moments of Type K90MC two-stroke cycle diesel engine

The values of Type *H* and Type *X* exciting moments of Type K90MC two-stroke cycle diesel engine provided by MAN B&W is shown in Table 5.2.5-1, which may be found:

① for the diesel engine with four to nine cylinders, the harmonic order of Type *H* exciting moment which is equivalent to the cylinder number is the main one, the Type *H* exciting moment of diesel engine with more than seven cylinders is not larger, particular attention is to be given to the effects of Type *H* exciting moment for diesel engine with four to six cylinders;

② with the difference of firing order, harmonic order and cylinder number, the larger value of Type *X* exciting moment is distinct, particular attention is to be given to the effects of Type *X* exciting moment for diesel engine with more than seven cylinders.

Values of Type *H* and Type *X* Exciting Moments of Type K90MC Two-Stroke Cycle Diesel Engine Table 5.2.5-1

Number of cylinder	4	5	6	7	8	9	10	11	12
Firing order	1-3-2-4	1-4-3-2-5	1-5-3-4-2-6	1-7-2-5-4-3-6	1-8-3-4-7-2-5-6	1-6-7-3-5-8-2-4-9			1-8-12-4-2-9-10-5-3-7-11-6
Harmonic order	Type *H* exciting moment (kNm)								
1	0	0	0	0	0	0	0	0	0
2	0	0	0	0	0	0	0	0	0
3	0	0	0	0	0	0	747	352	0
4	2437	0	0	0	0	0	1018	830	0
5	0	2342	0	0	0	0	325	403	
6	0	0	1680	0	0	0	97	406	0
7	0	0	0	1257	0	0	659	577	
8	426	0	0	0	852	0	167	439	0
9	0	0	0	0	0	460	89	37	0
10	0	145	0	0	0	0	103	59	0
11	0	0	0	0	0	0	43	131	0
12	59	0	88	0	0	0	15	34	176
Harmonic order	Type *X* exciting moment （kNm）								
1	977	317	0	188	82	650	116	80	0
2	132	164	114	33	0	37	1	5	0
3	180	635	1148	1256	1922	2306	2517	3274	4091
4	0	125	963	2738	1112	1387	1977	2526	1927
5	302	0	0	215	3220	1066	438	2009	0
6	511	57	0	34	0	2310	1503	171	0
7	116	408	0	0	10	93	1743	180	0
8	0	242	168	13	0	45	181	1015	337
9	33	10	210	23	3	33	69	127	748
10	53	0	46	131	0	12	149	95	0
11	12	4	0	86	132	10	112	148	0
12	0	33	0	7	27	121	49	56	0

(2) Example 2 —Type *H* and Type *X* exciting moments of RT-flex96C Type two-stroke cycle diesel engine

The values of Type *H* and Type *X* exciting moments of Type RT-flex96C two-stroke cycle diesel engine provided by WÄRTSILÄ is shown in Table 5.2.5-2, which may be found:

① the Type *H* exciting moment of diesel engine with more than seven cylinders is not larger, particular attention is to be given to the effects of Type *H* exciting moment for diesel engine with six cylinders;

② with the difference of firing order, harmonic order and cylinder number, the larger value of Type *X* exciting moment is distinct, particular attention is to be given to the effects of Type *X* exciting moment for diesel engine with more than seven cylinders.

Values of Type *H* and Type *X* Exciting Moments of Type RT-Flex96c Two-Stroke Cycle Diesel Engine Table 5.2.5-2

Number of cylinder	6	7	8	9	10	11	12	14
Power (kW）	34320	40040	45760	51480	57200	62920	68640	80080
Harmonic order	Type *H* exciting moment (kNm)							
1	0	0	0	0	59	0	0	22
2	0	0	0	0	2	0	0	1
3	0	0	0	0	73	0	0	34
4	0	0	0	0	253	0	0	969
5	0	0	0	0	455	0	0	180
6	2088	0	0	0	203	0	0	75
7	0	1596	0	0	191	0	0	141
8	0	0	1082	672	32	0	0	69
9	0	0	0	0	8	0	0	41
10	145	0	0	0	420	0	0	156
11	0	0	0	0	15	279	0	22
12	0	88	0	0	1	0	214	3
Ha	Type *X* exciting moment (kNm)							
1	0	314	364	1082	13	751	0	24
2	1083	314	0	354	261	284	0	1
3	951	1041	1483	1799	2229	2695	3228	4223
4	1232	3501	1423	1774	1750	673	2464	2117
5	0	256	3570	1269	272	1687	0	751
6	0	35	0	2258	679	880	0	505
7	0	0	29	104	1983	174	0	674
8	183	14	0	49	200	1348	366	38
9	249	28	9	0	55	25	0	121
10	61	173	0	16	36	11	0	123
11	0	105	149	12	15	0	0	450
12	0	7	29	122	18	5	0	25

5.3 Frame Vibration Calculation

5.3.1 Calculation of natural frequency for frame transverse vibration

The natural frequency of frame transverse vibration may be calculated by simple estimation method (semi-empirical method).

The engine frame is to be simplified to the calculation model as shown in Figure 5.3.1-1.

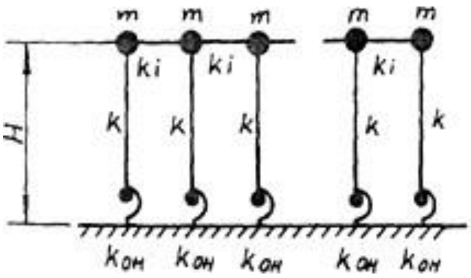


Figure 5.3.1-1
Simplified Model of Frame

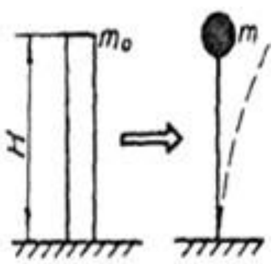


Figure 5.3.1-2
Equivalent Concentrated Mass of Frame

Symbols in the Figure are as follows:

m — equivalent to the concentrated mass of one cylinder;

k — equivalent to the stiffness of frame for one cylinder;

$k_{\theta H}$ — lateral torsional stiffness installed at the bottom;

k_i — connecting stiffness between cylinders;

H— height.

The equivalent mass *m* means that the homogeneously distributed mass of a plating of which one end is fixed is to be changed to the equivalent concentrated mass, m of a plating with one end fixed according to the principle of equivalent energy and similar deflection curve, as shown in Figure 5.3.1-1 and Figure 5.3.1-2, and calculated by the following formula:

$$m \approx 0.25 m_0 \qquad (5.3.1)$$

where: m_o — mass of grillage.

The natural frequencies of Type *H* and Type *X* vibrations are calculated as follows:

$$f_H = \frac{1}{2\pi}\sqrt{\frac{k_e}{m}} \quad \text{Hz} \qquad (5.3.2)$$

$$f_X = \frac{1}{2\pi}\sqrt{\frac{k_e + a_x k_i}{m}} \quad \text{Hz} \qquad (5.3.3)$$

where: k_e — synthesis of stiffness, to be calculated by:

$$k_e = k + \frac{1}{1 + H^2 (\frac{k}{k_{\theta H}})}$$

a_x — coefficient related to the number of cylinder *Z*, e.g. a_x of Sulzer RND diesel engine is given in Table 5.3.1.

The coefficients used for simply estimation method, k_e and k_i are to be obtained by backstepping on the basis of actual measurement, e.g. according to actual measured value f_H and estimated value *m*, substituting for formula (5.3.2) to backstep its synthesis of stiffness k_e. According to the actual measured values of f_H and f_X for various numbers of cylinder, substituting them for formulae (5.3.2) and (5.3.3) to backstep the connection stiffness k_i between cylinders and coefficient a_x.

From formulae (5.3.2) and (5.3.3), the following conclusions may be obtained:

(1) The natural frequency of Type *H* vibration is not related to number of cylinders *Z*;

(2) The natural frequency of Type *H* vibration is related to number of cylinders *Z*, and reduced obviously with the increase of *Z*.

Value of a_x **Table 5.3.1**

Number of cylinder Z	6	7	8	9	10	11	12
a_x	0.268	0.198	0.152	0.121	0.098	0.081	0.008

5.3.2 Calculation for vibration response of three-dimensional finite element model

(1) Basic data

Due to the symmetry of diesel engine, half side is taken for modeling, e.g., Type 4RLA90 diesel engine:

Power: *Ne* = 10000 kW;

Speed: *n* = 90 r/min;

Total mass: 578 t;

Firing order: 1—4—2—3.

(2) Modeling of finite element

To establish a coordinate network and convert the finite element model to six *i* -planes along the length direction, six *i* -planes along the vertical direction and three *k*-planes along the transverse direction, as shown in Figure 5.3.2-1.

The finite element characteristics of plating and girder are determined by the relevant drawings of diesel engines. The through bolts for connecting cylinder, engine frame and pedestal are not grouped to the finite elements for which are assumed as rigid connection. All moving parts and turbocharger are excluded in the model.

Mass of diesel engine is to be modeled by the point mass on the nodes. The mass of large type structural elements is to be determined by their actual positions, the mass of the other parts is distributed in linear on each node along the length direction and vertical direction.

The finite element characteristics and mass distribution of Type 4RLA90 engine frame is shown in Figure 5.3.2-2.

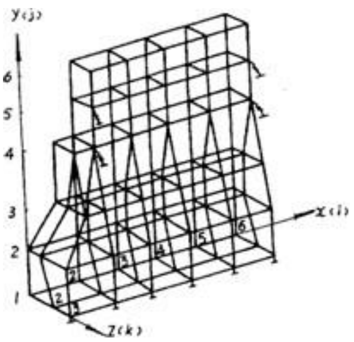


Figure 5.3.2-1
Three-dimensional Finite Element Model

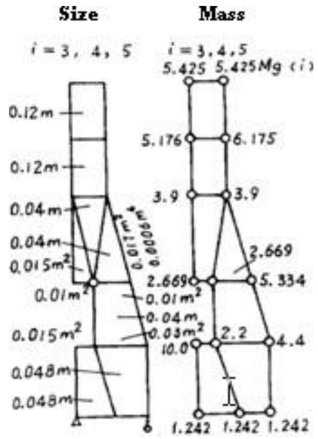


Figure 5.3.2-2
Typical Mass Distribution Model

(3) Principle of modeling

Nodes excluding cylinders are to be modeled by thin plate element with linear displacement and cylinders are to be modeled by thin shell element permitted for displacement and whirling. In order to obtain the asymmetric vibration mode, whirling around transverse axis and displacement along *x* axis and *y* axis in way of centerplane *k*=1, are to be fixed. The external base of model consists of six elastic supports with the stiffness determined by connection conditions of engine pedestal and test bed.

(4) Distribution principle of exciting forces

The amplitude of exciting force inducing Type *H* vibration at the rated working condition for the engine is $P_{H1} = 96.8$ kN and $P_{H8} = 13.2$ kN, the corresponding frequency is $f_4 = 6Hz$ and $f_8 = 12H_z$. The amplitude of exciting force inducing Type *X* vibration is $P_{H2} = 299.4$ kN and $P_{H6} = 148.3$ kN, the corresponding frequency is $f_2 = 8$ Hz and $f_6 = 9$ Hz.

The distribution of these exciting forces on finite element model is shown in Figure 5.3.2-3, the distribution principle is to decompose the force acting on the cross head to the node closer to the force. The point of action for exciting force is in way of connection for each cylinder, other than the centerline of each cylinder, and such distribution meets the condition of equivalent moment.

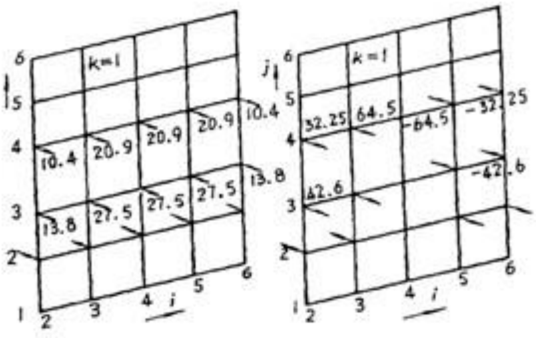


Figure 5.3.2-3　Distribution of Exciting Forces on Nodes

(5) Calculation of natural frequency

The value of natural frequency of vibration is to be calculated according to the finite element model established according to (2) to (4) mentioned above.

(6) Calculation of vibration response

The amplitude of forced vibration is to be calculated by superposition method for mode shape, in general, the frame response is only to use 1 to 5 node natural frequency. Based on the actual measured data, the damping ratio of frame without supporting may be taken as 0.03 to 0.04.

The response of constant force is to be first calculated based on exciting force at the rated working condition, then correction is to be carried out according to the actual exciting force. The frame is a complicated component, even in an ideal state, the amplitudes of Type *H* vibration for various cylinders are different, the amplitudes for the fore and aft cylinders are not the same for Type *X* vibration.

5.3.3　Finite element model of vibration system between frame and hull structure

A vibration system is formed between engine frame and hull structure, which is also called coupled vibration. Due to large mass difference between the engine and hull, assuming an elastic support to represent the effects on hull as the vibration of engine frame is analyzed.

For ships with the engine room located in the middle, the stiffness of support for main engine is primarily determined by bending stiffness of double-bottom, therefore, a finite element model of conventional double-bottom with peripheral support is sufficient to substitute hull structure. However, for ships with the engine room located in the stern, the conditions are complicated.

In the case of a bulk carrier with the engine room located in the stern, the lowest part of engine room structure to the first layer of platform is to be taken into account.

Both of the main diesel engine and engine room structure are taken half for modeling, the finite element model is converted to 13 i -planes along the length direction, seven i -planes along the vertical direction and four *k*-planes along the transverse direction, as shown in Figure 5.3.3. The model of main diesel engine is more simple than that on test bed.

The mass of main diesel engine is distributed on each node of model based on the principle in 5.3.2. The mass of bottom structures in engine room is distributed in the nodes at bottom. The auxiliary engine, equipment, joints and piping system located above and below the double-bottom respectively are to be considered as additional mass of inner bottom plate element.

The vibration response is to be calculated as same as 5.3.2(6). In the calculation, damping ratio may still be taken as 0.03 to 0.04.

The mass of entrained water m_W is to be divided into several sections and estimated by:

$$m_W = \rho \frac{\pi}{4} b^2 J \qquad (5.3.4)$$

where: ρ = 1025 kg/m^3 — density of water;

b — half breadth of inner bottom plate in way of waterline;

J— for immersed plate at the side of three-dimensional flow field, the correction coefficient for curve-type vibration J is taken as 0.8.

5.3.4 Calculation of longitudinal vibration

Finite element model is to be established for longitudinal vibration by reference of 5.3.1. The stiffness of connection between foundations is to take the value of longitudinal torsional stiffness $K_{\theta L}$ so as to calculate the natural frequency of frame longitudinal vibration.

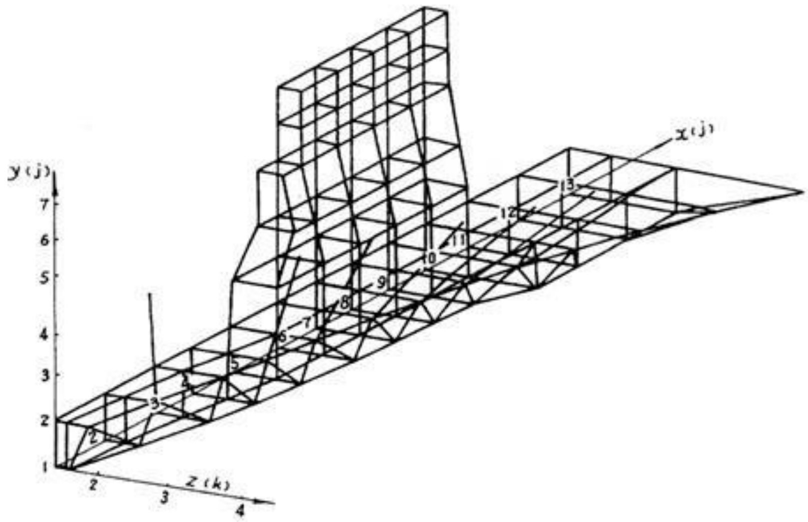


Figure 5.3.3 Finite Element Model of Main Diesel Engine and Hull Structure

5.4 Relationship among Engine Frame Vibration, Hull Vibration and Shafting Vibration

5.4.1 Introduction

Where the diesel engine is installed onboard, engine frame, double-bottom and bulkheads of engine room compose a complicated vibration system and the natural frequency will be reduced correspondingly. Meanwhile, shafting vibration may induce frame vibration, further to cause local vibration of hull and superstructure vibration. On the contrary, hull vibration may also lead to frame vibration.

5.4.2 Relationship between frame vibration and hull vibration

(1) Frame transverse vibration may induce local vibration of hull.

Where the frame induces Type *H* and Type *X* vibrations, the amplitude of vertical or transverse vibration of engine foundation and stern transmitted by engine frame is only 0.2 to 0.3 times the amplitude of transverse vibration on the frame top. However, in general, the effect of frame transverse vibration on the hull is not large, where local resonance occurs with the hull, it will seriously influence ship's seaworthiness.

(2) Hull transverse vibration may induce frame transverse vibration.

Hull transverse vibration including torsional composition has obvious influence on initiation of frame transverse vibration.

① Where the hull occurs low-frequency vibration, the engine frame may be regarded as a composition of hull structure and vibrate together with the hull.

② Where the hull occurs high-frequency vibration, the amplitude of induced transverse vibration on frame top will be about twice of the amplitude of stern transverse vibration, i.e. hull transverse vibration has obvious effect on frame transverse vibration.

(3) Frame longitudinal vibration may induce superstructure longitudinal vibration

Where the frame occurs Type *L* vibration, larger longitudinal vibration is caused at frame top, furthermore, superstructure longitudinal vibration is induced.

5.4.3 Effects of hull double-bottom stiffness on natural frequency of frame vibration

After diesel engine is installed onboard ships, due to the influence of double-bottom stiffness, the natural frequencies of Type *H* and Type *X* vibrations for engine frame will be reduced appropriately. Comparing the natural frequency measured on the test bed with that measured actually onboard, the transverse vibration frequency of engine frame after the engine installed onboard may be reduced about 10%, so the testing results on test bed may be used to calculate the natural frequency of transverse vibration for engine frame after the installation of engine.

For Type *L* longitudinal vibration of engine frame, due to the weak longitudinal stiffness of double-bottom, the natural frequency of longitudinal vibration for engine frame measured on test bed is much higher than that measured after installation onboard, with a difference above 30%. A resonance speed approximate to the normal speed range after installation onboard is likely to occur by the frame longitudinal vibration not being a problem on the test bed, particularly for engines with four or five cylinders, Type *L* longitudinal vibration with the larger amplitude of z^{th} order harmonic equivalent to the number of cylinders may occur.

Diagram of measurement results of Type *L* longitudinal vibration for a marine diesel engine is shown in Figure 5.4.3 to reflect the vibration of double-bottom.

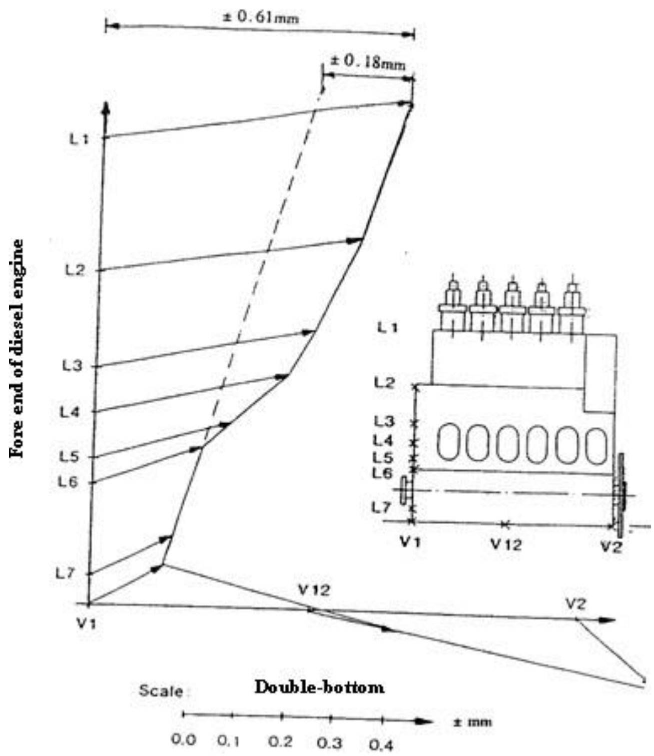


Figure 5.4.3 Diagram of Measurement Results of Type *L* Longitudinal Vibration for a Marine Diesel Engine

5.4.4 Frame vibration likely occurred by shafting vibration

(1) For diesel engines with less number of cylinders, where the shafting is designed to be short and thick, the alternating thrust of propeller induced by shafting torsional vibration may achieve about 50% of average thrust of propeller and aggravate Type *L* frame vibration.

(2) The torsional – longitudinal vibration caused by shafting may induce Type *L* frame vibration and further lead to superstructure vibration.

(3) The longitudinal vibration caused by shafting may induce Type *L* frame vibration and further lead to superstructure vibration.

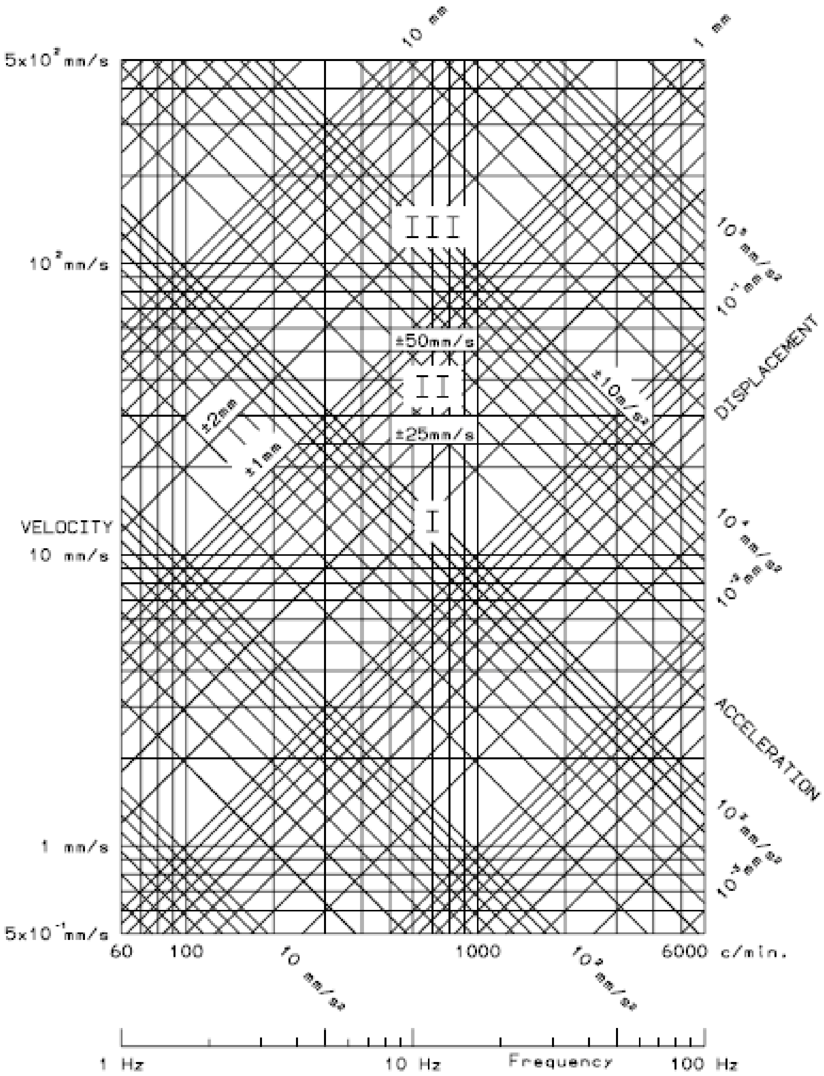
5.5 Criteria

5.5.1 The displacement or velocity of vibration for low-speed diesel engines are to be generally controlled within the range of requirements in Table 15.4.3 of Chapter 15.

5.5.2 The measuring position and direction of frame vibration is to be in compliance with the requirements in Table 14.4.1 of Chapter 14.

5.5.3 The criteria given by the manufacturers or other relevant standards may be adopted. Criteria of two-stroke cycle diesel engines made by MAN B&W and WÄRTSILÄ are shown in Figure 5.5.3-1 and Figure 5.5.3-2 respectively, which are applicable to the measured results of speeds corresponding to different working conditions of main engine.

5.5.4 Due to the fact that the measured results on test bed do not reflect the conditions after the installation of engine onboard, unless otherwise specified, the measuring results on test bed may not be regarded as the basis of inspection and acceptance by the both contracted parties.

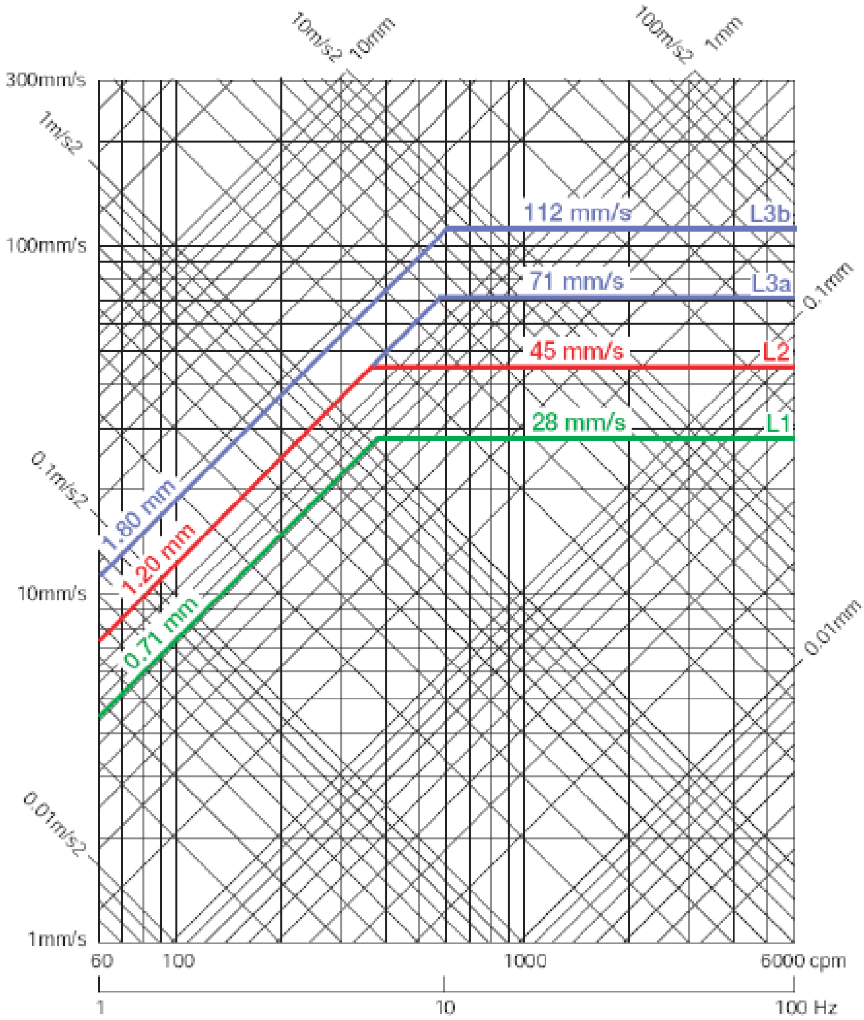


Zone I: Acceptable;

Zone II: Generally, the vibration will not endanger diesel engine. However, if it is serious, the vibration response may lead to damage of structural members connecting with diesel engine.

Zone III: Unacceptable.

Figure 5.5.3-1 Limiting Values of Single Harmonic Order Vibration for Diesel Engines



L1 – engine body, etc.;
L2 – platform girders, etc.;
L3 – high-pressure oil pipe (only for RT-flex).

Figure 5.5.3-2 Acceptable Vibration of Two-stroke Cycle Diesel Engines Made by WÄRTSILÄ (RMS)

5.6 Precautions

5.6.1 Evaluation of frame vibration conditions after installation of engine onboard

(1) Evaluation of design vibration

In the initial design, the natural frequency of frame vibration for the corresponding product types and the corresponding transverse exciting moments are to be obtained from the diesel engine manufacturers for carrying out evaluation of frame transverse vibration. As seen from the requirements of 5.3, it is difficult to accurately calculate the natural frequency of frame vibration for main diesel engines. Hence, it is recommended to install a transverse supporting on the top of main diesel engine in order to prevent or minimize frame transverse vibration, which is a general and effective antivibration measure.

In general, damper of shafting longitudinal vibration is to be provided at the free ends of diesel engine by the two-stroke diesel engine manufacturers to prevent Type *L* frame vibration which further leads to hull vibration, therefore, the Type *L* frame vibration induced by shafting longitudinal vibration has no problem. However, thrust force of propeller may cause frame longitudinal vibration, so longitudinal supporting is to be generally installed at the fore end of top of engine frame.

(2) Recommendations by MAN B&W

It is strongly recommended by MAN B&W that ships installed with two-stroke cycle diesel engines with the diameter of cylinder more than 26 cm are to be provided with transverse supporting for engine frames so as to effectively control the frame transverse vibration.

(3) Recommendations by WÄRTSILÄ

WÄRTSILÄ provides the countermeasures of frame transverse vibration for ships installed with Sulzer Type RT-flex96C two-stroke cycle diesel engines as shown in Table 5.6.1. Where:

A – compensating measures needed;

B – compensating measures likely;

C – compensating measures not relevant.

Countermeasures of Frame Transverse Vibration **Table 5.6.1**

Number of cylinder Z	6	7	8	9	10	11	12	14
Transverse supporting	B	C	A	B	B	A	B	A
Longitudinal supporting	C	C	C	C	C	C	C	C

5.6.2 Antivibration supporting installed on frame top

(1) Function of vibration damping for frame antivibration supporting

The frame antivibration supporting may increase stiffness of engine body, enhance natural frequency and avoid the low harmonic order resonance approximate to the rated speed, further to greatly reduce the amplitude of engine frame. Meanwhile, friction-type and hydraulic-type supporting has greater damping function to reduce the amplitude of whole system. The antivibration effects of different supporting are shown in Figure 5.6.2-1.

After the antivibration supporting is provided between the top of main engine frame and port and starboard, the amplitude of frame transverse vibration may generally be reduced about 50% and natural frequency will be raised 5% to 50%.

The longitudinal antivibration supporting installed between fore end of main engine and bulkhead of engine room may also reduce the amplitude of frame longitudinal vibration above 50% and raise the natural frequency about 25%.

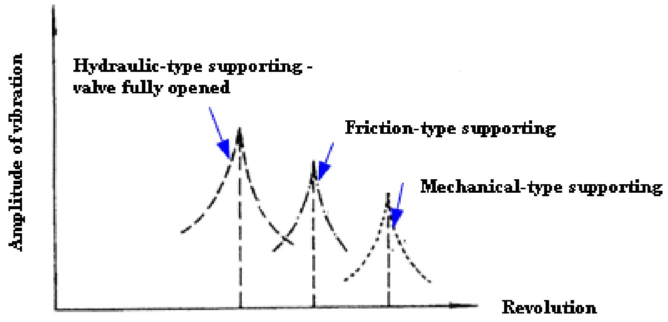


Figure 5.6.2-1 Effects of Supporting to Transverse Vibration

(2) Type of frame supporting

There three types of frame supporting as follows:

① Friction-type supporting:

The cross sectional shape of friction-type supporting is generally to be of U-shape, one end connects with engine frame through bolts while one piece of reed is inserted between two side plating at the other end of U-shaped girder to for a friction-type connection with the hull, the coefficient of friction is about 0.4. A torque bolt is to be used to adjust the tensioning force between friction plates to cause certain static friction force. Where the force on the girder is less than static friction force, the function of friction-type supporting is similar to mechanical-type supporting to increase the stiffness of assembly of engine body connected with girder grillage of engine room, the natural frequency is raised and amplitude is reduced. However, where the force on the girder exceeds static friction force, relative movement is caused between the side plating and reed, then partial energy of vibration system will convert to thermal energy, which is similar to a damper, to slightly raise the natural frequency and reduce the amplitude.

When the ship is navigating at sea, the friction-type supporting is to be adjustable to apply to various loads and structural deformation caused by ship's operation.

For the multi-cylinder diesel engine with 9 to 12 cylinders, the friction-type supporting may has sound antivibration effects.

Friction-type supporting has two structure types. U-shaped friction-type supporting is shown in Figure 5.6.2-2 and double-rod-type friction supporting is shown in Figure 5.6.2-3.

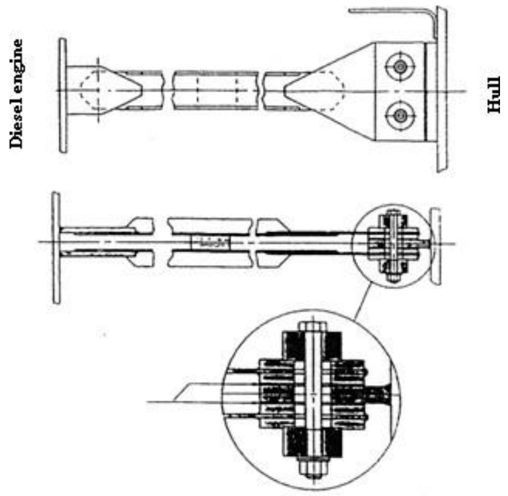


Figure 5.6.2-2 U-shaped Friction Supporting

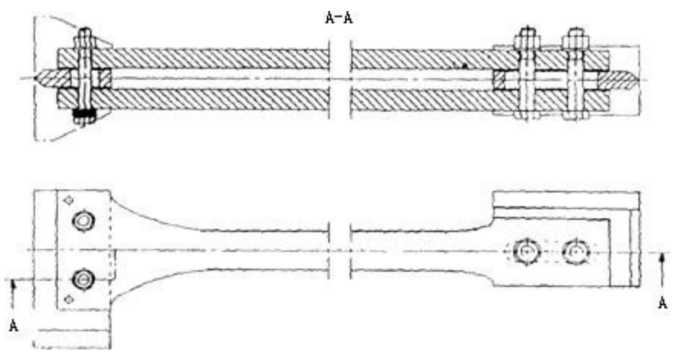


Figure 5.6.2-3 Double-rod-type Friction Supporting

② Hydraulic-type supporting

The hydraulic-type supporting consists of one energy storage device full of nitrogen, one throttle valve fitted with pressure gauge, one hydraulic cylinder fitted with differential piston which fixed on the hull and one piece of rod. A typical hydraulic-type supporting is shown in Figure 5.6.2-4.

The stiffness of hydraulic system may be changed by the adjustment of throttle valve. Where the throttle valve is fully closed, due to the incompressibility of oil, the characteristic of hydraulic-type supporting is then similar to that of the mechanical-type supporting. The supporting with throttle valve fully closed is not applicable for hull deformation with low-frequency vibration.

Where the throttle valve is fully opened, due to the fact that the change of pressure for hydraulic system induced by vibration is small, the force on the rod is similar to the normal value and only has a slight change when the hull deformation is larger. There has no influence on the natural frequency of frame, but due to damping function, the damping ratio may be triple for the frame without supporting, the amplitude will be reduced.

Where the throttle valve is partially opened, the vibration of piston means the oil pressure in cylinder is changed with the same frequency, due to the absorption of throttle valve, the high-frequency pressure change will unlikely reach to the hydraulic energy storage device. Corresponding to the full close condition of throttle valve, the performance of device is like a low stiffness spring, with the change of opening status for the throttle valve, the spring constant also changes within a certain range.

The final adjustment of throttle valve is to be finished at sea trial.

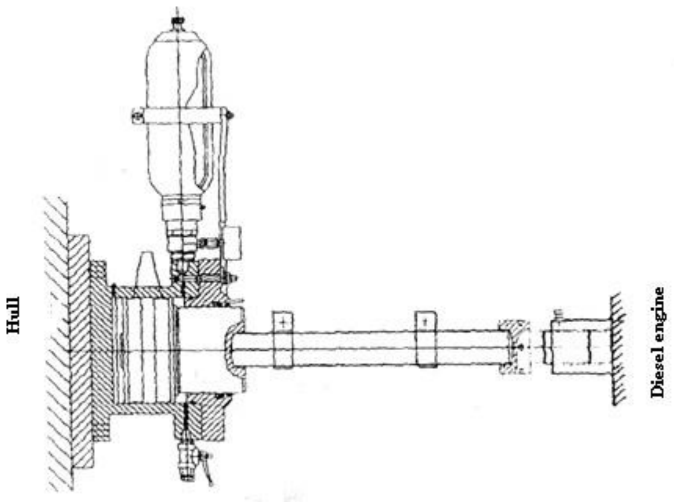


Figure 5.6.2-1 Hydraulic-type Supporting

③ Mechanical-type supporting

The mechanical-type supporting is shown in Figure 5.6.2-5. The one end of rod for mechanical-type supporting is to be freely inserted in the cylindrical support on the frame while the other end is to be contacted with hull by threaded sleeve, the rotational threaded sleeve may adjust the pre-tightening force of rod. The pre-tightening force is to be large enough to ensure that the static deformation of rod is more than the amplitude of transverse vibration for engine frame.

The mechanical-type supporting may obviously increase stiffness of main engine to raise the natural frequency and reduce the amplitude of frame. On the other hand, partial vibration energy of frame will be transmitted to hull through the supporting, which may aggravate the hull vibration.

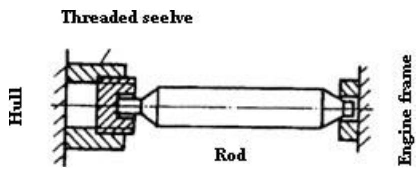


Figure 5.6.2-5 Mechanical-type Supporting

(3) Arrangement of supporting

Where the antivibration supporting is used, the correction of its arrangement and installation is to greatly influence the effect.

① Transverse supporting: lateral supporting is to be installed on the engine top to prevent or minimize transverse vibration of frame. In general, lateral supporting is to be so arranged on one side of diesel engine, or so installed at the side of exhaust pipe of engine or so installed at the side of fuel oil of engine that the engine has a certain freedom in the transverse direction; or installed at both sides of engine. Such arrangement of lateral supporting is shown in Figure 5.6.2-6.

② Longitudinal supporting: longitudinal supporting is to be installed at the fore end of frame top for diesel engine to prevent or minimize the Type *L* longitudinal vibration of the frame.

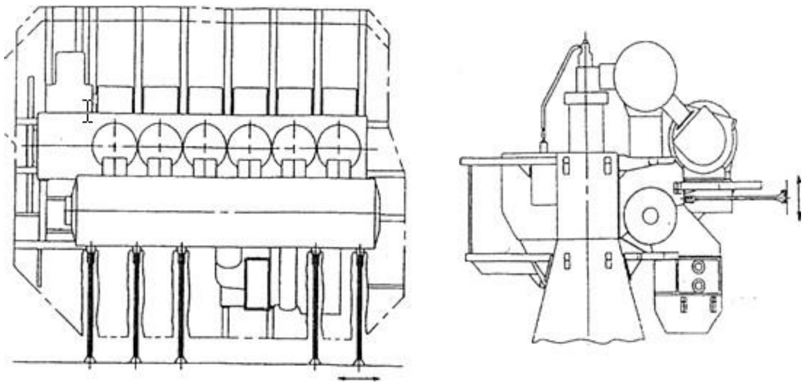


Figure 5.6.2-6 Arrangement of Lateral Supporting

③ Influence of hull stiffness on supporting effect: After antivibration supporting is provided, the additional stiffness of engine frame depends on the supporting stiffness, and also is related to the hull stiffness and installation position. Where the supporting stiffness exceeds 10^6 kN/m, the natural frequencies of Type *H*, Type *X* and Type *L* are to basically remain unchangeable, as shown in Figure 5.6.2-7.

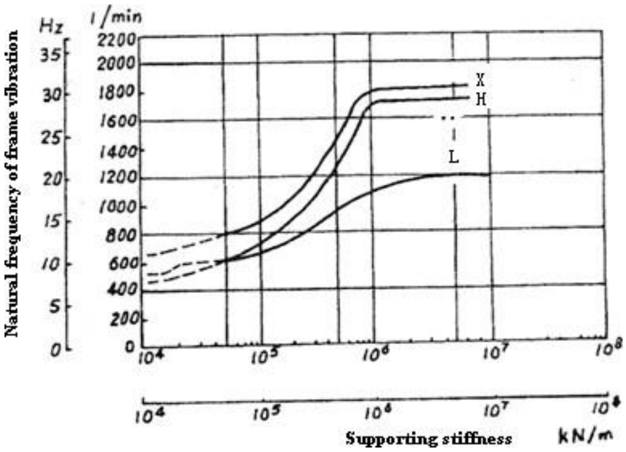
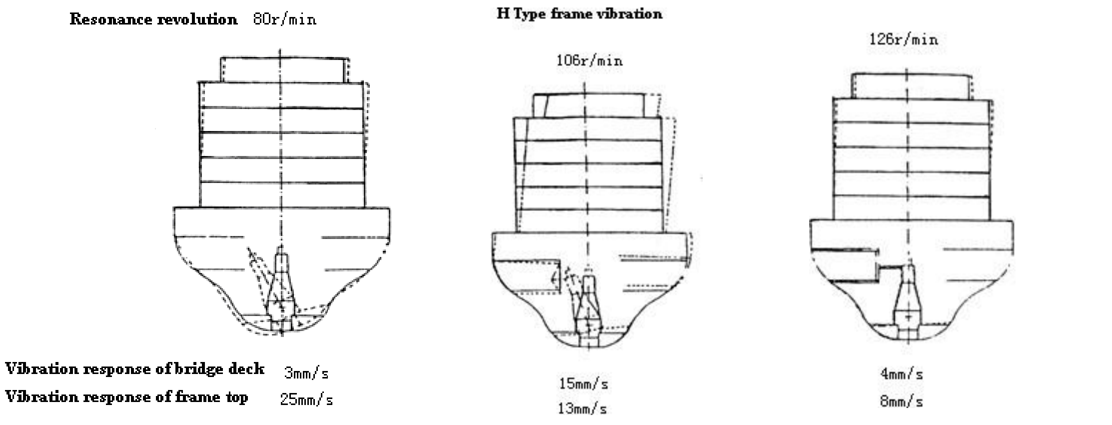


Figure 5.6.2-7 Relationship between Supporting Stiffness and Frame Natural Frequency

④ The hull connecting with supporting is to have sufficient stiffness: where the supporting is not installed correctly, it can not prevent frame vibration, and then deteriorates the vibration.

On a certain cargo ship, lateral supporting was provided for the main diesel engine, but installed on the partition of oil tank, due to weak stiffness of such partition, severe vibration occurred for the engine room and even the whole ship during the ship's navigation. After correct and rational installation of supporting, the problem was solved. Three different cases for connection between supporting and hull are shown in Figure 5.6.2-8. Where the transverse supporting is not installed for frame, the resonance speed of Type *H* vibration is 80 r/min, vibration velocity of engine top is 25 mm/s and vibration velocity of bridge deck is 3 mm/s. Where transverse supporting is installed but not at the oil tank, the resonance speed is raised to 106 r/min, vibration velocity of engine top is reduced to 13 mm/s but vibration velocity of bridge deck is raised to 15 mm/s. Where the transverse supporting is installed in way of stiffness of oil tank, the resonance speed is raised to 126 r/min, vibration velocity of engine top is reduced to 8 mm/s and vibration velocity of bridge deck is reduced to 4 mm/s, which is compliance with the relevant requirements.



Without transverse supporting | Transverse supporting in way of the position with insufficient stiffness | Transverse supporting in way of stiffness

Figure 5.6.2-8 Effect of Connection between Supporting and Hull on the Vibration

(4) Strength check of supporting rod

In the ship's design, the strength for mechanical-type and hydraulic-type supporting rods are to be checked. The force on the rod is to be determined depending on the maximum torque which may be induced by the selected engine type, Type *H* and Type *X* vibration, as well as the distance between frame top supporting and foundation of main diesel engine.

5.6.3 Electronic compensator provided for diesel engine

Where the diesel engine can not be provided with a transverse supporting for its frame, in order to prevent or minimize the effect of frame vibration on hull vibration, an electronic compensator may be installed to effectively control the severe Type *H* frame vibration induced by diesel engine.

Chapter 6 TORSIONAL VIBRATION OF SHAFTING

6.1 Introduction

6.1.1 Torsional vibration of shafting

If torsional deformation of shafting system occurs around its longitudinal axis under periodical torque excitation of diesel engine, bearing force for propeller, etc., it is called as torsional vibration of shafting.

For characteristics of torsional vibration for ship's propulsion shafting, the shafting may be regarded as a discrete system consisting of several concentrated masses and connected with torsional spring without mass, this system has several natural frequencies of torsional vibration. The gas pressure in cylinder of diesel engine and inertial force induced by reciprocating parts as of tangential component acting on crank pin through connecting rod may be decomposed to numerous tangential harmonic forces with different frequency and different amplitude and phase position, and the propeller also induces periodical tangential exciting force in inhomogeneous flow field. Where the frequency of a certain excitation is same or similar to the natural frequency of torsional vibration for certain node of propulsion shafting, a shafting torsional vibration phenomenon is caused.

For propulsion shafting of diesel engine, the exciting moment of shafting torsional vibration is mainly the gas pressure of diesel engine, however for turbine propulsion shafting and electric propulsion shafting, it is mainly the bearing force for propeller blade order.

For international navigating in ice strengthening areas, where resonance speed of 1st node blade order occurs within (0.8 to 1.2) rated speed range, influence of torque excitation for maximum ice blocks is to be evaluated.

The calculation software of shafting torsional vibration developed by ISC in 1980's has been updating and improving constantly and may be applied to design for shafting torsional vibration, plan approval and evaluation for shafting fault analysis.

6.1.2 Hazard of shafting torsional vibration

Vibration torque/stress induced by torsional vibration is alternating, where the shafting has operated at resonance speed or non-resonance speed with larger vibration torque/stress for a long time, the components of shafting may cause fatigue and damage, even lead to severe torsional vibration:

(1) twist-off of crankshaft, camshaft, thrust shaft, intermediate shaft, screwshaft and tube shaft;

(2) impact of teeth, pitting corrosion of teeth surface and broken teeth for reduction gearbox;

(3) cutting-off of connecting bolts for coupling and tearing of rubber coupling;

(4) quickly weardown of engine parts;

(5) unallowable voltage fluctuation output by diesel engine generating set;

(6) torsional – longitudinal vibration is caused by torsional vibration, which may further induce frame longitudinal vibration, double-bottom vertical vibration, hull girder vertical vibration, superstructure longitudinal vibration, local vibration, etc., through double-bottom.

6.2 Calculating Models

6.2.1 Basic principle of shafting torsional vibration

The equation of motion for forced damped vibration with single mass is:

$$I\ddot{\varphi}+C\dot{\varphi}+K\varphi=M\sin\omega t \qquad (6.2.1)$$

For torsional vibration system for forced damped vibration with multiple masses of n, the equation of motion for any mass at the certain circular frequency ω is:

$$I_k\,\ddot{\varphi}_k+C_k\,\dot{\varphi}_k+C_{k-1,k}(\dot{\varphi}_k-\dot{\varphi}_{k-1})+C_{k,k+1}(\dot{\varphi}_k-\dot{\varphi}_{k+1})+K_{k-1,k}(\varphi_k-\varphi_{k-1}) \qquad (6.2.2)$$
$$+K_{k+1,k}(\varphi_k-\varphi_{k+1})=M_k\sin(\omega t+\psi_k)$$

The φ_k in equation (6.2.2) is as follows:

$$\varphi_k=A_k\sin(\omega t+\varepsilon_k) \qquad (6.2.3)$$

where: φ_k, $\dot{\varphi}_k$, $\ddot{\varphi}_k$ — angular displacement, angular velocity and angular acceleration of mass k;

I_k — moment of inertia for mass k;

C_k — damping coefficient of mass k;

$K_{k-1,k}$ — torsional stiffness of shaft between mass (k -1) and mass k;

M_k — value of exciting moment acting on mass k;

ψ_k — initial phase angle of exciting moment;

ω — circular frequency of exciting moment;

t — time;

A_k — amplitude;

ε_k — phase angle.

For branch system, except the masses of branch points, the equations of motion for other masses are consistent with equation (6.2.2), but the equation for branch mass k is:

$$I_k\,\ddot{\varphi}_k+C_k\,\dot{\varphi}_k+\sum_{j=1}^{r}[C_{k,j}(\dot{\varphi}_k-\dot{\varphi}_j)+K_{k,j}(\varphi_k-\varphi_j)]=M_k\sin(\omega t+\psi_k) \qquad (6.2.4)$$

where: r — number of branch in way of point of k;

j — coefficient of j shaft connecting with the point k;

Other symbols are as same as above-mentioned.

6.2.2 Calculation method of response

(1) Amplification coefficient method

The amplification coefficient method is an empirical calculation way to calculate the response of torsional vibration. In calculation, assuming that the vibration type of forced vibration as resonance occurs is similar to that of free vibration with the same frequency, the energy of input system for exciting moment is fully consumed on damping for all components of the system, the resonance amplitude of 1st mass is to be determined on the basis of balance principle between exciting moment power and total damping power, then stresses or torques of various sections of shafting are to be calculated according the corresponding mode shapes.

For the most shafting systems onboard ships, amplification coefficient method used to carry out torsional vibration calculation may get satisfactory results which are basically in compliance with the actual conditions, therefore this method is commonly applied.

(2) Analytical calculation method

The analytical calculation method is a precise way to calculate equation set of torsional vibration, the transfer matrix method introduced in 6.6 of this Chapter is one of the analytical calculation. Analytical calculation method is usually to be adopted to calculate the torsional vibration in the following cases:

① PTO shafting of main engine driving generator or dedicated pump, etc.;

② shafting of multiple branch system, e.g. propulsion shafting with multi-engine and single propeller;

③ shafting with larger damping, e.g. shafting has several high-elastic couplings;

④ shafting of diesel engine firing at different intervals;

⑤ for torsional vibration calculation of diesel engine with one cylinder misfiring, comparing with the results obtained by amplification coefficient method, the results got by analytical calculation method is closer to the actual conditions;

⑥ amplification coefficient method is to be used, however, analytical calculation method may get more precise results when resonance speed wave slope occurs approximate to the normal speed of diesel engine and stress/torquecan not be ignored.

(3) The shafting and special systems of non-linear system which can not be treated according to the linear system are to be considered otherwise.

6.2.3 Equivalent system

In calculation of torsional vibration, the complicated shafting is to be first simplified to an equivalent system with linear lumped parameter, which consists of several rigid concentrated masses and connecting elastic axis without inertia.

(1) Conversion principle of equivalent

① The equivalence system is to represent the characteristics of torsional vibration of actual shafting, the natural frequency of free vibration calculation is to be basically the same as the actual frequency and the mode shape of torsional vibration is also similar to that of the actual shafting. Where the difference between measured natural frequency and calculated value is more than 5% and the actual measuring value is correct, the equivalent parameter is to be adjusted.

② Components with larger mass, such as flywheel, flange and elastic coupling or driving and driven parts of pneumatic clutch, etc., are to take their rotating plane center as the concentration point of inertia, and evenly add the inertia of their shaft on the concentration point at both ends.

③ For shafting installed with hydraulic coupling, shafting in front of or after the coupling are to be regarded as two independent torsional vibration system respectively.

④ The tested or verified torsional vibration parameters of diesel engine, elastic coupling, pneumatic clutch, variable gear device, damper are to be submitted by the manufacturers.

(2) Conversion of equivalent:

① Diesel engine

(a) The center of each crank plane is to be taken as the concentration point of unit cylinder moment of inertia, for the V-type generator with parallel connecting rods, the intersection point of cylinder centerline and axis is to be taken as the concentrated point, and then convert each crank to two concentration masses.

(b) The moment of inertia for unit cylincer is to take the average value within the whole revolving period, and consists of moment of inertia for all rotating parts (crank pin, journal, crank web, balance block, partial of connecting rod) and that of reciprocating parts (piston set, crosshead, reciprocating part of connecting rod) converted in way of radius of crank pin.

(c) Where the shafting fitted with a longitudinal damper, it is to be treated as an inertial mass.

② Torsional vibration damper: the equivalent conversion of torsional vibration damper is related to its type

(a) Elastic torsional vibration damper, the driving and driven inertial gears are to be regarded as two mass points, the elastic value is to take the dynamic stiffness of elastic component for the damper.

(b) Silicone oil damper, may generally be simplified to an equivalent inertia which consists of half shell inertia and half inertia gear, and also may be converted to two mass points.

③ Elastic coupling: the driving and driven parts of elastic coupling are to be divided into two mass points, the elastic value is to take the dynamic stiffness of elastic component. The stiffness of Geislinger coupling is not a fixed value and related with frequency.

④ Transmission system:

(a) The driven system is to be converted to equivalent system with the same speed of diesel engine, the equivalent value is to be calculated as follows:

$$I_1 = I_2 / i^2 \quad (6.2.5)$$

$$k_1 = k_2 / i^2 \quad (6.2.6)$$

where: I_2, I_1 —moment of inertia before and after conversion;

k_2, k_1 — stiffness before and after conversion;

i — transmission ratio, $i = n_1 / n_2$;

n_1 — speed of diesel engine;

n_2 — speed of driven shaft.

(b) Conversion of driving gear may convert the moment of inertia for driven gear to driving gear based on transmission ratio and to be a mass point; it may also convert the driving and driven gears to two mass points and assuming that the stiffness between them is large (generally 1000 times the maximum stiffness of the system to be taken). For each component such as gear, clutch, shaft in the driving gear device, equivalent parameters provided by the manufacturers are to be taken.

(c) Belt -driving propulsion shafting is to convert the system after the belt-driving device to the equivalent system of the same speed of diesel engine, and only the stiffness of belt is taken into account.

⑤ Propeller: the value of moment of inertia for propeller is to take the entrained water into consideration, the quantity of entrained water is related to the propeller type:

(a) Water propeller:

— propeller with fixed pitch, the quantity of entrained water is to take 25% to 30% of its inertia in air, if fitted with a sleeve, the quantity of entrained water is to take 35% of its inertia in air;

— propeller with controllable pitch, the quantity of entrained water is generally to take 50% to 55% of its inertia in air at full pitch and to take about 2% of its inertia in air for zero pitch;

— for the propeller with larger disk area ratio and larger pitch, larger quantity of entrained water may be taken.

(b) Air propeller without entrained water.

(c) Water-jet propeller, not taking entrained water into account.

⑥ Generator: rotor of generator is to be regarded as a mass point of inertia.

⑦ Lift fan: whether it is a double-inlet fan or a single-inlet fan, the lift fan is to be regarded as a mass point of inertia.

⑧ Hydraulic dynamometer: the moment of inertia for hydraulic dynamometer is to take the influence of entrained water into account. The quantity of entrained water is related to the loads absorbed by hydraulic dynamometer, which may be taken as 35% in the absence of detailed data.

⑨ Equipment such as belt-driving pumps, generators, etc.: due to the stiffness of belt being less and may lead to slightly slip, the equipment such as belt-driving pumps and generators for shafting may be regarded not related with the characteristics of torsional vibration of the original system.

⑩ Hydraulic coupling: where the shafting is transmitted by hydraulic coupling, it may be regarded that the stiffness of liquid is less, the driving parts in front of hydraulic coupling and the driven parts after hydraulic coupling may be regarded as two independent systems for the characteristics of torsional vibration.

(3) Empirical formula for shafting stiffness calculation: the following empirical formulas may be used for the stiffness of short axes, thrust shaft, intermediate shaft, propeller shaft, with less tolerance:

Thrust shaft: $$K = 9.116\frac{d^4}{L} \quad \text{Nm/rad} \qquad (6.2.7)$$

Short axes: $$K = 10.395\frac{d^4}{L} \quad \text{Nm/rad} \qquad (6.2.8)$$

Intermediate shaft: $$K = 8.643\frac{d^4}{L} \quad \text{Nm/rad} \qquad (6.2.9)$$

Propeller shfat: $$K = 10.604\frac{d^4}{L'} \quad \text{Nm/rad} \qquad (6.2.10)$$

Integral flange: $$K = 10.811\frac{d^4}{L''} \quad \text{Nm/rad} \qquad (6.2.11)$$

Combined flage: $$K = 9.058\frac{d^4}{L'} \quad \text{Nm/rad} \qquad (6.2.12)$$

$$K = 9.497\frac{d^4}{L''} \quad \text{Nm/rad} \qquad (6.2.13)$$

where: d — basic diameter of shaft, in mm;

L — total length of shaft, in mm;

L' — total length of propeller shaft (to the big end of the cone), in mm;

L'' — total length of propeller shaft (to the small end of the cone) in mm.

6.3 Free Vibration Calculation

6.3.1 Introduction

The natural frequency of torsional vibration is to be obtained by free vibration calculation. Currently, damping is not taken into account, it is a free vibration calculation without damping. When the free vibration calculation is done by computer, multiple methods may be used, however, the results are to be generally shown in Holzer table.

6.3.2 Calculation range

The rules specifies that the calculation is to be normally carried out to 12th order harmonic of shafting conditions within the speed range of $0.8n_{\min} \sim 1.2n_e$. For different shafting, the rated speed n_e and minimum steady speed $n_{\min}$ of diesel engine are different, the calculated number of nodes are also different, but in any case, the natural frequency is to be calculated to 14.4 times the rated speed of diesel engine.

For shafting of low-speed two-stroke cycle diesel engine, it is to be generally calculated to 2 node of vibration frequency, maximum to the 3 node.

For shafting of medium-speed four-stroke cycle diesel engine is to be generally calculated to 6 node of vibration frequency.

For shafting of high-speed four-stroke cycle diesel engine is to be generally calculated to 10 node of vibration frequency.

For individual cases, the characteristics of torsional vibration can not be fully examined until it is calculated beyond 10 node of vibration frequency.

6.3.3 Natural frequency

Where the values of damping and excitation in equation (6.2.2) are zero, the free torsional vibration is to be:

$$I_k\ddot{\varphi}_k + K_{k-1,k}(\varphi_k - \varphi_{k-1}) + K_{k,k+1}(\varphi_k - \varphi_{k+1}) = 0 \quad (k = 1, 2, 3, \ldots, n) \qquad (6.3.1)$$

For branch system, the free vibration equation is in consistent with that of (6.3.1) except the branch point:

$$I_k\ddot{\varphi}_k + \sum_{j=1}^{r}[K_{k,j}(\varphi_k - \varphi_j)] = 0 \qquad (6.3.2)$$

After the equation of circular frequency ω_n （rad/s）of natural vibration is solved, the value of natural frequency f is obtained:

$$f = \frac{\omega_n}{2\pi} \quad \text{Hz} \qquad (6.3.3)$$

6.3.4 Resonance speed nc

According to the natural frequency of each mode shape, the resonance speed n_c is to be calculated as follows:

$$n_c = \frac{60f}{v} \quad \text{r/min} \qquad (6.3.4)$$

where: f — natural frequency, in Hz;

v — harmonic order, to be calculated by $\frac{f}{1.2n_e} \le v \le \frac{f}{0.8n_{\min}}$;

for two-stroke cycle diesel engine, V =1, 2, 3, 4, …… 12;

for four-stroke cycle diesel engine, V =0.5, 1, 1.5, …… 12;

n_e— rated speed, in r/min;

$n_{\min}$ — minimum steady speed, in r/min.

6.3.5 Vector sum of relative amplitude

Diagram of equivalent system and mode shape for torsional vibration calculation is shown in Figure 6.3.5.

Based on free vibration, relative amplitude of mode shape in each node for each cylinder is to be calculated, and vector sum of relative amplitude of each mode shape may be obtained according to firing order.

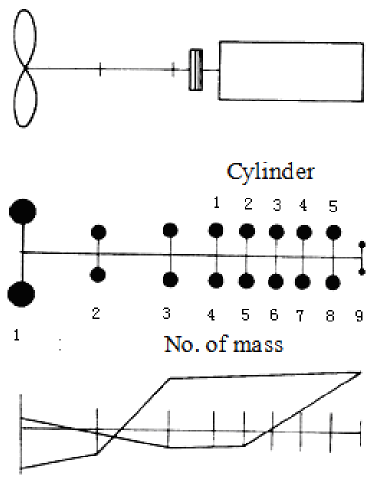


Figure 6.3.5 Diagram of Equivalent System and Mode Shape for Torsional Vibration

6.4 Excitation Source

6.4.1 Introduction

The exciting moments inducing shafting torsional vibration are as follows:

(1) exciting moment induced by the change of gas pressure in cylinder as the diesel engine is running;

(2) inertial moment induced by reciprocating mass as the crank and connecting rod mechanism is moving;

(3) exciting moment induced by wake of propeller;

(4) exciting moment induced by absorbing power components, such as generator, etc., can not homogeneously absorb torque;

(5) exciting moment induced by camshaft of oil fuel pump;

(6) 2nd order excitation induced by cardan;

(7) excitation induced by driving gear;

① excitation induced by gear engagement;

② excitation induced by manufacturing tolerance of gear;

③ 2nd order excitation induced by out-of-roundness of main wheel of reduction gear.

In the practical engineering, the former three excitations are taken into consideration. For engine propulsion shafting, only the excitation of diesel engine is to be taken into account, the exciting moment of propeller blade order is relatively less and may not be included. However, for the turbine propulsion shafting and electric propulsion shafting, only the excitation of exciting moment of propeller blade order is to be taken into account.

6.4.2 Mean indicated pressure of diesel engine Pi

(1) Diesel engine used for propulsion:

$$P_i = \frac{19100 N_e m_s}{ZD^2 R n_e}\left[\frac{1-\eta_m}{\eta_m} + \left(\frac{n_c}{n_e}\right)^2\right] \quad \text{N / mm}^2 \qquad (6.4.1)$$

(2) Diesel engine used for generating electric power:

$$P_i = \frac{19100 N_g m_s}{ZD^2 R n_e \eta_m \eta_g} \quad \text{N/mm}^2 \qquad (6.4.2)$$

(3) Diesel engine used for no load operaton:

$$P_i = \frac{19100 N_e m_s}{ZD^2 R n_e \left(\frac{1}{\eta_m - 1}\right)} \quad \text{N/mm}^2 \qquad (6.4.3)$$

where: N_e — rated power of diesel engine, in kW;

N_g — rated power of generator, in kW;

n_e — rated speed of diesel engine, in r/min;

n_c — resonance speed, in r/min;

Z — number of cylinders;

D — diameter of cylinder, in cm;

R — radius of crank, in cm;

m_s — stroke coefficient, m_s = 4 for four-stroke cycle diesel engine and m_s = 2 for two-stroke cycle diesel engine;

η_m — mechanical efficiency under rated condition of diesel engine;

η_g — efficiency of generator, to be taken as 0.9 for absence of precise data.

6.4.3 Exciting moment induced by gas pressure of diesel engine

(1) Exciting moment of diesel engine

The exciting moment induced by gas pressure M is a periodic function, one rotation of two-stroke engine is a cycle while two rotations of four-stroke engine is a cycle, which may be as follows:

$$M = M_o + \sum M_v \sin(v\omega t + \psi_v) \quad \text{Nm} \qquad (6.4.4)$$

The amplitude of v^{th} order exciting moment M_v is normally as follows:

$$M_v = \frac{\pi D^2 R c_v}{4} \qquad (6.4.5)$$

where: M_o — average moment of single cycle, in Nm;

M_v — amplitude of v^{th} order exciting moment, in Nm;

v — harmonic order;

ω — circular frequency of exciting force, in rad/s;

ψ_v — initial phase angle of v^{th} order, in rad;

c_v — harmonic coefficient, in N/mm^2.

(2) Harmonic coefficient of exciting moment for diesel engine

The harmonic coefficient of exciting moment c_v may be obtained by the following methods:

① parameter calculation provided by manufacturers: to be calculated according to the excitation parameters provided by manufacturers to get precise excitation values;

② empirical calculation: the harmonic coefficient c_v of v^{th} order exciting moment induced by gas pressure on unit piston area, to be calculated as follows:

$$c_v = a_v P_i + b_v \qquad (6.4.6)$$

where: a_v, b_v — coefficient of gas pressure, for four-stroke cycle diesel engine, to be obtained by Table 6.4.3, for two-stroke cycle diesel engine, to be obtained by redouble the value of c_v.

The coefficients of gas pressure given in Table 6.4.3 are the universal ones. Due to the development of super long stroke cycle diesel engines and continuous raising of power density, correction is generally to be needed as the harmonic coefficient c_v of two-stroke diesel engines is calculated by the values given in Table 6.4.3.

Coefficient of Gas Pressure **Table 6.4.3**

v	a_v	b_v	v	a_v	b_v
0.5	0.31625	0.06127	8.5	0.00963	0.01029
1	0.30705	0.13358	9	0.00875	0.00833
1.5	0.26875	0.15686	9.5	0.00820	0.00686
2	0.21125	0.14583	10	0.00770	0.00544
2.5	0.17250	0.12868	10.5	0.00713	0.00441
3	0.14000	0.11029	11	0.00650	0.00355
3.5	0.11050	0.09314	11.5	0.00600	0.00282
4	0.08500	0.07598	12	0.00550	0.0022l
4.5	0.06750	0.05882	12.5	0.00510	0.00182
5	0.04850	0.04902	13	0.00470	0.00142
5.5	0.03450	0.03970	13.5	0.00440	0.00l17
6	0.02625	0.03309	14	0.00410	0.00091
6.5	0.02075	0.02598	14.5	0.00380	0.00075
7	0.01675	0.02047	15	0.00360	0.00058
7.5	0.01433	0.01598	15.5	0.00340	0.00048
8	0.01138	0.01263	16	0.00320	0.00037

③ calculation based on diesel engine diagram: the shearing force P_T acting on crank pin is obtained from $P—\alpha$ diagram:

$$P_T = P\sin\alpha\left(1+\frac{\lambda\sin\alpha}{\sqrt{1-\lambda^2\sin^2\alpha}}\right) \quad \text{N/mm}^2 \qquad (6.4.7)$$

where: $\lambda = \frac{R}{L}$ — length ratio of connecting rod;

R— radius of crank;

L— length of connecting rod;

a— corner of crank.

The shearing force P_T is represented by trigonometric function:

$$P_T = c_0 + \sum(a_v \cos va + b_v \sin va) \quad \text{N/mm}^2 \qquad (6.4.8)$$

where:

$$\left.\begin{aligned}
c_0 &= \frac{1}{\tau}\int_0^{\tau} P_i(a)da \ \text{N/mm}^2 \\
a_v &= \frac{2}{\tau}\int_0^{\tau} P_i(a)\cos vada \ \text{N/mm}^2 \\
b_v &= \frac{2}{\tau}\int_0^{\tau} P_i(a)\sin vada \ \text{N/mm}^2 \\
c_v &= \sqrt{a_v^2 + b_v^2} \ \text{N/mm}^2 \\
\psi_v &= tg^{-1}(\frac{a_v}{b_v}) \ \text{rad}
\end{aligned}\right\} \quad (6.4.9)$$

where: τ being the cycle, $\tau = 2\pi$ for two-stroke cycle diesel engine and $\tau = 4\pi$ for four-stroke cycle diesel engine.

6.4.4 Exciting moment induced by reciprocating inertial force

The reciprocating inertial force caused when the moving parts such as piston, piston rod, connection rod, etc., is running leads to shearing force P_{IT} induced in way of crank pin and the corresponding exciting moment M_{IT}. After the shearing force P_{IT} is represented by trigonometric function, 5th harmonic order is usually to be taken into account, i.e.:

$$P_{IT} = \frac{4m_B R\omega^2}{\pi D^2}[\frac{\lambda\sin\omega t}{4} - \frac{\sin 2\omega t}{2} - (\frac{3\lambda}{4} + \frac{9\lambda}{32})\sin 3\omega t - \frac{\lambda^2\sin 4\omega t}{4} + \frac{5\lambda^3\sin 5\omega t}{32}]\times 10^{-4} \quad \text{N/mm}^2 \quad (6.4.10)$$

where: m_B— mass of reciprocating motion.

6.4.5 Harmonic coefficient of synthesis of shearing force for diesel engine

The harmonic coefficient $c_v^{'}$ of synthesis of shearing force for gas pressure and inertia force with the same order are to be calculated by:

$$c_v^{'} = \sqrt{c_v^2 + (\frac{m_B R n_c^2}{D^2 d_v})^2 + 2c_v\frac{m_B R n_c^2}{D^2 d_v}\cos\psi_v} \qquad \text{N/mm}^2 \quad (v = 1, 2, 3, 4, 5) \qquad (6.4.11)$$

The value of coefficient d_v is referred to Table 6.4.5.

Coefficient d_v **Table 6.4.5**

Harmonic order v	Coefficient d_v
1	$36.004\lambda\times10^{-9}$
2	-71.178×10^{-9}
3	$-108.758\lambda\times10^{-9}$
4	$-36.518\lambda^2\times10^{-9}$
5	$246.5\lambda^3\times10^{-9}$

6.4.6 Exciting moment of propeller

The exciting moment induced by propeller wake may be represented by trigonometric series based on angular velocity of rotation for propeller shaft. The excitation is less, only the excitation of blade order is taken into consideration.

In the absence of detailed data of propeller wake field, the alternating torque of excitation for propeller blade order M_t may be calculated as follows:

$$M_t = \beta \left(\frac{n_c}{n_e} \right)^2 M_e \quad \text{Nmm} \qquad (6.4.12)$$

where: β — coefficient of torque change for propeller, to be taken by the following, if not otherwise specified:

0.055 — for propeller with four blades;

0.015 — for propeller with five blades;

0.030 —for propeller with six blades;

M_e — average torque at rated speed, in Nmm;

n_c — resonance speed, in r/min;

n_e — rated speed, in r/min.

6.5 Response Calculation — Amplification Coefficient Method

6.5.1 Total amplification coefficient

The total amplification coefficient of system is related to that of each component, with the difference of shafting, the total amplification coefficient is also different, including:

(1) Propulsion shafting of diesel engine:

$$\frac{1}{M} = \frac{1}{M_e} + \frac{1}{M_d} + \frac{1}{M_p} + \frac{1}{M_h} + \frac{1}{M_r} \qquad (6.5.1)$$

(2) Shafting of generator:

$$\frac{1}{M} = \frac{1}{M_e} + \frac{1}{M_d} + \frac{1}{M_g} + \frac{1}{M_h} + \frac{1}{M_r} \qquad (6.5.2)$$

(3) Lift shafting (means the shafting driving lift fan for hovercraft):

$$\frac{1}{M} = \frac{1}{M_e} + \frac{1}{M_d} + \frac{1}{M_b} + \frac{1}{M_h} + \frac{1}{M_r} \qquad (6.5.3)$$

(4) Turbine propulsion shafting and electric propulsion shafting:

$$\frac{1}{M} = \frac{1}{M_p} + \frac{1}{M_h} \qquad (6.5.4)$$

6.5.2 Amplification coefficient of each component

(1) Amplification coefficient of diesel engine M_e:

$$M_e = \frac{\sum_{k=1}^{n} I_k a_k^2}{\mu_e \sum_{j=1}^{z} I_j a_j^2} \qquad (6.5.5)$$

where: Z — number of cylinder;

$\mu_e = \dfrac{C_k}{I_k \omega}$ — damping factor provided by manufacturers, generally to be taken as 0.04.

(2) Amplification coefficient of torsional vibration damper M_d:

The damping factor of torsional vibration damper μ_d or relative damping ratio (loss coefficient) ψ_d, are to be generally provided by manufacturers.

① Silicone oil damper M_d:

$$M_d = \frac{\sum_{k=1}^{n} I_k a_k^2}{\mu_d I_d a_d^2} \qquad (6.5.6)$$

② Elastic damping shock absorber M_d:

$$M_d = \frac{2\pi\omega^2 \sum_{k=1}^{n} I_k a_k^2}{\psi_d K_d (\Delta a_d)^2} \qquad (6.5.7)$$

where: I_d — inertia of damper, for silicone oil damper, it means the inertia of wheel;

a_d — relative amplitude of damper;

μ_d — damping factor;

K_d — stiffness of damper;

ψ_d — loss coefficient of damper;

Δa_d — relative amplitude tolerance between driving and driven ends of damper.

(3) Amplification coefficient of propeller M_p:

① Amplification coefficient of water propeller M_p:

$$M_p = \frac{v_p n_e^3 \sum_{k=1}^{n} I_k a_k^2}{91190 a N_p a_p^2}$$ — applicable to fixed pitch and controllable pitch propeller of full pitch (6.5.8)

$$M_p = \frac{v_p n_e^3 \sum_{k=1}^{n} I_k a_k^2}{273600 a N_p a_p^2}$$ — applicable to controllable pitch propeller of zero pitch

where: v_p — blade order;

N_p — absorption power of propeller at the rated speed, in kW;

a_p — relative amplitude of propeller;

a — coefficient, to be determined by the related propeller structure parameters, generally to be taken as 30.

② Amplification coefficient of water-jet propeller M_p:

$$M_p = \frac{\omega n_e^3 \sum_{k=1}^{n} I_k a_k^2}{10^7 \times N_e a_p^2} \qquad (6.5.9)$$

③ Due to the damping for air propeller being less and negligible, the amplification coefficient of air propeller may not be taken into account.

(4) Amplification coefficient of shaft M_h:

$$M_h = \frac{\omega n_e^2 \sum_{k=1}^{n} I_k a_k^2}{0.032 \sum K_{k,k+1} (a_k - a_{k+1})^2} \tag{6.5.10}$$

where: $\sum$— all shafts other than crankshaft and elastic elements such as elastic coupling, etc.

(5) Amplification coefficient of elastic coupling M_r:

$$M_r = \frac{\omega n_e^2 \sum_{k=1}^{n} I_k a_k^2}{0.5 \psi_r K_r (\Delta a)^2} \tag{6.5.11}$$

where: K_r — stiffness of elastic coupling, in Nm/rad;

Δa — relative amplitude difference between driving and driven ends of elastic coupling;

ψ_r — loss coefficient (relative damping ratio), $\psi_r = 2\pi\omega C_{K,KH}/K_{k,k+1}$, provided by manufacturer.

(6) Amplification coefficient of generator M_g is provided by manufacturer, or may be calculated by the following:

① M_g of D.C. generator:

$$M_g = \frac{\omega \sum_{k=1}^{n} I_k a_k^2}{C_g a_g^2} \tag{6.5.12}$$

where: $C_g = \xi_g T_g / n_g$ — damping coefficient of generator, where $\xi = 124 \sim 135$;

T_g — load of generator shaft, in Nm;

n_g — speed of generator shaft, in r/min.

② For A.C. generator, amplification coefficient may not be taken into account.

(7) Amplification coefficient of hydraulic dynamometer M_w:

$$M_w = \frac{v n_e^3 \sum_{k=1}^{n} I_k a_k^2}{91190 b N_w a_w^2} \tag{6.5.13}$$

where: b — coefficient, may be taken as $b = 5$;

N_w — absorption power of hydraulic dynamometer at rated speed, in kW;

a_w — relative amplitude of hydraulic dynamometer.

6.5.3 Resonance calculation

(1) Amplitude of resonance for propulsion shafting of diesel engine:

For propulsion shafting of diesel engine, the main exciting force is gas pressure of the engine. The amplitude of resonance for the 1st mass A_1 is to be calculated as follows:

$$A_1 = M A_0 \quad \text{rad} \tag{6.5.14}$$

where: M— total amplification coefficient of the system;

$A_0 = \dfrac{\pi D^2 c_v R \sum \vec{a}}{4\omega^2 \sum_{k=1}^{n} I_k a_k^2}$ — balance amplitude, in rad;

$\sum \vec{a}$ — vector sum of relative amplitude.

(2) Amplitude of resonance of turbine or electric propulsion shafting:

For the turbine or electric propulsion shafting, the main exciting force is the exciting torque of propeller. The amplitude of resonance for the 1st mass A_1 is to be calculated as follows:

$$A_1 = MA_0 \qquad \text{rad} \qquad (6.5.15)$$

where: M— total amplification coefficient of system;

$A_0 = \dfrac{M_t \alpha_p}{\omega^2 \sum_{k=1}^{n} I_k \alpha_k{}^2}$ — balance amplitude, in rad;

M_t — exciting torque of propeller blade order, calculated as formula (6.4.12);

α_p — relative amplitude of propeller.

(3) Resonance stress:

The resonance stress of shaft $\tau_{k,k+1}$ is calculated as follows:

$$\tau_{k,k+1} = A_1 \tau_0 \qquad \text{N/mm}^2 \qquad (6.5.16)$$

$$\tau_o = \frac{i\omega^2 \sum_{k=1}^{k} I_k a_k^2}{W_{k,k+1}} \qquad \text{N/mm}^2 \qquad (6.5.17)$$

where: $W_{k,k+1} = \dfrac{\pi d^3}{16[1-(\frac{d_o}{d})^4]}$ — section modulus of shaft $k, k+1$;

d_o and d — internal and external diameters of shaft, in mm.

(4) Vibration torque:

① In way of gear engagement:

(a) Where the driving and driven gears are treated as two masses, the vibration torque in way of gear engagement T_G is to be calculated as follows:

$$T_G = \omega^2 A_1 \sum_{k=1}^{G} I_k a_k{}^2 \qquad \text{Nm} \qquad (6.5.18)$$

where: G — No. of mass of driving gear.

(b) Where the driving and driven gears are treated as one mass, the vibration torque in way of gear engagement T_G is to be calculated as follows:

$$T_G = (\sum_{k=1}^{G-1} I_k a_k + I_{G1} a_G)\omega^2 A_1 \qquad \text{Nm} \qquad (6.5.19)$$

where: G — No. of mass for gear;

I_{G1}— inertia of driving gear;

a_G— relative amplitude of gear.

② Elastic coupling: the vibration torque of elastic coupling T_R is to be calculated as follows:

$$T_R = \omega^2 A_1 \sum_{k=1}^{R} I_k a_k^2 \qquad \text{Nm} \qquad (6.5.20)$$

where: R — No. of mass of driving end for elastic coupling.

③ Generator: the vibration inertia torque in way of rotor for generating unit T_g is to be calculated as follows:

$$T_g = \omega^2 A_1 I_g a_g^2 \qquad \text{Nm} \qquad (6.5.21)$$

where: I_g — inertia of rotor for generator;

α_g — relative amplitude of rotor of generator.

(5) Electric angular amplitude: resultant amplitude in way of rotor of A.C. generator θ_g (electric angular) is to be calculated by:

$$\theta_g = \frac{180}{\pi} P \alpha_g A_1 \qquad \text{deg} \qquad (6.5.22)$$

where: P — number of generator pole-pairs.

6.5.4 Non-resonance calculation

Approximate to the resonance speed n, amplitude of 1st mass point A_1' and vibration stress of shaft $\tau'_{k,k+1}$ may be calculated as follows:

$$A_1' = \frac{A_1}{\sqrt{[1-(\frac{n}{n_c})^2]^2 M^2 + (\frac{n}{n_c})^2}} \qquad \text{rad} \qquad (6.5.23)$$

$$\tau'_{k,k+1} = \frac{\tau_{k,k+1}}{\sqrt{[1-(\frac{n}{n_c})^2]^2 M^2 + (\frac{n}{n_c})^2}} \qquad \text{N/mm}^2 \qquad (6.5.24)$$

where: n — speed under the calculated working condition, in r/min;

M— total amplification coefficient.

6.5.5 Torsional vibration calculation with one cylinder misfiring of diesel engine

(1) The mean indicated pressure with one cylinder misfiring of diesel engine is approximately equal to 0, and the corresponding harmonic coefficient of gas is b_v.

(2) The mean indicated pressure of other cylinders P_{imis} :

$$P_{imis} = \frac{Z}{Z-1} P_i \qquad \text{N/mm}^2 \qquad (6.5.25)$$

where: Z — number of cylinders;

P_i — mean indicated pressure, calculated as formulae (6.4.1) or (6.4.2).

(3) Corresponding c_{vmis} :

$$c_{vmis} = a_v P_{vmis} + b_v \qquad \text{N/mm}^2 \qquad (6.5.26)$$

(4) Influence coefficient with one cylinder misfiring of engine γ :

$$\gamma = \frac{c_{vmis} \sum \vec{\alpha}_{mis}}{c_v \sum \vec{\alpha}} \qquad (6.5.27)$$

where: c_v , c_{vmis} — harmonic coefficients for normally fired and one cylinder misfiring, respectively;

$\sum \vec{\alpha}$ — vector sum of relative amplitude for normally fired;

$\sum \vec{\alpha}_{mis}$ — vector sum of relative amplitude for one cylinder misfiring, calculated as follows:

$$\sum \vec{\alpha}_{mis} = \sqrt{\left(\sum_{k=1}^{z} \beta_k \alpha_k \sin v\xi_{1,k}\right)^2 + \left(\sum_{k=1}^{z} \beta_k \alpha_k \cos v\xi_{1,k}\right)^2} \qquad (6.5.28)$$

β_k — coefficient, $\beta_k = {b_v}/{c_{vmis}}$ for cylinder misfiring, $\beta_k = 1$ for other cylinders;

$\xi_{1,k}$ — ignition angle.

(5) Amplitude, stress and torque with one cylinder misfiring of engine:

Amplitude of 1st mass A_{1miis}:

$$A_{1miis} = \gamma A_1 \quad \text{rad} \qquad (6.5.29)$$

Stress of shaft $\tau_{misk,k+1}$:

$$\tau_{misk,k+1} = \gamma \tau_{k,k+1} \quad \text{N/mm}^2 \qquad (6.5.30)$$

Vibratory torque in way of gear engagement T_{gmis}:

$$T_{gmis} = \gamma T_G \quad \text{N/mm}^2 \qquad (6.5.31)$$

Vibratory torque of elastic coupling T_{rmis}:

$$T_{rmis} = \gamma T_R \quad \text{N/mm}^2 \qquad (6.5.32)$$

6.6 Response Calculation — Transfer Matrix Method

6.6.1 Introduction

Response calculation is to be carried out by amplification coefficient method for the engineering, assuming that mode shape for the calculation is consistent with mode shape of free vibration, which has a close approximation to resonance calculation under less damping condition, and is simple for widely use, however, in the case of 6.2.2(2), analytical method is to be generally used for torsional vibration calculation. There are many analytical methods, this paragraph introduces simple transfer matrix methods with less computer memory.

6.6.2 State vector

The state vector $\{Z\}_i$ is shown by:

$$\{Z\}_i = \{X \ \ Y \ \ R_c \ \ R_s \ \ 1 \ \ 1\}^T \qquad (6.6.1)$$

where: X,Y — component of cosine and sine for angular displacement, in rad;

R_c, R_s — component of cosine and sine for elastic moment, in Nm;

i — suffix to indicate the number of element for the system.

6.6.3 Dot-matrix and field-matrix

(1) Dot-matrix and field-matrix for mass k without branch system:

$$[P]_k = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ -I_k\omega^2 & -C_k\omega & 1 & 0 & -M_{ck} & 0 \\ C_k\omega & -I_k\omega^2 & 0 & 1 & 0 & -M_{sk} \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{6.6.2}$$

Its inverse dot-matrix:

$$[P]'_k = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ I_k\omega^2 & C_k\omega & 1 & 0 & M_{ck} & 0 \\ -C_k & I_k\omega^2 & 0 & 1 & 0 & M_{sk} \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{6.6.3}$$

(2) Field-matrix for shaft between element k and element $k+1$:

$$[F]_{k,k+1} = \begin{bmatrix} 1 & 0 & \dfrac{K_{k,k+1}}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & \dfrac{C_{k,k+1}\omega}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & 0 & 0 \\ 0 & 1 & -\dfrac{C_{k,k+1}\omega}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & \dfrac{K_{k,k+1}}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{6.6.4}$$

Its inverse field-matrix:

$$[F]'_{k,k+} = \begin{bmatrix} 1 & 0 & -\dfrac{K_{k,k+1}}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & -\dfrac{C_{k,k+1}}{K_{k,k+1}^2 + C_{k,k+1}^2\omega} & 0 & 0 \\ 0 & 1 & \dfrac{K_{k,k+1}}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & -\dfrac{K_{k,k+1}}{K_{k,k+1}^2 + C_{k,k+1}^2\omega^2} & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \tag{6.6.5}$$

where: ω — vibration frequency, in rad/s;

I_k — moment of inertia for element k, in kgm^2;

C_k — external damping of element k, in Nms;

M_{ck} — component of cosine for element k subjected to external force, in Nm;

M_{sk} — component of sine for element k subjected to external force, in Nm;

$K_{k,k+1}$ — stiffness of shaft between element k and element k+1, in Nm;
$C_{k,k+1}$ — damping of shaft between element k and element k+1, in Nms.

(1) Calculation for elastic moment of each shaft

According to dot-matrix and field-matrix formed by each element (such as branch points, to be formed dot-matrix of branch point), the accumulative transfer of matrix may be obtained by:

$$[T]_n = [P]_n[F]_{n-1,n}[P]_{n-1}\cdots[F]_{k,k+1}[P]_k^B[F]_{k-1,k}\ldots[P]_2[F]_{1,2}[p]_1 \qquad (6.6.6)$$

Based on $[T]_n\{Z\}_1^L = \{Z\}_n^R$

and the elastic moment at fore and aft ends is equal to 0, the mass at the fore end $A_1(X_1,\ Y_1)$ is obtained, then by:

$$[F]_{k,k+1}[P]_k\{Z\}_k^L = \{Z\}_{k+1}^L \qquad (k=1,\ 2,\ 3,\ldots\ldots,\ n) \qquad (6.6.7)$$

to get each mass amplitude at main system $A_k(X_k, Y_k)$ and elastic moment of each shaft $U_k(R_{ck}, R_{sk})$.

After the branch point $A_k(X_k, Y_k)$ is obtained, according to

$$[F]_{k,r1}[P]_k\{Z\}_k^L = \{Z\}_{r_1}^L \qquad (6.6.8)$$

and

$$[F]_{r_j, r_{j+1}}[P]_{r_j}\{Z\}_{r_j}^L = \{Z\}_{r_j+1}^L \qquad (r_j = r_1, r_2, \ldots\ldots, n_r),(r=1,2,3,\ldots,m) \qquad (6.6.9)$$

to get each mass amplitude at branch system $A_{r_j}(X_{r_j}, Y_{r_j})$ and elastic moment of each shaft $U_{r_j}(R_{Cr_j}, R_{sr_j})$.

(4) Torsional vibration stress of each shaft τ_k:

$$\tau_k = \frac{\sqrt{R_{ck}^2 + R_{sk}^2}}{W_{k,k+1}} \times 10^{-6} \qquad \text{N/mm}^2 \qquad (6.6.10)$$

where: $W_{k,k+1}$ — torsional modulus of section for shaft, in m^3;
R_{ck}, R_{sk} — the same as 6.6.2.

6.6.4 Selection of excitation and damping

Torsional vibration calculation for shafting with large damping and which is complicated may be carried out by the analytical method, however, its precision will depend on the determination of values of excitation and damping, proper values are only obtained through large quantity of calculation and actual measurement, Generally, if calculated by analytical method, the values of excitation and damping may still be obtained by the relevant formulae in this Chapter.

6.7 Criteria

6.7.1 The rules requirements

The criteria of torsional vibration calculation is the rules specifications, which also requires that standards provided by the manufacturers or IACS UR may be used as criteria for some components.

The requirements of Section 2, Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships are as follows:

(1) The allowable torsional vibration stress for engine crankshaft of propulsion shafting is shown in Figure 6.7.1-1.

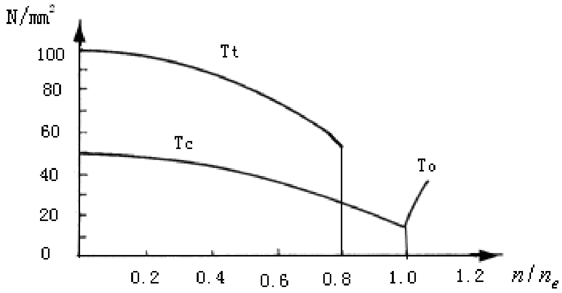


Figure 6.7.1-1 Allowable Torsional Vibration Stress for Engine Crankshaft of Propulsion Shafting

where: $r = n/n_e$;

n — speed of diesel engine, in r/min;

n_e — rated speed of engine, in r/min;

T — calculated torional vibration stress of crankshaft, in N/mm^2;

T_c — allowable torsional vibration stress of continuous operation, in N/mm^2;

T_t — allowable torsional vibration stress of transient operation, N/mm^2.

(2) The allowable torsional vibration stresses of thrust shaft, intermediate shaft, propeller shaft and tube shaft are shown in Figure 6.7.1-2, the meanings of symbols in the Figure are as mentioned as above.

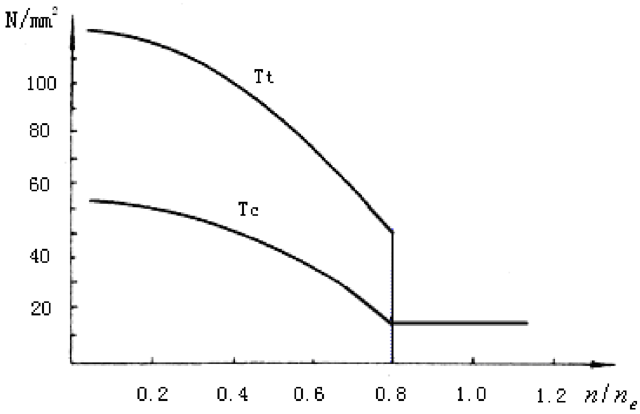


Figure 6.7.1-2 Allowable Torsional Vibration Stresses for Thrust Shaft, Intermediate Shaft, Propeller Shaft and Tube Shaft

(3) The allowable torsional vibration stresses of engine crankshaft for power generating and for main purposed auxiliary engine and that of driving shafting are shown in Figure 6.7.1-3, to be the allowable torsional vibration stress of continuous operation T_c within the speed range r = 0.95 to 1.10; and to be the allowable torsional vibration stress of transient operation T_t within the speed range r < 0.95.

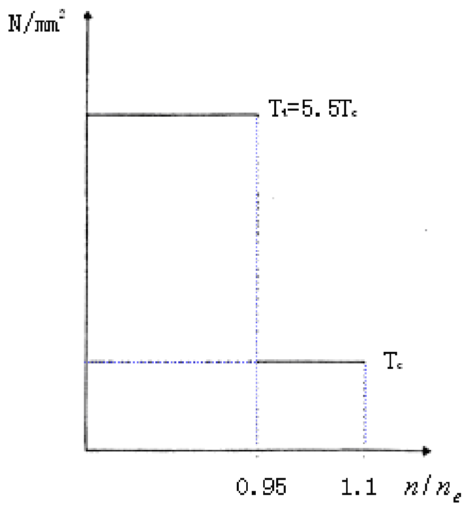


Figure 6.7.1-3 Allowable Torsional Vibration Stresses of Engine Crankshaft for Power Generating and Driving Shafting

(4) The allowable torsional torque of generator is shown in Figure 6.7.1-4, where: $M_e$is the rated torque of diesel engine. Setting of the restricted speed range is not allowed for such shaftingoperation.

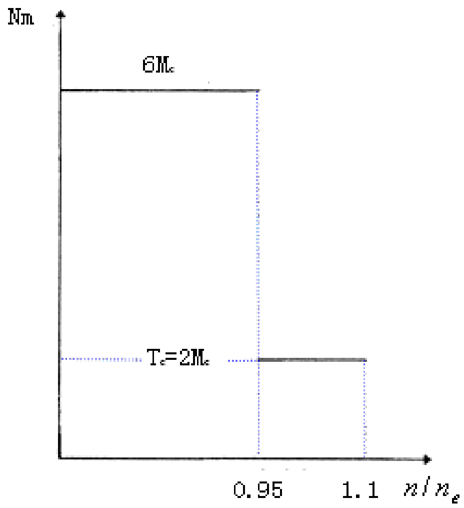


Figure 6.7.1-4 Allowable Vibration Torque of Generator

(5) The allowable vibration torque of gears is shown in Figure 6.7.1-5.

where: $M = 9550\frac{N_n}{n}$ — mean torque of diesel engine, in Nm;

N_n — power of engine at the speed n.

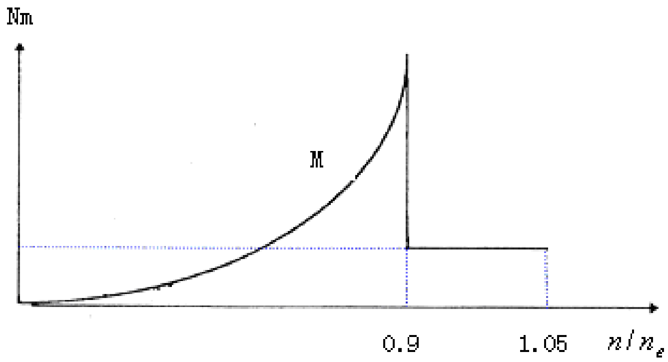


Figure 6.7.1-5 Allowable Vibration Torque of Gears

6.7.2 Allowable values in the rules applicable to speed range

(1) Restricted speed range of running for main propulsion shafting is shown in Figure 6.7.2.

(2) Two working conditions are for shafting, one is steady continuous operation condition and the other is transient operation condition. Hence, the crankshaft of main engine, intermediate shaft, propeller shaft, tube shaft and thrust shaft are to have two corresponding allowable values. The transient allowable value is only applicable to the range r≤0.8, the restricted speed range is not allowed to set within the normal service.

(3) In the case of one cylinder misfiring conditions of propulsion ships with single engine and single propeller, the restricted speed range is to be so set as to ensure the ship's safety navigation. The propulsion shafting of two-stroke cycle of diesel engine for propulsion ships with single engine and single propeller is to be at least of 40% of the rated speed.

(4) Only continuous allowable value is for the vibration torque in way of gear engagement. The restricted speed rage is to be set within r≤0.9.

(5) The allowable values of torsional vibration stresses of engine crankshaft driving generator and for main purposed auxiliary engine and that of driving shafting are continuous allowable ones with the normal service of r= 0.95 to 1.1. Hence, restricted speed range is not allowed to set within such range.

(6) The allowable value of vibration torque for generator is the continuous allowable one within the normal service of r= 0.95 to 1.1. Hence, the restricted speed range is not allowed to set within such range.

(7) The allowable value of resultant amplitude (electric angular) in way of rotor for generator is to be that under the rated working conditions.

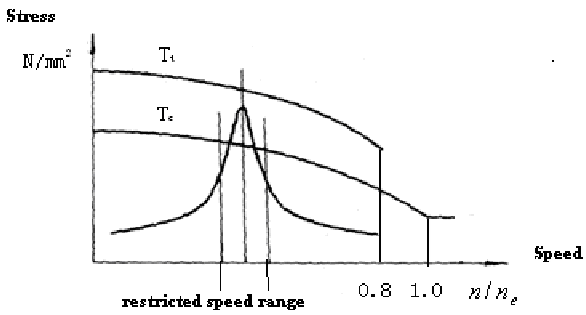


Figure 6.7.2 Determination of Restricted Speed Range

6.7.3 Allowable value of crankshaft torsional vibration stress

(1) The rules requires that the following are to be noted when the rules allowable values are used:

① The allowable value of engine crankshaft for propulsion shafting and crankshaft for generating set are different.

② When the allowable value of crankshaft torsional vibration stress is calculated, the basic diameter of shaft means the main journal diameter of the crankshaft.

③ Except the nodular cast iron crankshaft, in general, material correction may be carried out for the allowable value of crankshaft torsional vibration stress:

$$\tau_0 = \frac{\tau(R_m + 184)}{614} \quad \text{N/mm}^2 \qquad (6.7.1)$$

where: τ_o — allowable value of crankshaft torsional vibration stress after the material correction, in N/mm^2;

τ — allowable value of crankshaft torsinal vibration stress calculated according to the rules, in N/mm^2;

R_m — minimum tensile strength of crankshaft material, in N/mm^2; if $R_m > 600\ \text{N/mm}^2$, to be taken as $600\ \text{N/mm}^2$.

(2) Manufacturers specify that the manufacturers may provide allowable values of torsional vibration stresses for the corresponding engine types. However, if the given allowable stress exceeds the rules value, the calculations of fatigue acceptability factor for the corresponding crankshaft are to be submitted for examination according with the requirements of IACS UR.

(3) Miscellaneous: when the shafting calculation or actual measurement is carried out, if the value at rated speed is found to exceed the given allowable stress required in the rules or provided by the manufacturer, the measured torsional vibration stress may substitute to the formula of fatigue acceptability factor, where it still meets Q $\leq$1.15, the strength of crankshaft is safety. Similarly, the corresponding calculations are to be submitted for examination.

6.7.4 Allowable values of torsional vibration stresses for thrust shaft, intermediate shaft, propeller shaft and tube shaft

(1) The values specified by the rules are same as those required by IACS UR.

(2) In the following cases, correction is to be made for the allowable values of torsional vibration stress:

① Correction of shape coefficient C_k: the transition arc of intermediate shaft connected by the whole flange is designed as multiple arc transition and after the verification by photoelastic test, C_k may be increased appropriately so that the allowable value of torsional vibration stress is increased.

For example: the shape coefficient C_k of intermediate shaft designed by a diesel engine company approved by ISC is shown in Table 6.7.4.

Shape Coefficient of Intermediate Shaft C_k **Table 6.7.4**

Fillet transition	Relative coefficient of stress concentration a_T	Shape coefficient C_k
Single arc transition	1.47	1.0
Arc transition with larger radius 3	1.15	1.28
Arc transition with largest radius 3	1.03	1.43

Note: the relative coefficient of stress concentration a_T is the ratio between actual torsional stress and nominal stress.

② Tensile strength of shaft material: for the intermediate shaft, when it is made of carbon steel and manganese steel, the tensile strength is to be taken as 760 N/mm^2 for R_m> 760 N/mm^2; when it is made of alloy steel, the tensile strength is to be taken as 800 N/mm^2 for R_m> 800 N/mm^2. For propeller shaft and tube shaft, the tensile strength is to be taken as 600 N/mm^2 for R_m> 600 N/mm^2.

6.7.5 Allowable value of vibration torque for elastic coupling

In general, the allowable value of vibration torque for elastic coupling is a constant, being not related to the speed of shafting. However, in the general elastic couplings, the allowable torque of Geislinger coupling is related to the speed.

The allowable values of vibration torque for elastic couplings are to be provided by the manufacturers, which may be applicable to the determination of the results of torsional vibration calculation by amplification coefficient method and analytical calculation method.

(1) The allowable alternating torque of elastic coupling is the value being not to exceed the vibration torque of coupling as the shafting is running continuously.

(2) The maximum torque of elastic coupling is the value being not to exceed the vibration torque of coupling as the shafting is running transiently:

① the maximum torque for certain elastic coupling products is a fixed value;

② two maximum torque values for certain elastic coupling products, one is the maximum torque under the normal transient state, including the shafting operation conditions such as starting/stopping, which is the maximum torque of elastic coupling at transient operation condition specified by the rules; the other is the maximum torque under the abnormal transient state.

6.8 Precautions

6.8.1 Altering the natural frequency of shafting

(1) Change of inertia, stiffness, etc., of shafting equivalent system may alter the natural frequency of shafting vibration so as to remove the dangerous resonance resonance speed.

① Means to change the inertia of flywheel is to install a side flywheel at the free end of crankshaft or to equip the additional inertia in way of shafting with relative larger amplitude, which may alter 1st node and 2nd node resonance speed and mode shape.

② Means to change the stiffness of shaft:

—increase of the shaft diameter may raise 1st node resonance speed and reduce torsional vibration stress of shaft;

—decrease of shaft diameter may reduce 1st node resonance speed;

—increase of shaft length may reduce 1st node resonance speed and alter 2nd and 3rd mode shapes so as to increase the relative amplitude of intermediate shaft.

(2) Where the adverse resonance of torsional vibration occurs within the range of r = 0.9 to 1.05, frequency modulation method or frequency modulation and antivibration method is to be used to remove the resonance speed beyond the normal speed, the damper is not to be used to reduce amplitude as far as possible.

(3) Installation of elastic coupling may effectively reduce 1st node resonance speed to mitigate the

impact of gearbox, however attention is to be given to the removal of resonance speed for other vibration frequency.

6.8.2 Increasing the damping

(1) The installation of high-damping elastic coupling for shafting may have sound antivibration effect when the relative amplitude difference between driving and driven ends is larger. The damper is to be installed in way of the position with the relative amplitude is larger, and the design and manufacture are to be ensured safe and reliable. For damper supplied by circulating oil, reliable oil pressure indicator and controlling device are to be provided. During the service, periodical inspection is to be carried out to prevent shafting damage from the failure of damper.

(2) Installation of side flywheel at the free end of crankshaft or increase of inertia of main engine flywheelk may increase the relative amplitude of 1st node propeller vibration, raise the damping function of propeller and reduce amplitude of 1 node vibration.

6.8.3 Reducing excitation

(1) Change of firing order may reduce secondary harmonic order of straight-type diesel engine and vector sum of relative amplitude of main and secondary harmonic orders for V-type diesel engine so as to reduce its excitation energy, but particular attention is to be given to the increase of vector sum of relative amplitude of other harmonic orders.

(2) Installation of side flywheel at the free end of crankshaft and adjust the inertia of flywheel on main engine may alter the position of nodes for crankshaft so as to reduce the vector sum of relative amplitude of main harmonic order and reduce the excitation energy.

6.8.4 Miscellaneous

Increase of diameter of dangerous shaft may reduce torsional vibration stress or use of high tensile strength material may raise the allowable stress value of torsional vibration and increase the capability of torsional vibration resistance for shafting.

6.9 Plan Approval and Inspection

6.9.1 Plan approval

(1) The compliance of calculation parameters with the submitted drawings, calculated documents, etc., is to be examined

(2) The accuracy of equivalent parameters of shafting torsional vibration is to be examined.

(3) Re-check by ISC-COMPASS software is to be carried out.

(4) Plan approval comments are to be made.

(5) In general, measurements are required in the following cases:

① within the normal speed range as the rules specified, the calculated torsional vibration value is found to reach 70% or above the allowable one under continuous operation condition;

② within the speed range r < 0.8, the calculated torsional vibration stress is 90% or above the allowable value under transient operation condition;

③ certain novel shafting containing unconventional parts.

6.9.2 Inspection

(1) Where measurements are required by plan approval comments, confirmation is to be made by the surveyor in the approval of navigation test program, however, measurements are not necessary in the case of one cylinder misfiring.

(2) The qualifications of measuring unit and personnel are to be confirmed.

(3) Measurements of shafting torsional vibration are to be in compliance with the relevant requirements of Chapter 14 of the Guidelines.

(4) Measurement reports of torsional vibration are to be examined.

(5) The difference between measured frequency and calculated frequency not exceeding 5% is to be confirmed.

(6) The measurement results complying with the rules requirements are to be confirmed.

(7) Where necessary, the measurement report of torsional vibration is to be transferred to the plan approval center for recheck.

(8) Where the measured value exceeds the allowable one under continuous operation condition, the restricted speed range may be certified according to the measured value, and to confirm such range has been marked in red in the revolution meter of main engine by the shipyard.

Chapter 7 LONGITUDINAL VIBRATION OF SHAFTING

7.1 Introduction

7.1.1 Longitudinal vibration of propulsion shafting

If periodical motion of shafting system occurs around its axial direction under periodical torque excitation of diesel engine, bearing force for propeller, etc., it is called as longitudinal vibration of shafting.

For the characteristics of longitudinal vibration for ship's propulsion shafting, the shafting may be regarded as a discrete system consisting of several concentrated masses and connected with longitudinal spring without mass, this system has several natural frequencies of longitudinal vibration. The gas pressure in cylinder of diesel engine and inertial force induced by reciprocating compoments as well as radial component acting on crank pin through connecting rod may be decomposed to numerous radial harmonic forces with different frequencies, amplitudes and phases, the propeller also induces periodical axial exciting force in inhomogeneous flow field. Where the frequency of a certain excitation is same or similar to the natural frequency of longitudinal vibration for certain node of propulsion shafting, a longitudinal vibration phenomenon is caused.

The periodical axial force induced by propeller may also lead to shafting longitudinal vibration phenomenon.

For the gear-driven shafting system of turbine and gear-driven shafting system of diesel engine, the main exciting force is the axial force of propeller, which may lead to shafting longitudinal vibration from the gearbox extended to the propeller.

The calculation software of shafting longitudinal vibration developed by ISC in 1980's has been updating and improving continuously and may be applied to design for shafting longitudinal vibration, plan approval and evaluation for shafting fault analysis.

7.1.2 Hazard of shafting longitudinal vibration

Severe shafting longitudinal vibration may induce the following mechanical faults and vibration phenomenon:

(1) excessive alternating bending stress and tensile stress are induced in way of crank, even may lead to bending fatigue damage of crankshaft;

(2) excessive alternating load is caused on teeth of driving gear so that the teeth surface is worn rapidly, even damaged;

(3) excessive alternating load is caused due to thrust bearing;

(4) longitudinal vibration of engine frame is induced and further vertical vibration of hull girder or superstructure longitudinal vibration will be induced through double-bottom;

(5) vertical vibration of double-bottom structural members, local vibration of structural members in engine room, vertical vibration of hull girders and superstructures longitudinal vibration are induced through thrust bearing.

7.1.3 Range of natural frequency for shafting longitudinal vibration

The characteristics of shafting longitudinal vibration are complicated because the natural frequency of longitudinal vibration is related to the moving parts of shafting, and also to thrust ring, thrust bearing and associated hull structural member. In addition to the dimension of structure, the longitudinal stiffness of crank is related to the arrangement of crank. When the radial harmonic force of gas pressure acts on crank pin, the induced longitudinal deformation of crank is complicated, the longitudinal deformation of crank induced by torsional vibration and coupling of longitudinal vibration are also complicated. These complicated factors have greater effects on accuracy of natural frequency calculation for shafting longitudinal vibration and response calculation for longitudinal vibration. Therefore, in the shafting design, it is necessary to understand the harmonic orders of main engine within the service speed range in order to carry out longitudinal vibration evaluation and take the effective precautions for the issues on adverse longitudinal vibration.

When longitudinal vibration occurs, if no nodes of longitudinal vibration in the shafting, it is called 0 node shafting longitudinal vibration; if 1st node of longitudinal vibration in the shafting, it is called 1st node shafting longitudinal vibration, and so on and so forth.

For longitudinal deformation of crank caused by shafting torsional vibration, its alternating order is the same as harmonic order of torsional vibration. Where the natural frequency of shafting torsional vibration is equivalent to or similar to that of longitudinal vibration, torsional-longitudinal coupled vibration phenomenon is caused.

The longitudinal stiffness of crankshaft for large low-speed two-stroke cycle diesel engine is less, and the natural frequency of shafting will be reduced, therefore longitudinal vibration phenomenon by gas pressure of diesel engine may occur. However, the gas pressure of four-stroke cycle diesel engine will not induce shafting longitudinal vibration.

The range of natural frequency of propulsion shafting longitudinal vibration for large low-speed two-stroke cycle diesel engine is as follows:

0 node longitudinal vibration frequency: f_o =4.5 Hz～23.0 Hz;

1st node longitudinal vibration frequency: f_1 =20.0 Hz～42.0HZ.

For engines with less number of cylinders and short shafting, only 0 node longitudinal vibration generally occurs, but for engines with multiple number of cylinders and long shafting, 1st node longitudinal vibration may also occur.

7.1.4 General relationship of natural frequencies between shafting longitudinal vibration and torsional vibration

The 0 node longitudinal vibration frequency is higher than 1st node torsional vibration frequency. In the case of same number of cylinders, the longer shafting, the more frequency decrease of 0 node longitudinal vibration and 1st node torisional vibration, but the decrease of 0 node longitudinal vibration is slightly less.

The 0 node longitudinal vibration frequency is generally lower than 2nd node torsional vibration frequency. But for long shafting, the both frequencies may be similar or equivalent, thus, the torsional-longitudinal coupled vibration phenomenon is caused.

In general, the range of natural frequency of shafting longitudinal vibration required to study is not to be less than 14 times the rated speed of main engine.

7.1.5 Harmonic orders to be noted

The scale of excitation energy for the system induced by radial harmonic force of gas pressure is related to the firing order, number of cylinder, stroke number and longitudinal mode shape of diesel engine. In the shafting design, if the larger harmonic order of vector sum of relative amplitude difference for longitudinal vibration (see 7.4.6) may be avoided, the adverse longitudinal vibration will not be caused. The harmonic orders to be noted are listed in Table 7.1.5. The effect of shafting longitudinal vibration may not be taken into consideration if damper for longitudinal vibration is installed.

Harmonic Orders of Longitudinal Vibration to be Noted Table 7.1.5

Number of cylinder	0 node	1st node
5	5	—
6	9, 6	—
7	7	7
8	8, 5	8
9	6, 5	9
10	6, 5	10
12	6, 5	12

7.2 Calculating Models

7.2.1 Mechanical model of longitudinal vibration

(1) Equivalent model of shafting longitudinal vibration

The mechanical model of longitudinal vibration is to reflect the characteristics of actual vibration system as far as possible. In general, it is to be treated as a discrete system consisting of finite mass m_k and spring without mass (stiffness k_k).

(2) Mechanical model of longitudinal vibration for propulsion shafting system directly driven by diesel engine means the whole shafting from the free end of crankshaft to the propeller, as shown in Figure 7.2.1-1. The conversion principle of equivalent parameters is as follows:

① concentrating 1/2 mass of the two adjacent cranks on the middle of main journal;

② setting a concentrating point in way of thrust ring and flywheel individually, it may concentrate the mass of last half crank on the thrust ring or concentrate the masses of last half crank and flywheel on the thrust ring;

③ the mass of shafting may be concentrated in way of both ends of mass or the adjacent concentrated mass respectively, the masses of intermediate shaft, tube shaft (if fitted) and screwshaft may also be concentrated in the middle of corresponding shaft respectively;

④ the mass of propeller may be concentrated in the center of propeller;

⑤ the longitudinal stiffness of shaft between two adjacent concentrated masses is to be regarded as that of spring between the concentrated masses;

⑥ the equivalent stiffness k_{th} is to be used to connect thrust ring with hull and to be regarded as a supporting point; the mass of thrust bearing and its seat may also be regarded as a concentrated mass, the mass before or after is to be connected with thrust ring or hull by film stiffness or relevant hull structural stiffness respectively;

⑦ where the damper for longitudinal vibration is installed at the free end, it is to be regarded as a fixed point, 1/2 mass of the first crank and the mass before the crank are to be concentrated in the middle of main journal as the first mass; the first mass point is to be connected with the fixed point by damper stiffness k_d.

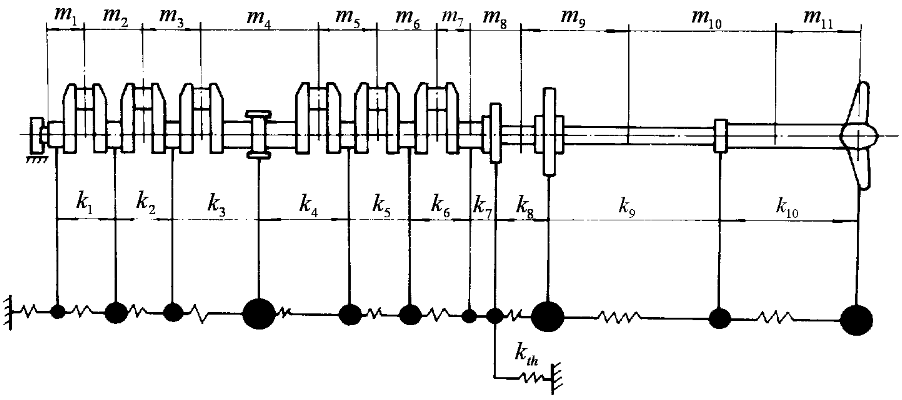


Figure 7.2.1-1 Mechanical Model of Longitudinal Vibration for Propulsion Shafting of Diesel Engine

(3) Mechanical model of longitudinal vibration for gear-driven shafting system

The gear-driven shafting system includes gear-driven shafting system of turbine, gear-driven shafting system of diesel engine and electric propulsion shafting. Its mechanical model of longitudinal vibration means the whole shafting from wheel to propeller, as shown in Figure 7.2.1-2. The conversion principle of equivalent parameters is as follows:

① the wheel of driving gear is to be regarded as the first mass;

② the others are same as (2)③ to ⑥ above.

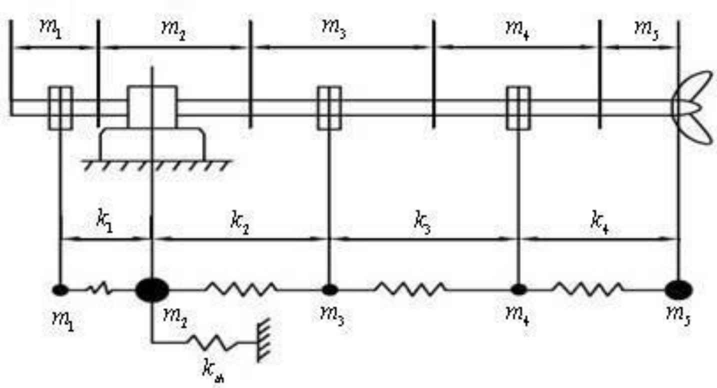


Figure 7.2.1-2 Mechanical Model of Longitudinal Vibration for Gear-driven Shafting System

7.2.2 Mass calculation

The mass in the shafting longitudinal vibration model may be calculated by a general formula, however, the effects of associated moving parts may be taken into account respectively by the following conditions:

(1) in calculating unit crank longitudinal vibration mass, the effect of moving parts, such as connecting rods, etc., may not be taken into account;

(2) when thrust ring and hull are simplified as an equivalent spring, the effect of thrust block mass may generally not be taken into account;

(3) the effect of entrained water of propeller is to be taken into account, the entrained water mass

of longitudinal vibration may be obtained by theoretical formula, or be obtained by experience. The entrained water mass is generally 60% to 100% of the propeller mass (exposed in the air); if not otherwise specified, it may be taken as 60%.

7.2.3 Calculation of longitudinal stiffness

(1) Longitudinal stiffness of direct shaft

The longitudinal stiffness of direct shaft k_i is calculated by:

$$k_i = \frac{\pi E d_i^2}{4L_i} \quad \text{N/mm} \qquad (7.2.1)$$

where: E — modulus of elasticity, 20.60×10^4 N / mm^2 for carbon steel;

d_i — diameter of shaft, in mm;

L_i — length of shaft, in mm.

(2) Longitudinal stiffness of conical shaft

The longitudinal stiffness of conical shaft k_t is calculated by:

$$k_t = \frac{\pi E d_1 d_2}{4L_t} \quad \text{N/mm} \qquad (7.2.2)$$

where: d_1— diameter of small end of cone, in mm;

d_2— diameter of big end of cone, in mm;

L_t— length of cone, in mm.

(3) Longitudinal stiffness of crank

The value of longitudinal stiffness of crank has the greater effects on the natural frequency and mode shape of shafting longitudinal vibration. Due to the fact that the structural type of crankshaft and stress are complicated, the accurate value of crankshaft longitudinal stiffness is to be tested and determined by the manufacturers or natural frequency of longitudinal vibration is to be measured on the test bed, combining with the stiffness of the thrust bearing, the value of longitudinal stiffness is to be calibrated.

(4) Longitudinal stiffness of thrust bearing

The longitudinal stiffness of thrust bearing is actually the equivalent value of stiffness of structural members, such as thrust ring, oil film, thrust block, thrust bearing housing and double bottom. Due to the complicated structures and more influence factors, it is generally to be determined by measured results.

Normally, after the stiffness of thrust bearing is more than one certain value, the influence on the natural frequency of longitudinal vibration is less, hence, in estimation, the stiffness of thrust bearing k_{th} may be taken as:

$$k_{th} = (1.0 \sim 5.0) \times 10^6 \quad \text{N/mm} \qquad (7.2.3)$$

(5) Longitudinal stiffness of other shafting

The reciprocal of longitudinal stiffness of tandem shaft is the sum of reciprocals of stiffness for each shaft; the longitudinal stiffness of parallel shaft if the sum of stiffness for each shaft and the longitudinal stiffness of hollow shaft is the difference between external diameter and internal diameter of direct shaft.

7.2.4 Equation of motion for longitudinal vibration

The longitudinal vibration system with n masses is shown in Figure 7.2.4.

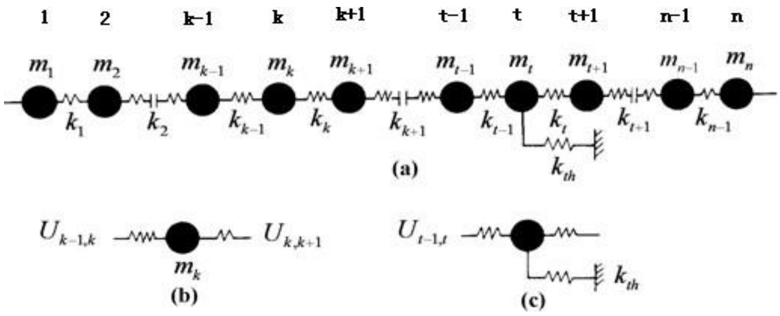


Figure 7.2.4　Longitudinal Vibration System with *n* masses

Taking any one of k mass as a detached body, the forces acting on m_k are: inertial force, longitudinal elastic force induced by fore and aft shafts, external damping force, internal damping force of fore and aft shafts and longitudinal exciting force. According the balance principle, the equation of motion of damping longitudinal vibration for the k_k^{th} mass:

$$\begin{aligned}&m_k\ddot{x}_k - k_{k-1}(x_{k-1}-x_k) - c_{ak-1}(\dot{x}_{k-1}-\dot{x}_k) + k_k(x_k - x_{k+1}) + c_{ak}(\dot{x}_k - \dot{x}_{k+1})\\ &+ c'a_k = P^*{}_{ak}e^{i\omega t}\,(k=1,2\cdots\cdots,\ \ n,\ \ k\neq t,\ \ k_0=c_0=0)\end{aligned} \tag{7.2.4}$$

$$\begin{aligned}&m_k\ddot{x}_k - k_{t-1}(x_{t-1}-x_t) - c_{at-1}(\dot{x}_{t-1}-\dot{x}_t) + k_t(x_t - x_{t+1})\\ &+ c_{at}(\dot{x}_t - \dot{x}_{t-1}) + k_{th}\dot{x}_t + c'_{at}\dot{x}_t = P^*{}_{at}e^{i\omega t}\end{aligned} \tag{7.2.5}$$

where: k_{k-1} — longitudinal stiffness between the $k-1^{\text{th}}$ mass and the k^{th} mass;

k_k — longitudinal stiffness between the k^{th} mass and $k+1^{\text{th}}$ mass;

x_k — longitudinal displacement of the k^{th} mass changed with the time;

$\dot{x}_k$ — longitudinal vibration velocity of the k^{th} mass;

$\ddot{x}_k$ — longitudinal vibration acceleration of the k^{th} mass;

c_{ak-1} — longitudinal internal damping coefficient of the shaft between the $k-1^{\text{th}}$ mass and the k^{th} mass;

c'_{ak} — longitudinal external damping coefficient of the k^{th} mass;

P^*_{ak} — complex amplitude of longitudinal exciting force acting on the k^{th} mass;

k_{th} — longitudinal stiffness of thrust bearing;

ω — circular frequency of longitudinal excitation;

t — time.

Footnote t is the corresponding parameter in way of thrust ring.

Where the complex amplitude of longitudinal vibration is of A^*_{ak}, the particular solution of the formula (7.2.5) is:

$$x_k = A^*_{ak}e^{i\omega t} \qquad (k=1,2,3\ldots,n) \tag{7.2.6}$$

7.2.5　Calculation method of longitudinal vibration

In general, the exciting force and damping force are to be taken as zero in the formulas (7.2.4) and (7.2.5) so as to get the natural frequency and corresponding mode shape of free longitudinal vibration without damping, then the longitudinal vibration response is to be calculated by energy method or power amplification method.

7.3 Calculation of Natural Frequency

7.3.1 Holzer table

Where all of the damping force and exciting force are taken as zero in the formulas (7.2.4) and (7.2.5), the equation of free longitudinal vibration without damping may be obtained: i.e. equation of shafting longitudinal deformation (amplitude) and equation of elastic (inertial) force are to be as follows:

Equation for deformation:

$$\left.\begin{aligned} \alpha_{ak} &= \alpha_{ak-1}\frac{U_{k-1,\ k}}{k_{k-1}} \\ \alpha_{at} &= \alpha_{at-1}-\frac{U_{t-1,\ t}}{k_{k-1}} \end{aligned}\right\} \tag{7.3.1}$$

Equation for elastic force:

Equation for deformation:

$$\left.\begin{aligned} \alpha_{ak} &= \alpha_{ak-1}\frac{U_{k-1,\ k}}{k_{k-1}} \\ \alpha_{at} &= \alpha_{at-1}-\frac{U_{t-1,\ t}}{k_{k-1}} \end{aligned}\right\} \tag{7.3.1}$$

Equation for elastic force:

$$\left.\begin{aligned} U_{k,\ k+1} &= \sum_{k=1}^{k} m_k\omega_a^2\alpha_{ak} && \text{for } k<t; \\ U_{t,\ t+1} &= \sum_{k=1}^{k} m_k\omega_a^2\alpha_{ak}-\alpha_{ak}k_{th} && \text{for } k\ge t; \\ U_{n,\ n+1} &= \sum_{k=1}^{k} m_k\omega_a^2\alpha_{ak}-\alpha_{ak}k_{th}=0 && \text{for } k=n \end{aligned}\right\} \tag{7.3.2}$$

The symbols in equation (7.3.1) and (7.3.2) are as follows:

- ω_a — natural circular frequency of longitudinal vibration for a certain node, in rad/s;
- α_{ak} — amplitude of longitudinal vibration for the k th mass, in mm;
- a_{at} — relative amplitude of longitudinal vibration in way of thrust ring, in mm;
- $U_{k,k+1}$ — elastic force of longitudinal vibration between the k^{th} mass and $k+1^{th}$ mass, in N/mm;
- $U_{t,t+1}$ — elastic force of longitudinal vibration between thrust ring and the mass after it, in N/mm;
- m_k — the k^{th} concentrated mass, in kg;
- k_{k-1} — longitudinal stiffness between the $k-1^{th}$ mass and the k^{th} mass in N/mm;
- k_{th} — longitudinal stiffness of thrust bearing, in N/mm.

Where the damper for longitudinal vibration is installed at the free end of crank, the equation (7.3.2) is changed to:

$$\left.\begin{aligned} U_{k,k+1} &= \sum_{k=1}^{k} m_k\omega_a^2\alpha_{ak}-k_a\alpha_{a1} && \text{for } k<t; \\ U_{k,k+1} &= \sum_{k=1}^{k} m_k\omega_a^2\alpha_{ak}-k_a\alpha_{a1}-k_{th}\alpha_{at} && \text{for } k\ge t; \\ U_{n,n+1} &= \sum_{k=1}^{k} m_k\omega_a^2\alpha_{ak}-k_a\alpha_{a1}-k_{th}\alpha_{at}=0 && \text{for } k=n \end{aligned}\right\} \tag{7.3.3}$$

According to equations (7.3.1) to (7.3.3) may be listed in Holzer table of shafting free longitudinal vibration.

7.3.2 Calculation of resonance speed

(1) The natural frequency of longitudinal vibration f_a is:

$$f_a = \frac{1}{2\pi}\omega_a \quad \text{Hz} \tag{7.3.4}$$

(2) The resonance speed induced by radial harmonic force of gas pressure n_c is:

$$n_c = \frac{60 f_a}{\nu} \quad \text{r/min} \tag{7.3.5}$$

where: ν — harmonic order of radial harmonic force of diesel engine.

(3) The resonance speed induced by harmonic force of propeller blade order n_c is:

$$n_c = \frac{60 f_a}{iZ_p} \quad \text{r/min} \tag{7.3.6}$$

where: i — harmonic of 1 times blade order, harmonic of twice blade order, respectively, only the harmonic of blade order is calculated;

Z_p — number of propeller blade.

7.4 Calculation of excitation power

7.4.1 Exciting Force

The exciting force inducing shafting longitudinal vibration is:

(1) radial harmonic force induced by engine gas pressure;

(2) radial harmonic force induced by reciprocating inertial force;

(3) exciting force of propeller induced in the inhomogeneous field.

7.4.2 Radial harmonic force induced by gas pressure

It is shown from Figure 4.1.2, the gas radial force P_i acting in way of crank pin changes with the time and tensile and bending are caused in way of crank. In general, the radial harmonic force is to be obtained through the measurement of diagram of diesel engine under different working conditions and the calculation together with the harmonic analysis, which are usually provided by the manufacturers.

The relationships among the amplitude of v^{th} order radial harmonic force for two-stroke cycle diesel engine P_{nv}, initial phase angle φ_{nv} and mean indicating pressure P_i are shown in Figures 7.4.2-1 to 7.4.2-4.

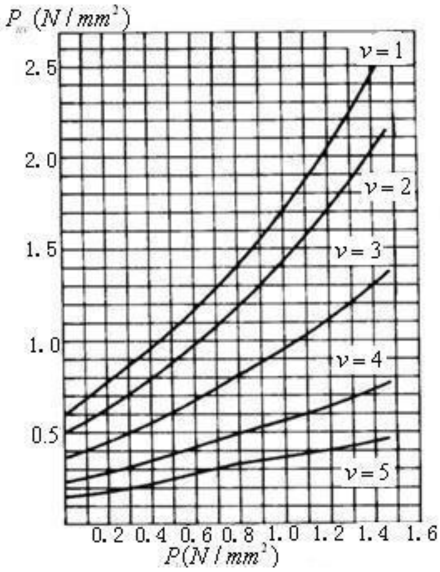


Figure 7.4.2-1 Relationship between P_{nv} and P_i（v =1 ~ 5）

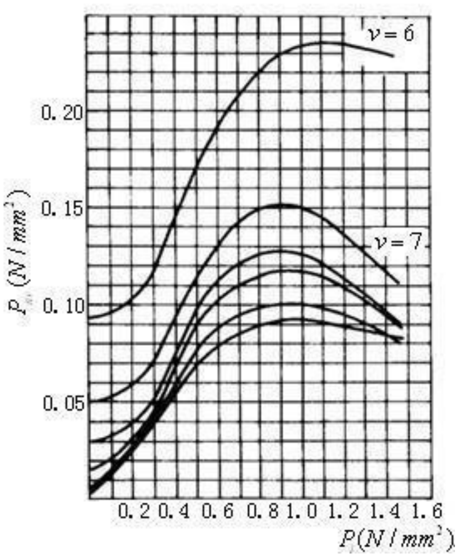


Figure 7.4.2-2 Relationship between P_{nv} and P_i（v =6 ~ 11）

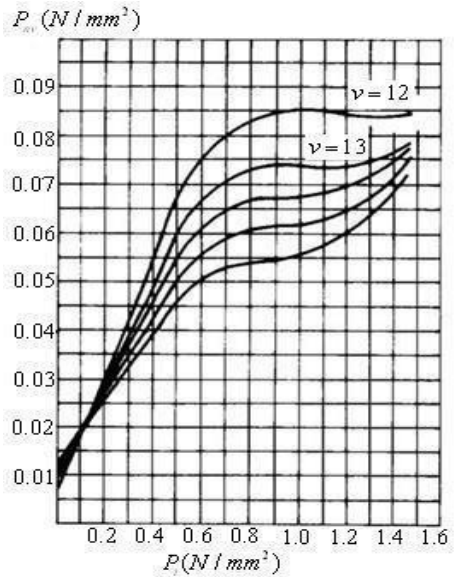


Figure 7.4.2-3 Relationship between P_{nv} and P_i (ν =12 to **16)**

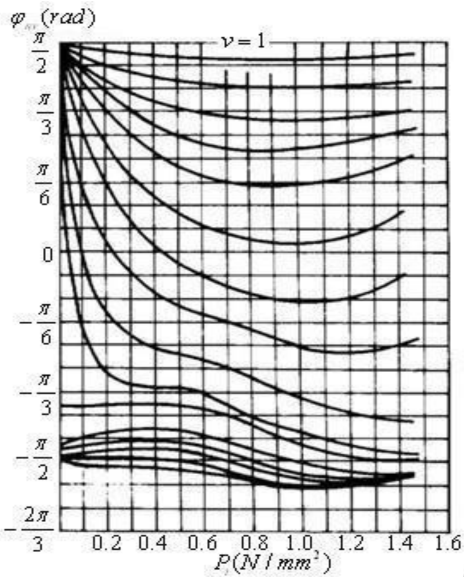


Figure 7.4.2-4 Relationship between φ_{nv} and P_i

7.4.3 Radial harmonic force induced by reciprocating inertial force

The reciprocating inertial force $P_j^{'}$ induced by moving parts of diesel engine such as piston, piston rod and connecting rod during the motion may be similarly decomposed to tangential force and radial force, and the radial force may also lead to shafting longitudinal vibration.

Where the harmonic order $\nu \geq 5$, the harmonic force is much less and may be neglected. However, the ν = 1st to 4th order radial harmonic force $P_{j\nu}$ and its initial phase angle $\varphi_{j\nu}$ are as follows:

$$\left.\begin{aligned}
&P_{j1} = 0.25P_{j0} && \mathrm{N/mm^2} \\
&P_{j2} = 0.5(1-\lambda)P_{j0} && \mathrm{N/mm^2} \\
&P_{j3} = 0.75\lambda P_{j0} && \mathrm{N/mm^2} \\
&P_{j4} = 0.25\lambda^2 P_{j0} && \mathrm{N/mm^2} \\
&P_{j0} = 9.81\frac{m_j R\omega^2}{\pi D^2/4} && \mathrm{N/mm^2} \\
&\varphi_{j1-4} = -\frac{\pi}{2} && \mathrm{rad}
\end{aligned}\right\} \qquad (7.4.1)$$

where: D — diameter of cylinder, in mm;

m_j — mass of reciprocating parts, in kg;

R — radius of crank, in mm;

λ —— length ratio of connecting rod.

7.4.4 Combined radial harmonic force

The combined radial harmonic force P_r is calculated by:

$$P_r = \sqrt{P_{nv}^2 + P_{jv}^2 - 2P_{nv}P_{jv}\cos(\varphi_{nv} - \frac{\pi}{2})} \quad \mathrm{N/m^2} \qquad (7.4.2)$$

7.4.5 Equivalent axial harmonic force

By the effect of radial harmonic force P_r, the deformation of crank is shown in Figure 7.4.5(1), the scale of axial δ_r along the direction of centerline of crankshaft is related to structural dimension and angle of adjacent cranks, if β_r is used, as follows:

$$\delta_r = \beta_r P_r \qquad (7.4.3)$$

Where an axial force P_a is acting along the direction of centerline of crankshaft, the axial deformation of crank δ_a is shown in Figure 7.4.5(2), its scale is also related to structural dimension and angle of adjacent cranks, if β_r is used, as follows:

$$\delta_a = \beta_a P_a \qquad (7.4.4)$$

Taking $\delta_r = \delta_a$ to get the relationship between equivalent axial harmonic force P_a and radial harmonic force P_r:

$$P_a = \frac{\beta_r}{\beta_a}P_r = \beta P_r \qquad \mathrm{N/mm^2} \qquad (7.4.5)$$

where: β — conversion factor, generally to be calculated by:

$$\beta = 0.125\frac{l_p}{R\theta_z} \qquad (7.4.6)$$

where: l_p — length of crank pin, in mm;

R — radius of crank, in mm;

$\theta_Z = \frac{1}{Z}\sum_{k=1}^{k}\theta_k$;

Z — number of cylinder;

$\theta_k = \frac{1}{2}\cos^2\frac{a_{av}}{2}$;

a_{av} — mean angle between adjacent cylinders, in °;

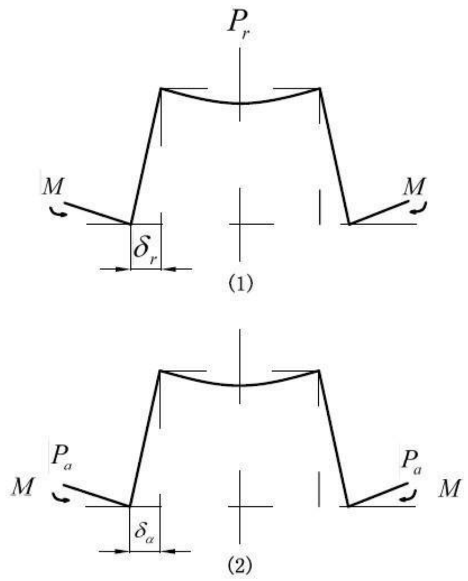


Figure 7.4.5 Deformation of Crank

7.4.6 Power made by radial harmonic force

Due to the fact that the mass of each crank is concentrated in the middle of main journal, and the axial forces P_a acting on the k^{th} mass and $k+1^{th}$ mass are in opposite directions, as shown in Figure 7.4.6.

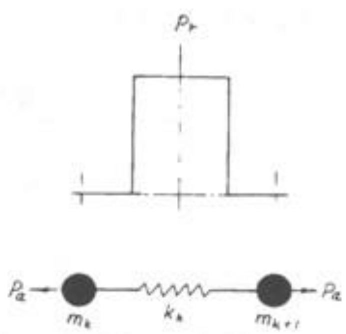


Figure 7.4.6 Direction of Axial Force

Where $v\omega = \omega_a$, the power W_e done by v^{th} order harmonic force of Z cylinders in a cycle is calculated by:

$$W_e = \frac{\pi^2}{4} D^2 \beta P_r A_{a1} \sum \Delta a_k \qquad \text{N/mm} \qquad (7.4.7)$$

where: $\sum \Delta a_k$ is called vector sum of relative amplitude difference, to be calculated by the following:

$$\sum \Delta a_k = \sqrt{(\sum \Delta a_k \sin v\psi_{1k})^2 + (\sum \Delta a_k \cos v\psi_{1k})^2} \qquad (7.4.8)$$

where: Δa_k — relative amplitude difference for longitudinal vibration, $\Delta a_k = a_{ak} - a_{ak-1}$;

ψ_{1k} — firing angle between the k th cylinder and 1st cylinder.

7.4.7 Excitation power of propeller

The exciting force of propeller has been analyzed in Chapter 3, F_x is the exciting force inducing the shafting longitudinal vibration. Similarly, the power W_t done in a cycle is as follows:

$$W_t = \pi (F_x)_v A_{a1} a_{ap} \quad \text{Nmm} \qquad (7.4.9)$$

where: A_{a1} — amplitude of longitudinal vibration for the 1st mass, in mm;

a_{ap} — amplitude of longitudinal vibration for propeller;

$(F_x)v$ — amplitude of v^{th} order exciting force for propeller, in N. If detailed data of wake field is provided, it may be estimated by:

$$(F_x)_v = \xi_p (\frac{n}{n_e})^2 F_{x0} \qquad (7.4.10)$$

where: ξ_p — thrust fluctuation coefficient of propeller, to be obtained from Table 7.4.7. If not specified, it may be taken the intermediate value.

Thrust Fluctuation Coefficient of Propeller **Table 7.4.7**

Number of blade Z_p	Blade order (v)	Thrust fluctuation coefficient ξ_p	Twice blade order (2 v)	Thrust fluctuation coefficient ξ_p	3 times blade order (3 v)	Thrust fluctuation coefficient ξ_p
4	4	0.090～0.130	8	0.020～0.040	12	0.015～0.005
5	5	0.025～0.035	10	0.020～0.030	15	0.008～0.002
6	6	0.050～0.090	12	0.020～0.030	18	0.015～0.005

F_{xo} — mean thrust at the rated speed, to be provided by the designer or calculated as follows:

$$F_{x0} = 1943.3 \frac{\eta \eta_t N_e}{V_s (1 - t_1)} \quad \text{N} \qquad (7.4.11)$$

η — propulsion efficiency;

η_t — transmission efficiency, η_t=0.95 for engine room located in midship and η_t=0.97 for engine room located in stern;

t_1 —thrust deduction factor, generally to be taken as t_1=0;

N_e — rated power, in kW;

V_s — velocity, in kn;

n — any speed, in r/min;

n_e — rated speed, in r/min.

$k+1^{\text{th}}$ mass, in

7.5 Calculation of Damping Power

7.5.1 Damping force

The damping force of shafting longitudinal vibration is:

(1) hysteretic damping force induced by axial compressive deformation of shaft;

(2) damping force of bearing;

(3) damping force induced by propeller in water;

(4) damping force of thrust bearing, including film damping force, hysteretic damping force, etc.

7.5.2 Hysteretic damping power of crankshaft

The hysteretic damping power W_{ac} of crankshaft is:

$$W_{ac} = 0.725 \times 10^{-7} (C_{al} + C_{a2} + C_{a3}) \pi \sum_{k=1}^{Z} U_{k,k+1}^{2} A_{a1}^{2} \tag{7.5.1}$$

where: $C_{a1} = 0.405 \dfrac{l_j}{d_j^2}$, in l/mm;

$C_{a2} = 0.405(8R + d_p)^2 \dfrac{l_p}{d_p^4}$, in l/mm;

$C_{a3} = 22.918 \dfrac{R^3}{BH^3}$, in l/mm;

$U_{k,k+1}$ — elastic force of longitudinal vibration between the k^{th} mass and $k+1^{\text{th}}$ mass, in N/mm;

Z — number or cylinder;

A_{a1} — amplitude of longitudinal vibration of the 1st mass, in mm;

l_j — length of main journal, in mm;

d_j — diameter of main journal, in mm;

l_p — length of crank pin, in mm;

d_p — diameter of crank pin, in mm;

B — breadth of crank web, in mm;

H — thickness of crank web, in mm;

R — radius of crank, in mm.

7.5.3 Hysteretic damping power of driving shaft

The hysteretic damping power W_{as} of all shafts after the crankshaft is:

$$W_{as} = 0.725 \times 10^{-7} \pi \sum_{k=f-1}^{n} C_{as} U_{k,k+1}^{2} A_{a1}^{2} \quad \text{Nmm} \tag{7.5.2}$$

where: $C_{as} = 0.4505 \dfrac{l_{k,k+1}}{d_{k,k+1}^2}$, in l/mm;

$l_{k,k+1}$ — length of shaft between the k^{th} mass and $k+1^{\text{th}}$ mass, in mm;

$d_{k,k+1}$ — diameter of shaft between the k^{th} mass and $k+1^{\text{th}}$ mass, in mm;

$\sum_{k=f-1}^{n}$ — total sum of shafts calculated from a mass before flywheel.

7.5.4 Damping power of bearing

The damping power W_b of bearing for the whole propulsion shafting in a cycle is:

$$W_b = 2\times10^{-3}\pi\xi_b\omega_a^2A_{a1}^2\sum_{k=1}^{n}m_ka_{ak}^2 \quad \text{Nmm} \qquad (7.5.3)$$

where: ξ_b — damping ratio of bearing, to be taken as $\xi_b = 0.03 \sim 0.85$ for propeller and shaft damping not taken into consideration;
It is recommended that $\xi_b = 0.03$ for all damping taken into consideration;

m_k — mass of the k th point, in kg.

7.5.5 Damping power of propeller

The damping power W_p of propeller longitudinal vibration is:

$$W_p = \pi c_{ap}\omega_a\alpha^2{}_{ap}A_{a1}^2 \quad \text{Nmm} \qquad (7.5.4)$$

where: c_{ap} — damping coefficient of propeller longitudinal vibration, to be calculated by:

$$c_{ap} = 8.2047\times10^{-2}\frac{\omega_a}{v}H_pD_p^2\frac{dC_t}{dS} \quad \text{Ns / mm} \qquad (7.5.5)$$

v — harmonic order;

H_p — pitch of propeller, in m;

D_p — diameter of propeller, in m;

$\frac{dC_t}{dS}$ — related to propeller expanded area ratio and pitch ratio, to be obtained from Figure 7.5.5.

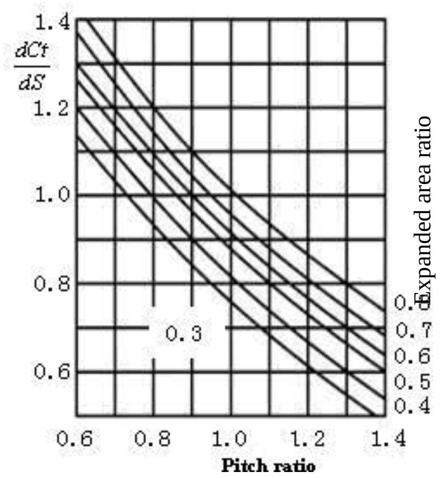


Figure 7.5.5 Relationship between $\frac{dC_t}{dS}$ and Pitch Ratio

7.6 Response Calculation

7.6.1 The response calculation of longitudinal vibration for propulsion shafting system directly driven by diesel engine

(1) Energy method of longitudinal vibration calculation

Where the resonance of shafting longitudinal vibration occurs, the excitation power inducing longitudinal vibration is equal to damping power, the resonant amplitude A_{a1} of the 1st mass may be calculated.

(2) The amplitude of longitudinal vibration as the exciting force of diesel engine acts independently

The excitation power induced by gas pressure and inertial force of diesel engine is equal to all damping powers, i.e.:

$$W_a = W_{ac} + W_{as} + W_b + W_p \qquad \text{Nmm} \tag{7.6.1}$$

(3) The amplitude of longitudinal vibration as propeller excitation acts independently

Then the excitation power induced by propeller is equal to all damping power, i.e.:

$$W_t = W_{ac} + W_{as} + W_b + W_p \qquad \text{Nmm} \tag{7.6.2}$$

(4) Estimation of amplitude of shafting longitudinal vibration in the initial design

In the initial design, due to the structural parameters of crankshaft and propeller for diesel engine being unknown, the amplitude of longitudinal vibration may be estimated by:

$$W_e = W_{as} + W_b \qquad \text{Nmm} \tag{7.6.3}$$

Where the estimated amplitude of longitudinal vibration exceeds the allowable value, detailed calculation is to be carried out.

(5) Amplitude of longitudinal vibration for the 1st mass in non-resonance zones

The amplitude of longitudinal for the 1st mass A_a approximate to the resonance speed n_c:

$$A_a = \frac{A_{a1}}{\sqrt{[1-(\frac{n}{n_c})^2]^2 M_a^2 + (\frac{n}{n_c})^2}} \tag{7.6.4}$$

where: n — calculated speed, in r/min;

$M_a = \frac{A_{a1}}{A_{a0}}$ — amplification coefficient of total powers;

A_{a0} — balance amplitude of longitudinal vibration, calculated by:

$$A_{a0} = \frac{1000\frac{\pi}{4}D^2 \beta R_r \sum \Delta a_k}{\omega_a^2 \sum_{k=1}^{n} m_k a_{ak}^2} \qquad \text{mm} \tag{7.6.5}$$

7.6.2 Response calculation for gear-driven shafting system

(1) Energy method of longitudinal vibration calculation

For the gear-driven shafting systems of turbine, electric propulsion and diesel engine, the calculation model of longitudinal vibration is set up from the propeller to the driving wheel, and the main exciting force is the axial thrust induced by propeller. Then the excitation power induced by propeller is equal to all damping powers, i.e.:

$$W_a = W_{as} + W_b + W_p \qquad \text{Nmm} \tag{7.6.6}$$

(2) Amplitude of longitudinal vibration in way of gear within the resonance zone.

The amplitude of longitudinal vibration in way of gear (the 1st mass) A_{a1} approximate to resonance speed n_c is:

$$A_{a1} = \frac{\xi_p (n/n_c)^2 F_{x0}\alpha_{ap}}{0.725\times10^{-7}\sum_{k=1}^{n} C_{as}U_{k,k+1}^2 + 2\times10^{-3}\xi_p\omega_a^2\sum_{k=1}^{n} m_k\alpha_{ak}^2 + C_{ap}\omega_{ap}^2} \quad \text{mm} \qquad (7.6.7)$$

7.6.3 Secondary exciting force induced by shafting longitudinal vibration

(1) Exciting force acting in way of thrust bearing and induced by shafting longitudinal vibration

Where the shafting longitudinal vibration occurs, the value of secondary exciting force F_{th} with the same frequency and harmonic order of the shafting longitudinal vibration in way of thrust bearing is:

$$F_{th} = k_{th} a_{at} A_{al} \quad \text{N} \qquad (7.6.8)$$

where: k_{th} — longitudinal stiffness of thrust bearing, in N/mm;

a_{at} — relative amplitude of longitudinal vibration for thrust ring.

(2) Effects of secondary exciting force on other vibrating body

Within the normal speed range, where the frequency of secondary exciting force induced by shafting longitudinal vibration is equivalent to or similar to that of other vibrating body, it will lead to strong vibration of other vibrating body. The secondary exciting force F_{th} induced by shafting longitudinal vibration may induce frame longitudinal vibration through thrust bearing housing, and further to cause the superstructure longitudinal vibration, vertical vibration of hull girder, local vibration of structural members in engine room, etc.

Where the point of action for secondary exciting force F_{th} leaves the physical longitudinal axis of vibrating body, it will induce alternating moment to the vibrating body with the same frequency and harmonic order of secondary exciting force and to cause vertical vibration of double-bottom, vertical vibration of hull girder, etc.

7.7 Criteria

7.7.1 Introduction

(1) As mentioned in 7.1, severe shafting longitudinal vibration may lead to mechanical fault of propulsion shafting itself, and also cause the severe ship vibration. In order to prevent and eliminate the adverse shafting longitudinal vibration phenomenon, in addition to the calculation for shafting longitudinal vibration in the design, the rules requirements or the requirements of manufacturers are to be complied with. Here, the criteria have been provided only with the angle to prevent crankshaft bending and fatigue damage and the weardown of gear.

(2) If the manufacturer provides the corresponding criteria of amplitude for longitudinal vibration, the calculations for acceptability factor of fatigue strength is to be submitted for examination according to the requirements of Appendix 3, Chapter 9 of PART THREE of ISC Rules for Classification of Sea-going Steel Ships.

7.7.2 The rules requirements

The requirements of shafting longitudinal vibration in 12.3.3 of PART THREE of ISC Rules for Classification of Sea-going Steel Ship are as follows:

(1) The amplitude of longitudinal vibration at the free end of crankshaft under continuous running

condition induced by shafting longitudinal vibration for the propulsion shafting of diesel engine within the range of r = 0～1.0 is not to exceed the value calculated as follows:

$$[A_{a1}] = \frac{R[\Delta a_0]}{2(\Delta a_k)_{\max}(R + \frac{d_j}{2})} \qquad \text{mm} \qquad (7.7.1)$$

where: $[A_{a1}]$ — allowable amplitude of longitudinal vibration at the free end of crankshaft under continuous running condition, in mm;

R — radius of crank, in mm;

$(\Delta a_k)_{max}$ — maximum value of relative amplitude difference of crankshaft for longitudinal mode shape to be calculated, in mm;

d_j — diameter of main journal for crankshaft, in mm;

$[\Delta a_0]$ — maximum value of allowable crankshaft deflection, in mm.

(2) The allowable amplitude of longitudinal vibration under transient running state is generally to be 1.5 times the allowable value under continuous running condition.

(3) Where the allowable value under continuous running condition is exceeded, a restricted speed range is to be set. In general, the amplitude of longitudinal vibration induced by resonance or upper wave slope is not to exceed the allowable value under continuous running condition for r =0.85, and the amplitude of longitudinal vibration induced by resonance or lower wave slope is also not to exceed the allowable value under continuous running condition for r =1.0.

(4) According to the empirical data or detailed calculation material provided by manufacturers, the allowable amplitude of longitudinal vibration provided may be adopted.

7.7.3 Criteria of longitudinal vibration for gear-driven shafting system

(1) Criteria of longitudinal vibration for turbine propulsion shafting system

Where the acceleration in way of gear is limited not to exceed 0.10g, the allowable amplitude of longitudinal vibration in way of gear $[A_{a1}]$ is:

$$[A_{a1}] = 0.089\left(\frac{100}{6f_a}\right)^2 \qquad \text{mm} \qquad (7.7.2)$$

where: f_a — frequency of shafting longitudinal vibration, in Hz.

(2) Criteria of longitudinal vibration for diesel engine propulsion shafting with transmission gearing

Where the acceleration in way of gear is limited not to exceed 0.15g, the allowable amplitude of longitudinal vibration in way of gear is:

$$[A_{a1}] = 0.134\left(\frac{100}{6f_a}\right)^2 \qquad \text{mm} \qquad (7.7.3)$$

7.7.4 Miscellaneous

(1) The criterion value of shafting longitudinal vibration provided in this Section is applicable to either the response calculation or the measurement results of shafting longitudinal vibration. However, where the theoretical calculated value of natural frequency for shafting longitudinal vibration is more than 5% of the measured result, necessary correction is to be made to meet the

requirements.
(2) Where the measured amplitude of longitudinal vibration exceeds the rules specified value or the certified value given by manufacturer, the measured value may be substituted for acceptability factor of fatigue strength check, where the requirement is complied with, the safety of crankshaft may be acceptable. For details, see Appendix 3, Chapter 9 of PART THREE of ISC Rules for Classification of Sea-going Steel Ships.

7.8 Precautions

7.8.1 Introduction

In order to prevent the shafting fault and stern and superstructure vibration induced by shafting longitudinal vibration, the shafting is to be rationally designed so that the amplitude of shafting longitudinal vibration for the main engine within its normal speed range or special service speed range will not exceed the criterion value, otherwise, necessary measures of frequency modulation are to be taken to remove the resonance speed of adverse longitudinal vibration rapidly. If it is difficult to take the measures of frequency modulation due to constrained conditions, appropriate vibration damping measures are to be taken into consideration. Here, the precautions are briefly described only from point of view of shafting longitudinal vibration.
It is to be pointed out that for final solution of the adverse shafting longitudinal vibration, the calculated results of shafting torsional vibration, whirling vibration and shafting alignment, as well as the change of propeller surface force, are also to be considered. Therefore, a general consideration is to be taken.

7.8.2 Frequency modulation

(1) Altering the shafting longitudinal stiffness
Alteration of the length or diameter of shafting may raise or reduce the natural frequency of shafting longitudinal vibration so as to remove the resonance speed of adverse longitudinal vibration. The effects of shafting longitudinal stiffness on the natural frequency of longitudinal vibration is varied by the change of device, mode shape and mode shape curve, which is to be determined by calculation.
(2) Additional mass installed in shafting
Installation of an additional mass in way of the larger relative amplitude of shafting longitudinal vibration or regulation of the flywheel mass of main engine may reduce the natural frequency of shafting longitudinal vibration, and also change the mode shape for the purpose to prevent the resonance speed of adverse longitudinal vibration or decrease the amplitude. The position and scale of additional mass are to be determined according to the related mode shape to be solved.
In general, as the relative amplitude of 0 node longitudinal vibration in way of flywheel is less, the influence of flywheel mass change on the natural frequency and mode shape of 0 node longitudinal vibration is not great. Therefore, where the issues for characteristics of shafting torsional vibration are solved or improved by inertia method of engine flywheel, the influence of 0 node shafting longitudinal vibration may not be considered. However for 1st node longitudinal vibration, a certain influence may occur with the change of shafting arrangement, which is to be noted.
The relative amplitude of 0 node longitudinal vibration at the free end of crankshaft is larger, if

side flywheel is installed at the free end, the frequency of 0 node shafting longitudinal vibration may be reduced rapidly, and the mode shape also changes. This indicates that the influence on longitudinal vibration is to be noted where the issues for characteristics of shafting torsional vibration are solved or improved by installation of side flywheel so as to prevent resonance speed of adverse longitudinal vibration being within the normal speed range or causing torsional-longitudinal coupled vibration phenomenon.

(3) Altering of propeller blade number

Where the longitudinal vibration is induced by propeller excitation, alteration of blade number may remove the resonance speed of adverse longitudinal vibration.

(4) Altering of engine firing order

In addition to the structural dimension, the longitudinal stiffness of crankshaft is related to angle between adjacent cranks, hence, alteration of engine firing order may change the value of longitudinal stiffness of crankshaft, further to change the natural frequency of shafting longitudinal vibration.

7.8.3 Installing FM damper

From the formula (7.4.8), the vector sum of relative amplitude difference $\sum \Delta a_k$ is related to engine firing order. Where adverse longitudinal vibration response is induced by larger secondary harmonic, the alteration of engine firing order may decrease $\sum \Delta a_k$ so as to reduce the excitation energy input system. However, particular attention is to be given to the change of $\sum \Delta a_k$ for the other harmonics.

If it is induced by main harmonic, a FM damper for longitudinal vibration is to be installed at the free end of crankshaft. Due to the fact that relative motion is caused with the disk connected with crankshaft as a whole within a hydraulic cylinder of the damper which is fixed on the girder of main bearing housing for diesel engine, a certain longitudinal constraint will be acted on the free end of crankshaft so as to raise the natural frequency of shafting longitudinal vibration, at meanwhile, due to the damping function of hydraulic oil, the amplitude of longitudinal vibration will be decreased for the purpose of to minimize the vibration.

The low-speed two-stroke cycle diesel engine propulsion shafting may cause 0 node shafting longitudinal vibration, as a convention, FM damper of longitudinal vibration has been designed to be as a part for the main engine to supply by the diesel engine manufacturers currently so that no resonance speed of shafting longitudinal vibration will be caused within the normal speed range. For ships with longer shafting, such as container ships, 1st node shafting longitudinal vibration may be caused, it is necessary to install longitudinal damper in way of the position with larger amplitude in order to prevent the strong frame longitudinal vibration and/or superstructure vibration caused by shafting longitudinal vibration.

The diagram of FM damper for longitudinal vibration and damper for torsional vibration installed by RTA diesel engine are shown in Figure 7.8.3-1. The effects of FM damper are shown in Figure 7.8.3-2.

In order to prevent the crankshaft fault or frame longitudinal vibration due to insufficient oil supply for damper, inlet pressure indicating and controlling device or alarming device for excessive amplitude of longitudinal vibration is to be provided for the damper. Maintenance is to be carried out in the daily work.

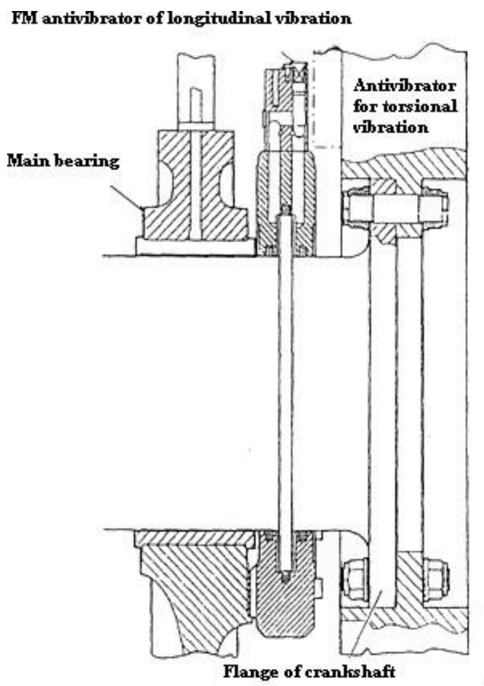


Figure 7.8.3-1 Diagram of FM Damper for Longitudinal Vibration and Damper for Torsional Vibration

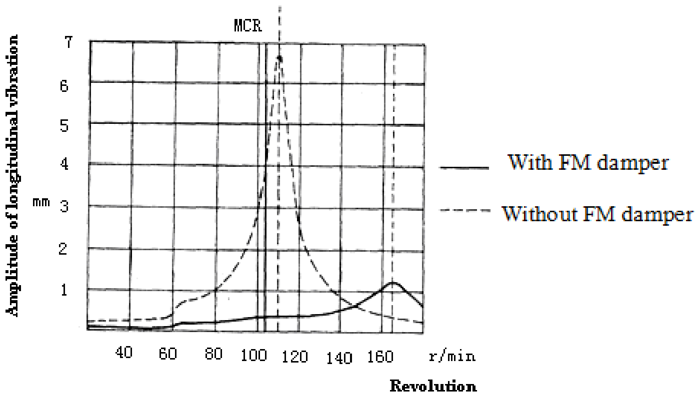


Figure 7.8.3-2 Effects of FM Damper

7.8.4 Miscellaneous

(1) Where the amplitude of longitudinal vibration measured at the free end of crankshaft is larger and caused by shafting torsional vibration, measures are to be taken to eliminate the shafting torsional vibration.

(2) Where the individually calculated frequency of shafting torsional vibration is similar to or equivalent to natural frequency of longitudinal vibration, strong torsional–longitudinal coupled vibration may occur, in this case, measures are to be taken to keep the relevant two frequencies as far as possible. Such circumstance is only to be considered in the design of diesel engine, consideration of the shafting coupled vibration in the full-scale shafting calculation is not necessary.

(3) Where strong vibration response of other vibrating bodies is induced by secondary excitation caused by shafting longitudinal vibration, measures are to be taken to keep the secondary exciting frequency and natural frequency of vibrating bodies as far as possible, or change the natural frequency of shafting longitudinal vibration or minimize the secondary excitation induced by shafting longitudinal vibration.
(4) Where the frequency of shafting longitudinal vibration can not be distanced from that of a certain order vertical vibration of hull girder and the exciting force is larger, the thrust shaft of diesel engine is to be so arranged in the loop of vertical mode shape of corresponding hull girder as far as possible.
(5) Although the longitudinal vibration level for axial shafting does not exceed the rules specified value, FM damper of shafting longitudinal vibration is generally to be installed for low-speed two-stroke cycle diesel engine in order to prevent the secondary excitation induced by shafting longitudinal vibration, which is further to induce the frame longitudinal vibration or superstructure longitudinal vibration or vertical vibration of hull girder. Meanwhile, maintenance instructions of FM damper of longitudinal vibration are to be provided by the manufacturers for crew use.

7.9 Plan Approval and Inspection

7.9.1 Plan approval

(1) The compliance of calculation parameters with the submitted drawings and calculation documents is to be examined.
(2) The accuracy of equivalent parameters of shafting longitudinal vibration is to be examined.
(3) Recheck by ISC-COMPASS software is to be carried out.
(4) Plan approval comments are to be made.
(5) Where amplitude of longitudinal vibration is found to reach 70% or above the rules specified value in the calculation, measurements are to be required.

7.9.2 Inspection

(1) Where measurements are required by plan approval comments, confirmation is to be made by the surveyor in the approval of navigation test program.
(2) The qualification of measuring unit and personnel are to be confirmed.
(3) Measurements of shafting longitudinal vibration are to be in compliance with the relevant requirements of Chapter 14 of the Guidelines.
(4) Measurement reports of longitudinal vibration are to be examined.
(5) The measurement results complying with the rules requirements are to be confirmed.
(6) Where necessary, the measurement report of longitudinal vibration is to be transferred to the plan approval center for recheck.
(7) Where the measured value exceeds the allowable one under continuous operation condition, the restricted speed range may be certified according to the measured value, and to comfirm such range has been marked in red in the revolution meter of main engine by the shipyard.

Chapter 8 WHIRLING VIBRATION OF SHAFTING

8.1 Introduction

8.1.1 Whirling vibration of propulsion shafting

A vibration phenomenon of procession caused by an axis rotating around its static balance curve under the function of transverse moment for propulsion shafting revolving on propeller or rotating shaft is called shafting whirling vibration.

When the propeller is running in the inhomogeneous wake filed, periodic bending moment will be induced on the propeller shaft, the harmonic order of exciting force is $v_p = iZ_P$ (i=1, 2, …), Z_P is the number of blades. The main harmonic order is the blade order $v = Z_p$.

Second, the exciting force induced by unbalance of propeller or fan, bending of shaft, misalignment of flange is normally 1st order harmonic. However, different modulus of section for the main shafts along shafting length direction may cause exciting force of 2nd order harmonic. For shafting with cardan, exciting force of 2nd order harmonic may be caused.

In addition, the external exciting force may induce shafting whirling vibration through foundation and bearing, the harmonic order is same as the number of external exciting force.

The calculation software of shafting whirling vibration developed by ISC in 1980's has been updating and improving continuously and may be applied to design for shafting whirling vibration, plan approval and evaluation for shafting fault analysis.

8.1.2 Hazard of shafting whirling vibration

Severe shafting whirling vibration will cause the following mechanical faults and vibration phenomenon:

(1) the aftermost sterntube bearing is overheated or early worn and causes the corrosion of shaft bushing;

(2) additional alternating bending stress is caused in way of screwshaft, particularly within the front area of keyways of tapered ends, even fatigue damage such as crake, broken, etc.;

(3) the power inducing bearing reaction is increased to further cause the vibration of hull structures at the stern;

(4) the sealing device of sterntube leaks, even early damaged.

8.1.3 General characteristics of shafting whirling vibration

For propulsion shafting with low-speed high-power diesel engine onboard the civilian merchant ship, resonance induced by centrifugal force of unbalanced mass for propeller will not occur due to the first natural frequency of its 1st order whirling vibration is rather higher than the rated speed.

However, for shafting with brackets, shafting with more number of blades (five blades or six blades), or shafting of passenger ship and high-speed craft, the first natural frequency of whirling vibration with blade order may be reduced within the range of rated speed to cause the shafting resonance.

In shafting whirling vibration design, even if the resonance speed of whirling vibration is beyond the working speed range, the natural frequency of whirling vibration will be reduced due to poor shafting alignment and voidable load of sterntube forward bearing, the resonance speed may still be within or approximate to the working speed range, to which due attention is to be given.

For large type ships, even if the resonance of whirling vibration will not occur in the working speed range, the whirling vibration response may be increased to an unnegligible degree due to larger exciting force of flow. In general, with the raise of speed of shafting, the amplitude of whirling vibration is increased, and the maximum achieves at the highest speed.

8.2 Calculating Models

8.2.1 Coordinate system

The shafting whirling vibration is described by a fixed three-dimensional coordinate system ($x - y - z$) shown in Figure 8.2.1. The static balance location of propeller center is to be taken as the original of coordinate o, x axis is for the bow direction, y axis is for horizontal port direction and z axis is upward.

The projections of propulsion shafting on the horizontal and vertical planes are shown in Figure 8.2.1. The vibration of any point on axial is shown by y and z coordinates. The corners of propeller plane θ_y, θ_z are to be taken as positive value from the positive direction of coordinate axis according to the right-hand rule.

The forces acting on propeller or shafting F_y, F_z are to be taken as positive value from the positive direction of coordinate axis, the moment (or bending moment) $M_y M_z$ is to be taken as positive value from the positive direction of coordinate axis according to the right-hand rule.

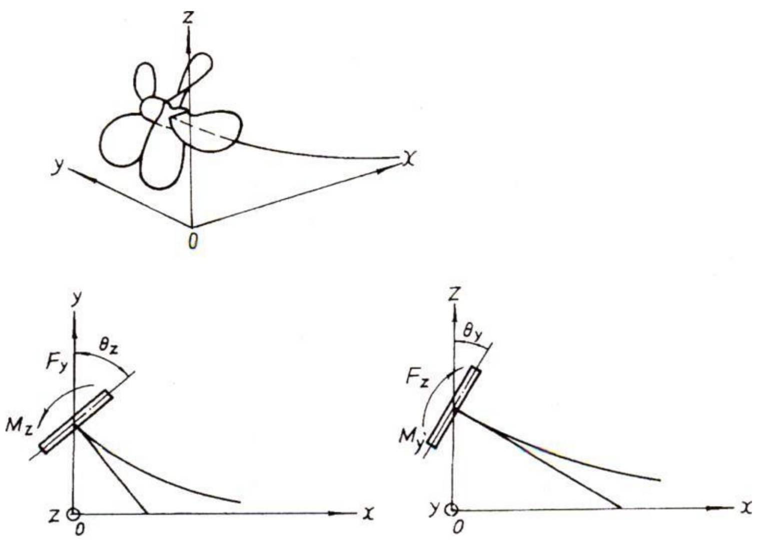


Figure 8.2.1 Projection of Propulsion Shafting in the Horizontal and Vertical Planes

8.2.2 Modes of shafting whirling vibration

The whirling vibration of propulsion shafting is a procession motion of multi-point supporting revolving shaft at the end of cantilever with propeller. The propeller and axis rotate around its geometrical axis with the angular velocity ω_s while the axis revolves around the ox axis with the angular velocity ω_n. The former is equivalent to a rotation while the latter is a revolution. The angular velocity ω_n is also called precession rate or whirling angular velocity.

The absolute angular velocity ω of any section on rotating axis is equal to the vector sum of rotating angular velocity ω_s and whirling angular velocity ω_n. For the micro amplitude vibration, the absolute angular velocity on any section of rotating axis may be similar to:

$$\omega = \omega_s + \omega_n \qquad (8.2.1)$$

Where the whirling angular velocity ω_n is in the same direction of rotating angular velocity ω, it is called as forward whirling, on the contrary, it is called as backward whirling, as shown in Figure 8.2.2.

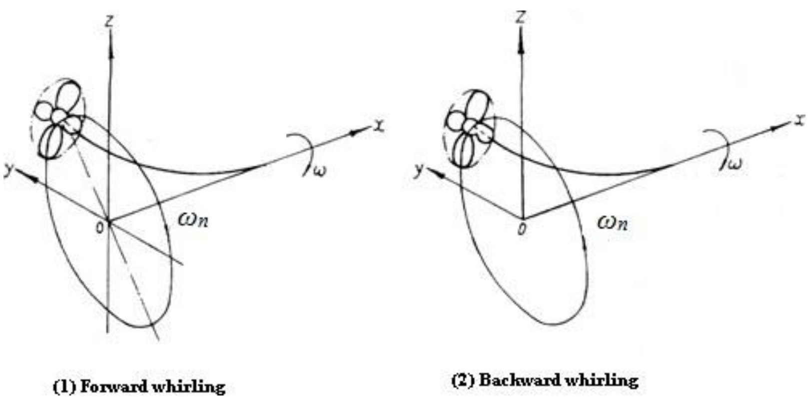


Figure 8.2.2 Shafting Forward Whirling and Backward Whirling

8.2.3 Equation of whirling free vibration

The simplified models of whirling free vibration shown in Figure 8.2.1 are discussed first. The propeller in the diagram has been simplified to a homogeneous thin disk, the mass and moment of inertia has included the function of entrained water, the rotating axis is simplified to an elastic axis without mass, and the damping is ignored. The equation of frequency of whirling vibration is as follows:

$$\begin{bmatrix} 1-m\omega_n^2\delta_{Mz} & J_d\omega_n^2\delta_{Mz} & 0 & J_p\omega\omega_n\delta_{Mz} \\ m\omega_n^2\phi_{wy} & 1-J_d\omega_n^2\phi_{Mz} & 0 & -J_p\omega\omega_n\phi_{Mz} \\ 0 & -J_p\omega\omega_n\delta_{Mz} & 1-m\omega_n^2\delta_{wz} & -J_d\omega_n^2\delta_{Mz} \\ 0 & -J_p\omega\omega_n\phi_{Mz} & -m\omega_n^2\phi_{wz} & 1-J_d\omega_n^2\phi_{My} \end{bmatrix} = 0 \qquad (8.2.2)$$

where: J_p, J_d — polar moment of inertia and radial moment of inertia of propeller respectively, in kgm^2;

ω — angular velocity of rotation for propeller, in rad/s;

θ_Z, θ_Y — calculated corner of axis in way of center of propeller within $x-y$ plan and $x-z$ plane, in rad;

m — mass of propeller (including the function of entrained water), in kg;

δ_w — deflection caused in way of geometric center as the unit of force is acting on propeller, in m/N;

ϕ_w — corner of axis caused in way of geometric center as the unit of force is acting on propeller, in rad/N;

δ_M — deflection caused in way of geometric center as the unit of moment is acting on propeller, in m/N;

ϕ_M — corner of axis caused in way of geometric center as the unit of moment is acting on propeller, in rad/N;

ω_n — frequency of whirling vibration, in rad/s.

Where the shaft does not revolve, i.e. $\omega = 0$, the formula (8.2.1) is the frequency equation for shafting transverse vibration, ω_n obtained by the equation is the natural frequency of shafting transverse vibration.

8.2.4 Natural frequency of whirling vibration

From the equation (8.2.1), the natural frequency of whirling vibration ω_n is related to m, J_p, J_d, δ_M, δ_w, ϕ_M, ϕ_w and ω, among which the other parameters are unchangeable except the operation parameter of angular velocity of propeller ω.

This shows that the natural frequency of shafting whirling vibration ω_n is changed with the angular velocity of shafting rotation. This is because the scale of moment of inertia for propeller (gyroscopic moment) is changed with the different ω.

The simplified model shown in Figure 8.2.1 has four freedoms, i.e. two removing degrees of freedom and two rotating degrees of freedoms. Therefore, for each angular velocity of rotation ω, four natural frequencies may be obtained by equation (8.2.1), the curve of relationship between natural frequency ω_n and angular velocity ω is shown in Figure 8.2.4-1, corresponding to 1st backward whirling, 1st forward whirling, 2nd backward whirling and 2nd forward whirling, respectively. The four points of intersection between curve and vertical coordinate ω_{y1}, ω_{z1}, ω_{y2}, ω_{z2} are the natural frequencies of transverse vibration for shafting within horizontal and vertical plane.

Where the supporting is isotropic, the curve of relationship between natural frequency of whirling vibration ω_n and angular velocity of rotation ω_n is shown in Figure 8.2.4-2. In practice, it is sufficient to calculate several natural frequencies corresponding to the points of intersections between frequency curves and straight lines $\omega_n = \omega$ and $\omega_n = kZ_p\omega$ (Z_p is the number of propeller blades, k =1, 2,…) in Figure 8.2.4-1 and Figure 8.2.4-2.

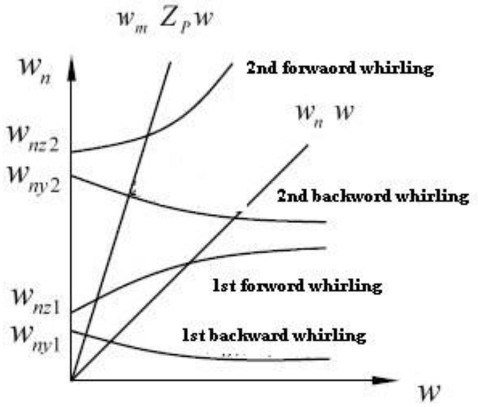


Figure 8.2.4-1 $\omega_n - \omega$ **for Anisotropy of Supporting Stiffness**

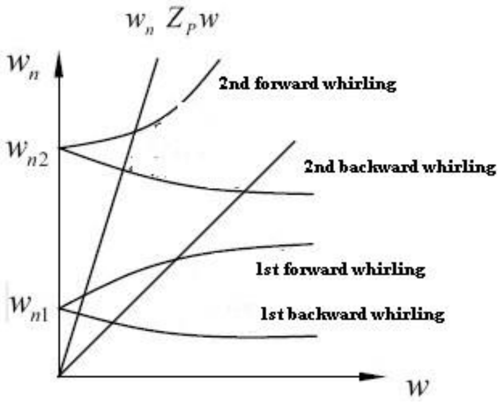


Figure 8.2.4-2 $\omega_n - \omega$ **for Isotropic of Supporting Stiffness**

8.2.5 Factors to influence the natural frequency of shafting whirling vibration

(1) Positions of supporting points for bearings

In the calculation and analysis of whirling vibration, the bearing is simplified to a supporting point, for the intermediate bearing and sterntube forward bearing, it may be regarded that the bearing reaction is evenly distributed along the axial direction and the supporting point is in way of the center of bearing.

For aftermost sterntube bearing, due to the function of propeller on cantilever end, the bearing reaction is distributed unevenly along the direction of bearing length and the point of action for resultant of bearing reaction is close to the stern and is influenced by the degree of wear of bearing, load and deformation of ship, etc. For the calculation of whirling vibration, the distance from supporting point of aftermost sterntube bearing to aft end of bearing liner L according to the following scope:

for bearing lined with white-metal: L= (1/7～1/3) L_b, in mm;
for bearing lined with lignum vitae: L= (1/4～1/3) L_b, in mm;
for bearing lined with rubber: L= (1/3～1/2) L_b, in mm;
for ball bearing: L= L_b/2, in mm.

where: L_b — length of aftermost sterntube bearing liner, in mm.

The positions of supporting points for bearing made of the approved compound material, such as Thordon, Feroform and Vesconite may be referred to bearing lined with white-metal for oil lubrication and referred to bearing lined with lignum vitae for water lubrication.

(2) Coefficient of entrained water for propeller

Two functions are for the entrained water of propeller, such as mass and moment of inertia, the precise value may be obtained by theoretical formula, for primary evaluation, coefficient of entrained water may be used:

coefficient of entrained water for mass: 1.10 to 1.30;
coefficient of entrained water for polar moment of inertia: 1.25 to 1.30;
coefficient of entrained water for radial moment of inertia: 1.50 to 2.0.

(3) Shafting alignment condition

Where the shafting alignment is poor, a negative bearing reaction occurs at certain bearing of shafting (particularly at the sterntube forward bearing), i.e. when there is voidable bearing load, the natural frequency of whirling vibration will be rapidly reduced, and may lead to that the critical speed of whirling vibration which is not a problem originally drops off and is within the operating speed range.

In addition, where the supporting point of aftermost sterntube bearing is worn out and moves forward gradually, the natural frequency of whirling vibration may be reduced.

(4) Supporting stiffness

Where the whirling vibration is calculated, an equivalent spring is to be used as the supporting stiffness. The stiffness of aftermost sterntube bearing is generally to be (1～3)×10^9 N/m; For the aftermost sterntube bearing for water lubrication, its stiffness may be obtained as 0.2×10^9 N/m. Stiffness of sterntube forward bearing and intermediate shaft bearing can be obtained as 0.1×10^9 N/m.

8.3 Calculation of Natural Frequency

8.3.1 Jasper formula

The simplified model of Jasper formula is shown in Figure 8.3.1.

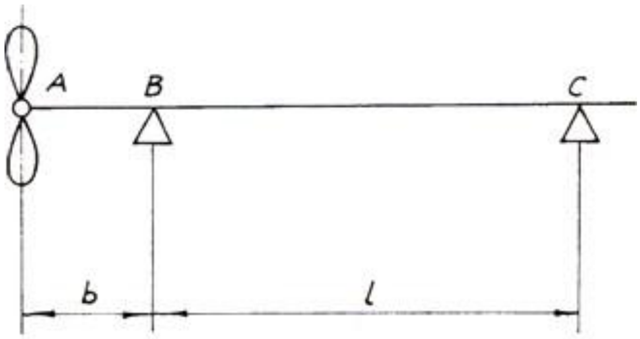


Figure 8.3.1 Simplified Model

The gyroscopic moment of propeller is to be taken into consideration by this method, the propeller is to be simplified as a thin disk. The 1st natural frequencies and 2nd natural frequencies of shafting whirling vibration are to be calculated by formula (8.3.1), if the denominator in radical is obtained as "+", it is the 1st natural frequency f_{h1}, if the denominator in radical is obtained as "–", it is the 2nd natural frequency f_{h2}:

$$f_{h1,2} = \frac{1}{2\pi}\sqrt{\frac{2}{Q_2 \pm \sqrt{Q_2^2 - Q_1 Q_3}}} \quad \text{Hz} \qquad (8.3.1)$$

where: $Q_1 = \delta_w \phi_M = \delta_M \phi_w$;

$\delta_w, \phi_w, \delta_M, \phi_M$ — same as 8.2.1;

$Q_2 = m_e \delta_\omega + G_g \phi_M$;

$m_e = m_p + 0.38 m_s$ — equivalent mass of propeller, in kg;

m_p — mass of propeller, considering function of entrained water, in kg;

m_s — mass of shaft, in kg;

$G_g = (1 - j_o h) J_d$, in kgm^2;

j_o — ratio of moment of inertia, $j_o = J_p / J_d$;

J_p — polar moment of inertia for propeller, considering function of entrained water, the coefficient of entrained water may be taken as 1.30 in the absence of exact data;

J_d — radial moment of inertia of propeller, considering function of entrained water, the coefficient of entrained water may be taken as 1.60 in the absence of exact data;

h — ratio of frequency, $h = \omega / \omega_n$;

ω — angular velocity of shafting rotation, in rad/s;

ω_n — angular frequency of whirling vibration, in rad/s;

$Q_3 = 4 m_e G_g$.

As seen from the above, the scales of Q_2, Q_3 in formula (8.3.1) are related to ratio of frequency h. The value of h is to be selected for calculation in advanced. In general, the value of h is only to be taken as equal to ± 1, $\pm \frac{1}{Z_p}$.

h is a forward whirling when in positive, h =1 for the 1st order forward whirling, $h = \frac{1}{Z_p}$ for the blade order of forward whirling.

h is a backward whirling when in negative, h =−1 for the 1st order backward whirling, $h = -\frac{1}{Z_p}$ for the blade order of backward whirling.

The critical speed of 1st order forward and backward whirling vibration $n_{h=\pm1}$ is to be calculated by:

$$n_{h=\pm1} = 60 f_{h=\pm1} \quad \text{r/min} \qquad (8.3.2)$$

The critical speed of forward and backward whirling vibration of blade order $n_{h=\pm\frac{1}{Z_p}}$ is to be calculated by:

$$n_{h=\pm\frac{1}{Z_p}} = 60 f_{h=\pm\frac{1}{Z_p}} \frac{1}{Z_p} \qquad (8.3.3)$$

8.3.2 Transfer matrix method

(1) The shafting whirling vibration is to be calculation by transfer matrix method, simplified model combining lumped element with distribution parameter element is generally to be used, as shown in Figure 8.3.2-1.

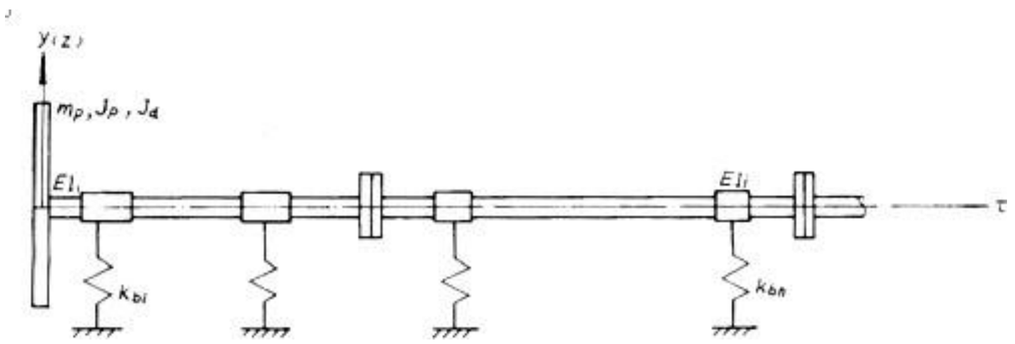


Figure 8.3.2-1 Simplified Model

The detailed simplification principles are as follows:

① The propeller is simplified as a homogeneous rigid disk element with the mass and moment of inertia as lumped parameter, the function of entrained water is to be considered.

② The propeller shaft, sterntube shaft, intermediate shaft are to be divided to homogeneous shaft elements with constant cross-sections according to natural assemblies. In order to draw the curve of mode shape, the shaft elements may be subdivided appropriately, in general, shaft whirling effect and the influences of shearing force and radial thrust force may be ignored.

③ Where the bending stiffness of flange connection is larger than that of shaft, its bending deformation may not be taken into consideration, the two connecting flanges may be directly processed as homogeneous rigid disk or shaft element. Where the connecting blot is bent or deformed and can not be ignored, a spring may be used to substitute.

④ Bearing is to be treated as an elastic hinge.

⑤ The boundary of propeller at shafting end is a free end, the boundary conditions of fore end are divided to flywheel, gearbox and wheel according to it fore end elements, and are respectively to be fixed end, hinge end or free end. For the long shafting onboard ships which the engine room is located in the middle, the boundary conditions of fore end has less effect on the value of natural frequency.

For lift shafting of high-speed craft:

① The fan may be simplified as rigid disk and its gyroscopic effect is to be taken into consideration.

② The rigid coupling is to be considered as a concentrated mass or distributed mass. Where the semi-rigid coupling is used between fans, the coupling is to be divided into driving and driven units and the bending and shearing effects are to be considered.

③ Calculation is to be carried out from the pump to the high-elastic coupling, the boundary conditions of high-elastic coupling is to be regarded as free end.

(2) Element type and state vector

The simplified model shown in Figure 8.3.2-1 includes two types of two-terminal elements and one type of three-terminal element.

The first type of two-terminal element is: homogeneous rigid disk element and homogeneous shaft element. The second type of two-terminal element is each element of supporting branch system, i.e. lumped mass element and spring element. The four-terminal element is supporting pseudo element which two terminals are connected with the left and right shaft elements respectively, and the other two terminals are connected with supporting systems in horizontal and vertical direction.

The definition of state vector Z_i for the first type of two-terminal element and three-terminal element is:

$$Z_i = \{y \quad \theta_z \quad M_z \quad Q_y \quad z \quad \theta_y \quad M_y \quad Q_z\}_i^T \qquad (8.3.4)$$

where: y, z — deflection in horizontal or vertical direction, in m;

θ_z, θ_y — inclination in the horizontal or vertical plane, in rad;

M_z, M_y — bending moment in horizontal or vertical plane, in Nm;

Q_y, Q_z — shearing force in horizontal or vertical plane, in N;

i — suffix, indicating the No. of elements in the system.

The definition of state vector Z_i for the second type of two-terminal element is:

$$Z_i = \{y \ F_y \quad z \ F_z\}_i^T \qquad (8.3.5)$$

where: y, z — displacement in horizontal or vertical direction, in m;

F_y, F_z — force in horizontal or vertical direction, in Nm;

i — suffix, indicating the No. of elements in the system.

Two superscripts L and$R$ are for the state vector and its each element, indicating left and right ends of the element respectively.

Where the anisotropy of supporting is not taken into consideration for calculation, the state vector Z_i for the first type two-terminal element and four-terminal element may be simplified to:

$$Z_i = \{y \ \theta \ M \ Q\}_i^T \qquad (8.3.6)$$

where: y — deflection, in m;

θ — inclination, in rad;

M — bending moment, in N.m;

Q — shearing force, in N.

The state vector Z_i for the second two-terminal element may be simplified to:

$$Z_i = \{y_o \;\; F_y\}_i^T \tag{8.3.7}$$

(3) Transfer matrix of element

Where the whirling vibration is calculated, anisotropy of supporting is generally not considered, the transfer matrix of each element is as follows:

① Homogeneous rigid disk element

The force of homogeneous rigid disk element in the projection of horizontal plane is shown in Figure 8.3.2-2.

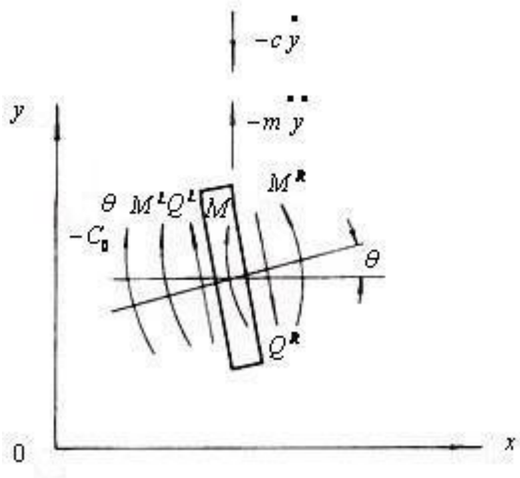


Figure 8.3.2-2 Force of Homogeneous Rigid Disk Element

The formula of state vectors at its left and right ends may be as follows:

$$Z_p^R = T_p Z_p^L \tag{8.3.8}$$

where: Z_p^R, Z_p^L — state vectors of left and right ends of disk element;

T_p — transfer matrix of disk element, i.e.:

$$T_p = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & (j_0 h - 1) J_d \omega_n^2 & 1 & 0 \\ m\omega_n^2 & 0 & 0 & 1 \end{bmatrix} \tag{8.3.9}$$

where: J_p — polar moment of inertia of disk element, in kgm^2;

J_d — radial moment of inertia of disk element, in kgm^2;

m — mass of disk element, in kg;

ω_n — circular frequency of whirling vibration, in rad/s;

j_o, h — same as 8.3.1.

Where the moment of inertia of disk is not considered, and only treated as a lumped mass, its transfer matrix T_p may be simplified to:

$$T_p = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ m\omega_n{}^2 & 0 & 0 & 1 \end{bmatrix} \tag{8.3.10}$$

② Homogeneous shaft element

The homogeneous shaft element is a straight circular shaft with constant cross-section, the force of projection in horizontal plan is shown in Figure 8.3.2-3.

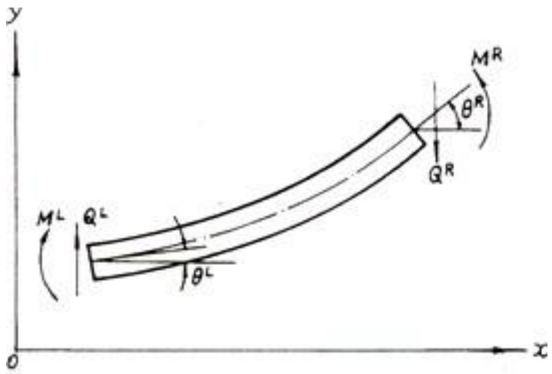


Figure 8.3.2-3 Force of Homogeneous Shaft Element in the Projection of Horizontal Plane

According to the relationship of displacement and force between left and right ends, the formula of state vector at both ends of element may be:

$$Z_s^R = T_s Z_s^L \tag{8.3.11}$$

where: T_s — transfer matrix of homogeneous shaft element, i.e.:

$$T_s = \begin{bmatrix} T_{11} & T_{12} & T_{13} & T_{14} \\ T_{21} & T_{22} & T_{23} & T_{24} \\ T_{31} & T_{32} & T_{33} & T_{34} \\ T_{41} & T_{42} & T_{43} & T_{44} \end{bmatrix} \tag{8.3.12}$$

where: $T_{11} = c_o - P_3 c_2$;

$T_{12} = \dfrac{1+P_1}{B}[B - P_2 P_3]c_1 + (P_2 P_3^2 - BP_3 - BP_4)c_3]$;

$T_{13} = c_2(1+P_1)(B + P_2 K_2^2 - P_2 P_3)(B - P_2 K_1^2 - P_2 P_3) / BEL(B + P_2 P_4 - P_2 P_3)$;

$T_{14} = (1+P_1)^2[-P_3 c_1 + (B + P_3^2)c_3] / BEI$;

$T_{21} = Bc_3 /(1+P_1)$;

$T_{22} = c_o - P_4 c_2$;

$T_{23} = \dfrac{1}{EI}[c_1 - (P_2 + P_4)c_3]$;

$T_{24} = c_2(1+P_1)/EI$;

$T_{31} = BEIc_2 / (1+P_1)$;

$T_{32} = EI[-P_4 c_1 + (B - P_2 P_3 + P_2 P_4 + P_4^2)c_3]$;

$T_{33} = c_o - (P_2 + P_4)c_2$;

$T_{34} = (1+P_1)[c_1 - (P_2 + P_3 + P)C_3]$;

$T_{41} = [BEI(c_1 - (P_2 + P_3)C_3] / (1+P_1)^2$;

$T_{42} = EI[B + P_2 P_4 - P_2 P_3]c_2 / (1+P_1)$

$T_{43} = [-P_2 c_1 + (B + P_2^2 + P_2 P_4) c_3]/(1 + P_1)$;

$T_{44} = c_o (P_2 + P_3) c_2$;

where: $c_o = [K_2^2 ch(K_1 l) + K_1^2 \cos(K_2 l)]/(K_1^2 + K_2^2)$;

$c_1 = [\frac{K_2^2}{K_1} sh(K_1 l) + \frac{K_1^2}{K_2} \sin(K_2 l)]/(K_1^2 + K_2^2)$;

$c_2 = [ch(K_1 l) - \cos(K_2 l)]/(K_1^2 + K_2^2)$;

$c_3 = [\frac{1}{K_1} sh(K_1 l) - \frac{1}{K_2} \sin(K_2 l)]/(K_1^2 + K_2^2)$;

$P_1 = F_x k / AG$;

$P_2 = F_x (1 + P_1) / EI$;

$P_3 = \rho k \Omega^2 / G$;

$P_4 = -\rho\ (2h - 1)\ \Omega^2 / E$;

$B = \rho A \Omega^2 (1 + P_1)^2 / EI$;

$K_{1,2} = [\sqrt{B + \frac{1}{4}(P_2 + P_4 - P_3)^2} \mp \frac{1}{2}(P_2 + P_3 + p_4)]^{\frac{1}{2}}$;

$h = \omega / \Omega$;

$I = \frac{\pi}{64}(D^4 - d^4)$;

$A = \frac{\pi}{4}(D^2 - d^2)$;

$k = 1.11$;

G — modulus of elasticity of shearing for material, in N/m^3;

E — modulus of elasticity of material, in N/m^3;

ρ — density of material, in kg/m^3;

D, d — external and internal diameters, in m.

③ Elastic element of flange connecting bolt

The formula of state vector of both sides for elastic element of connecting bolt between two shaft flanges is:

$$Z_F^R = T_F Z_F^L \tag{8.3.13}$$

where: Z_F^R, Z_F^L — state vectors of left and right ends of connecting blot;

T_F — transfer matrix of connecting bolt, i.e.:

$$T_F = \begin{bmatrix} 1 & 0 & 0 & 1/k_f \\ 0 & 1 & 1/k_m & 0 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{8.3.14}$$

where: k_f — shearing stiffness of connecting spring element, in N/m;

k_m — bending stiffness of connecting spring element, in Nm/rad.

④ Supporting element

The supporting element is a three-terminal one, its left and right ends are connected with shaft element and the third one is connected with supporting branch system.

In the lumped parameter system model, the supporting element is generally a homogeneous rigid disk element (or lumped mass element). The force and deformation are shown in Figure 8.3.2-4.

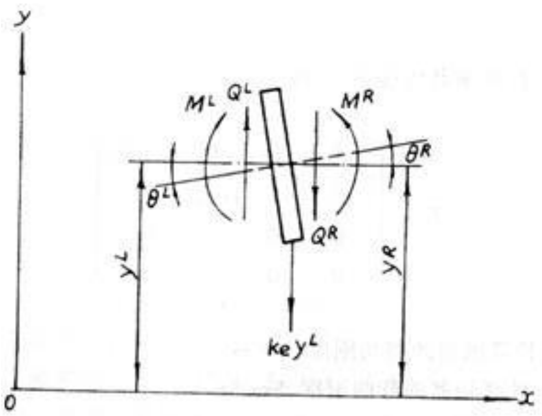


Figure 8.3.2-4 Force/deformation of Supporting Element

Where harmonic vibration of system occurs with an angular frequency ω_n, the formula of state vector at both ends is:

$$Z_b^R = T_b Z_b^L \tag{8.3.15}$$

where: T_b — transfer matrix of supporting element, i.e.:

$$T_b = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & (j_0 h - 1) J_d \omega_n^2 & 1 & 0 \\ (m - k_e / \omega_n^2)\omega_n^2 & 0 & 0 & 1 \end{bmatrix} \tag{8.3.16}$$

In the mixed model, shaft is usually treated as the shaft element with evenly distributed mass, in this case, the supporting element is a pseudo one without mass and elasticity, the transfer matrix T_b may be simplified to:

$$T_b = \begin{bmatrix} 1 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 \\ 0 & 0 & 1 & 0 \\ -k_e & 0 & 0 & 1 \end{bmatrix} \tag{8.3.17}$$

where: k_e — equivalent stiffness of bearing, in N/mm;

m — mass of disk, in kg;

other symbols are same as 8.3.1.

⑤ Semi-rigid elastic coupling

Semi-rigid elastic coupling may be grouped into driving and driven parts, the formula of state vector is as follows:

$$Z_t^R = T_t Z_t^L \tag{8.3.18}$$

where: T_t — transfer matrix of semi-rigid elastic coupling, i.e.:

$$T_t = \begin{bmatrix} 1 & 0 & 0 & 1/k_j \\ 0 & 1 & 1/k_w & 1 \\ 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 1 \end{bmatrix} \tag{8.3.19}$$

where: k_j — shearing stiffness of semi-rigid elastic coupling, in N/m;

k_w — bending stiffness of semi-rigid elastic coupling, in Nm/rad.

(4) Transfer matrix method (MP method)

Because of being simple, flexible and easy to program, the requirements for computer memory being not so strict, and taking a short working time, the transfer matrix method (MP method) is a traditional and basic method for shafting vibration analysis and widely used up to now.

① Frequency equation

For the i^{th} element, transfer matrix of state vector between left and right ends:

$$\{Z\}_i^R = [T]_i \{Z\}_i^L \tag{8.3.20}$$

where: $[T]_i$ — transfer matrix of the i^{th} element;

$\{Z\}_i^L$、$\{Z\}_i^R$ — state vectors between left and right ends of the i^{th} element.

According to transfer matrix of each element $[T]_i$, cumulative matrix of the system $[T]_n$ may be obtained by:

$$[T]_n = \prod_{i=1}^{n} [T]_i \tag{8.3.21}$$

The transfer matrix of system is obtained by:

$$\{Z\}_n^R = [T]_n \{Z\}_1^L \tag{8.3.22}$$

where: $\{Z\}_n^R$、$\{Z\}_1^L$ — state vectors of both ends of the main branch, respectively.

Therefore, where the main branch end is free, the frequency equation is:

$$R_{es}(\omega_n) = \begin{vmatrix} T_{r13} & T_{r14} \\ T_{r23} & T_{r24} \end{vmatrix} \tag{8.3.23}$$

As the same way, where the main branch end is fixed, the frequency equation is:

$$R_{es}(\omega_n) = \begin{vmatrix} T_{r33} & T_{r34} \\ T_{r43} & T_{r44} \end{vmatrix} \tag{8.3.24}$$

Where the main branch end is a hinge, the frequency equation is:

$$R_{es}(\omega_n) = \begin{vmatrix} T_{r13} & T_{r14} \\ T_{r33} & T_{r34} \end{vmatrix} \tag{8.3.25}$$

② Calculation of natural frequency

The natural frequency is to be obtained by trial-and-tolerance method. In the calculation, the effect of damping may not be taken into account, the procedures are as follows:

(a) defining the relationship between equations ω and ω_n, i.e. giving the value of $h = \omega / \omega_n$, in general, only 1st order and blade order are required, maximum to the natural frequency of forward/backward whirling for twice blade order;

(b) assuming a serial of test frequency $\omega_0, \omega_0 + \Delta\omega, \omega_0 + 2\Delta\omega \ldots\ldots$, where ω_0 is the initial value of calculated frequency and $\Delta\omega$ is the step length of test frequency;

(c) for each calculated frequency, calculating the cumulative matrix of main branch according to formula (8.3.21). According to formulae (8.3.23) to (8.3.25), calculating the value of frequency polynomial Res (ω_n). Where the values of frequency polynomial (residual value) obtained from the former and the latter calculated frequencies are different, there is a zero on the band between the two calculated frequencies, which meets the boundary condition, i.e. the natural frequency of the system. Interpolation method may also be used.

③ Calculation of inherent mode shape:

For the free end of boundary, after the natural frequency is obtained:

$$\theta_1^L = -\frac{T_{r13}}{T_{r14}} y_1^L = a y_1^L$$

where: $a = -\frac{T_{r13}}{T_{r14}}$

may be as follows:

$$Z_1^L = \begin{Bmatrix} y \\ \theta \\ M \\ Q \end{Bmatrix} = \begin{Bmatrix} 1 \\ \alpha \\ 0 \\ 0 \end{Bmatrix} y_1^L \tag{8.3.26}$$

The state vector of right end of each element may be obtained from the cumulative matrix T_n:

$$Z_1^R = T_1 Z_1^L \begin{Bmatrix} \beta_1 \\ \beta_2 \\ \beta_3 \\ \beta_4 \end{Bmatrix}_1 y_1^L \tag{8.3.27}$$

$$Z_2^L = T_{r2} Z_1^L = \begin{Bmatrix} \beta_1 \\ \beta_2 \\ \beta_3 \\ \beta_4 \end{Bmatrix}_2 y_1^L \tag{8.3.28}$$

....

$$Z_n^R = T_{rn} Z_1^L = \begin{Bmatrix} \beta_1 \\ \beta_2 \\ \beta_3 \\ \beta_4 \end{Bmatrix}_n y_1^L \tag{8.3.29}$$

where: $\beta_1, \beta_2, \beta_3, \beta_4$ are the relative ratio between the deflection, inclination, bending moment, shearing force of right end section for each element and the calculated propeller deflection respectively. The y_1^L is assumed and the inherent mode shape is to be obtained from the deflection of right end of each element β_1. Similarly, the relative amplitude curves of inclination, bending moment and shearing force are to be obtained by the $\beta_2, \beta_3, \beta_4$ of right end of each element.

(5) Improved transfer matrix method (RMP method)

① Frequency equation

In the case of longer shafting, more bearing supports and disks and larger supporting stiffness and more natural frequencies, the transfer matrix method (MP method) may cause unsteady of values, i.e. irregular pulsation of frequency polynomial due to large number and small difference of residual value in the system. In order to improve the unsteady values by MP method, the improved transfer matrix method (RMP method) will be adopted. The four elements in state vectors of the system are to be divided to two groups according to the left end boundary condition, one is the element having zero on the left end boundary while the other is non-zero element.

$$\{Z\}_i = \begin{Bmatrix} M \\ Q \\ y \\ \theta \end{Bmatrix}_i = \begin{Bmatrix} U \\ V \end{Bmatrix}_i \tag{8.3.30}$$

where: $\{U\}_i = \begin{Bmatrix} M \\ Q \end{Bmatrix}_i$; $\{V\}_i = \begin{Bmatrix} y \\ \theta \end{Bmatrix}_i$.

Establishing Riccati converting relationship between U_i^L and V_i^L in the state vector Z_i^L of the same end, i.e.:

$$[U]_i^L = [R]_i^L [V]_i^L \tag{8.3.31}$$

where: $[R]_i^L$ —Riccati transfer matrix of the i th end of element.

For the left end of n^{th} element, may obtain:

$$\begin{Bmatrix} M \\ Q \end{Bmatrix}_n^R = \begin{bmatrix} R_{11} & R_{12} \\ R_{21} & R_{22} \end{bmatrix}_n^R \begin{Bmatrix} y \\ \theta \end{Bmatrix}_n^R \tag{8.3.32}$$

The frequency equation by RMP method is:
For free end:

$$R_{es}(\omega_n) = \begin{vmatrix} T_{r13} & T_{r14} \\ T_{r23} & T_{r24} \end{vmatrix} = \left|R_n^R\right| \left(\prod_{i=n-1}^{0} \left|[T_{21}R + T_{22}]_i^R\right| \right) = 0 \tag{8.3.33}$$

For fixed end:

$$R_{es}(\omega_n) = \begin{vmatrix} T_{r33} & T_{r34} \\ T_{r43} & T_{r44} \end{vmatrix} = \prod_{i=n-1}^{0} \left|[T_{21}R + T_{22}]_i^R\right|) = 0 \tag{8.3.34}$$

For hinge end:

$$R_{es}(\omega_n) = \begin{vmatrix} T_{r13} & T_{r14} \\ T_{r33} & T_{r34} \end{vmatrix} = (-1)^k \left|(R_{12})_n^R\right| \left(\prod_{i=n-1}^{0} \left|[T_{21}R + T_{22}]_i^R\right| \right) = 0 \tag{8.3.35}$$

where: k — order of $[R_{12}]_n^R$.

② Calculation of natural frequency

The value of frequency polynomial $R_{es}(\omega_n)$ is to be calculated according to frequency equation. Where the values of frequency polynomial (residual value) obtained from the former and latter calculated frequencies are different, there is a value between them, that is the natural frequency of the system. Interpolation method may also be used.

③ Calculation of inherent mode shape

After the frequency is obtained, the proportional relationship of state vectors of right end for each non-zero element in the system is to be defined, then state vectors of shaft end elements are to be ascertained from the right to the left in order to get the inherent mode shape.

(6) Calculation of resonance speed

The resonance speed n_h is to be calculated by:

$$n_h = 9.55|h|\omega_n \quad \text{r/min} \qquad (8.3.36)$$

where: h— frequency ratio used for calculating the natural frequency;

ω_n — natural frequency as the frequency ratio is h, in rad/s.

(7) Calculation of natural frequency of shafting with rigid supporting

In certain ships, the stiffness of bearing, particularly for the intermediate bearing, is higher than that of shafting system, in order to ensure the accuracy of calculation and not to lead to data overflow in the calculation, the calculation method different from that of elastic supporting is preferred.

The rigid supporting means the deflection of this supporting point is equal to 0. A sudden change of the shearing force in way of the supporting point cross-section may occur due to bearing reaction.

Several methods are used to process rigid supporting, only one of which is introduced to a modified transfer matrix method, i.e. transfer matrix is modified at supporting point according to the supporting conditions.

The processing method of rigid supporting is to remove an unknown element (θ or y) of state vector from the left end of rigid supporting and change to another unknown element (support reaction R). In the transferring, change will be done for each rigid supporting till to the right end of the system, then the frequency equation is obtained according to the right end boundary condition.

The calculation procedures are the same as 8.3.2(4)②.

8.4 Criteria

8.4.1 The rules requirements

The requirements of shafting whirling vibration in 12.4.3 of PART THREE of ISC Rules for Classification of Sea-going Steel Ships are as follows:

For shafting with brackets and cardan shafts, the blade order forward whirling resonance speeds are not to appear in the range of r = 0.85 to 1.0, and 1st order forward whirling resonance speeds are to be 20% more than the rated speed.

8.5 Precautions

8.5.1 Frequency Modulation

(1) Altering the number of propeller blades

Alteration of the number of blade (usually the number of blade is reduced) may greatly change the critical speed of blade order and the results are often better than other measures, but except for the 1st order critical speed modulation.

(2) Altering the size of shafting system

Alteration of the size of shafting system is normally to increase the diameter of propeller shaft and raise the stiffness, meanwhile, to increase the mass and moment of inertia with the effect to raise the natural frequency.

(3) Adjusting interval between bearings

Adjustment of interval between bearings, particularly for the distance between sterntube forward and afterward bearings or distance between the last two bearings closer to the propeller, has greater influence on the natural frequency of whirling vibration. Decrease of the interval between bearings may raise the natural frequency, however the unreasonable load distribution of bearings due to small interval is to be prevented. Increase of the interval between bearings may cause shafting whirling vibration or larger bearing load. See Chapter 9, 9.2.2 Recommendation of control for interval between bearings for details.

(4) Reasonable shafting alignment

The shafting alignment is to meet the rules requirements so as to make the load of each bearing for shafting is positive, particularly for the loads of sterntube bearing and intermediate shaft bearing are positive under various working conditions of the ship.

8.5.2 Reducing of exciting force

For large-type ships, due to the increase of main engine power, the excitation of propeller is raised, even if resonance is not caused within the range of running speed, the response of whirling vibration may be increased to an unnegligible degree. The basic means to resolve the vibration is to reduce the excitation of propeller.

8.6 Plan Approval and Inspection

8.6.1 Plan approval

(1) The compliance of calculation parameters with the submitted drawings and calculation documents is to be examined.

(2) The accuracy of equivalent parameters of shafting whirling vibration is to be examined.

(3) Recheck by ISC-COMPASS software is to be carried out.

(4) Plan approval comments are to be made.

(5) Where blade order forward whirling resonance speed is found to approximate to 0.85 rated speed or 1st order forward whirling resonance speed approximate to the rated speed, measurements are to be required.

8.6.2 Inspection

(1) Where measurements are required by plan approval comments, confirmation is to be made by the surveyor in the approval of navigation test program.

(2) The qualification of measuring unit and personnel are to be confirmed.

(3) Measurements of shafting whirling vibration are to be in compliance with the relevant requirements of Chapter 14 of the Guidelines.

(4) Measurement reports of whirling vibration are to be examined.

(5) The measurement results complying with the rules requirements are to be confirmed.
(6) Where necessary, the measurement report of whirling vibration is to be transferred to the plan approval center for recheck.

Chapter 9 SHAFTING ALIGNMENT

9.1 Introduction

9.1.1 Propulsion shafting alignment

The shafting alignment means to lay the screwshaft, tube shaft (if any), intermediate shaft, thrust shaft and engine shaft (diesel engine or gearbox) of propulsion shafting in a certain state by the corresponding method according to the relevant requirements.

The installation of propulsion shafting is an important link in the ship's manufacture, the shafting is to be installed in accordance with the approved calculation results of shafting alignment and corresponding installation process. Whether or not to design and install shafting alignment correctly and reasonably will directly influence the safety operation of shafting system onboard ships. Faults have occurred by poor shafting alignment sometimes, particularly in recent years, shafting faults also occur for large low-speed diesel engine due to poor shafting alignment.

This Chapter is developed by ISC based on the research results of years and in conjunction with the practical experience of shafting alignment from both at home and abroad, as a guidance for reference of the parties concerned.

In addition to those covered by rules in this Chapter, some issues are clarified on design, application, plan approval and survey for shafting alignment, to ensure the compliance of implementation with rules requirements . Furthermore, this Chapter provides the means to assist the industry in improving the design of shafting alignment.

The calculation software of shafting alignment developed by ISC in 1980's has been updating and improving continuously and may be applied to design for shafting alignment, plan approval and evaluation for shafting fault analysis.

9.1.2 Hazard of poor propulsion shafting alignment

The poor shafting alignment may lead to the following hazards on different degrees:

(1) the bearing load which support the propeller is excessive, particularly for the excessive local load in way of bearing aft end will accelerate the wear or damage of bearing, as shown in Figure 9.1.2;

(2) the load of sterntube forward bearing is small, even in negative, change of the interval between bearings will reduce the natural frequency of shafting whirling vibration and its resonance speed may be in the normal service range;

(3) the sealing device of sterntube forward bearing is damaged;

(4) the bearing of intermediate shaft is worn or damaged;

(5) the last 1 to 3 main bearings of the diesel engine are worn or damaged;

(6) the load difference between two bearings forward or afterward the reduction gear for the gear-driven shafting system is so big as to damage the establishment of oil film leading to poor engagement of bearing, and when severely, hammering of gears, overheating of thrust block and thrust bearing, melting of bearing alloy, etc., will occur;

(7) the stern vibration is induced.

Figure 9.1.2 Melting of Tube Afterward Bearing due to Poor Shafting Alignment

9.2 Number and Arrangement of Bearings

9.2.1 The rules requirements

The relevant requirements for shafting vibration and alignment in Chapter 12 of PART THREE of ISC Rules for Classification of Sea-going Steel Ships are as follows:

The alignment of main propulsion shafting systems and the arrangement of bearing are to be such as to give reasonable bending moments and bearing reactions and minimize the effects of hull deformation or bearing weardown on shafting alignment.

The number and arrangement of gears are to be reasonable. The interval between bearings is neither bigger nor smaller. Hull deformation, change of cold and heat conditions and weardown of bearing will have great influence on the loads of adjacent bearings and likely to cause inhomogeneous loads, even overload. Where the interval between bearings is bigger, shafting whirling vibration will be caused or loads of bearing will be larger.

9.2.2 Recommendation of control for interval between bearings

For the number and arrangement of bearing in the shafting design, the interval of bearing may be controlled as follows:

(1) The minimum interval between bearings $L_{\min}$ is:

$$L_{\min} \geq 5.5\sqrt{d} \quad \text{m} \qquad (9.2.1)$$

where: d— diameter of shaft between bearings, in m.

(2) The maximum interval between bearings

① The interval between bearings L is generally to be:

$$L \leq K_1\sqrt{d} \quad \text{mm} \qquad (9.2.2)$$

where: d— diameter required by the rules, in mm;

K_1 = 450 — for bearing with white-metal lined and oil lubricated;

$K_1 = 280$ — for tube bearing with grey cast iron lined and oil lubricated;

$K_1 = 280\text{-}350$ — for tube and strut bearing with rubber lined and water lubricated.

For shaft with speed more than 350r/min, the interval between bearings *L* is generally to be:

$$L \le K_2 \sqrt{\frac{d}{n}} \quad \text{mm} \qquad (9.2.3)$$

where: *d*— diameter required by the rules, in mm;

n — speed of shaft, in r/min;

$K_2 = 8400$ — for bearing with white-metal lined and oil lubricated;

$K_2 = 5200$ — for tube bearing with grey cast iron lined and oil lubricated, tube and strut bearing with rubber lined and water lubricated.

② The interval between sterntube forward and afterward bearings *L* is:

$$L \le 0.3\sqrt{d} \quad \text{m} \qquad (9.2.4)$$

where: *d*— basic diameter of screwshaft, in mm.

(3) Distance between thrust bearing (or wheel shaft afterward bearing) and its first afterward bearing

The distance between thrust bearing/wheel shaft afterward bearing and their first afterward intermediate shaft bearing or screwshaft forward bearing (without intermediate shaft) *L* has great influence on shafting alignment, particularly the hull deformation or bearing weardown have greater influence on the loads of bearing.

The distance between thrust bearing (or wheel shaft afterward bearing) and its first afterward bearing *L* is:

Short shafting of transmission gearings: $10d < L < 12d$ mm (9.2.5)

Shafting system directly driven by diesel engine: $6d < L < 12d$ mm (9.2.6)

where: *L*— distance between the center of thrust shaft bearing (or wheel shaft afterward bearing) and that of its first afterward bearing, in mm;

d— diameter of shaft, in mm.

(4) Interval between bearings recommended by WÄRTSILÄ

For the interval between intermediate shaft bearings, including interval between the intermediate shaft bearing and the last bearing of the diesel engine, 70% of the formula (9.2.2) is recommended by WÄRTSILÄ.

9.2.3 Example

Calculation is carried out for the interval between bearings with the diameters of intermediate shaft of 200 mm and screwshaft of 250 mm, the results are shown in Table 9.2.3.

Calculation of Interval between Bearings **Table 9.2.3**

Diameter of shaft (mm)	Min. interval between bearings (m)	Max. interval between bearings (for bearing with white-metal lined and oil lubricated with the speed below 350 r/min) (m)	Max. interval between intermediate shaft bearings recommended by WÄRTSILÄ (m)	Distance between the center of thrust shaft bearing and that of its first afterward bearing (m)	Distance between the center of wheel shaft afterward bearing and that of its first afterward bearing (m)
For intermediate shaft 200	2.4 to 2.46	4.2	3	1.2 to 2.4	2.0 to 2.4
For screwshaft 250	2.74 to 3.0	4.7 to 6.3			

9.3 Principle of Shafting Alignment and Methods

9.3.1 Introduction

(1) Shafting alignment

The shafting alignment means to lay the screwshaft, tube shaft (if any), intermediate shaft, thrust shaft and engine shaft (diesel engine or gearbox) in a certain state according to the relevant requirements and methods so as that the bearing load or/and additional bending stress of shaft or/and slope of screwshaft, etc., are within the allowable ranges of design and installation process to ensure the safety application of marine shafting systems. Shafting arrangement for a certain container ship is shown in Figure 9.3.1-1.

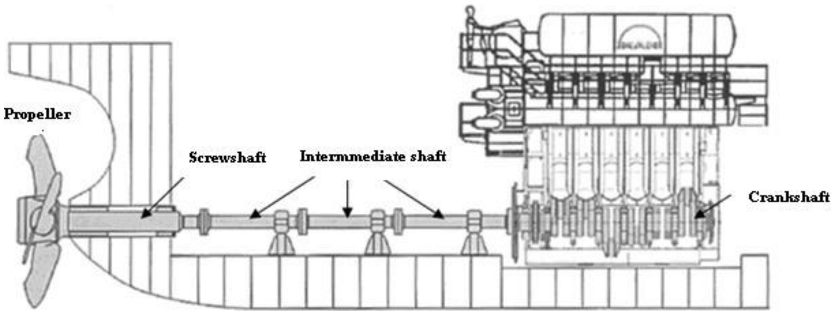


Figure 9.3.1-1 Shafting Arrangement for a Certain Container Ship

(2) Installation process of shafting alignment

According to the selected shafting alignment principles and methods, the installation process mainly includes, as appropriate:

— defining the reference line of shafting;

—slop-bored sterntube bearing or bearing inclining;

— pre-opening for diesel engine foundation;

— preliminarily defining the displacement of intermediate shaft bearing, gearbox bearing and main engine bearing;

— adjusting the sag and gap of flange;

— determining and measuring the flange bolts, bearing load; or measuring and adjusting the bearing load;

— defining the height of gaskets for intermediate shaft bearing, gearbox and main engine;

— fixing the bolts of intermediate shaft bearing, gearbox and main engine foundation;

— rechecking and measuring the bearing load;

— checking the engagement of gear;

— checking the difference between crank webs.

(3) Reference line of shafting

Based on the demand of ship's design, the propulsion shafting may adopt different arrangement type. The propulsion shafting is generally to be arranged in a line or inclined line, the primary working of shafting alignment is to define the reference line of shafting – theoretical centerline, i.e. define the centerline of shafting and the arrangement type of hull baseline.

The determination of reference line of shafting rests upon the shipyards, for the final shafting centerline is determined by the shafting alignment principles and its methods.

The reference line of shafting may be determined by optical instrument, laser collimator or pulling method, etc., as shown in Figures 9.3.1-2 and 9.3.1-3. The calibration plate of aft datum point is to be located in way of the aftermost sterntube bearing, the calibration plate of fore datum point is generally to be located in the fore bulkhead of engine room.

After determination of the reference line of shafting, tube pipe may be processed or installed. Then, the shafting system is to be aligned and installed in accordance with the shafting alignment principles and methods.

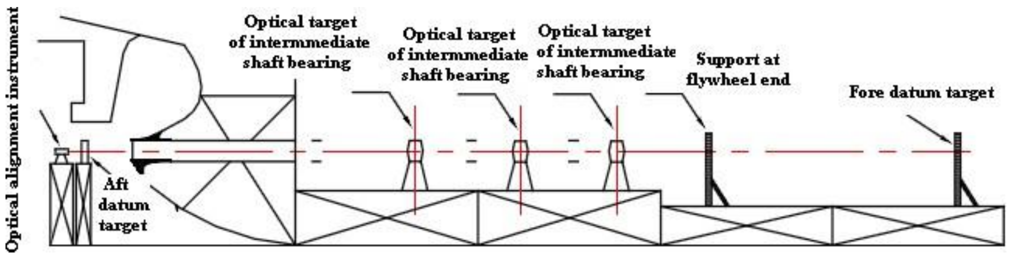


Figure 9.3.1-2 Reference Line of Shafting Determined by Optical Method

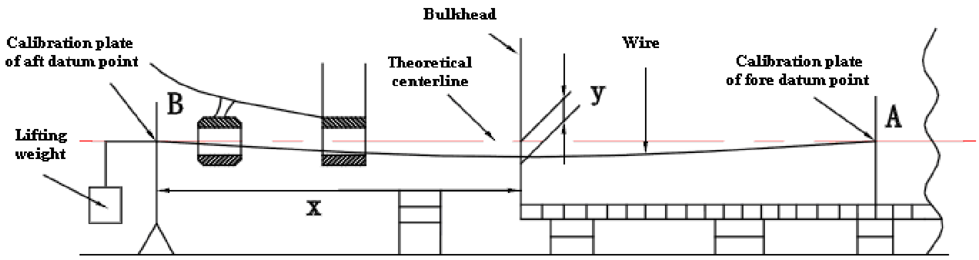
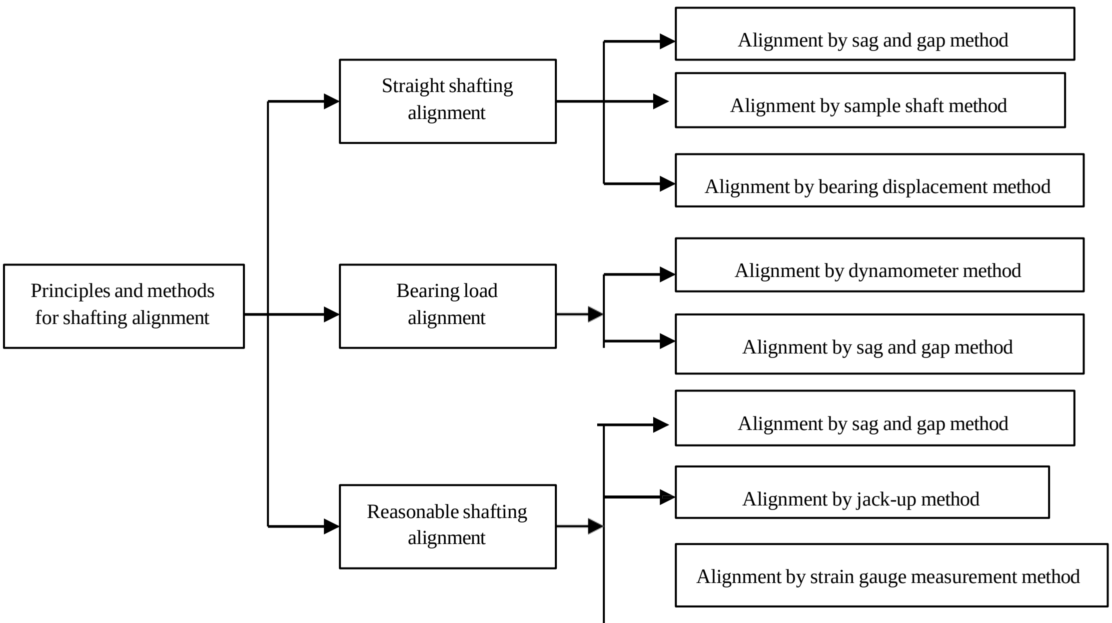


Figure 9.3.1-3 Reference Line of Shafting Determined by Pulling Method

(4) Shafting alignment principles

According to the shafting alignment principles, the main propulsion shafting alignment currently may be divided into three types, straight shafting alignment, bearing load alignment, reasonable shafting alignment. For civil ships, the straight shafting alignment and reasonable shafting alignment are mainly applied.

The shafting alignment principles and methods are shown in Figure 9.3.1-4.



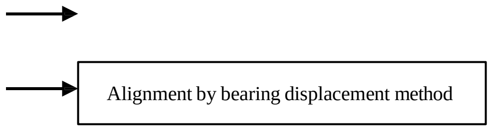


Figure 9.3.1-4 Shafting Alignment Principles and Methods

(5) Shafting alignment methods

For different shafting alignment principles, several alignment methods may be adopted; on the contrary, one alignment method may be applicable to different alignment principles: e.g. alignment by sag and gap method may apply to straight shafting alignment, bearing load alignment and reasonable shafting alignment; alignment by bearing displacement method may apply to straight shafting alignment and reasonable shafting alignment.

(6) Displacement of bearing

The vertical distance between bearing centerline and theoretical axial centerline is called displacement of bearing. According to the specifications of coordinate system, for the displacement of bearing related to the shafting – theoretical centerline, in general, downward is positive, upward is negative, but in some cases, downword is negative and upward is positive.

(7) Sag and gap

Theoretically, where two unconnected flanges are concentric, the adjacent two shafts with such flanges are also concentric, vice versa.

Due to the self-weight, displacement of bearing, installation tolerance, hull deformation, etc., the unconnected flanges is deviated from theoretical axial centerline. Although the names of flange parameters indicating the sap of theoretical axial centerline are different, the meanings are same. The definition of the Guidelines is shown in Figure 9.3.1-5.

Gap: means the axial centerlines of two unconnected flanges intersect at an angle, the product of the angle and diameter of flange is called gap. Where the axial distance between upper edges of two flanges are greater than that between lower edges, it is called as upper gap (negative value), on the contrary, it is called as lower gap (positive value).

Sag: means the axial centerline of two unconnected flanges are not coincided, the distance between upper edges or lower edges of two flanges is called sag. Where the flange of aft axis (left) is higher than that of fore axis (right), the sag is positive, on the contrary, it is negative.

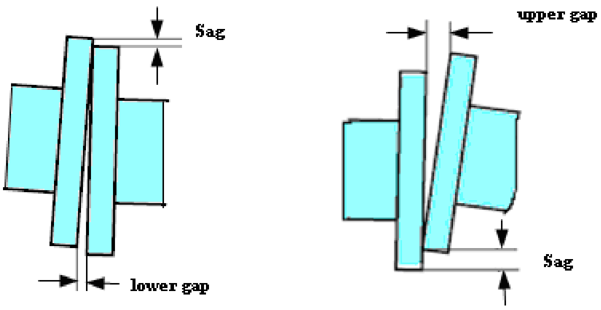


Figure 9.3.1-5 Gap and Sag of Flange

9.3.2 Straight shafting alignment

(1) Principles of straight shafting alignment

The principles of straight shafting alignment is an installation process to arrange the scerwshaft, tube shaft (if any), intermediate shaft, thrust shaft and gearbox or diesel engine in a line.The straight shafting alignment was ever commonly used, for people regarded as reasonable for the whole shaft to be arranged in a line. The straight shafting alignment, i.e. 0-0 alignment, the screwshaft and tube shaft, intermediate shaft, gearbox or diesel engine shaft are located in a theoretical axial centerline with the centerline is in straight, as shown in Figure 9.3.2.

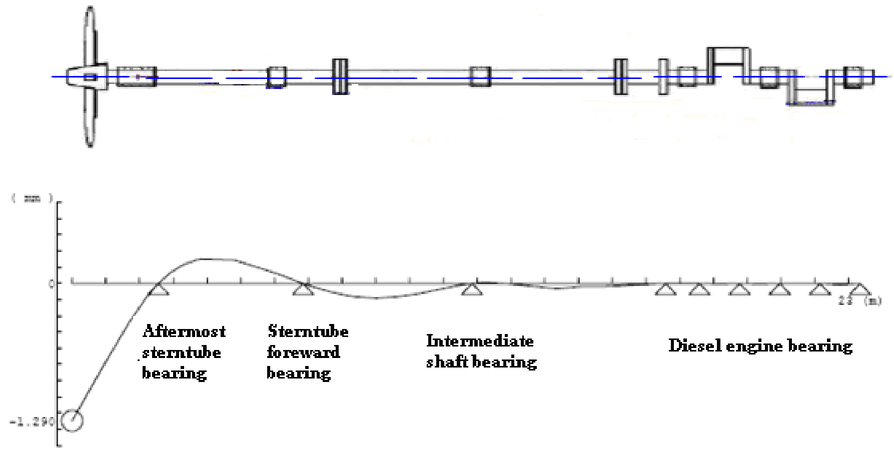


Figure 9.3.2 Shafting Centerline State for Straight Shafting Alignment

Theoretical alignment calculation for straight shafting alignment is not necessary, therefore the actual force and deformation of shafting alignment and actual bearing load can not be reflected. Also in the shafting design, it can not predict the poor loading distribution of each bearing for shafting system caused by unreasonable factors and with some hidden danger and risk. Therefore, alignment by gap and sag method may generally be adopted for shafting of civil small-type ships. Where the theoretical axial centerline method is used, straight shafting alignment is applicable.

(2) Implementation of straight shafting alignment

Three methods may be adopted for the implementation of straight shafting alignment:

① alignment by gap and sag method, the gap and sag values of each unconnected flange pairs are to be controlled within the range of 0.08 mm respectively, whether the displacement of bearing is in a straight line or not is determined based on the gap and sag values;

② alignment by sample shaft method;

③ alignment by displacement of bearing method, pulling method or optical instrument, etc., may be adopted to make the displacement of each bearing in zero.

(3) Elements of control for straight shafting alignment

For straight shafting alignment, the following four elements may be controlled:

① the length of bearing is to meet the rules requirements or in compliance with the design practice;

② the interval between bearings is to be reasonably arranged for neither longer nor shorter;

③ in order to minimize the effects of self-weight of shaft and flange, during the installation, temporary supporting is to be provided at appropriate places and the gap and sag values are to be controlled, or means such as pulling method or optical instrument or laser collimator are to be taken to make the displacement of bearing to be in zero;

④ the difference between crank webs or gear engagement are to be in compliance with the specifications of manufacturers.

9.3.3 Bearing load alignment for shafting systems

(1) Principles of bearing load alignment

For the principles of bearing load alignment, it is regarded that the self-weight of all intermediate shaft W is to be evenly born by its supported intermediate shaft bearing n, i.e. mean load $\pm P = W/n$. Meanwhile, the allowable additional load of sterntube forward bearing is $\pm P_{\text{stern}}$ (P_{stern} is the design load of sterntube forward bearing), the allowable additional load of thrust bearing is $\pm 0.5P_{\text{pushing}}$ (P_{pushing} is the design load of thrust bearing), which are not to be more than the load values as the rules specified.

For the bearing load alignment, it is also called as alignment by dynamometer method, and is an installation process to carry out primary alignment for screwshaft, tube shaft and gearbox or diesel engine, then to connect all of the bolts of shafting flanges, to install a dynamometer under the bearing housing through the adjustment of displacement for intermediate shaft bearing to make the actual loads of sterntube forward bearing, intermediate shaft bearing and thrust shaft bearing are within the specified range. For the bearing load alignment, the screwshaft and tube shaft, box gear shaft or diesel engine shaft are located in a straight line of theoretical centerline for shafting system, however the intermediate shaft bearing is not in the straight line of theoretical centerline for shafting system; the shafting centerline in way of intermediate shaft is in a curved state, as shown in Figure 9.3.3.

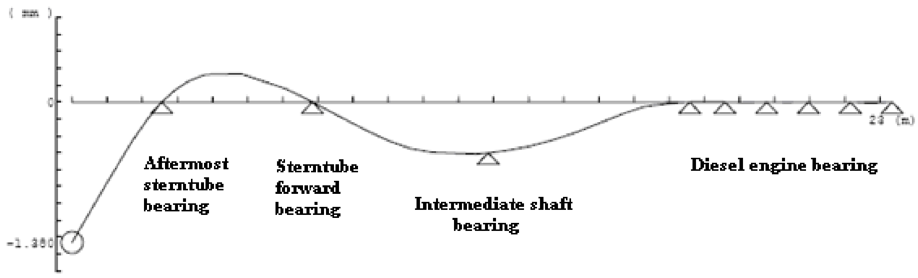


Figure 9.3.3 Shafting Centerline State for Bearing Load Alignment

Theoretical alignment calculation for bearing load alignment is not necessary, therefore the actual force and deformation of shafting alignment and actual bearing load can not be reflected. Also in the shafting design, it can neither predict the poor loading distribution of each bearing for shafting system caused by unreasonable factors nor judge the bending condition of shaft, inclination of aftermost sterntube bearing, etc., and with some hidden danger and risk. Therefore bearing load alignment may still be adopted for shafting of small-type ships; for naval ships bearing load alignment are more adopted.

Sometimes the bearing load alignment is confused with jack-up of reasonable alignment method, which are actually different in essence. The former is a method only to carry out measurement and adjustment of load for intermediate shaft bearing according to the calculated result of mean load for intermediate shaft bearing $P = W/n$ while the latter is a method only to carry out measurement and adjustment of load for relevant bearings according to the calculated result of reasonable shafting alignment.

(2) Implementation of bearing load alignment

Two methods may be adopted for the implementation of bearing load alignment:

① the alignment by dynamometer method is to carry out the measurement and adjustment through the dynamometer installed on each intermediate shaft bearing housing so as to make the loads of relevant bearings are to be within the allowable range based on the relevant loads of bearings;

② the alignment by gap and sag method is to determine the gap and sag values of each connecting flange pairs according to the allowable loads of bearings so as to make the gap and sag values of unconnected flanges are within a certain range based on the gap and sag values.

(3) Elements of control for bearing load alignment method

For bearing load alignment, the following four elements may be controlled:

① the length of bearing is to meet the rules requirements or in compliance with the design practice;

② the interval between bearings is to be reasonably arranged for neither longer nor shorter;

③ the load of intermediate shaft bearing measured by dynamometer is to be in compliance with the rules requirements; the load of intermediate shaft bearing is to be as evenly as possible, in general, is to be 0.5 to 1.5 times the mean load or is to be converted to control the gap and sag values of flanges through calculation;

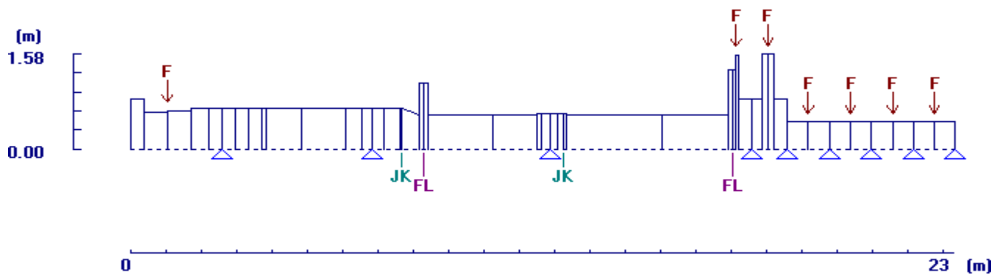
④ the difference between crank webs or gear engagement are to be in compliance with the specifications of manufacturers.

9.3.4 Reasonable shafting alignment

(1) Principles of reasonable shafting alignment

The principles of reasonable shafting alignment is a design and installation process to determine the shafting centerline through the theoretical calculation based on the conditions of bearing load, relative inclining angle of screwshaft in way of aftermost sterntube bearing and bending stress of shaft, and covert to displacement of each bearing.

For the reasonable shafting alignment, in general, the screwshaft and tube shaft are located in a straight line of theoretical centerline for shafting system, however, the intermediate shaft bearing and gearbox shaft or diesel engine shaft are not located in a straight line of theoretical centerline for shafting system, the shafting centerline is in a curved state, as shown in Figure 9.3.4.



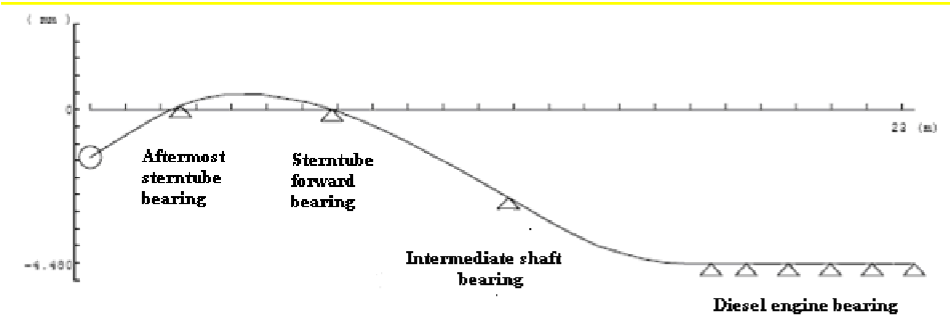


Figure 9.3.4 Shafting Centerline State for Reasonable Shafting Alignment

In general, the reasonable shafting alignment is carried out by adjusting and reasonably arranging the position of shaft bearing. The propulsion shafting system is to be arranged in a longitudinal straight line direction, and the position of bearing only changes within two degrees of freedom of the plane. After the arrangement of shafting system is determined, the longitudinal position of bearing is generally not to be changed, the position of bearing for shafting alignment may only be adjusted in the vertical bearing displacement. The reasonable alignment for propulsion shafting is a design of shafting alignment actually, including shaft bearing arrangement, shafting alignment calculation, shafting alignment instructions (alignment conditions, alignment procedures), shafting alignment inspection. Therefore, reasonable alignment applies to all propulsion shafting system for ships, particularly for the propulsion shafting system with the diameter of screwshaft of 250 mm and above.

The calculation of reasonable shafting alignment is to make the shafting alignment parameters meet the rules requirements and obtain the corresponding displacement value for each bearing so as to determine displacement value of each bearing in installation conditions, and to complete the calculation of the whole shafting alignment under hot condition or operating condition, cold condition and installation condition.

The design of reasonable shafting alignment is a general term for bearing arrangement, alignment calculation, including the calculation of appropriate installation process parameters and installation for shafting alignment. The design and installation of shafting alignment are closed tightly as an integral.

Therefore, the shafting parameters reflecting the calculated shafting centerline are: gap and sag for unconnected flanges, load and displacement of each bearing.

(2) Implementation of reasonable shafting alignment

The following four methods may be used for reasonable shafting alignment, as appropriate:

① Alignment by gap and sag method, which is to adjust the gap and sag values of unconnected flanges as to meet the requirements of alignment calculations; after connection of the bolts of flanges for shafting system, jack-up test is to generally be carried out, one or two typical bearings are to be chosen to perform the loading test based on the gap and sap values.

② Alignment by jack-up method, which is to adjust the gap and sag values of unconnected flanges only as an intermediate process; After connection of the bolts of flanges for shafting system, jack-up test is to generally be carried out, measurement and adjustment are to be made for the loads from the sterntube forward bearing to the last 3rd main bearing of main engine or gearbox afterward bearing as to meet the requirements of the alignment calculations based on the measured loads of bearings.

③ Alignment by strain gauge measurement method, which is to convert the directly measured strain capacity to bearing load as to meet the requirements of alignment calculations based on the bearing load.

④ Alignment by bearing displacement method, which is to adjust the bearing displacement by optical instrument, laser collimator, etc., as to meet the requirements of the calculations based on the bearing displacement.

9.3.5 Application of reasonable shafting alignment

Since 1980's, the reasonable shafting alignment has been applied in domestic China, the alignment by gap and sap method was used and introduced by the guidance notes of ship's standards Alignment of Ship's Propulsion Shafting System (CB/Z338－1984), ZC Prevention for Shipboard Adverse Vibration (1986), ISC Guidelines for Shipboard Vibration Control (2000). In recent years, due to the fact that the diesel engine manufacturers have the corresponding requirements for loads of bearings and adjustment is to be required to carry out after shafting alignment, jack-up method is directly applied more than before, but shafting alignment conditions have been changed. Some are carried out shafting positioning by straight shafting alignment method, the displacement of intermediate shaft bearing, displacement of gearbox bearing or/and displacement of main engine bearing are determined by gap and sag method, then the loads of bearings are to be measured and adjusted by jack-up method, but this is likely to cause inconsistent or unclear boundaries.

The reasonable shafting alignment includes:

(1) Shafting static alignment

Currently, static alignment is basically applied for the reasonable shafting alignment. Assuming that each bearing of shafting system is a rigid supporting point, and the propulsion shafting system is to be regarded as a continuous girder placed on rigid hinge. Three conditions of shafting system are to be considered for static alignment:

① Hot condition

It is the condition when the shafting system is running. The hot condition requires to consider the value of temperature raise of crankshaft bearing due to the temperature raise of diesel engine pedestal, or the value of temperature raise of gearbox base being more than that of gear shaft bearing and the value of oil temperature raise of double bottom below the intermediate shaft bearing (if any).

The loads under hot condition are forces of screwshaft, shaft, diesel engine flywheel, chain drive, weight of wheel of reduction gear for gearbox, etc., as well as transfer torque of reduction gear of gearbox when the shafting system is running, etc.

② Cold condition

It is the static condition after connection of the shafting system. The loads are forces of propeller shaft, shaft, diesel engine flywheel, weight of wheel of reduction gear of gearbox, force of chain drive, etc.

③ Installation condition

It is the conditions that all shafts of the shafting system have not yet been connected or some shafts have been connected for installation. The values of bearing displacement and loads are the same as those under cold condition; additional temporary support (T.S), auxilaiary external loads, etc., may be considered to facilitate installation.

(2) Shafting dynamic alignment

The shafting dynamic alignment is closer to the actual working condition of shafting system and capable of improving the shafting alignment quality and shafting operational reliability.

In addition to considering the effects of static force of shafting system and working temperature raise in the shafting dynamic alignment calculation, the effects of dynamic force and couple acting on the shafting system, oil film and structural elastic of bearing, hull loading deformation, etc., are to be taken into account.

Nowadays, shafting dynamic alignment complying with the definition in the engineering application has not yet been found.

(3) Shafting alignment in operational condition

① Contents to be considered for alignment in operational condition

The shafting alignment in operational condition is the one being more reasonable, closer to the actual situations.

The calculation for shafting alignment in operational condition means the alignment calculation including force and moment of force caused in shafting operational condition. Compared with the alignment calculation in hot condition, the calculation of alignment in operational condition is to take the following into account:

— moment of force caused by hydrodynamic of propeller;

— thrust acting on the hull;

— bending moment acting on the thrust bearing;

— gear force.

② Moment of force caused by hydrodynamic of propeller

When the ship is under full load condition, the eccentric of thrust caused by propeller is generally at the upper part of propeller, as shown in Figure 9.3.5-1; After the bending moment of propeller is included, the load of aftermost sterntube bearing is reduced while the load of sterntube forward bearing is increased, and it will bring benefit to improve the lubrication and force of sterntube bearing. However, when the ship is under ballast condition, the eccentric of thrust caused by propeller is generally at the lower part of propeller, after the bending moment of propeller is included, the load of aftermost sterntube bearing is increased while the load of sterntube forward bearing is reduced. In general, consideration of the effects of bending moment of propeller is not necessary.

Nowadays, some calculation procedures of shafting alignment have included the shafting alignment calculation for bending moment of propeller under full load and ballast conditions. The bending moment induced by eccentric of thrust of propeller M_{tp} may be calculated as follows:

$$M_{tp} = 0.7 T_p E_t \quad \text{kNm} \qquad (9.3.1)$$

where: T_p— axial thrust of propeller, in kN;

E_t — eccentric of thrust of propeller at the rated speed, in m; it is in the upper part of propeller under full load condition while in the lower part of propeller under ballast condition.

For reference, some service providers take 40% of mean transmission torque for upward eccentric bending moment and 5% of mean transmission torque for downward eccentric bending moment in the calculation.

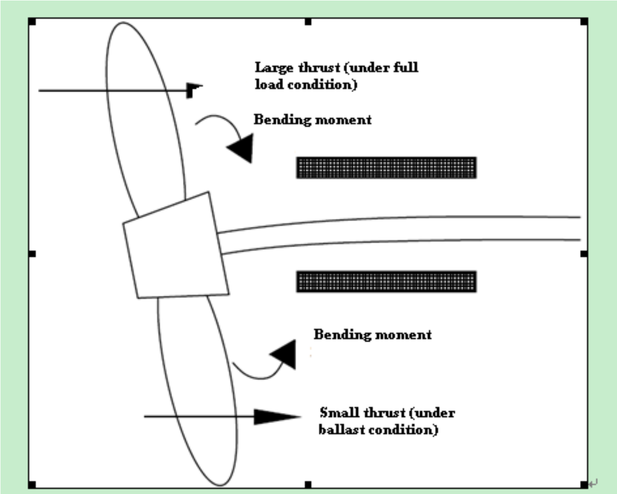


Figure 9.3.5-1 Hydrodynamic of Propeller

③ Bending moment acting on thrust bearing

The bending moment acting on thrust bearing is induced by thrust of propeller. The analysis carried out by WÄRTSILÄ is shown in Figure 9.3.5-2. From point of view of counterbalance for additional load of the last bearing of main engine induced by hull deformation, the bending moment acting on thrust bearing has the improvement effect. WÄRTSILÄ gives the bending moment M_{tb} as follows:

$$M_{tb} = 0.7T_pE_{tb} \quad \text{kNm} \qquad (9.3.2)$$

where: T_p — axial thrust of propeller, in kN;

E_{tb} — vertical eccentric displacement of thrust bearing, in m.

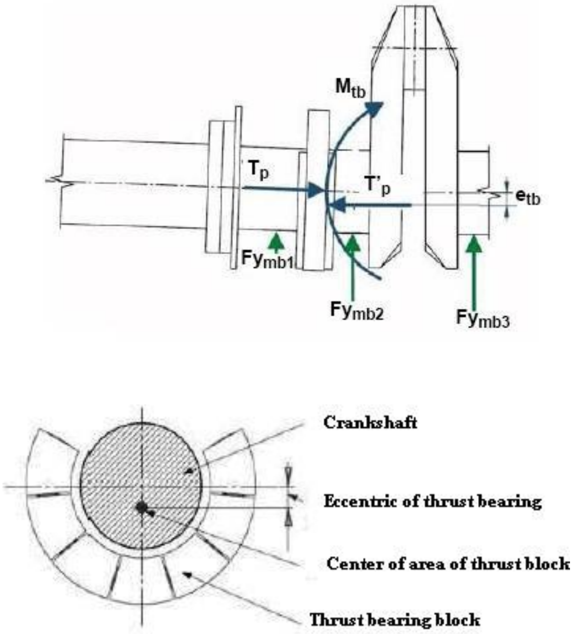


Figure 9.3.5-2 Effect of Bending Moment Acting on Thrust Shaft

④ Thrust acting on hull

The thrust acting on hull will induce the hogging strain of hull, just as the increase of draught, but the value is less, so the effect of thrust may be not considered.

⑤ Gear force

For the alignment calculation of gear force, see Appendix 1, Chapter 10 of PART THREE of ISC Rules for Classification of Sea-going Steel Ships.

9.4 Integrity of Shafting Alignment

9.4.1 Contents of integrity of shafting alignment

The requirements of shafting alignment in ISC Rules for Classification of Sea-going Steel Ships have been fully changed to completely reflect the integrity of shafting alignment, mainly including:

(1) application;

(2) shafting alignment calculation;

(3) shafting alignment requirements (criteria);

(4) shafting alignment conditions;

(5) shafting alignment procedures;

(6) shafting alignment inspection.

9.4.2 Scope of reasonable shafting alignment

The scope of reasonable shafting alignment specified in Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships is as follows:

(1) The alignment of main propulsion shafting systems and the arrangement of bearings are to be such as to give reasonable bending moments and bearing reactions and minimize the effects of hull deformation or bearing weardown on shafting alignment.

(2) For shafting alignment of large ships, the effects of hull deformation when the ship is ballasted or fully loaded are to be taken into consideration.

(3) Shafting alignment calculations together with shafting instructions for the following main propulsion shafting systems are to be submitted for approval:

① shafting systems where the screwshaft has a diameter (hereinafter referred to as screwshaft diameter) of 250 mm or greater in way of the aftermost sterntube bearing:

② shafting systems fitted with no sterntube forward bearing where the screwshaft diameter is 200 mm or greater;

③ shafting systems with speed reduction gear, of which the wheel is driven by two or more than two pinions;

④ shafting systems for which the sterntube bearings are to be slop-bored or inclined;

⑤ shaft generator or electrical motor as an integral part of the low-speed shaft in diesel engine propulsion.

(4) Shafting alignment instructions are to correctly reflect the calculation results for the alignment in cold condition, containing at least the alignment conditions and alignment procedures.

9.4.3 Alignment instructions

Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships specifies that “shafting alignment calculations together with shafting instructions for the main propulsion shafting systems are to be submitted for approval".

The correct technology of shafting alignment can ensure the shafting alignment quality. Otherwise, the sterntube bearing or/and intermediate shaft bearing accidents will be caused in sea trial or after a period of usage.

Therefore, in order not to confuse with the procedure specifications of shafting alignment in shipyards, alignment instructions is provided and is defined at least to include the contents of shafting alignment conditions and shafting alignment procedures.

9.4.4 Shafting alignment conditions

Whether the shafting alignment is carried out in dock or berth or during ship's afloat, the basic conditions of shafting alignment are specified in Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships, mainly including:

(1) Requirements before boring

Before boring the sternframe, the ship's structures are to be generally completed to the upper deck and to the engine-room forward bulkhead.

(2) The essential equipment are to have been hoisted to their respective places.

The ship's superstructures, main engines, boilers, generators and other pieces of essential equipment are to have been hoisted to their respective places. There is to be no movement of essential equipment or change of ballasting in the alignment, installation and inspection onboard. It is mainly to prevent the excessive welding to cause final structural deformation and effect the shafting alignment.

(3) The welding of hull portions within shafting areas are to have been completed.

The processing and welding of hull portions within shafting areas are to have been completed. It is mainly to prevent the welding temperature change to effect the shafting alignment.

(4) The temperatures of hull structures are to be stable.

The temperatures of hull structures are to be stable and even in so far as practicable. For this purpose, the reference line is to be determined in the morning before sunrise, at least in the evening, to ensure the temperatures of hull structures distribute evenly. Measurement in daytime will induce the structural deformation to influence the hull structures inhomogeneously, and the measurement readings may be greatly different. Therefore, to understand such differences or measurement to be carried out in the morning is important.

(5) Ship's working conditions for shafting alignment

Where the ship is in a normal floating condition, the immersion of the propeller is to be so close as practicable to that indicated in the calculations. At such stage, the ship's systems have not been completed, i.e. no ballast water, the immerse condition of propeller is to be determined as close as practicable to that indicated in the calculations. Where the immersion condition of the propeller has great difference from that indicated in the calculations, it has less effect on the gap and sag of flange, even only 0.01 mm, which is still within the tolerance range, however, it has greater influence on the load of sterntube forward bearing. The calculation results under different immersion conditions of propeller for a certain large-type oil tanker are shown in Table 9.4.4-1, the load of sterntube forward bearing may achieve twice of the value.

Calculation Results Under Different Immersion Conditions of Propeller for a Certain Large-Type Oil Tanker Table 9.4.4-1

	50％ propeller	100 ％	Difference (%)

	immersed	propeller immersed	
Loads of sterntube forward bearing (N)	11885	24498	Increased 206
Loads of intermediate shaft bearing (N)	180127	178167	Reduced 1.1

For a certain large-type oil tanker, where the propeller is fully immersed during the installation, the ambient temperature is 10℃, the data as shown in Table 9.4.4-2.

Calculation Results Under Full-immersed Condition of Propeller Table 9.4.4-2

Conditions	Displacement & bearing load	Aftermost sterntube bearing	Sterntube forward bearing	Gearbox afterward bearing	Gearbox forward bearing	Slope of aftermost sterntube bearing (10-4 rad)	Gap (mm)	Sag (mm)
Cold condition	Bearing displacement (mm)	0	0	-0.82	-0.9	2.04	0.183	-0.238
	Bearing load (kN)	88	28	9.8	34			
Hot condition	Bearing displacement （mm)	0	0	-0.673	-0.753	1.92		
	Bearing load (kN)	96.2	19	24.4	24			

During the installation, where the propeller is partially immersed, the ambient temperature is 10℃, the data as shown in Table 9.4.4-3.

Calculation Results Under Partial-immersed Condition of Propeller Table 9.4.4-2

Conditions	Displacement and bearing load	Aftermost sterntube bearing	Sterntube forward bearing	Gearbox afterward bearing	Gearbox forward bearing	Slope of aftermost sterntube bearing （10-4 rad）	Gap (mm)	Sag (mm)
Cold condition	Bearing displacement （mm)	0	0	-0.82	-0.9	2.14	0.187	-0.256
	Bearing load (kN)	95.5	24.7	10.6	33.7			
Hot condition	Bearing displacement (mm)	0	0	-0.673	1.92	2.02		
	Bearing load (kN)	96.2	19	24.4	24			

As seen from Table 9.4.4-2 and Table 9.4.4-3, the shafting alignment for propeller fully immersed under hot condition meet the rules requirements, however, the data of shafting alignment for propeller fully or partially immersed under cold conditions (according to the temperature of engine room when installation is carried out) are different.
Therefore, in shafting alignment calculation, the immersion of propeller is to be determined by the ship's floating conditions and shafting alignment is to be carried out according to the conditions

9.4.5 Shafting alignment procedures

Shafting alignment may be carried out only after confirming the shafting alignment conditions are complied with. Otherwise, alignment tolerance may occur, and even become one of reasons to cause shafting faults. Hence, the shafting alignment procedures specified in Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships include:

(1) Where the shafting alignment conditions are met, shafting alignment may be carried out according to the approved alignment calculations or alignment technology.

(2) Where any noncompliance with calculation conditions, e.g. the ambient temperature, propeller immersion, ship's condition, load adjustment of main bearings of engines, may affect the alignment results during the installation of shafting systems, the alignment calculations together with alignment instructions are to be resubmitted for approval, unless such effects have been included in the calculations.

(3) Where the load of the sterntube forward bearing is not positive for unconnected shafting flanges, the force applied downward from the screwshaft forward flange is to comply with the calculations. The axial distance or temporary supports, if fitted, is to comply with the calculations.

(4) For shafting alignment, the screwshaft forward flange is to be used as the reference for positioning bearings or temporary supports and the engine (or gearbox) from aft to fore by adjusting the sag and gap of unconnected flange pairs, in compliance with the relevant requirements.

(5) Upon completion of the shafting alignment, the gear engagement is to be examined or the difference between crank webs of engine measured, and the results are to comply with the requirements of manufacturers and appropriate records are to be made.

(6) After rechecking of the bearing parameters, coupling bolts are to be provided to flanges, chocks provided or epoxy resin cast to diesel engines (or gearboxes) and engine pedestal bolts provided, followed by connection and fixing.

(7) The gap and sag tolerances of flanges pairs for straight alignment shafting are to comply with the relevant rules requirements or standards.

(8) For reasonable alignment shafting, if alignment by gap and sag method is used, at least 1 or 2 bearings are to be selected for load verification with the ship afloat based on the values of gap and sag of flanges,.

(9) For reasonable alignment shafting, if alignment by jack-up method is used, the shipyards may adjust the gap and sag values of each unconnected flange pairs in dock or berth. After the flanges are connected, load is to be measured and adjusted from the sterntube forward bearing to the last 1 to 3 bearings of main engine with the ship afloat. The difference between the measured value under cold condition and calculated value within ±20% of the range will be acceptable. However, the values of bearings of main engine are at least in positive.

9.5 Calculation Models for Reasonable Shafting Alignment

9.5.1 Simplification of reasonable shafting alignment calculation

(1) Calculating coordinates

① The coordinates of alignment calculation is a plane coordinate system, the end of screwshaft is to be taken as origin of coordinates, theoretical centerline of shafting system is X axis, stem is the positive direction.

② The line which is via the origin of coordinates and vertical to X axis is the Z axis; the bearing displacement is represented by coordinating value of Z axis, which indicates the vertical distance between the bearing and X axis — displacement, if the bearing displacement is lower than X axis, it is negative, and if the bearing displacement is higher than X axis, it is positive.

③ The concentrated load and homogeneously distributed load are shown by coordinating values of Z axis, which indicates the scale of load, the downward load is positive while the upward load is negative.

(2) Simplification of parts for shafting system

① Propeller

The propeller is to be considered as a concentrated external load. The intersection point of center of gravity of propeller and axis of propeller is to be taken as a point of action for propeller weight, in the case that the center of gravity of propeller can not be determined, the intersection point of middle line of blade in way of 0.7*r* (*r* is the radius of blade) and the axis, or estimate the middle point of blade shell may be taken. The scale of propeller weight used for alignment calculation is related to the propeller immersion:

(a) The weight of propeller under full-immersed condition W_p is to be calculated by:

$$W_p = W_a(\rho_\rho - \rho_s)/\rho_\rho \quad \text{kg} \qquad (9.5.1)$$

where: W_a — weight of propeller in the air, in kg;

ρ_ρ — density of propeller material, in kg/m^3;

ρ_s — density of sea water, in kg/m^3;

In general, it may be approximately obtained by:

$$W_p = 0.87W_a \quad \text{kg} \qquad (9.5.2)$$

(b) The weight of propeller under partial-immersed condition W_p is to be calculated by:

$$W_p = W_s(\rho_\rho - \rho_s)/\rho_p + W_n \quad \text{kg} \qquad (9.5.3)$$

where: W_s — weight of propeller part immersed, in kg;

W_n — weight of propeller part not immersed, in kg.

In general, it may be approximately obtained by:

$$W_p = (0.935 \sim 0.947)W_a \quad \text{kg} \qquad (9.5.4)$$

② Screwshaft and tube shaft

The weight of screwshaft and tube shaft is to be processed as a homogeneous load. Mean diameter may be obtained for conical shafts and the weight of shaft may be calculated according to homogeneous load. The nuts and shaft liners are to be calculated as homogeneous loads for corresponding shafts.

The effect of buoyancy for shaft immersed in sea water or lubricating oil is to be taken into consideration. Approximate 90% of the weight in the air may be taken for shafts immersed in lubricating oil, and approximate 87% of the weight in the air may be taken for shafts immersed in sea water.

③ The intermediate shaft, thrust shaft, wheel shaft of gearbox and its flange are to be processed as homogeneously distributed loads.

④ Thrust ring, flywheel of diesel engine, wheel of gearbox

They are generally to be processed as concentrated loads. The parts with equivalent shaft diameter of the corresponding shafts are to be calculated as the homogeneously distributed loads, the other parts are to be calculated as concentrated loads, the point of action is the intersection point of their middle transverse section and axis.

⑤ Supporting position of aftermost sterntube bearing

The center of actual pressure for aftermost sterntube bearing moves backward by the action of straddle for propeller, for the alignment calculation, the distance between supporting point of aftermost sterntube bearing and after edge of bearing liner L is to be determined according to the following range:

For bearing lined with white-metal: $L = (\frac{1}{7} \sim \frac{1}{3})L_b$ mm

For bearing lined with lignum vitae: $L = (\frac{1}{4} \sim \frac{1}{3})L_b$ mm

For bearing lined with rubber: $L = (\frac{1}{3} \sim \frac{1}{2})L_b$ mm

where: L_b is the length of aftermost sterntube bearing liner, in mm.

Some take the aftermost sterntube bearing as two supporting points.

The positions of supporting points for bearing made of the approved compound material, such as Thordon, Feroform and Vesconite may be referred to bearing lined with white-metal for oil lubrication and referred to bearing lined with lignum vitae for water lubrication.

The positions of supporting points for bearing made of other material, the middle point of length of bearing liner is to be taken.

(3) Division of calculating sections

When the simplified shafting is discretely processed, the positions in way of section of supporting point for bearing, point of action of concentrated loads for shafting system, change of shaft section, etc., are to be divided as calculating sections.

Alignment calculation model of propulsion shafting system for a certain two-stroke cycle diesel engine is shown in Figure 9.5.1.

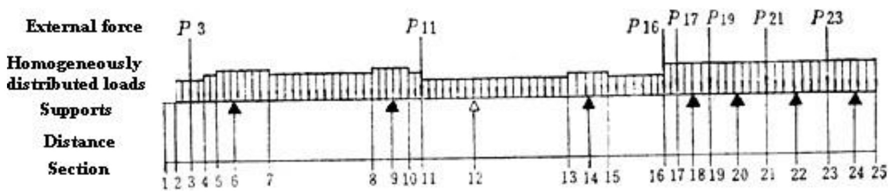


Figure 9.5.1 Alignment Calculation Model of Propulsion Shafting System for a Certain Two-stroke Cycle Diesel Engine

(4) Value of bearing displacement under hot condition

The raising value of expansion of heated bearing after the operation of shafting system is to be taken into consideration for the alignment calculation. The raising values of diesel engine bearings, gearbox bearings and intermediate shaft bearings (if oil tanks are located below) are to be included.

The raising values of diesel engine bearings and gearbox bearings at stable speed are to be provided by the manufacturers.

The raising value of bearing may be calculated as follows:

$$\Delta h = \rho\ CH\ (t_1 - t_2) \qquad (9.5.5)$$

where: Δh — the raising value of bearing from installation condition to hot condition, in mm;

ρ — coefficient of expansion of material, in 1/C°;

C— correction coefficient, it is generally to be taken as 1, or may be taken in accordance with the experience of manufacturers;

H— distance between engine pedestal or surface plate of bottom of oil tank to the centerline of bearing, in mm;

t_1— the highest temperature as the shafting system is running, may be taken as 55℃ for diesel engine bearing and gearbox bearing and as 45℃ for intermediate shaft bearing (if double-bottom oil tanks fitted below);

t_2— different ambient temperatures of engine room installed, may be taken as 0℃, 10℃, 20℃ depending on the possible temperatures of engine room installed, .

(5) Supporting stiffness

The effects of oil film of bearing are not to be taken into consideration for alignment calculation, in general, the bearing is a rigid body without elasticity, the effects of hull deformation are also not to be taken into account.

9.5.2 General calculation model for diesel engine crankshaft alignment

In the diesel engine shafting alignment calculation, the crankshaft may be simplified to a shaft with the equivalent diameter of main shaft journal and processed as a homogeneously distributed load; the weight of reciprocating and revolving parts for each cylinder of diesel engine, including piston, crosshead, connecting rod, etc., is to be processed as concentrated loads which are superimposed on the beam span corresponding to the middle point of crank pin, deducing the weight of crank web with the equivalent part of main journal.

9.5.3 MAN B&W calculation model for diesel engine crankshaft alignment

MAN B&W provides calculation model diagram of corresponding crankshaft alignment for each type of marine low-speed two-stroke cycle diesel engines, as shown in Figure 9.5.3. The equivalent crankshaft model is to be composed by the following elements:

(1) structural dimension of diesel engine crankshaft equivalent system;

(2) loads acting on the crankshaft, including the equivalent concentrated loads of each cylinder, chain driving loads;

(3) the bending moment and shearing stress in way of flywheel are to be within the specified block diagram;

(4) the crankshaft bearing load is to be in positive, but the load of last bearing may be equal to 0.

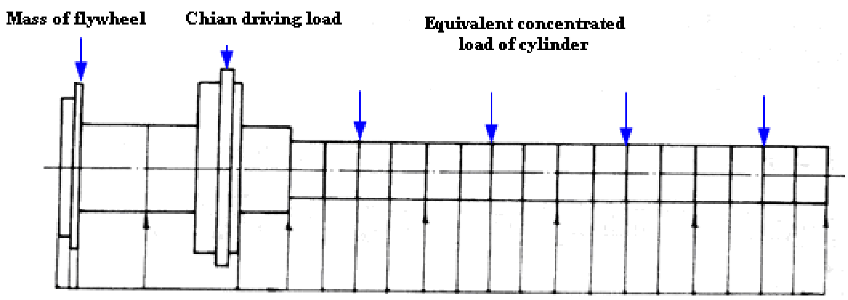


Figure 9.5.3 Calculation Model for Crankshaft Alignment

9.5.4 WÄRTSILÄ calculation model for diesel engine crankshaft alignment

WÄRTSILÄ provides calculation model diagram and table of corresponding crankshaft alignment for marine low-speed two-stroke cycle diesel engines.

The calculation model diagram for crankshaft alignment may be applied commonly for each type, as shown in Figure 9.5.4. The equivalent crankshaft model is to be composed by the following elements:

(1) connecting flange and thrust shaft;

(2) three-dimensional finite element model is used for crankshaft, an equivalent rigid cylinder within the range of crank;

(3) mass of crank, equivalent forces of driving mechanism, thrust ring and gear;

(4) elasticity of main bearing.

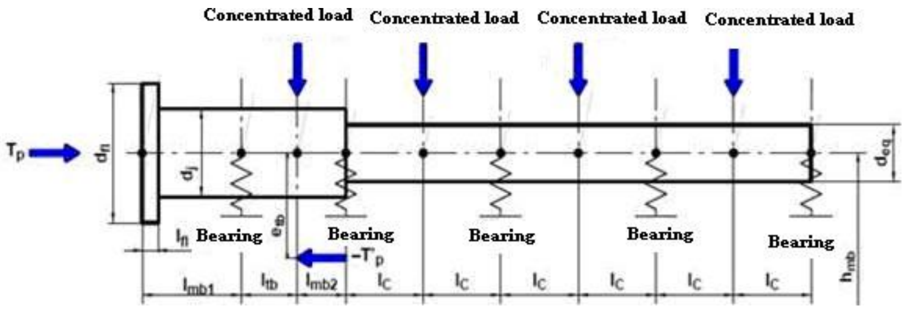


Figure 9.5.4 Model for Diesel Engine Crankshaft Alignment

9.6 Calculation for Reasonable Shafting Alignment

9.6.1 Calculation range for reasonable shafting alignment

With the difference of arrangement for ship's propulsion shafting system, the calculation range of reasonable shafting alignment is also different:

(1) Propulsion shafting system directly driven by diesel engine

For the propulsion shafting system directly driven by low-speed two-stroke cycle diesel engine, the screwshaft and intermediate shaft are to be connected with crankshaft of diesel engine. In alignment calculation, the 1st main bearing at the free end of diesel engine is generally to be calculated, but at least to the 5th main bearing counted forward from the output end of flywheel of diesel engine. The calculation model for the propulsion shafting system directly driven by diesel engine is shown in Figure 9.6.1-1.

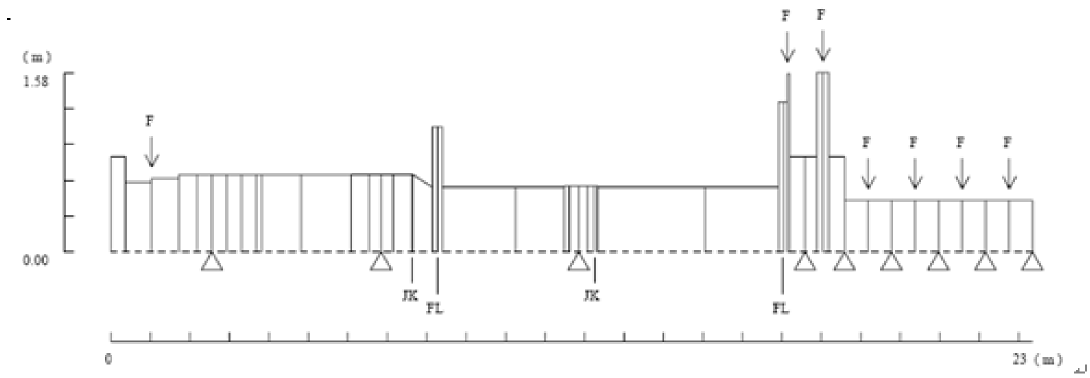


Figure 9.6.1-1 Calculation Model for the Propulsion Shafting System Directly Driven by Diesel Engine

(2) Propulsion shafting system with driving gearbox

For the propulsion shafting system with driving gearbox, the screwshaft or/and intermediate shaft/short shaft is to be connected with the wheel shaft flange of gearbox. In alignment calculation, the fore end of wheel shaft of reduction gear is to be calculated. The diesel engine output flange is to be directly connected with input gear shaft flange in a position of zero to zero, excluding in the alignment calculation. Calculation model for the gear-driven propulsion shafting system is shown in Figure 9.6.1-2.

The gearbox with PTO device may be processed in the same way. Most of such shafting systems are CPP shafting systems.

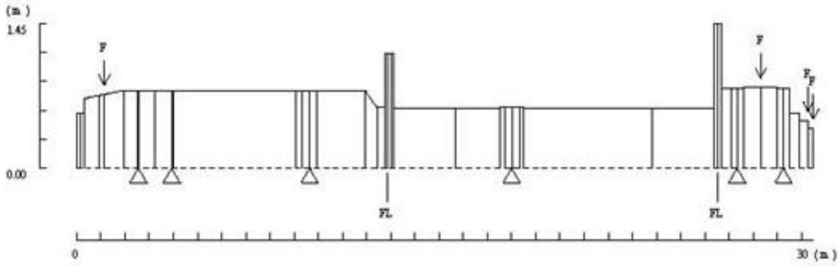


Figure 9.6.1-2 Calculation Model for the Gear-driven Propulsion Shafting System

9.6.2 Calculation method of reasonable shafting alignment

The calculation of reasonable shafting alignment may be by transfer matrix method, three-moment method and finite element method, the theoretical basis is the method of classical mechanics, and no essential difference is among shafting alignment results calculated by various methods.

(1) Transfer matrix method

Assuming that the state vector of the i th calculated section Z_i is:

$$Z_i = \{-H \quad \theta \quad M \quad Q\}^t \qquad (9.6.1)$$

where: H— deflection of section;

θ— corner of section;

M— bending moment;

Q— bearing reaction.

Then, the state vector of the $i+1$ th section Z_{i+1} is to be:

$$Z_{i+1} = T_{i,i+1} Z_i \qquad (9.6.2)$$

The transfer matrix of shaft $T_{i,i+1}$ is to be:

$$T=\begin{bmatrix} 1 & L & \frac{L^3}{2EI} & \frac{L^3}{6EI} & -\frac{qL^4}{24EI} \\ 0 & 1 & \frac{L}{EI} & \frac{L^2}{2EI} & -\frac{qL^3}{6EI} \\ 0 & 0 & 1 & L & -\frac{qL^2}{2}-M \\ 0 & 0 & 0 & 1 & -qL-P \\ 0 & 0 & 0 & 0 & 1 \end{bmatrix} \qquad (9.6.3)$$

where: L — length of shaft;

E— modulus of elasticity of shaft material;

I— moment of inertia of shaft section;

q— homogeneously distributed load;

P— concentrated load.

(2) Three-moment method

The actual bearing in the shafting system is to be acted as solid support and assuming that support is also provided in the other sections other than the solid one and treated as virtual support. Three-moment equation is to be listed for each middle supporting position, equation in way of the i^{th} supporting position (see Figure 9.6.2) is as follows:

$$\begin{gathered} \frac{L_{i-1}}{E_{i-1}I_{i-1}}M_{i-1}+2\left(\frac{L_{i-1}}{E_{i-1}I_{i-1}}+\frac{L_i}{E_{i-1}I_i}\right)M_i+\frac{L_i}{E_iI_i}M_{i+1}-\frac{6}{L_{i-1}}H_{i-1}+ \\ 6\left(\frac{1}{L_{i-1}}+\frac{1}{L_i}\right)H_i-\frac{6}{L_i}H_{i+1}=-\frac{1}{4}\frac{q_{i-1}L_{i-1}^3}{E_{i-1}I_{i-1}}+\frac{q_iL_i^3}{E_iI_i} \qquad (i=1,2,3,\ldots\ldots,n) \end{gathered} \qquad (9.6.4)$$

where: L_i—— span between the i^{th} and $i+1^{th}$ supports, in mm;

M_i——bending moment of the i^{th} supporting section, kN.mm;

E_i—— elastic modulus of shaft material between the i^{th} and $i+1^{th}$ supports, in kN/mm^2;

I_i— inertia moment of shaft section between the i^{th} and $i+1^{th}$ supports, in mm^4;

q_i— homogenously distributed loads on shaft between the i^{th} and $i+1^{th}$ supports, in kN/mm;

H_i— deflection of the i^{th} section, in mm;

P_i— concentrated load, in kN.

If the boundary condition is given, the result may be obtained by the equation.

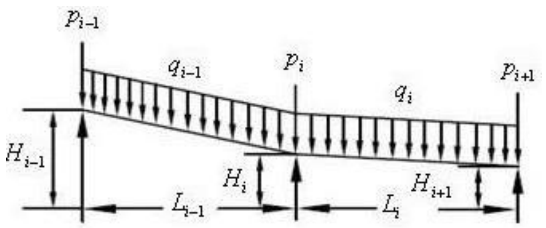


Figure 9.6.2 Unit of Three Bending moments

9.6.3 Reasonable shafting alignment calculation

(1) Calculation for shafting alignment under hot condition

It means the calculation which is to ensure that each parameter of shafting alignment under hot condition is to meet the relevant requirements of ISC Rules for Classification of Sea-going Steel Ships through the adjustment of bearing displacement on the basis of preliminary regulating for quantity and longitudinal positions of bearings as appropriated (see 9.2 of this Section).

(2) Calculation for shafting alignment under operating condition

It means the calculation which is to ensure that each parameter of shafting alignment under operating condition is to meet the relevant requirements of ISC Rules for Classification of Sea-going Steel Ships when the related forces and moments induced under shafting operating condition are included.

(3) Calculation for shafting alignment under cold condition

It means the calculation which includes the bearing thermal expansion and meets the relevant requirements of ISC Rules for Classification of Sea-going Steel Ships. If possible, the effects of hull deformation under different working conditions are to be taken into account.

(4) Calculation for shafting alignment under installation condition

It means the calculation which converts theoretical axial centerline obtained from the calculation for shafting alignment under cold condition to shafting parameters under installation condition and is determined by the alignment methods, such as:

① gap and sag of each unconnected flange pair;
② loads of each bearing;
③ displacement of each bearing;
④ jack-up coefficient of bearing load;
⑤ temporary support.

Calculating the gap and sag values for each unconnected flange pair of shafting system under installation condition, to specify that downward is positive for gap values and is positive for sag values when the aft flange is higher than the fore one.

The calculation for shafting alignment under installation condition will be varied with the different arrangement of shafting system. Where different alignment methods are used for the same shafting system, the shafting alignment calculation is also different. Theoretically, calculation may be carried out for shafting system under any installation conditions, however as a matter of fact, not all alignments can be carried out correctly under any installation conditions. Therefore, sound and operable shafting installation conditions are to be selected for calculation.

9.6.4 Gap and sag installation conditions of typical shafting system

Whether gap and sag method or jack-up method is used, the control of gap and sag of unconnected flange is a simpler way for shafting installation, some gap and sag installation conditions of typical shafting system are listed as follows for reference.

(1) Shafting system with two bearings arranged both for screwshaft and each intermediate shaft

For shafting with two bearings arranged both for screwshaft and each intermediate shaft, installation condition is that each shaft of the system is in an unconnected state. In this case, gap and sag values are to be calculated for each flange pair, as shown in Figure 9.6.4-1.

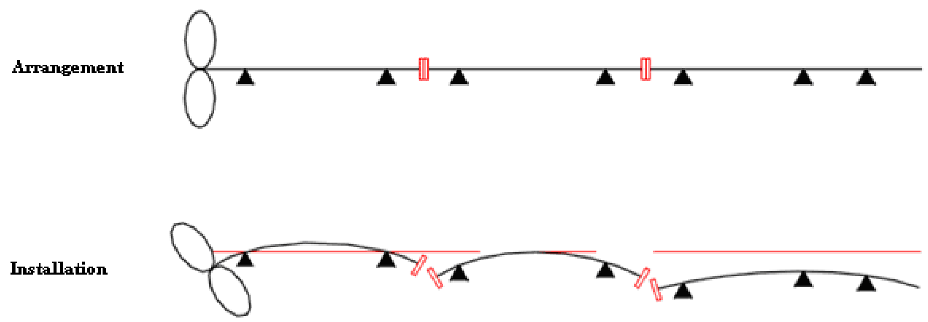


Figure 9.6.4-1 Shafting System with Two Bearings Arranged Both for Screwshaft and Each Intermediate Shaft

(2) Shafting system containing two or more intermediate shafts with only one bearing arranged for each shaft

For shafting system containing two or more intermediate shafts with only one bearing arranged for each shaft, the common and simpler way is to connect the two intermediate shafts as one directly before the shafting alignment installation so as to make the installation condition as same as that indicated in Figure 9.6.4-1. In this case, the connected flanges are not to be taken into account for shafting alignment calculation under installation condition, as shown in Figure 9.6.4-2.

However, where two temporary supports are provided for each intermediate shafts, and the bearing of intermediate shaft is not to be installed, or as the case, the bearing of intermediate shaft is to be installed and only one temporary support is to be provided for each intermediate shaft. In this case, adjustment and installation may be carried out in sequence for each flange.

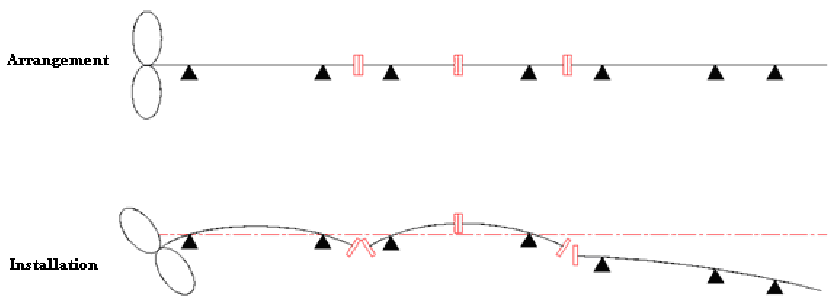


Figure 9.6.4-2 Shafting System Containing Two or More Intermediate Shafts with Only One Bearing Arranged for Each Shaft

(3) Shafting system with two bearings arranged for screwshaft and only one bearing arranged for its connected intermediated shaft

For shafting system with two bearings arranged for screwshaft and only one bearing arranged for its connected intermediated shaft, normally three methods for alignment installation may be used, thus three installation conditions exist. The common way is to add one temporary support (T.S) for the intermediate shaft, shown as installation condition 1 in Figure 9.5.4-3. In the case of the displacement values of sterntube bearing and intermediate shaft bearing being equal to 0, the screwshaft and intermediate shaft may be connected as one, shown as installation condition 2 in Figure 9.6.4-3. Alternatively, two temporary supports may be fitted on intermediate shaft, shown as installation condition 3 in Figure 9.6.4-3.

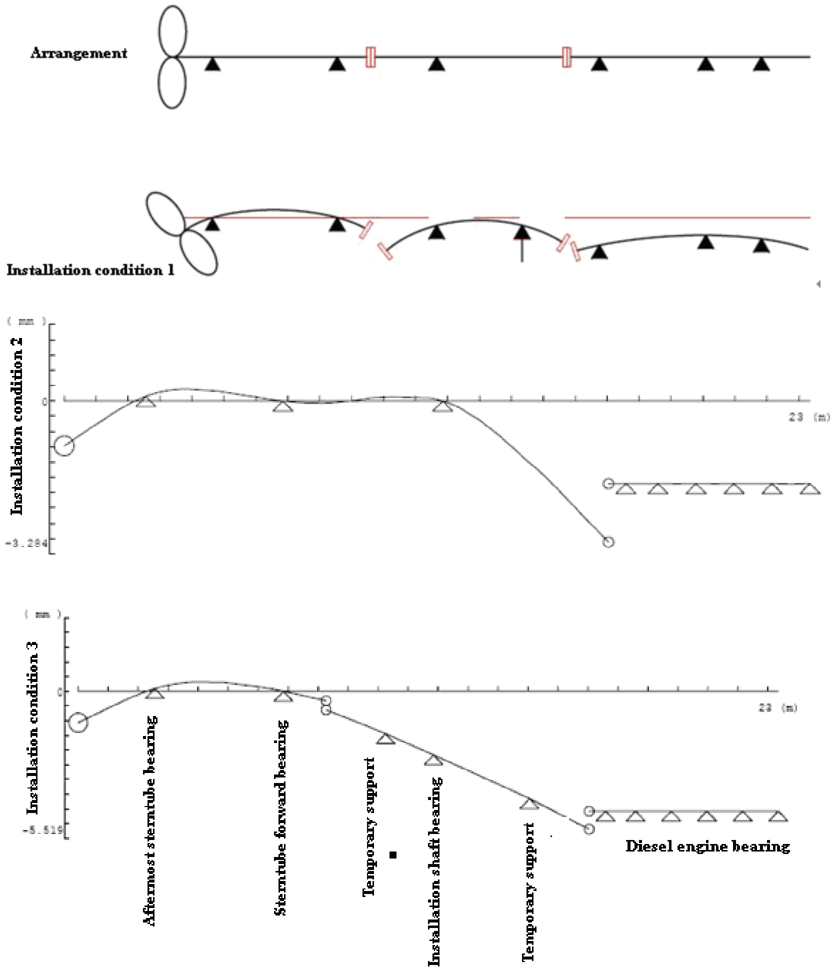


Figure 9.6.4-3 Shafting System with Two Bearings Arranged for Screwshaft and Only One Bearing Arranged for Its Connected Intermediated Shaft

(4) Shafting system with only one bearing arranged for screwshaft and only one bearing arranged for its connected intermediated shaft

For shafting system with only one bearing arranged for screwshaft and only one bearing arranged for its connected intermediated shaft, two installation methods may be used, one is to provide temporary supports in an appropriate place respectively before the screwshaft and after the intermediate shaft, similar to installation condition 1 as shown in Figure 9.6.4-3. The other way is to connect the screwshaft and the intermediate shaft as one, regardless of bearing displacement.

(5) Shafting system with two bearings arranged for screwshaft and directly connected to gearbox shaft

For shafting system with two bearings arranged for screwshaft without intermediate shaft bearing, one short shaft is to be used to connect the screwshaft with gearbox shaft directly, the installation condition is similar to that shown in Figure 9.6.4-4.

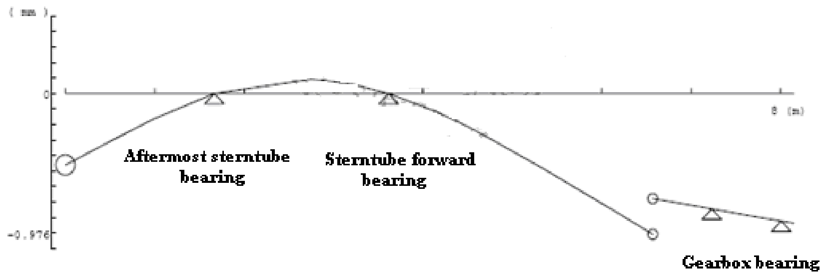


Figure 9.6.5-5 Shafting System with Two Bearings Arranged for Screwshaft and Directly Connected to Gearbox Shaft

9.7 Concerning Shafting Alignment in Dock

As mentioned above, the reasonable alignment shafting by gap and sag method has been used in China since 1980's based on the values of gap and sag of flanges, and 1 or 2 bearings are to be selected for load verification with the ship afloat.

In order to shorten the ship's construction period, shafting alignment in dock/berth has been tried to use by some domestic shipyards in recent years. No hull deformation may be regarded as a ship is in docking/berthing condition; however when a ship is in afloat condition, effects of hull deformation can not be ignored.

Hence, if the inspection point for shafting alignment is moved forward as the ship is in dock, the effects of hull deformation is to be taken into account, however, only test verification may be carried out as the ship is afloat. Therefore, ISC Rules for Classification of Sea-going Steel Ships requires that if shafting alignment is carried out in dock, effect of vessel deformation is to be included and shafting alignment calculation in dock is to be submitted.

For the purpose of achieving a short period of construction and readjustment of main engine bearing and intermediate shaft bearing as necessary, jack-up alignment method is produced. Under the premise of the reasonable alignment, the related bearing loads are to be adjusted and measured to achieve the results of alignment calculation. In this case, preliminary shafting alignment is to be carried out in accordance with the gap and sag values of shafting alignment in floating or dock conditions to adjust the axis, and then connect the flanges. The shafting alignment under such condition is not a real meaning of alignment in dock, therefore the gap and sap of flanges are not to be considered as inspection points – i.e. not as the inspection requirements, but to be controlled by shipyards, and to meet the basic conditions of shafting alignment.

Where the jack-up method is used as the ship is afloat, the loads are to be measured and adjusted from the sterntube forward bearing to the last 3rd main engine bearing to meet the requirements of the calculations.

Where the gap and sag method is used as the ship is afloat, the gap and sag values are to be based and a test verification is to be carried out for 1 or 2 bearings (sterntube forward bearing, intermediate shaft or last bearing of main engine/gearbox).

Three differences are in the shafting alignment as the ship is afloat or in dock, such as propeller weight, screwshaft/tube shaft weight and hull deformation. Comparing the shafting alignment calculation taking consideration of effects of propeller weight and screwshaft/tube shaft weight with the calculating results of shafting alignment as the ship is afloat, the gap and sag of flanges is still within the installation error range, but the loads of bearings are varied largely. Due to the preliminary shafting alignment at the docking stage, and the gap and sag of flanges being taken for reference, gap and sag values as the ship is afloat or in dock may be adopted.

As aforementioned, an uncertainty and risk may occur if loads of bearing are not adjusted according to the alignment calculation results, and the failure of sterntube bearings for some ships may be related. This is the reason why "the minimum load of main bearings of main engines is in general not to be less than 10% of the allowable load on main bearings. Alternatively, the minimum value specified by the engine manufacturer may be accepted, but this is to be reflected in shafting alignment calculations", and "where any noncompliance with calculation conditions (e.g. the ambient temperature, propeller immersion, ship's condition, load adjustment of main bearings of engines) may affect the alignment results during the installation of shafting systems, the alignment calculations together with alignment instructions are to be resubmitted for approval, unless such effects have been included in the calculations".

9.8 Concerning Shafting Alignment Requirements by Diesel Engine Manufacturers

9.8.1 Shafting alignment requirements provided by MAN B&W

(1) Bearing loads of crankshaft

MAN B&W requires that the bearing load values of crankshafts for each engine type are to be in positive, and puts forward the specific provision. For a certain type of diesel engine, the provisions of bearing load is shown in Figure 9.8.1.

Provisions for Bearing Load for a Certain Type of Diesel Engine Manufactured by MAN B&W **Table 9.8.1**

Permissible bearing reaction force (kN)	Maximum	Minimum
Last main engine bearing (aftermost main bearing)	573	0%
Last main bearing (last 2nd main bearing)	573	5%

(2) Recommended aftermost and last 2nd bearing loads

It was recommended by MAN B&W that the minimum static load of aftermost main engine bearing may be equal to 0 or approximate to 0 and that of last 2nd main engine bearing may be 5% of the specified value.

(3) Shearing force and bending moment applied on flywheel

It provides the requirements for shearing force and bending moment on output flange of diesel engine.

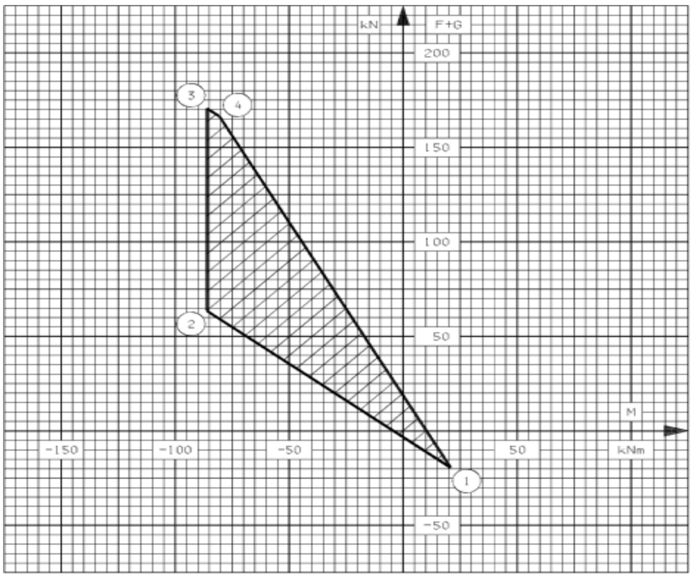
For a certain type of diesel engine, the following bending stress and moment on the diesel engine shaft are:

— maximum permissible bending stress: 15 N/mm^2;

— maximum permissible bending moment: 86 kNm.

(4) Diagram of bending moment and shearing force

The shearing force applied on the flywheel and obtained by alignment calculation under hot and cold conditions, plus the intersect of coordinating values between flywheel weight and bending moment are to be located in the specified diagram of corresponding diesel engine shown in Figure 9.8.1.



	M	F
1	20.9	-19.5
2	-86.1	63.6
3	-86.1	170.7
4	-80.6	166.5

Figure 9.8.1 Permissible Diagram of Bending Moment and Shearing Force for a Certain Type of Diesel Engine Manufactured by MAN B＆W

9.8.2 Shafting alignment requirements by WÄRTSILÄ

(1) Recommended relationship among the last 1 to 3 main engine bearing loads

In shafting alignment calculation, the relationship among the last 1 to 3 main engine bearing loads recommended by WÄRTSILÄ is shown in Figure 9.8.2. Taking the calculated static load of Type RT—flex82 main engine bearing for example, the minimum load of the aftermost main bearing mb1may be 30 kN while that of the last 2nd main bearing mb2 may be (220 ~ 470) kN and the load of aftermost main bearing is less than that of the 2nd one, the load of 3rd main bearing mb3 is to be also more than or equal to that of 2nd one (220 ~ 470) ×0.9 kN.

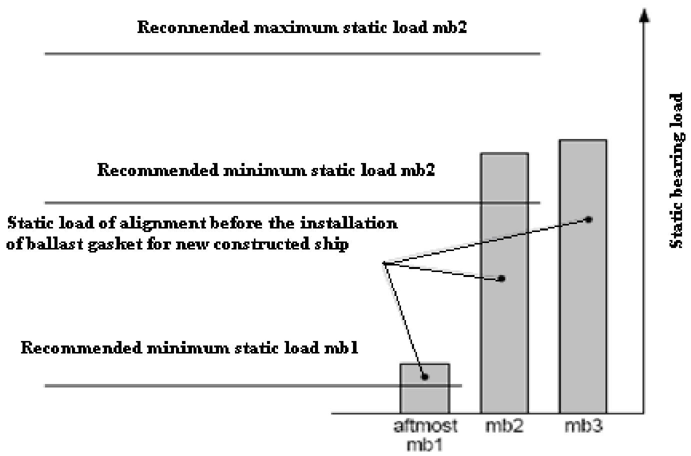


Figure 9.8.2 Recommended Relationship among the Last Three Main Engine Bearings Loads

(2) Difference range of static loads obtained by between calculation and jack-up testing
The static loads of 2nd and 3rd bearings obtained by alignment calculation does not match with that obtained by actual jack-up testing, when the crank angle is 0°, the 2nd bearing is reduced 20% to 30% while the 3rd bearing (measured at the after end of bearing) is increased 20% to 30%.
(3) The effects of bending moments for thrust bearing and propeller thrust are to be taken into consideration.
(4) No requirements are provided for the shearing force and bending moment of output flange. Previously, in shafting alignment calculation, the shearing force and bending moment of the output flange for diesel engine are to meet the requirements of the applicable engine type, but it has been cancelled by WÄRTSILÄ.
(5) The stiffness and clearance of bearings are to be taken into account.

9.8.3 Relevant illustrations
(1) These measures taken by the diesel engine license manufacturers are related to the effects of hull deformation on the diesel engine bearings, without involving the effects of other bearing loads, bending stress and slope of bearing for shafting system, particularly for tube bearing load and slope of aftermost sterntube bearing of which the effects are not clear, therefore some uncertainty and risk exist, which may cause the failure of tube bearing. An overall evaluation is required through calculation.
(2) If the above-mentioned recommendations provided by diesel engine license manufacturers are adopted by the shipyards, the shafting alignment calculations are to be complied with, and in addition, the installation to be made in accordance with the approved technical documents in the shafting alignment technical procedures. Otherwise, readjustment in the installation of shafting system may alter the shafting alignment condition and with some uncertainty and risk.
(3) Due to design of shafting system, where the temperature of intermediate shaft bearing is higher, the displacement of main engine bearing is to be appropriately raised, i.e. the load of main engine bearing is to be increased so as to reduce the load of intermediate shaft bearing.
(4) Particular attention is to be given to the alteration of shafting alignment requirements of the license manufacturers.

9.9 Concerning the Effects of Ambient Temperature of Installation on Shafting Alignment

9.9.1 Requirements of ISC Rules for Classification of Sea-going Steel Ships
The relevant provisions of Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships are as follows:
(1) In shafting alignment calculation, the influence of thermal expansion of heated bearings (e.g. 55℃) during operation of the engine or gearbox and so far as possible, the influence of expansion of heated tanks (e.g. 45℃) in double bottom that are located below intermediate shaft bearings, are to be taken into account.
(2) In shafting alignment calculation, the influence of different ambient temperatures (taken as 0℃, 10℃ or 20℃) on the installation of shafting systems is to be taken into account in so far as

practicable.

(3) In shafting system installation, where any noncompliance with calculation conditions (e.g. the ambient temperature, immersion of propeller, ship's condition, load adjustment of main engine bearings of engines) may affect the alignment results, the alignment calculations together with alignment instructions are to be resubmitted for approval, unless such effects have been included in the calculations.

9.9.2 Analysis for effect of ambient temperature of installation of diesel engine in engine room on shafting alignment results

The effect of ambient temperature of installation on shafting alignment will be varied by different ship types, particularly for the influence on short shafting system, sterntube forward bearing and the last main engine bearing will be greater.

Taking the 7S80MC diesel engine for example, the ambient temperature given by the manufacturer is 20℃, the thermal expansion of main engine at hot condition of 55℃ is 0.38 mm, and those at other ambient temperatures are shown in Table 9.9.2-1.

Effect of Ambient Temperature of Installation on Expansion Table 9.9.2-1

Ambient temperature of installation (℃)	Difference of temperature (℃)	Thermal expansion at 55℃ (mm)	Ratio with expansion at the ambient temperature of 20℃ (%)
20	35	0.38	100
10	45	0.49	129
0	55	0.60	158

As shown from Table 9.9.2-1, the thermal expansion of main engine may be increased to 158% during the installation at 0℃, if the expansion is calculated according to the ambient temperature of installation in engine room is 20℃.

Where tanks are located below intermediate shaft bearings, the bottom structure of bearings will expand due to the constantly raise of oil temperature, and the bearing displacement may be influenced.

In order to analyze the influence of different ambient temperatures of installation on alignment under hot and cold conditions, for a certain large bulk carrier, shafting alignment calculation is to be carried out at the ambient temperatures of installation in engine room of 20℃ (difference of 35℃ between cold and hot conditions), 10℃ (difference of 45℃ between cold and hot conditions) and 0℃ (difference of 55℃ between cold and hot conditions) respectively.

As shown from Table 9.9.2-2, the loads of sterntube forward bearing and main bearing 1 are 70.43 kN and 25.19 kN respectively if the shafting alignment calculations is based on ambient temperature under normal cold condition of 20℃, however, if the actual ambient temperature of installation is at 0℃, the bearing load verification carried out after the installation of shafting system is to be in accordance with the loads of sterntube forward bearing and main bearing 1of 75.91 kN and 12.31 kN respectively. Where the verification is carried out still based on the temperature of 20℃, the conditions between shafting system installation and alignment calculation are completely different.

Results of Alignment Calculation at Different Ambient Temperatures of Installation Under Heat and Cold Conditions for a Certain Larger Bulk Carrier Table 9.9.2-2

Conditions	Hot	Cold	Cold	Cold

	(ambient temp. 55℃)		(ambient temp. 20℃)		(ambient temp. 10℃)		(ambient temp. 0℃)	
Bearing and relevant parameters	Displacement (mm)	Load (kN)	Displacement (mm)	Load (kN)	Displacement (mm)	Load (kN)	Displacement (mm)	Load (kN)
Aftermost sterntube bearing	0.11	527.62	0.11	531.24	0．11	529.8	0．11	528.40
Sterntube forward bearing	0.00	84.83	0.00	70.43	0．00	73.21	0．00	75.91
Intermediate shaft bearing	-2.55	96.37	-2.55	125.46	-2.60	124.92	-2.65	124.56
Main bearing 1	-4.47	111.68	-4.8	25.19	-4.894	19.01	-4.99	12.31
Main bearing 2	-4.47	191.02	-4.8	260.84	-4.894	266.33	-4.99	272.26
Main bearing 3	-4.47	249.44	-4.8	247.37	-4.894	247.20	-4.99	247.20
Main bearing 4	-4.47	230.10	-4.8	230.66	-4.894	230.70	-4.99	230.75
Main bearing 5	-4.47	292.94	-4.8	292.80	-4.894	292.78	-4.99	292.78
Main bearing 6	-4.47	83.24	-4.8	83.26	-4.894	83.26	-4.99	83.26
Slope of aftermost sterntube bearing(rad) $\times 10^{-4}$		4.066		4.015		4.035		4.055

The following conclusions may be obtained through calculation and analysis:

(1) The ambient temperature of installation in engine room has a certain influence, particularly for the influence on short shaft system, sterntube forward bearing and the last main engine bearing will be greater. Therefore, if the shafting system is installed at lower ambient temperature, the effect of ambient temperature on shafting alignment is to be taken into account.

(2) In shafting alignment calculations, shafting alignment data are to be calculated at the ambient temperature of installation at 20℃, 10℃ and 0℃ respectively, in shafting alignment, shafting system is to be installed according to the calculated values approximate to ambient temperature.

(3) The influence of intermediate shaft bearing displacement on shafting alignment is also greater. Therefore, in shafting alignment calculation, the influence of expansion of tanks in double bottom that are located below intermediate shaft bearings is to be taken into account, as far as possible. Therefore, if the shafting system is installed at a lower ambient temperature, the effect of ambient temperature of installation on shafting alignment is to be taken into consideration. In alignment calculation, the thermal expansion is to be calculated according to the actual installation temperatures at 0℃, 10℃ and 20℃ with the difference of 55℃ for the maximum temperature of main engine or gearbox. In order to reduce the working loads, the possible minimum temperature provided by shipyard may be used for deformation calculation, such as 0℃, the difference is (55－0) ℃. In fact, it increases the bearing displacement as appropriate so as to resist the adverse effect of hull deformation on shafting alignment. During the site construction and inspection, the alignment calculation results at the approximate temperatures is to be used and confirmed.

9.9.3 Analysis for effect of ambient temperature of installation of intermediate shaft on shafting alignment results

In order to analyze the effect of different intermediate shaft bearing displacement on shafting alignment, shafting alignment calculation is carried out with the intermediate shaft bearing displacement is at -1.9 mm, -1.8 mm, -1.7 mm, -1.6 mm, -1.5 mm and -1.4 mm respectively in the event of only changing the intermediate shaft bearing displacement for a certain large bulk carrier with unchangeable displacement of each bearing of shafting system, the calculation results are shown in Table 9.9.3.

As shown from Table 9.9.3, the change of intermediate shaft bearing has less influence on the loads of aftermost sterntube bearing, main bearing 3 to main bearing 6, but has greater influence on the loads of sterntube forward bearing, intermediate shaft bearing and main bearing 1. If the intermediate shaft bearing displacement is at -1.9 mm, the loads of sterntube forward bearing and main bearing 1 are to be of 59.23 kN and 51.37 kN respectively and if the intermediate shaft bearing displacement is at -1.7mm, the load of main bearing 1 is negative and not to meet the alignment requirements. If the intermediate shaft bearing displacement is at -1.4 mm, the loads of sterntube forward bearing and main bearing 1 are -9.613 kN and -133.22 kN respectively, and there is voidable sterntube forward bearing load.

Therefore, if the shafting system is installed at lower ambient temperature, the effect of ambient temperature of installation on shafting alignment is to be taken into consideration. In alignment calculation, the thermal expansion is to be calculated according to the actual installation temperatures at 0℃, 10℃ and 20℃ with the difference of 45℃ for the tanks below the intermediate shaft bearing. In order to reduce the working loads, the possible minimum temperature provided by shipyard may be used for deformation calculation, such as 0℃, the difference is (45 — 0) ℃. In fact, it increases the bearing displacement as appropriate so as to resist the adverse effect of hull deformation on shafting alignment. During the construction and inspection on site, the alignment calculation results at the approximate temperatures is to be used and confirmed.

Effect of Various Intermediate Shaft Displacement on Shafting Alignment for a Certain Large Bulk Carrier **Table 9.9.3**

Conditions	Displacement under hot condition (mm)	Load of each bearing with intermediate shaft bearing displacement at -1.9mm (kN)	Load of each bearing with intermediate shaft bearing displacement at -1.8mm (kN)	Load of each bearing with intermediate shaft bearing displacement at -1.7mm (kN)	Load of each bearing with intermediate shaft bearing displacement at -1.6mm (kN)	Load of each bearing with intermediate shaft bearing displacement at -1.5mm (kN)	Load of each bearing with intermediate shaft bearing displacement at -1.4mm (kN)
Aftermost sterntube bearing	0.0	541.7	546.64	551.57	556.50	561.43	566.36
Sterntube forward bearing	0.0	59.23	45.46	31.70	17.92	4.157.92	-9.613
Intermediate shaft bearing	Changed	110.31	127.97	145.62	163.28	180.93	
Main bearing 1	-2.7	51.37	14.45	-22.47	-59.38	-96.30	
Main bearing 2	-2.48	292.38	321.16	349.94	378.71	407.49	
Main bearing 3	-2.26	195.62	194.76	193.90	193.05	192.19	191.33
Main bearing 4	-2.04	238.34	238.57	238.79	239.02	239.25	
Main bearing 5	-1.82	298.52	298.46	298.41	298.35	298.30	298.24
Main bearing 6	-1.61	79.75	79.77	79.77	79.78	79.79	
Slope of aftermost sterntube bearing(rad) $\times 10^{-4}$		4.123	4.053	3.983	3.913	3.844	

9.10 Concerning the Effects of Hull Deformation on Shafting Alignment

9.10.1 The rules requirements

Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships specifies that the effects of hull deformation under ballast and full-load conditions are to be taken into consideration for shafting alignment of large ships.

9.10.2 Calculation principles of hull deformation on shafting alignment

Although the effects of hull deformation on shafting alignment are required by classification societies' rules, actually, effect of hull deformation has not been taken into account for shafting alignment of all ships, including the large ships. Considering effects of hull deformation means the effects of hull deformation under ballast and full-load conditions are to be taken into account in shaft alignment design so as to be included in alignment calculation. In addition, different calculation model of hull deformation may obtain various hull deformation results and testing is to be carried out to verify the rules requirements.

Therefore, when the situations of shafting system centerline deformation under ship's floating, ballast and full-load conditions are researched through finite element calculation and analysis of effects of hull deformation on relative deformation of shafting system centerline in vertical, i.e. deformation under each loading condition and relative position under various loading conditions in order to provide reference for shafting alignment design.

The hull deformation of stern – engine room – cargo hold and vertical deformation shape of shafting system centerline may be predicted by the relative difference of deformations between the full-load and ballast conditions and floating condition based on the reference line through the establishment of a "stern- engine room – cargo hold" three-dimensional finite element model (as shown in Figure 9.10.2) and inclusion of distributed loads of gravity and buoyancy under various main loading conditions.

The general principles to determine the hull deformation calculation are as follows:

(1) Ship's operating conditions (states)

① Floating condition: means the state after ship is launched, all the ballast tanks are void, main hull has been completed with the superstructure, main engine and main equipment have been installed in place, it is a working condition for shafting alignment.

② Light load (or ballast) condition: means all cargo holds are void, all ballast tanks are full-loaded or in the actual loading condition specified by the loading manual.

③ Full load condition: means all cargo holds are full-loaded, all ballast tanks are void or in the actual loading condition specified by the loading manual.

(2) Calculating loads

① Considering weights of all typical characteristics in ship model zones, such as weights of hull structure (including superstructure), main engine, main equipment, rudder and propeller.

② Various oil and water, ballast water, cargoes.

③ External hydrostatic pressures under draughts corresponding to various working conditions.

④ The effects of wave load, propeller dynamic force and propeller thrust are neglected.

(3) Requirements of hull modeling

① The model can correctly reflect the vertical deformation of double-bottom of engine room.

② The scope at least includes model of all hull structure parts (including superstructure) backward from fore transverse bulkhead of the cargo hold before the engine room, model of engine room and model of part after engine room.

③ The effect of additional stiffness of main engine may not be taken into consideration.

④ Taking the supporting point of bearing as a calculating point: supporting point is located in $d_p/3$ for aftermost sterntube bearing lined with white-metal, d_p being the diameter of propeller shaft (or $L/4$ is taken, L being the length of bearing); supporting point is located in $d_p/2$ for other bearings, d_p is the diameter of bearing. In the calculation, assuming that the relative deformation between sterntube forward bearing or aftermost sterntube bearing and the 1st main bearing at free end of crankshaft is equal to 0, and it is regarded as a reference line.

⑤ Where the relative deformation is calculated between ballast and full load conditions, the unchangeable load between light load and full load conditions (such as superstructure, main engine, main equipment, etc.) and the rudder and propeller weights which has less influence on analysis results may not be taken into account in order to reduce work.

(4) Conditions of boundary

All of nodes of longitudinal continuous structural members on the fore transverse bulkhead of model end face are to be constrained by hinges, i.e. the displacements of node along longitudinal, transverse and vertical directions are equal to 0.

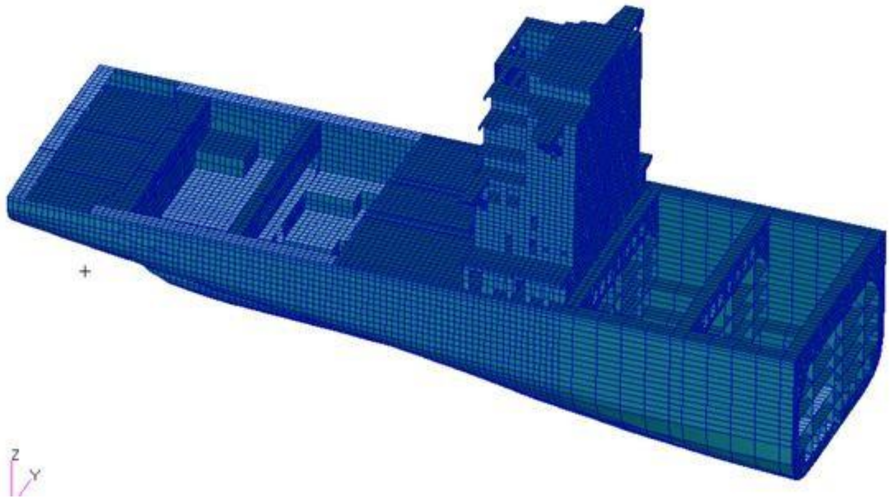


Figure 9.10.2 “Stern - Engine Room – Cargo Hold” Finite Element Model

9.10.3 Basic evaluation for hull deformation finite element calculation with effect on shafting alignment

(1) The hull deformation of stern – engine room – cargo hold and vertical deformation shape of shafting system centerline may be predicted by establishing a “stern – engine room – cargo hold” three-dimensional finite element model and including finite element analysis and calculation methods and working flows for distributed loads of gravity and buoyancy under various main loading conditions.

(2) The calculation methods and flows have generality and not related to the ship’s type, however, the calculated deformation value and deformation situation of shafting system centerline have particularity and associated with specific ship’s type and actual loading condition.

9.10.4 Effect of shape for typical hull deformation on shafting alignment

(1) Propulsion shafting systems for container ships

A certain larger container ship is shown in Figure 9.10.4-1, there is one cargo hold after the engine room, the vertical relative deformation shape of the ship under afloat, ballast and full load conditions are as follows:

① the relative deformation of shafting system centerline is distributed in a convex curve, i.e. the middle part of shafting system centerline is higher than two ends;

② the relative deformation under afloat condition is the minimum and that under full load condition is the maximum;

③ inflection point occurs in way of thrust bearing and crankshaft afterward bearing under afloat and ballast conditions;

④ the maximum relative deformation value under each working condition occurs at the middle of curve, the shafting system looks more flexible;

⑤ the relative deformation difference between ballast and afloat conditions is less, but the difference between full load and afloat is larger.

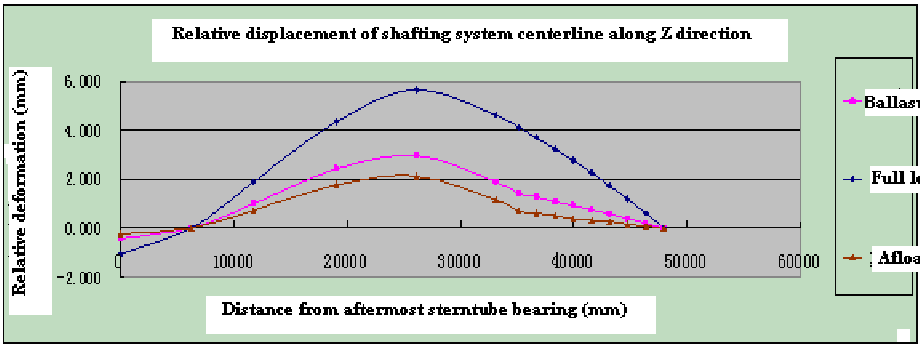


Figure 9.10.4-1 Vertical Relative Deformation Shape of Shafting System Centerline for a Certain Larger Container Ship

(2) Propulsion shafting systems of tankers and bulk carriers with engine installed at stern

A certain oil tanker is shown in Figure 9.10.4-2, the engine is installed at stern, the vertical relative deformation shape of the ship under afloat, ballast and full load conditions are as follows:

① The relative deformation of shafting system centerline under full load condition is distributed in a convex curve, i.e. the middle part of shafting system centerline is higher than two ends; however those under afloat and ballast conditions are distributed smoothly and in a concave curve.

② The relative deformation under ballast condition is the minimum and that under full load condition is the maximum.

③ The absolute deformation under ballast condition is larger than that under afloat condition, however the relative deformation under ballast condition is less than that under afloat condition, the relative deformation is not proportional to the absolute deformation under the same condition.

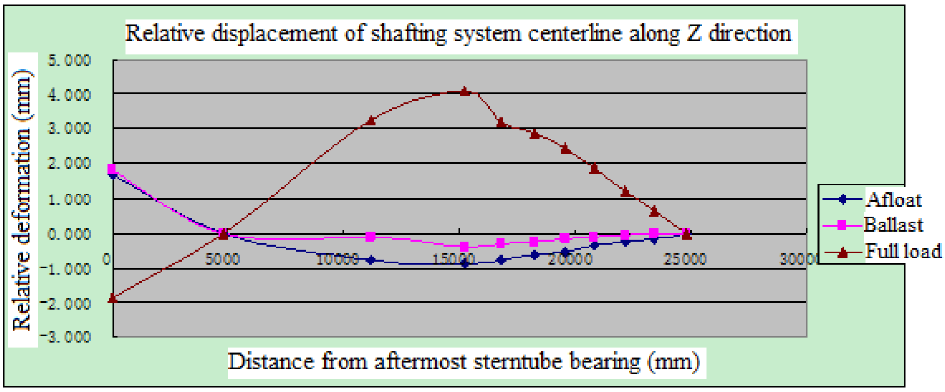


Figure 9.10.4-2 Vertical Relative Deformation Shape of Shafting System Centerline for a Certain Oil Tanker

9.10.5 Basic principles to considering effects of hull deformation on shafting alignment

(1) Application of hull deformation calculation with effects on shafting alignment

The hull deformation finite element calculation with effects on shafting alignment may be used to predict the hull deformation, although the method takes considerable time and effort, it may be used for design reference of the shafting alignment for the first ship in series and also used for qualitative analysis of accident cases.

(2) Inclusion of accurate values of hull deformation

If there are accurate values of hull deformation, the effects of hull deformation on shafting alignment are just included as a group of input parameters – changing the displacement value of bearing.

(3) Measures taken by diesel engine manufacturers

In order to prevent the damage of large low-speed diesel engine main bearings and crankshafts and neutralize or reduce the effects of hull deformation on main engine bearing loads, measures are to be taken by the diesel engine manufacturers when the engine is being installed onboard ships:

① anti-deformation method is to be taken so as to make a certain deflection of engine pedestal downward in advanced, i.e. making the displacement of main bearing in a concave state, the installation condition of 7S80MC diesel engine of MAN B&W onboard ships is shown in Figure 9.10.5;

② engine pedestal is to be arranged trim by stern, the shafting alignment calculations for ships with engine located in stern for two different tonnages are shown in Figure 9.10.5.

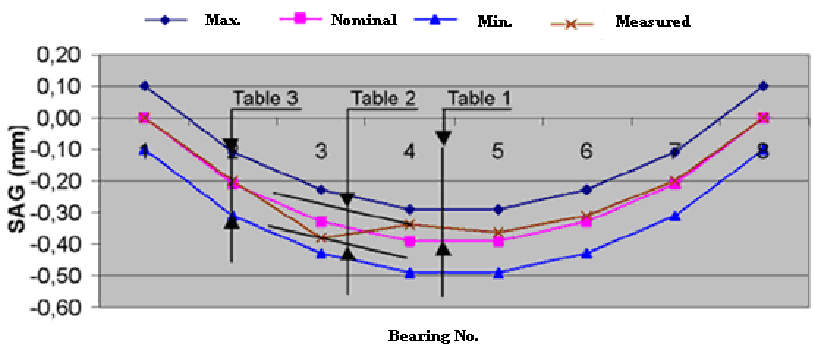


Figure 9.10.5 Concave State of Diesel Engine Installed Onboard

Arrangement of Main Engine **Table 9.10.5**

Tonnage (ten thousands)	Type of main engine	Displacement of intermediate shaft (cold condition) (mm)	Displacement of last shaft of main engine (cold condition) (mm)	Expansion of main engine bearing (mm) (55—20) ℃	Position of main engine
18	6S70MC	2.55	4.80	0.33	Placed flatly
17.7	6S70MC	1.90	3.03	0.33	Inclining by stern
29.7	7S80MC	1.60	2.60	0.38	Inclining by stern

(4) Basic principles for considering effects of hull deformation on shafting alignment design Although the effects of hull deformation is not taken into account in the shafting alignment calculation currently, as a matter of fact, some are considered of expansion of intermediate shaft bearing while some use oblique bores of aftermost sterntube bearing. The adoption of increase of displacement value between main engine and intermediate shaft is equivalent to take the hull deformation into consideration, and it is in compliance with the requirements of shafting alignment.

Based on the research for shafting system faults of some ships, it is regarded that, in the shafting alignment calculation, the adverse influence of hull deformation on shafting alignment may be resisted provided sufficient displacement of main engine is included, therefore the time-consuming hull deformation calculation and measurement is not necessary, so as to simplify the complicated technical issues.

Hence, the following basic principles may be used in the shafting alignment design:

① The bearing is to be arranged in a rigid pedestal and kept as far away from maximum position of hull deformation as possible so as to reduce the adverse effect of hull deformation on bearing load.

② For propulsion shafting system directly driven by diesel engine, the displacement of main engine bearing under hot condition is recommended at least 1.20 mm lower than theoretical centerline.

③ For propulsion shafting system directly driven by diesel engine, the loads of last 1st and 2nd main bearings under hot condition may be controlled within the range of 10% to 40% allowable load in order to prevent the excessive hull deformation which may cause overload of the last main bearing.

④ The maximum displacement value is to be determined by shape of relative deformations of shafting system centerline under ship’s afloat, ballast full load conditions according to various ship types.

9.11 Shafting Alignment for the Screwshaft with the Diameter Less Than 250 mm

9.11.1 The rules requirements

The relevant provisions in Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships are as follows:

For shafting systems where the screwshaft diameter is less than 250 mm, shafting alignment calculations are in general to be submitted for reference.

Where shafting alignment calculations are not submitted, the span of bearings etc., are to be included in the shafting strength calculations as required by Chapter 11 of that PART, reference may be made to the relevant requirements for shafting alignment in Chapter 9 of ISC Guidelines for Shipboard Vibration Control. In addition, the technological documents submitted by the shipyard regarding shafting alignment are to contain alignment instructions.

9.11.2 Implementation of shafting alignment for screwshaft with the diameter less than 250 mm

As mentioned above, for shafting alignment for small ships with the screwshaft diameter less than 250 mm and sterntube having a bearing with the screwshaft diameter less than 200 mm, straight alignment is to be applied by shipyards as a convention.

This is a new requirement raised by classification societies' rules for shafting alignment of small ships, but not mandatory, i.e. the submission of shafting alignment calculations is not required. Where the shafting alignment calculations are not submitted, the span of bearings, etc., may be included in the submitted instructions for shafting strength calculations.

If the bearings are found being arranged unreasonably, the shafting alignment calculations including alignment instructions is to be submitted for approval by shipyards. The span of bearings referred here is for the normally arranged main propulsion shafting systems provided with sterntube devices, which is not applicable to propulsion shafting systems with cardan or rudder and propeller unit.

Where the shafting alignment calculations is not submitted, the contents of shafting alignment instructions are to be included in shafting alignment processing specifications submitted by shipyards.

For straight alignment of shafting system, i.e. previous flange alignment with zero to zero, due to the shaft weight, the unconnected flanges are to be in drooping state, therefore, appropriate temporary supporting point is to be taken in order to ensure the axial is in a straight line state, i.e. flange alignment with zero to zero, as shown in Figure 9.11.2. For normally designed flange dimension, the distance from the middle point of temporary supporting to the end of flange A is calculated as follows:

$$A = (0.18 \sim 22)L \qquad m \qquad (9.11.1)$$

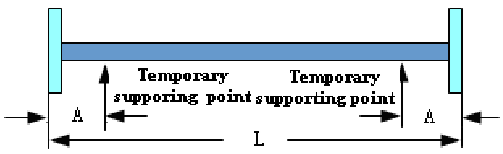


Figure 9.11.2 Positions of Temporary Supporting Points

9.12 Criteria

9.12.1 The rules requirements

The relevant provisions in Chapter 12, PART THREE of ISC Rules for Classification of Sea-going Steel Ships are as follows:

(1) In the static condition, all bearings loads are to be positive, i.e. there is to be no voidable bearing load. The bearing load is generally not to be less than 20% of the sum of all weights between two adjacent spans.

(2) Bearing loads are in generally not to exceed the values specified below or by the manufacturer:

aftermost sterntube bearing bearings: as specified in 11.2.5.1 of PART THREE of ISC Rules

for Classification of Sea-going Steel Ships;
sterntube forward bearings: 0.8 N/mm^2;
nonmetallic sterntube bearings: 0.3 N/mm^2;
intermediate shaft bearings: 0.8 N/mm^2;
gear shaft bearings: 1.0 N/mm^2;
main bearings of diesel engines: as specified by the engine manufacturer.

(3) Additional bending stresses of shafts are generally not to exceed the following values:
screwshaft and sterntube shafts: 20 N/mm^2;
intermediate shaft: 20 N/mm^2;
thrust shaft: 15 N/mm^2;
gear wheel shafts: 10 N/mm^2 or as specified by the gearbox manufacturer.

(4) The bending moments and shearing forces applied to the output flange of diesel engines are not to exceed those as specified by the engine manufacturer (if required). The minimum load of main bearings of main engines is in general not to be less than 10% of the allowable load on main bearings. Alternatively, the minimum value specified by the engine manufacturer may be accepted, but this is to be reflected in shafting alignment calculations.

(5) The load difference between fore and aft bearings of the gear wheel of the gearbox is to meet the relevant requirements of the manufacturer, generally not exceeding 20% of the sum of the weight of the shaft portion between the two bearings and that of the gear wheel.

Where the alignment calculation results in running condition are provided and the bearing structure is confirmed to be determined according to the acting angle of the resultant force in the running condition, the load difference between fore and aft bearings may not be limited to 20% as specified above provided that the relevant requirements of Appendix 1, Chapter 10 of PART THREE of ISC Rules for Classification of Sea-going Steel Ships are complied with.

(6) The relative angle between the screwshaft and the aftermost sterntube bearing at the supporting point of the bearing is, in general, not to exceed 3.5×10^{-4} rad in the static condition.

9.12.2 Relevant illustrations

(1) Due to the arrangement of shafting system, it may be acceptable for sterntube forward bearing load or intermediate shaft bearing load which do not meet the requirement of not to be less than 20% of the sum of all weights between two adjacent spans, but not less than 10% allowable load of intermediate shaft bearing. However, the measured value and calculated value of bearing load is to be strictly controlled within the range of ±20%.

(2) The shafting alignment requirements provided by diesel engine manufacturer are to be included in the shafting alignment calculations as the conditions. Where the shafting alignment calculations are approved by ISC, the displacement of main engine bearing can not be adjusted randomly during installation in order to avoid the excessive large or small loads of intermediate shaft bearing or sterntube bearing, even to cause faults.

(3) Although the maximum value of bearing loads are specified by the rules, a certain margin is to be remained for the number of bearings and the arrangement of the axial positions so as to prevent overloading of bearings due to hull deformation, it is recommended to control within the range of 80% allowable loading of bearings.

9.13 Plan Approval and Inspection

9.13.1 Plan approval of shafting alignment

(1) General requirements:

① to examine the arrangement of shafting system;

② to confirm whether reasonable shafting alignment needs to carry out;

③ to confirm the alignment method to be used;

④ to confirm the length of bearing to be in compliance with the rules requirements;

⑤ to confirm the shafting alignment instructions to be in compliance with the rules requirements.

(2) Straight shafting alignment:

① to confirm the span of bearings to be in compliance with the requirements (if shafting alignment calculations is not submitted);

② to confirm temporary supporting to be in compliance with the requirements.

(3) Alignment of bearing load for shafting system:

① to confirm the span of bearings to be in compliance with the requirements (if shafting alignment calculations is not submitted);

② to confirm the calculated bearing load to be in compliance with the requirements of 9.3.3.

(4) Reasonable shafting alignment:

① to confirm the contents of shafting alignment calculations to be complete and in compliance with the rules requirements;

② to confirm the diagram of shafting alignment to be in consistent with the submitted drawings of shafting system;

③ to examine the diagram of shafting alignment, including all of the concentrated load acting on the shafting system, the main structural dimension, etc.;

④ to confirm the shafting alignment calculation under hot condition to be in compliance with the rules requirements;

⑤ to confirm the shafting alignment calculation under running condition to be in compliance with the rules requirements (if applicable);

⑥ to confirm the shafting alignment under cold condition to be correct:

— considering the change of value of bearing displacement corresponding to hot condition;

— confirming the correctness of raising value of bearing displacement;

— confirming the expansions of different installation temperatures.

⑦ to confirm the shafting calculation under installation condition to be in compliance with the calculation results of alignment under cold condition:

— confirming the alignment instructions to be in compliance with the rules requirements;

—confirming the propeller condition during installation.

9.13.2 Shafting alignment survey

(1) General requirements:

① to confirm the technological documents of shafting alignment installation provided by shipyard to be in compliance with the approved shafting alignment calculation;

② to confirm the shafting alignment instructions to be in compliance with the rules requirements;

③ for reasonable shafting alignment, where the calculation conditions are not complied with and likely to influence the alignment results, such as ambient temperature, immersion of propeller, ship's operating condition, adjustment of engine main bearing, etc., in the shafting installation, the shipyard is to be required to resubmit the alignment calculations for approval together with the alignment instructions unless such influences have been included in the calculations;

④ to confirm the measures to be taken to prevent screwshaft change (if required);

⑤ where the measured or verified bearing load of reasonable shafting alignment exceeds ±20% of the calculated value, shafting alignment calculation is to be carried out according to the measured results, and may be accepted if the other rules requirements are satisfied.

(2) Straight shafting alignment:

① to confirm the positions of temporary supporting to be in compliance with the requirements;

② to examine gap and sag of each flange pair where the gap and sag method is used under the ship's afloat condition, the tolerance is generally not to exceed ±0.08 mm.

(3) Alignment of bearing load for shafting system:

① to confirm the actual load of each intermediate shaft bearing to be correct;

② to confirm the actual load of each intermediate shaft bearing to be in compliance with the requirements.

(4) Gap and sag alignment method is to be used by reasonable alignment shafting systems:

① to confirm the axial positions of temporary supporting to be in compliance with the requirements of calculations;

② to examine gap and sag of each flange pair where the gap and sag method is used under the ship's afloat condition, the tolerance is generally not to exceed ±0.08 mm.

③ to select 1 to 2 bearings with typical characteristics for loading verification, such as the measurement of sterntube forward bearing, one intermediate shaft bearing and the aftermost main engine or gearbox bearing; in the absence of sterntube forward bearing, one intermediate shaft bearing and aftermost gearbox bearing or aftermost main engine bearing are to be measured, and generally not to exceed ±20% calculated value under cold condition.

(5) Jack-up alignment method is to be used for reasonable alignment shafting systems:

① to adjust the gap and sag of unconnected flanges according to the requirements of alignment calculations, to be only as an intermediate process of shafting alignment by shipyards;

② to measure and adjust the actual bearing load under cold condition by jack-up method under the ship's afloat condition as to confirm the jack-up position to be the same as that in the calculations;

③ for propulsion shafting system directly driven by diesel engine, measurement and adjustment are to be carried out for sterntube forward bearing load, intermediate shaft bearing load and last 1 to 3 main engine bearing loads as to confirm to be in compliance with the requirements in the alignment calculations; for the propulsion shafting system with gearbox, measurement and adjustment are to be carried out for stertube forward bearing load, intermediate shaft bearing load and aftermost gearbox bearing load (as far as possible) as to confirm the requirements of alignment calculations to be complied with, the difference between the measured load and the calculated value is generally not to exceed ±20％ calculated value under cold condition.

(6) Bearing displacement alignment method is to be used for reasonable alignment shafting systems:

① to adjust the displacement of each bearing under cold condition when the ship is afloat by optical instrument or laser collimator, etc., as to comply with the requirements of alignment calculations;

② to select 1 to 2 bearings with typical characteristics for loading verification, such as the measurement of sterntube forward bearing and one intermediate shaft bearing or the aftermost main engine or gearbox bearing; in the absence of sterntube forward bearing, one intermediate shaft bearing and aftermost gearbox bearing or aftermost main engine bearing are to be measured, and generally not to exceed ±20% calculated value under cold condition.

(7) After completion of shafting installation:

① to inspect the engagement of gears;

② to inspect the tolerance of crank webs of diesel engine and to meet the relevant provisions of the diesel engine manufacturers.

(8) After completion of sea trial, bearings are to be dismantled for inspection, if necessary.

Chapter 10 MECHANICAL VIBRATION

10.1 Introduction

10.1.1 Characteristics of ship's mechanical vibration

There are many diffenrent types and specifications of ship machinery, which may be divided into two parts, such as main engine and auxiliary engine according to the function; and may be divided into two types, such as reciprocating machinery which are diesel engine, plunger pump, compressor and rotary machinery which are turbine, pump, fan, motor, etc., according to the operation. Periodical exciting force is always caused when the machinery is running, thus to lead to the mechanical vibration of itself. As seen from the vibration spectrum, the width of frequency band for these mechanical vibration is 2 Hz to 8000 Hz, mainly distributed within the frequency range of 2 Hz to 1000 Hz.

The equipment onboard ships includes boilers, incinerators, etc., which do not induce any exciting force by itself, but may lead to equipment vibration subject to the action of exciting force transmitted by pedestals.

The pipes onboard ships may also induce pipe vibration due to internal fluid or external exciting actions.

The machinery onboard ships includes mechanism, equipment and pipes, the mechanism and equipment may be called machinery unless otherwise specified.

10.1.2 Hazard of ship's machinery vibration

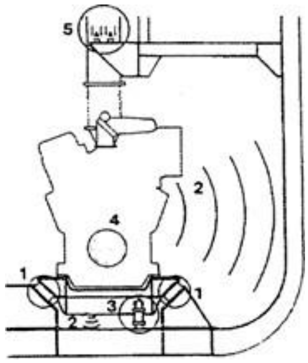
The strong mechanical vibration may induce the fatigue damage of itself, and also transmit the vibration to the relevant mechanism and equipment through the components connected with the hull. The strong vibration of ship's equipment may also influence the normal operation.

The strong vibration of pipes may induce fatigue damage of pipes and their attachments, and may aggravate vibration and noise around the pipes and influence the normal operation of the connected machinery.

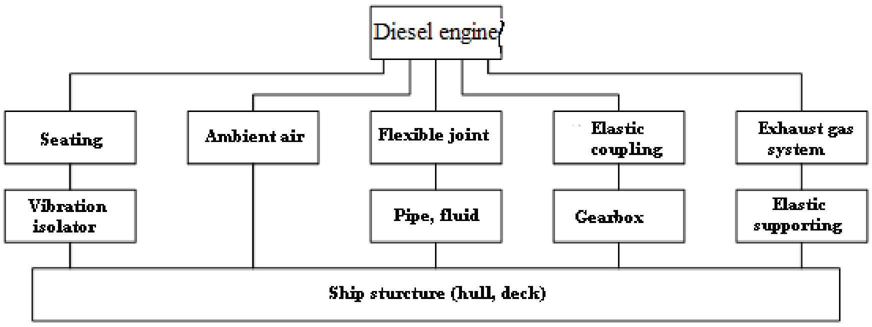
10.2 Basic Principles

10.2.1 Transmission path of mechanical vibration

The vibration caused by mechanism is transmitted to the base, deck and bulkhead plating then to induce vibration and propagate with hull structure in a elastic wave type to lead to structural vibration (also called structural noise). The diagram of transmission path of diesel engine vibration is shown in Figure 10.2.1.



(1)



(2)

Figure 10.2.1 Diagram of Transmission Path of Diesel Engine Vibration

As seen from the diagram, the engine vibration is transmitted through the following five paths:
(1) base: vibration is transmitted to hull structure directly through base or vibration isolator;
(2) air: radial noise on the surface of diesel engine is transmitted directly to the whole compartment through air and will cause the secondary excitation for hull structure;
(3) flexible joint: vibration is transmitted to hull structure, such as bulkheads, etc., through various flexible joints, pipes and fluids (water, oil) accumulated in pipes;
(4) elastic coupling: vibration at output of diesel engine is transmitted to hull structure through couplings, gearboxes and their supporting;
(5) exhaust gas system: Vibration of exhaust pipes and air vibration in pipes is transmitted to hull structure directly or through elastic supporting.
As a convention, the transmission path from pedestal to base is called the first path, the others are called the second path.

10.2.2 Basic parameters of vibration isolation

The vibration isolation technology is to control three basic parameters in mechanical vibration system: vibration isolation mass, stiffness and damping. The function of these three basic parameters is as follows:
Mass: under the action of fixed exciting force, the responding frequency amplitude becomes smaller as the mass of isolated object is larger.
Stiffness: under a same exciting frequency, the effect of isolation becomes better as the stiffness of vibration isolator is smaller, , on the contrary, the effect of isolation becomes poor. The stiffness determines the vibration isolation efficiency of whole system and meanwhile, it is related to swing degree of the system.

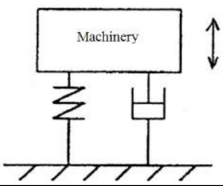
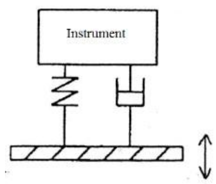
Damping: resonance peak is to be reduced in resonance zone so as to restrain the resonance amplitude, however, additional connection is to be provided for the system in isolation area to lead to the spring short circuit so as to raise the stiffness of supporting and reduce the efficiency of vibration isolation. Hence, a careful analysis for damping is required in the design.

10.2.3 Positive/active vibration isolation and negative/passive vibration isolation

In order to reduce or control the transmission of machinery equipment or structure vibrations, elastic element (normally called vibration isolator) is to be inserted on the transmitting path between the two structures so that the dynamic excitation or movement excitation transmitted from one structure to the other will be reduced, this is the vibration isolation (shortly called isolation). According to different vibration excitations, two types of positive isolation (or called active isolation) and negative isolation (or called passive isolation) are to be grouped. The purpose and application for these two isolations are listed in Table 10.2.3.

Although the concepts of positive isolation and negative isolation are different, the implementation methods are same, i.e. isolator (elastic element consisting of stiffness and damping) is to be installed between the isolated object and base. The difference is that the positive isolation reduces the force transmitted on the base, one portion of periodical exciting force is resisted to the inertial force of the machinery equipment itself, and the other portion is absorbed and consumed by isolator, but for the negative isolation, most portions of base vibration are absorbed and consumed by isolator and the isolated object remains basically static depending on inertia.

Positive Isolation and Negative Isolation **Table 10.2.3**

Type	Diagram	Purpose	Application
Positive isolation		Isolating or reducing the vibration transmitted to the base through pedestal, support caused by machinery equipment so as to avoid the influence of vibration on surrounding environment or adjacent structures	Various equipment, such as movable equipment, rotary machinery, reciprocating machinery, punching machine
Negative isolation		Preventing the vibration around transmitting to the instruments, equipment, precision machinery need protection through support and pedestal	Electronic instrument, navigational equipment, electric closet, transported valuables, etc.

10.2.4 Absolute transmission rate

The absolute transmission rate T_A means the ratio between the exciting force transmitted to base and that of rigid connection if isolator is provided.

The curve of absolute transmission rate T_A varied with frequency ratio ω/ω_n is shown in Figure 10.2.4, as follows:

(1) Regardless of the damping ratio, the absolute transmission rate T_A is less than 1 only when the frequency ratio $\omega/\omega_n > \sqrt{2}$. Hence, the selection of stiffness for single degree of freedom isolation system is to meet the requirement of $\omega/\omega_n > \sqrt{2}$, otherwise, the vibration will be amplified.

(2) When $\omega/\omega_n > \sqrt{2}$, the value of the absolute transmission rate T_A is getting smaller with the increase of frequency ratio ω/ω_n, and the isolation effect becomes better. However, the frequency ratio ω/ω_n should not be too greater, for the isolator requires large static compression, i.e. the spring becomes so soft that the machinery is liable to swing, and when the frequency ratio ω/ω_n > 5, the absolute transmission ratio T_A changes little, therefore the actually adopted frequency ratio ω/ω_n should be appropriately between 2.5 and 4.5.

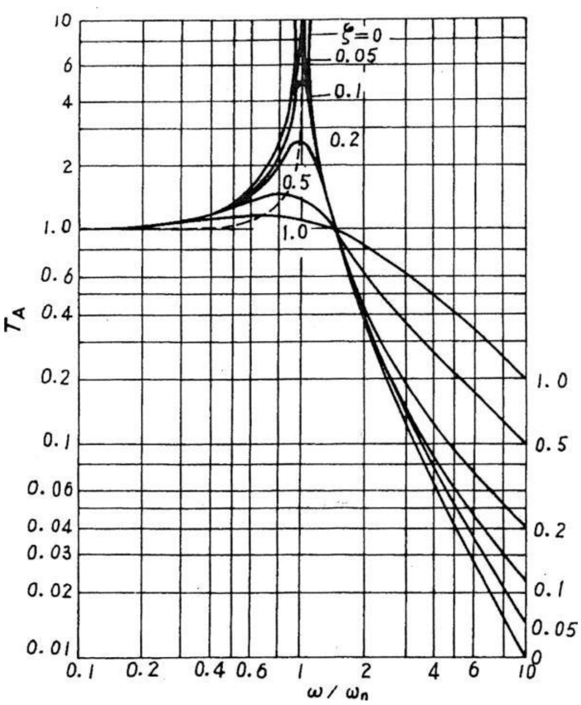


Figure 10.2.4 Curve of Absolute Transmission Rate T_A Varied with Frequency Ratio ω/ω_n

10.3 Mechanical Vibration Calculation

10.3.1 Introduction

In order to prevent the mechanical vibration exceeding criteria, mechanical vibration calculation is to be carried out and meet the requirements of design criteria.

Single-layer isolation system is normally used for ship's machinery, sometimes double-layer isolation system is also used. The calculation of single-layer isolation system design may be divided into single degree of freedom system with only one direction and multi degree of freedom system with six directions.

10.3.2 Preparation of isolation design information

In mechanical isolation design, only vertical vibration is normally taken into consideration, so it may be calculated according to single degree of freedom design. Prior to the isolation design, the

following information is to be prepared:
(1) type, specification and speed range of machinery;
(2) mass, position of centroid weight, installation position and scantling;
(3) structural characteristics and environmental conditions of installation base;
(4) requirements and indexes of ship's structural vibration and noise provided by shipowners.

10.3.3 Single degree of freedom isolation system
(1) Analysis of exciting force
First, the positive isolation or negative isolation is to be determined. In the case of a negative isolation, the dominant frequency amplitude and direction of vibration under the environment are to be analyzed. For the machinery, most are positive isolation, the frequency, amplitude and direction of main exciting force or exciting moment for machinery are to be calculated and analyzed. Where the machinery or vibration environment is a vibration source for several frequencies of excitation, excitation spectrum is to be drawn.
(2) Natural frequency of system
The natural frequency of system is to be determined by absolute transmission rate or isolation efficiency according to the design requirements. In general, the determination of natural frequency for isolation system is to take both isolation effect and swing stability of system into account at meanwhile. On the premise that the isolation efficiency is satisfied, the higher natural frequency may be designed to increase the stability of the system so as not to cause machinery swing due to lower natural frequency and soft isolator.
(3) Isolation design of pedestals
In order to reduce the vibration amplitude of isolated object and adjust the centroid of system, the machine is generally to be installed on an vibration isolation pedestal made of steel or reinforced concrete with sufficient stiffness and mass, then to support elastically on the ship. The vibration isolation pedestal is to:
① homogeneously force on vibration isolation element and control the vibration amplitude of the equipment;
② reduce the centroid of isolation system and raise its stability;
③ reduce coupled vibration induced by calculation error of equipment centroid position so that the system vibrates only along vertical direction as far as possible;
④ restrain the vibration amplitude as resonance speed.
(4) Determination of masses and the centroid positions for machine and vibration isolation pedestal
For an isolation system only considering vertical vibration, all masses of machinery and vibration isolation pedestal are to be obtained and the position of centroid is to be determined. Where the vertical (x), longitudinal (y) and horizontal (z) straight vibrations and whirling vibration around these three directions are required to consider at meanwhile, positions of three main inertia axes and moments of inertia around these three axes are to be calculated. The main inertia axes are to be located within the horizontal and vertical planes as far as possible through the adjustment of mass distribution of vibration isolation pedestal.
(5) Calculation of vibration amplitude
The vibration amplitude of machinery A is to be calculated by the following formula:

$$A = \frac{F_0}{\omega_n^2 m}\left|\frac{1}{1-(\omega/\omega_n)^2}\right| \tag{10.3.1}$$

where: F_0— amplitude of exciting force, in N; mainly for unbalanced force and moment of diesel engine;

m — total mass of machinery and vibration isolation pedestal, in kg;

ω — frequency of exciting force, in 1/s;

ω_n — natural frequency of vibration isolation system, in 1/s.

Where the calculated vibration amplitude A exceeds the permitted value of machinery equipment, m may be increased to reduce A, the mass of vibration isolation pedestal is generally to be increased. Where the exciting force is a moment and the moment of inertia I is used instead of m in formula (10.3.1), angular amplitude may be obtained.

(6) Selection of isolator

Stiffness, damping and performance of resistance to environmental conditions are mainly to be considered. In order to be installed and maintained conveniently, isolators with a same kind and type are to be used as far as possible.

(7) Arrangement of isolator

The isolators are to be arranged in compliance with the following principles:

① in an isolating device, isolators with a same type are to be selected as far as possible, to make an equal force acting on each isolator and a consistent deformation;

② isolators are to be arranged symmetrically along the main inertial axis;

③ when isolators with different types have to be used due to the special mechanical shapes and mass distribution, the deformation of each supporting point of isolators is to be consistent, to ensure that the vibration isolation system keeps vibration independence in vertical direction as vibration occurs;

④ in order to prevent the inconsistence of static compression for isolators due to calculation error, the installation positions of isolators may be designed as movable and adjustable in installation, to ensure the consistence of static compression for isolators.

(8) Selection of damping or limiter

Damping or amplitude limiter may be considered to provide in order to prevent swing of vibration isolation system or excessive amplitude as passing through resonance zone during starting.

(9) Flexible connection of other parts

All piping, power circuits and instrumentation wires of vibration isolation system are to be connected flexibly between the upper and lower parts of vibration isolation pedestal so as to reduce the vibration transmission of the second path.

(10) Inspection and plan comparison

After completion of the vibration isolation design, the mechanical vibration isolation system is to be checked for compliance with the design index, sometimes, several different plans are necessary to make comparison in order to meet the economic requirements. Meanwhile, compliance with the marine use requirements for all selected equipment or fittings are to be checked.

10.3.4 Multiple degree of freedom vibration isolation system

(1) Equation of motion for multiple degree of freedom vibration isolation system

When the machinery withstands exciting force and exciting moment from different directions simultaneously, the vibration isolation design is to be carried out in accordance with six degrees of freedom. See Figure 10.3.4, the mechanical and general vibration isolation pedestal are to be simplified to a rigid body with the mass m , the centroid position is the origin of coordinates, three coordinating axes are three main inertial ones, the equation of motion is as follows:

$$[M]\{\ddot{X}\}+[C]\{\dot{x}\}+[K]\{X\}=\{F\} \qquad (10.3.2)$$

where: $[M]$, $[C]$ and $[K]$— matrixes of mass, damping and stiffness respectively;

$\{X\}=[x,y,z,a,\beta,\gamma,]^T$ — displacement vector, x,y,z are linear displacements, a,β,γ are angular displacements;

$\{F\}=[F_x,F_y,F_Z,M_x,M_yM_z]^T$ — force vector, F_x,F_y,F_z are exciting forces, M_x,M_y,M_z are exciting moments.

The equation of natural frequency for vibration isolation system corresponding to equation (10.3.2) is as follows:

$$\left|[K]-[M]\omega^2\right|=0 \qquad (10.3.3)$$

(2) Response calculation for multiple degree of freedom vibration isolation system

The calculation of natural frequency, inherent mode shape and vibration response for multiple degree of freedom vibration isolation system is normally carried out by computer based on specialized program.

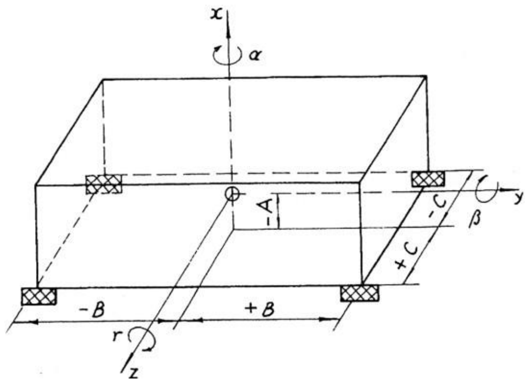


Figure 10.3.4 Multiple Degree of Freedom Vibration Isolation System

10.3.5 Double-tier vibration isolation system

(1) Introduction

In general, the level difference of single-tier vibration isolation system for ship’s machinery is 10dB～20dB. Therefore, double-tier vibration isolation system is to be used for the machinery onboard ships with higher vibration and noise requirements, i.e. an immediate seating is inserted between the isolated machinery and pedestal. The diagram of double-tier vibration isolation system is shown in Figure 10.3.5. In order to reduce air noise and the secondary structural vibration induced by it in the compartments, acoustic shield is to be installed to cover the engine, and generally to be located on the immediate seating. The level difference of double-tier vibration isolation system may be 30 dB to 40 dB in low frequency area and above 50dB in high frequency area.

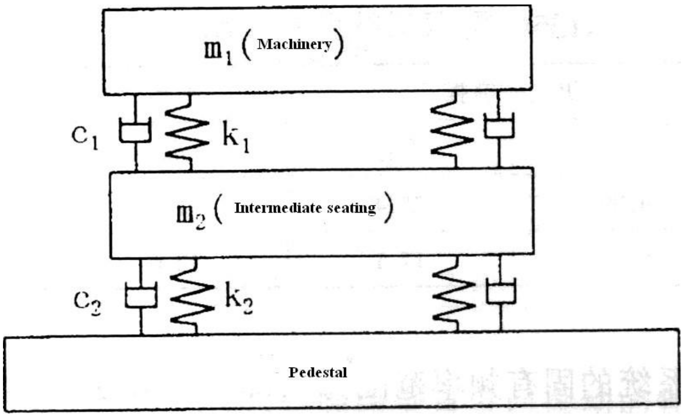


Figure 10.3.5 Diagram of Double-tier Vibration Isolation System

(2) Calculation of double-tier vibration isolation system

The double-tier vibration isolation system includes two parts such as machinery and intermediate seating, considering its non-rigidity, the natural frequency of intermediate seating is generally to be obtained by finite element method or testing mode method in order to avoid the resonance induced by coincidence with natural frequency. Normally, the intermediate seating is to be processed as rigidity, according to 10.3.3, it is known that such system has 12 degrees of freedom, i.e. six for machinery and six for intermediate seating. The equation of motion and formula of natural frequency are as same as those in 10.3.2 and 10.3.3, only the dimension and element in matrixes $[M]$, $[C]$ and $[K]$ are different, all of the calculation are to be carried out by specialized program.

Vertical vibration is usually concerned in the engineering projects, the double-tier vibration isolation system may be simplified to two degrees of freedom vibration system, its natural frequency without damping ω_n is to be as follows:

$$\omega_n^4 - [\frac{k_1}{m_1} + \frac{k_2}{m_2} + \frac{k_1}{m_2}]\omega_n^2 + \frac{k_1 k_2}{m_1 m_2} = 0 \tag{10.3.4}$$

The absolute transmission rate of double-tier vibration isolation system T_A is as follows:

$$T_A = \frac{F_{T0}}{F_0} = \left| \frac{(a^2 - 4\zeta_1\zeta_2 a\lambda^2) + j\lambda(2\zeta_1 a^2 + \zeta_2 a)}{A - jB} \right| \tag{10.3.5}$$

where: F_{T0} — force transmitted to the pedestal;

F_0 — exciting force;

$A = \lambda^4 - \lambda^2(a^2 + 4\zeta_1\zeta_2 a + \mu + 1) + a^2$;

$B = \lambda^3(2\zeta_2 a + 2\zeta_1\mu + 2\zeta_1) - \lambda(2\zeta_1 a^2 + 2\zeta_2 a)$;

$\mu = m_1 / m_2$;

$\omega_1 = \sqrt{k_1 / m_1}$;

$\omega_2 = \sqrt{k_2 / m_2}$;

$\lambda = \omega / \omega_1$;

$a = \omega_2 / \omega_1$;

$\zeta_1 = c_1 / 2\sqrt{k_1 m_1}$;

$\zeta_2 = c_2 / 2\sqrt{k_2 m_2}$;

$c_1 = a^2 - 4\zeta_1\zeta_2 a\lambda^2$;

$c_2 = \lambda(2\zeta_1 a^2 + 2\zeta_2 a)$.

when $\zeta_1 \ll 1, \zeta_2 \ll 1$ and $\frac{\omega}{\omega_1} \gg 1$:

$$T_A = \frac{\omega_1^2 \omega_2^2}{\omega^4} \tag{10.3.6}$$

The absolute transmission rate of double-tier vibration isolation system is inversely proportional to ω^4, hence, where exciting circular frequency is increased twice, the absolute transmission rate of double-tier vibration isolation system will be reduced 16 times, but the absolute transmission of single-tier vibration isolation system is only be reduced four times. This indicates the absolute transmission rate curve of double-tier vibration isolation system will be greatly dropped off after passing resonance peak, i.e. the effect of double-tier vibration isolation system is better than that of single-tier vibration isolation system in high frequency area.

10.4 Vibration Isolation Element

10.4.1 Performances of vibration isolation element

The vibration isolation element is generally divided into two categories such as vibration isolating pad and vibration isolator. The pad may be rubber vibration isolating pad, foam rubber, felt, glass fiber, mineral wool, etc. The isolator may be metal helical spring, rubber vibration isolator, wire rope isolator, air spring, etc. The comparison of performances for various vibration isolation elements is shown in Table 10.4.1.

It is to be pointed out that the rubber vibration isolation element has creep properties, i.e. the deformation of vibration isolation element is still increased in a period of time at the rated load, in general, hysteresis deformation in 48 h may achieve 90% of the creep. Therefore, vibration isolation devices of ship's machinery, such as essential equipment, i.e. main propulsion power equipment, diesel generators, etc., are to be aligned and installed with external devices after 48 h since the equipment is loaded on the vibration isolator, and general machinery vibration isolation devices are to be installed after 24 h.

Comparison of Performances for Various Vibration Isolation Elements Table 10.4.1

Performance	Metal helical spring	Stainless wire rope isolator	Rubber isolator/ isolating pad	Air spring	Wire mesh isolator	Foam rubber	Felt	Glass fiber and mineral wool
Applicable frequency (Hz)	2 ~ 10	5 ~ 10	5 ~ 100	0 ~ 5	20 ~ 25	2 ~ 5	25	>10
Multi-directional	○	▲	▲	○	○	○	○	○
Simplicity	○	▲	▲	△	○	○	○	○
Damping characteristic	×	▲	○	▲	○	△	△	○
High-frequency vibration isolation and sound isolation	×	○	○	▲	△	○	○	○
Linearity of load characteristic	△	○	○	○	×	×	×	×
High/low temperature resistance	▲	▲	△	△	▲	△	○	○
Oil resistance	▲	▲	△	△	▲	×	○	○

Aging resistance	▲	▲	△	△	▲	×	○	○
Uniformity of product quality	▲	○	△	○	○	×	△	△
Relaxation resistance	▲	○	○	○	△	△	△	○
Heat expansion resistance	▲	○	△	○	○	○	○	○
Price	Cheap	High	Middle	High	Middle	Middle	Cheap	Middle
Mass	Heavy	△	Middle	Heavy	Middle	Light	Light	Light
Consistence with calculation of characteristic values	▲	○	○	○	△	×	×	×
Difficulty level of design	▲	○	○	×	△	○	○	○
Difficulty level of installation	△	○	○	×	○	▲	▲	○
Service life	▲	▲	△	○	○	×	△	△

Note: ▲—— excellent; ○——good; △—— middle; ×—— poor.

10.4.2 Selection of vibration isolation element

According to the basic principle, the ratio between exciting frequency and natural frequency of vibration isolation system is to be more than $\sqrt{2}$, it is generally to be recommended as 2.5 to 4.5, for this purpose:

felt, rubber vibration isolating pad and some rigid rubber isolator, wire mesh isolator may be selected for the natural frequency f_0 = 20Hz ~ 30Hz;

metal spring, wire rope isolator, rubber isolator, foam rubber, foam plastic, etc., may be selected;

air spring isolator may be selected for the natural frequency f_0 = 2Hz ~ 10Hz for the natural frequency f_0 = 0.5Hz ~ 2Hz.

Another key point to select the vibration isolation element is the load, in general, the static load forced on the element is to be 80% to 90% allowable load, the sum of dynamic and static loads is not to exceed the maximum allowable load. For vibration isolating pad, the allowable load or recommended load are the load per unit area, and the load on each element is to be required homogeneously.

The characteristics and application of general vibration isolators for ship's machinery are listed in Table 10.4.2.

Characteristics and Application of General Vibration Isolators for Ship's Machinery

Table 10.4.2

Type	Characteristic	Application	Notice
Rubber vibration isolator	It has strong load bearing capacity and large rigidity, damping ratio is 0.05 ~ 0.15, may be made in various shapes, stiffness may be freely selected from three directions and has creeping effect	It is applicable to positive vibration isolation for main power equipment onboard ships and various machinery equipment	According to environmental condition, such as oil resistance, wear resistance, heat resistance, acid and alkali resistance,etc., different rubber materials are to be selected for vibration isolators
Stainless wire rope isolator	It has well elastic and damping and strong load bearing characteristic, good impact resistance, the horizontal and vertical rigidity are greatly different	It is applicable to positive vibration isolation for various equipment, negative vibration isolation for electronic instrument, as well as impacting environmental conditions	It is arranged in cross direction during installation so as that the rigidity of horizontal and vertical directions are closer

Metal spring	It has strong load bearing capacity, large deformation, small rigidity, with the damping ratio about 0.01, rigidity in horizontal direction is less than that in vertical direction, liable to swing	It is applicable to negative vibration isolation for instruments and positive vibration isolation for machinery with larger exciting forces	Where large damping is required, damper may be added or combined to use together with the rubber isolator
Air spring	The rigidity depends on the energy of compressed air, the damping ratio is 0.15 ~ 0.50	It is usually used for negative vibration isolation of precision instrument and equipment with special requirements, now used for positive vibration isolation of ship's main power equipment with special acoustics requirements	Air pressure is required to be stable, constant pressure air source is required
Foam rubber	It has small rigidity and high elasticity, damping ratio being 0.1 ~ 0.15, small load bearing capacity, unstable and liable to aging	It is used for negative vibration isolation of small instruments	The allowable stress is low, the relative deformation value is to be controlled in the range of 20%~35%, it is prohibited to be exposed in the sun and rain prevent contacting with acid, alkali and oil

10.4.3 Strength check for vibration isolation element

(1) The strength of spring element in metal spring vibration isolator is to be checked in accordance with the design method for mechanical parts.

(2) The allowable stress of rubber vibration isolator is not to exceed the following:

compression: 1 N/mm^2;

shearing: 0.4 N/mm^2.

(3) The allowable stress of rubber vibration isolator is not to exceed the following:

static load compression: 15%; shearing: 25%;

dynamic load compression: 5%; shearing: 10%.

(4) Due to ship's inclination and swing, the vibration isolation element will withstand the additional dynamic load in addition to the static load (weight of itself), hence, a dynamic strength check is to be carried out.

In general, only the maximum swing angle is taken into account and is to be treated as a static state. According to the rules provisions, the allowable maximum angle is as follows:

Trim is 7.5°and rolling is 22.5°for auxiliary engines related to ship's propulsion and safety;

Trim is 10°and rolling is 22.5°for emergency generator and equipment to be installed according to statutory requirements.

For small ships, inertia effect of machinery is to be taken into consideration due to the shorter swing period, as shown in Figure l0.4.3.

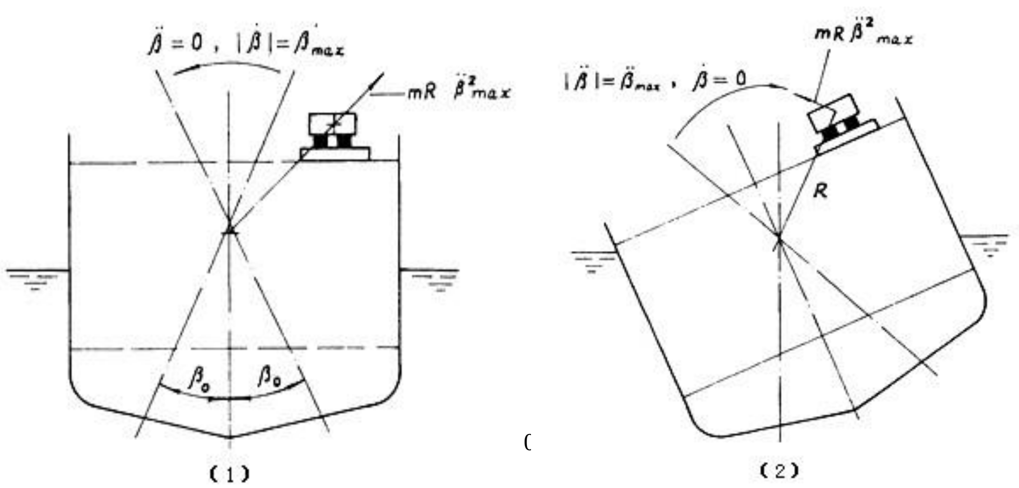


Figure 10.4.3 Ship's Dynamic Effect

Where m_0 is mass, swing angular displacement $\beta = \beta_0 \sin\omega t$. In (1) of the Figure shows the situations of β =0 and $|\beta| = \beta_{max}$ for swing angular displacement β =0; In (2) of the Figure shows the situations of $|\beta| = \beta_{max}$ and β =0 for swing angular displacement $\beta = \beta_0$. Therefore, the scantlings and directions of inertia forces are different, the dynamic strength check is to be carried out by considering several situations.

10.4.4 Arrangement of vibration isolation element

The arrangement mode of vibration isolation element is generally to be divided into supporting type and suspension type:

(1) Supporting type

According to the different supporting angles of vibration isolation element, it is mainly divided into flat type and oblique type.

The flat type is a usually and traditional arrangement mode with simple layout and easy installation. In such arrangement mode, three rigid axes of each vibration isolation element vertically each other are parallel to the corresponding selected referenced coordinating axes respectively.

In oblique type, three rigid axes of each vibration isolation element vertically each other are arranged corresponding to the referenced coordinating axes in such a way that one axis is parallel to the referenced coordinating axis while the both other two have a angle with the referenced coordinating axes respectively. In general, the oblique type vibration isolation elements are arranged at both sides of the vertical centerline in pairs, but the angle of each pair may be different. The most advantage of such arrangement is that it has either stronger lateral stiffness or sufficient rolling flexibility, usually used for vibration isolation of diesel engines, so as to both ensure greater lateral stability and meet the requirements of lower rolling natural frequency for isolating the rolling vibration induced by inhomogeneous torque.

(2) Suspension type

According to the difference of compartment spaces, it is divided into overhanging type and side mounting type. The former is usually used for vibration isolation of precision equipment while the latter is used for vibration isolation of ship's auxiliary engine with small power.

10.5 Piping Vibration

10.5.1 Introduction

The vibration and noise induced by mechanical vibration, pulsation and unstable flow of fluid in piping may cause severe piping vibration and fatigue damage of piping and its accessories, and also aggravate the vibration and noise around and influence the normal operation of associated machineries.

Therefore, appropriate means are to be taken to prevent the piping vibration at in ship's design and installation.

10.5.2 Analysis and calculation of piping vibration

The piping system is a continuous elastic structure and forced mechanical vibration will be induced by the excitation of fluctuation pressure of fluid in the pipes. In the design, analysis and calculation for natural frequency and mechanical vibration of piping system are recommended, to prevent resonance and estimate the vibration of piping system to determine the safety and reliability level of operation.

For simple piping system, such as single span piping with different supporting modes and mass distributions, the calculation method of natural frequency for cantilever rectangular pipe and plan bend at arbitrary angle, reference may be madeto vibration manual.

It is difficult to obtain the accurate values for actual piping system due to its complicated structures. Currently, the approximate value is generally to be obtained by various theory methods to meet the engineering calculation requirements. The finite element method is one the useful tool for piping system vibration analysis, the software may be used to calculate the natural frequency for complicated piping systems and hydrodynamics response.

10.5.3 Antivibration measures for piping

(1) Avoiding piping resonance

The natural frequency of piping vibration is related to the piping length between two supportings, the natural frequency becomes lower as the the piping is longer. Hence, different lengths may be used for changing the natural frequency of piping to avoid the occurrence of resonance.

(2) Bending and supporting of piping

The turning of piping is to be avoided as far as possible. If it is unavoidable, excessive spacing of supporting and suspension for piping are to be avoided, otherwise, bracing structure with sufficient stiffness is to be provided.

(3) Piping accessories

Accessories such as valves and instruments in piping are not to be installed in the loop positions of piping vibration, but are located adjacent to supporting points or fixed point as far as possible.

(4) Piping vibration isolation

In addition to the transmission along hull structures via the base, mechanical vibration may be transmitted to adjacent structures to radiate noises via piping, working medium in pipes and structural members of fixed piping. In order to isolate the transmission of piping vibration, the piping is to be isolated for vibration, which is generally to be realized through connection by elastic element between equipment and piping, i.e. elastic joint connection is to be used between fan, pump, air compressor, diesel engine inlet/outlet flanges and piping flanges, and the piping is to be supported by deck or bulkheads via spring hanger or bracing structure. Figure 10.5.3 shows the diagram of typical piping vibration isolation.

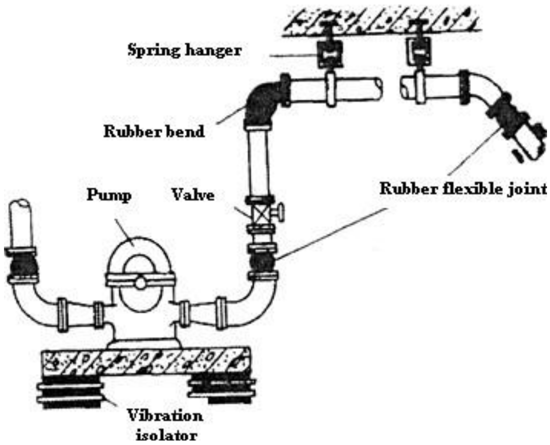


Figure 10.5.3 Diagram of Typical Piping Vibration Isolation

(5) Antivibration elastic element of piping

According to the different pressures and temperatures of machineries and fluid in piping, antivibration elastic elements, i.e. flexible joint, are divided into three tyeps as follows:

① flexible joint made of canvas or artificial leather applicable to ventilator piping vibration isolation with lower pressure in inlet/outlet;

② rubber flexible joint applicable to piping vibration isolation with working pressure less than $3N/mm^2$ and temperature of fluid medium below 100℃;

③ stainless steel corrugated metal pipe applicable to the working conditions with temperature higher than 100℃ and pressure higher than ambient pressure, such as exhaust vents of diesel engine, air compressor, vacuum pump, etc.

(6) Notice

In order to obtain a good effect of piping vibration isolation, the following notices are necessary in the design of flexible joint installation:

① the flexible joint is to be provided in both vertical and horizontal directions;

② the flexible joint is to be provided on the piping adjacent to the equipment, usually to be provided in way of inlet/outlet of equipment and piping;

③ the flexible joint can not withstand the axial tension, with less effect of lateral displacement compensation, therefore the limited rating of product is not to be exceeded during installation.

10.6 Design Criteria

10.6.1 Currently, the response calculation of mechanical vibration has not yet achieved the applicable level, hence, the essential factor of mechanical vibration control is to determine the exciting frequency and natural frequency of vibrating body and control them within a certain range. Resonance is to be avoided within the range of 85% to 100% maximum running speed.

10.6.2 In the positive vibration isolation, the ratio between exciting frequency and mechanical vibration natural frequency ω/ω_n is to meet the requirement of $\omega/\omega_n > \sqrt{2}$, in general ω/ω_n is equal to 2.5 ~ 4.5.

10.6.3 For medium or low-speed machinery, as the exciting frequency is very low, the natural frequency of system may be higher than certain exciting frequencies, which can not meet the requirement of $\omega / \omega_n > \sqrt{2}$, in such a case, means are to be taken to reduce the exciting force of related exciting frequency.

Chapter 11 OVERALL VIBRATION OF HULL GIRDERS

11.1 Introduction

11.1.1 Mode shape of hull girder vibration

The overall vibration of hull girders means the whole hull will cause vibration like a variable section beam type, which may have the following types:

(1) vertical bending vibration, see Figure 11.1.1-1;

(2) horizontal bending vibration, see Figure 11.1.1-1;

(3) torsional vibration, see Figure 11.1.1-2;

(4) longitudinal vibration, see Figure 11.1.1-3.

In general, the longitudinal vibration may not be taken into consideration, for ships with large openings, the coupling of horizontal vibration and torsional vibration is to be taken into account.

11.1.2 Main exciting force inducing hull girder vibration

The unbalanced moment of diesel engine will induce vertical overall vibration of hull girders, the relationship between exciting frequency and speed of diesel engine is:

the exciting frequency of 1st unbalanced moment being equal to the speed of diesel engine numerically;

the exciting frequency of 2nd unbalanced moment being twice the speed of diesel engine numerically;

the exciting frequency of 4th unbalanced moment being four times the speed of diesel engine numerically.

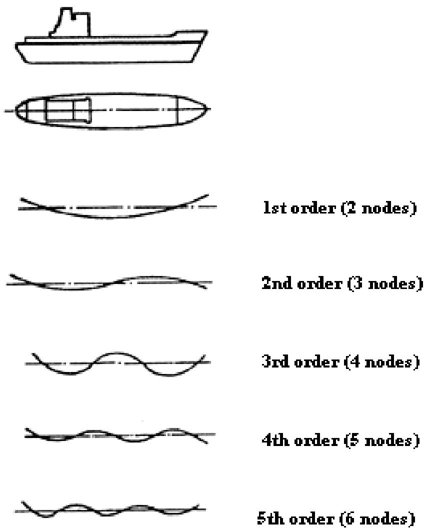


Figure 11.1.1-1 Mode Shapes of 1st to 5th Order Hull Girder Vertical and Horizontal Bending Vibrations

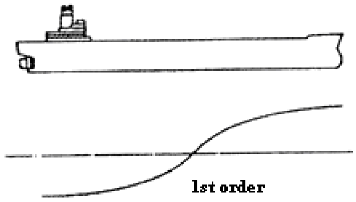


Figure 11.1.1-2 Mode Shape of 1st order Hull Girder Torsional Vibration

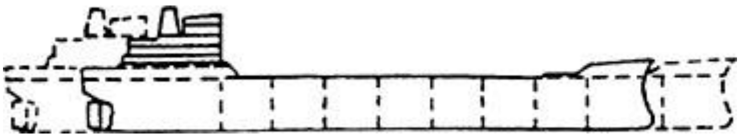


Figure 11.1.1-3 Mode Shape of Hull Girder Longitudinal Vibration

11.2 Estimation for Natural Frequency of Hull Girder Vibration

11.2.1 Estimation for natural frequency of vertical bending vibration

(1) Where the principle dimensions and displacement are known, the natural frequency f_{iv} of 1st order (2 node) and 2nd (3 node) hull girder vertical bending vibration may be calculated as follows:

$$f_{iv} = a_{iv} K_{iv} E_{iv} C_{vm} \frac{D}{L}\sqrt{\frac{B}{\Delta_v}} + b_{iv} \quad \text{Hz} \qquad (11.2.1)$$

where: i — number of nodes of hull girder vertical bending vibration, i =2 for 2 node vibration and i =3 for 3 node vibration;

f_{iv} — the natural frequency of hull girder vertical vibration with the number of nodes of i, in Hz;

D — molded depth, in m, the height measured from baseline to strengthen deck, for passenger-cargo ship, to be calculated to the upper deck;

L — length between perpendiculars, in m, applicable for ships of $L \leq 230$ m;

B — molded width, in m;

a_{iv}, b_{iv} — non-dimensional coefficient determined by ship's type and number of nodes;

K_{iv} — non-dimensional correction coefficient of effect on natural frequency due to the variation of moment of inertia of ship's transverse section to neutral axis along the ship length;

E_{iv} — non-dimensional correction coefficient of effect of bridge on natural frequency;

Δ_v — total mass of ship including mass of entrained water, in t;

C_{vm} — coefficient of effect of material types on hull vibration.

The above-mentioned coefficients may be determined by:

①a_{iv} and b_{iv} are to be determined by Table 11.2.1-1.

Coefficients of a_{iv} and b_{iv} **Table 11.2.1-1**

Number of nodes i	Coefficient	Oil tanker	Dry cargo ship	Bulk carrier	Ore carrier	Passenger-cargo ship
$i = 2$	a_{2v}	0.447×10^3	0.437×10^3	0.383×10^3	0.515×10^3	0.335×10^3
	b_{2v}	0.371	0.327	0.408	0.263	0.580

$i=3$	a_{3v}	1.290×10^3	0.690×10^3	0.775×10^3	1.123×10^3	0.710×10^3
	b_{3v}	0.238	1.208	0.782	0.442	0.842

② K_{iv}: $K_{2v}=0.90+0.10C_b$ for 2 node vibration (11.2.2)

$K_{3v}=0.85+0.15C_b$ for 3 node vibration (11.2.3)

where: C_b — block coefficient.

③ E_{iv}:

$$E_{2v}=\frac{D_e}{D} \quad \text{for 2 node vibration} \tag{11.2.4}$$

where: D_e is the equivalent molded depth, in m, calculated as follows:

$$D_e=\sqrt{D^2(1-x_1)+0.85kD_1^2(x_1-x_2)+0.67kD_2^2(x_2-x_3)+0.54kD_3^2x_3} \tag{11.2.5}$$

where: L_1, L_2, L_3 — lengths of bridge on each layer, in m, see Figure 11.2.1;

D_1, D_2, D_3 — the corresponding eights from ship's bottom to each top of the bridges, respectively, in m;

k— coefficient of different bridge types, k = 1.00 for the bridge being a superstructure, k = 0.95 for the bridge being a deckhouse;

x_1, x_2, x_3 are as follows respectively:

$$x_1=\frac{L_1}{L_,}, \quad x_2=\frac{L_2}{L}, \quad x_3=\frac{L_3}{L}$$

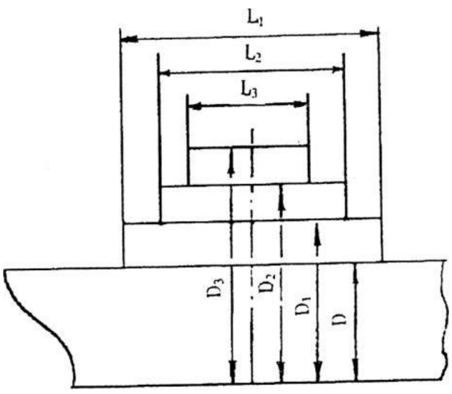


Figure 11.2.1 Diagram of Molded Depth and Dimension of Bridge

$E_{3v}=1.0$ for 3 node vibration.

④ $\Delta_v=\Delta(1+\tau)$ (11.2.6)

where: Δ — displacement, in t;

τ — coefficient of entrained water, to be determined by:

$$\tau=(0.2+\frac{B}{3d})(C_b^2+0.15) \tag{11.2.7}$$

where: d — mean draught, in m;

B — breadth of ship, in m;

C_b — block coefficient.

⑤ C_{vm}:

C_{vm} =1.0 for general steel used for hull;

C_{vm} is to be calculated by the following formulae, whichever is the greater, for high-strength steel being used for main hull structures within amidship areas:

$$C_{vm} = \sqrt{K_m} \qquad (11.2.8)$$

$$C_{vm} = 0.243\sqrt{L/D} \qquad (11.2.9)$$

C_{vm} is to be calculated by the following formulae, whichever is the greater, for high-strength steel being used for decks within the amidship areas:

$$C_{vm} = \sqrt{(1+K_m)/2} \qquad (11.2.10)$$

$$C_{vm} = \sqrt{\frac{0.059L/D}{2-0.059L/D}} \qquad (11.2.11)$$

The correction coefficient of materials is as follows:

$$K_m = 24f_a / R_{eH} \qquad (11.2.12)$$

where: R_{eH}— the indicated minimum upper yield point of high-strength steel, in kg/mm^2;

f_a — coefficient;

f_a = 1.056 for R_{eH} = 313.0 MPa;

f_a = 1.092 for R_{eH} = 325.8 MPa.

(2) Where the moment of inertia of midship section is known, the natural frequencies f_{iv} of 2 and 3 nodes of hull girder vertical bending vibration may be calculated by:

$$f_{iv} = A_{iv}K_{iv}E_{iv}\sqrt{\frac{I_{ov}}{\Delta_v L^3}} + B_{iv} \quad \text{Hz} \quad (i=2,3) \qquad (11.2.13)$$

where: I_{ov} — moment of inertia of midship section corresponding to horizontal axis, in m^4;

A_{iv},B_{iv} — coefficient determined by ship's type and number of nodes, obtained from Table 11.2.1-2.

Other symbols are same as 11.2.1(1).

Coefficients of A_{iv},B_{iv} **Table 11.2.1-2**

Number of nodes i	Coefficient	Oil tanker	Dry cargo ship	Bulk carrier	Ore carrier	Passenger-cargo ship
i =2	A_{2v}	0.530×10^5	0.423×10^5	0.340×10^5	0.432×10^5	0.357×10^5
	B_{2v}	0.162	0.337	0.455	0.337	0.537
i =3	A_{3v}	1.247×10^5	0.562×10^5	0.738×10^5	1.028×10^5	0.753×10^5
	B_{3v}	0.100	1.283	0.772	0.451	0.762

(2) Error and application of formulae: the formulae given in 11.2.1(1) and (2) are concluded according to the information of more than 100 ships with different types, and applicable to oil tankers, dry cargo ships, bulk carriers, ore carriers and passenger-cargo ships of less than 230 m in length, the error is generally not to be more than 7%.

(3) High-order vibration:

The natural frequency f_{iv} of hull girder vertical bending vibration above 3rd order is to be calculated by:

$$f_{iv} = a_{iv} f_{2v} \quad \text{Hz} \qquad (11.2.14)$$

where: a_{iv}— coefficient determined by number of nodes, obtained from Table 11.2.1-3;

i— number of nodes, $i \geq 4$;

f_{2v}— natural frequency of 1st order vibration, in Hz.

Coefficient of a_{iv} **Table 11.2.1-3**

Ship.1-3ype / Number of nodes i	Oil tanker	Ore carrier and bulk carrier	Cargo ship
i =4	3.07	3.00	2.53
i =5	4.11	4.00	3.23
i =6	5.16	5.00	3.90
i =7	6.22	6.00	4.55
i =8	7.28	7.00	5.18

Where the high-order vibration occurs, the distributions of mass and stiffness and shearing effects are increased, the accuracy of natural frequency calculated by the above-mentioned formulae will be decreased. Therefore, due caution is to be taken to estimate the natural frequency of high-order hull girder bending vibration in the absence of empirical data of similar ships.

11.2.2 Natural frequency of vertical bending vibration estimated by Kumai formula

Where the moment of inertia of midship section is known, the natural frequencies f_{2v} of 1st order of hull girder vertical bending vibration may be calculated by Kumai formula:

$$f_{2v} = 5.117 \times 10^4 \sqrt{\frac{I_{ov}}{\Delta_v L^3}} \quad \text{Hz} \qquad (11.2.15)$$

where: I_{ov}— moment of inertia of midship section corresponding to horizontal axis, in m^4;

L— length between perpendiculars, in m;

Δ_v — total mass of ship, including mass of entrained water, in t, calculated by:

$$\Delta_v = \left(1.2 + \frac{B}{3d}\right)\Delta$$

where: Δ—— displacement, in t;

B— molded breadth, in m;

d— mean draught, in m.

The natural frequency f_{iv} of hull girder vertical bending vibration more than 2nd order (3 node) may be calculated by:

$$f_{iv} \approx (i-1)^{\alpha} f_{2v} \quad \text{Hz} \qquad (11.2.16)$$

where: f_{2v}— natural frequency of 1st order (2 node) vibration, in Hz;

i— number of nodes, $i \geq 3$;

$\alpha = 0.845$ for general cargo ship;

$\alpha = 1.0$ for bulk carrier;

$\alpha = 1.02$ for oil tanker.

The relationship between natural frequency of 2 to 5 node vibration for general cargo ship and displacement shown in Figure 11.2.2-1 and that between natural frequency of 2 to 6 node vibration for oil tanker shown in Figure 11.2.2-2 may be combined with Kumai formula.

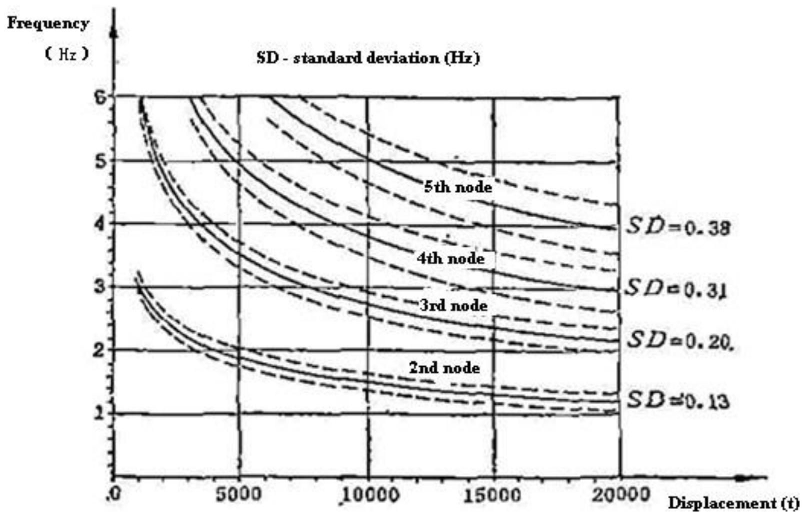


Figure 11.2.2-1 Relationship Between Natural Frequency of Hull Girder Vibration for General Cargo Ship and Displacement

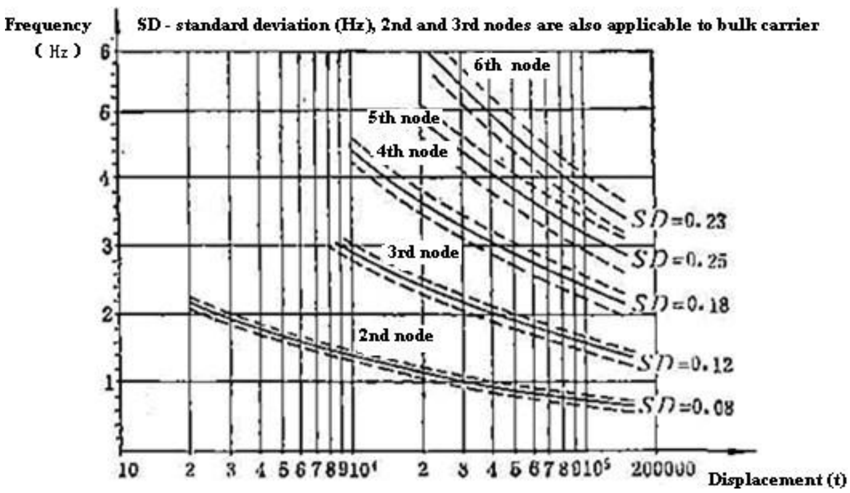


Figure 11.2.2-2 Relationship Between Natural Frequency of Hull Girder Vibration for Bulk Carrier/Oil Tanker and Displacement

11.2.3 Natural frequency of horizontal bending vibration

(1) Where the principle dimensions and displacement are known, the natural frequency f_{ih} of hull girder for each order horizontal bending vibration may be calculated as:

$$f_{ih} = a_{ih} K_{ih} E_{ih} C_{hm} \frac{B}{L} \sqrt{\frac{D}{\Delta_h}} + b_{ih} \quad \text{Hz} \qquad (11.2.17)$$

where: K_{ih}— non-dimensional correction coefficient of effect on natural frequency due to the variation of moment of inertia of ship's section to vertical axis along the ship length;

$a_{ih}, K_{ih}, E_{ih}, \Delta_h$ — same as the related parameters in 11.2.1, but change the suffix v to h in the formula to show the horizontal vibration;

C_{hm} — influence coefficient of steel types;

i — number of nodes;

L, B, D — same as 11.2.1.

The above related values may be determined as follows:

① a_{ih}, b_{ih}: obtained from Table 11.2.3-1.

Coefficient a_{ih}, b_{ih} **Table 11.2.3-1**

Number of nodes i	Coefficient	Oil tanker	Dry cargo ship	Bulk carrier
i =2	a_{2h}	0.547×10^3	0.542×10^3	0.493 ×10^3
	b_{2h}	0.373	0.120	0.290
i =3	a_{3h}	1.045×10^3	1.150×10^3	1.133×10^3
	b_{3h}	0.808	0.088	0.287

② K_{ih}: $K_{2h} = 0.90 + 0.10C_b$ for 2 node vibration (11.2.18)

$K_{3h} = 0.80 + 0.20C_b$ for 3 node vibration (11.2.19)

where: C_b — block coefficient.

③ E_{ih}: $E_{2h} = E_{3h} = 1.0$ (11.2.20)

④ $\Delta_h = \Delta + 0.57d^2LC_b + \frac{d}{2B}\Delta$ (11.2.21)

The other symbols are same as 11.2.1.

⑤ C_{hm}:

C_{hm} = 1.0 for general steel used for hull;

The influence coefficient C_{hm} of high strength steel may be calculated as follows for different ship's type when the high strength steel is used for hull:

(a) Cargo ship, bulk carrier and ore carrier:

$$C_{hm} = 9.28\sqrt{\frac{\sqrt{dK_m}}{L+115}}$$ for the transverse frame sides (11.2.22)

$$C_{hm} = 9.22\sqrt{\frac{\sqrt{dK_m}}{L+115}}$$ for the longitudinal frame sides (11.2.23)

(b) Oil tanker:

$$C_{hm} = 9.28\sqrt{\frac{\sqrt{dK_m}}{L+115}}$$ for the transverse frame sides (11.2.24)

$$C_{hm} = 8.80\sqrt{\frac{\sqrt{dK_m}}{L+115}}$$ for the longitudinal frame sides (11.2.25)

where: L — length of ship, in m;

d — mean draught for calculating the working conditions, in m;

K_m — conversion coefficient of material, obtained by formula (11.2.12).

(2) Where the moment of inertia of midship section is known, the 1st order and 2nd order natural frequencies f_{ih} of horizontal bending vibration for hull girder are to be calculated as follows:

$$f_{ih} = A_{ih} K_{ih} E_{ih} \sqrt{\frac{I_{oh}}{\Delta_h L^3}} + B_{ih} \quad \text{Hz} \quad (i = 2,3) \qquad (11.2.26)$$

where: I_{oh} — moment of inertia of midship section to vertical axis, in m^4;

L, Δ_h — same as 11.2.3(1);

$A_{ih}, B_{ih}, K_{ih}, E_{ih}$ — same as related parameters in 11.2.1(2), the suffix h in the formula indicates the horizontal direction, to be determined by the following respectively:

① K_{ih}: to be obtained by formulae (11.2.18) and (11.2.19);

② E_{ih}: to be obtained by formula (11.2.20);

③ A_{ih}, B_h: to be obtained from Table 11.2.3-2.

(3) Formula error and application: the above-mentioned formula is induced from information of more than 100 ships with different types and applies to oil tankers, dry cargo ships and bulk carriers of not more than 230 m in length, the error is generally not to be more than 7%.

Coefficients of A_{ih}, B_h **Table 11.2.3-2**

Number of node i	Coefficient	Oil tanker	Dry cargo ship	Bulk carrier
i =2	A_{2h}	0.473×10^5	0.468×10^5	0.417×10^5
	B_{2h}	0.362	0.175	0.473
i =3	A_{3h}	0.827×10^5	0.907×10^5	0.875×10^5
	B_{3h}	0.987	0.620	0.988

(4) High order vibration: the natural frequencies f_{ih} of hull horizontal bending vibration above 3rd order may be calculated by:

$$f_{ih} = a_{ih} f_{2h} \quad \text{Hz} \quad (i \geq 4) \qquad (11.2.27)$$

where: a_{ih}— coefficient determined by the number of node, to be obtained from Table 11.2.3-3.

Where the high order natural frequency of vibration is calculated by the methods specified in this Section, particular attention is to be given as same as those in 11.2.1(3).

Coefficient of a_{ih} **Table 11.2.3-3**

Ship's type / Number of node i	Oil tanker	Ore tanker
i =4	2.69	2.93
i =5	3.48	3.89
i =6	4.26	4.84
i =7	5.02	5.79
i =8	5.76	6.73

11.2.4 Natural frequency of torsional vibration

(1) Where the principle dimensions and displacement are known, the natural frequency of 1st order hull torsional vibration f_{1t} may be calculated by:

$$f_{1t} = 2.53\times10^4 \sqrt{\frac{B^2 D^2 t}{\Delta(B^2 + D^2)(B + D)L}} \quad \text{Hz} \qquad (11.2.28)$$

where: t— mean thickness of shell plating, in m;

B , D , L ,△— same as 11.2.1(1).

(2) Where the moment of inertia of free torsional pole for midship section is known, the natural frequency f_{1t} of 1st order torsional vibration for hull girder is to be calculated by the following formula:

$$f_{1t} = 0.141 \times 10^3 \sqrt{\frac{I_t}{\Delta (B^2 + D^2) L}} \quad \text{Hz} \qquad (11.2.29)$$

where: B , D , L ,△ —— same as 11.2.1(1);

I_t— moment of inertia of free torsional pole for midship section, in m^4, to be calculated by:

① For single-perimeter, the moment of inertia of free torsional pole I_t may be calculated by:

$$I_t = \frac{8F^2}{\sum_n \frac{S_n}{t_n}} \quad \text{mm}^2 \qquad (11.2.30)$$

where: F— half of the area surrounded by perimeter, in m^2;

S_n— length of each plating with the equivalent thickness, in m; due to the symmetric midship section, may be calculated only along the half section (see Figure 11.2.4);

n— number of plating;

t_n— thickness of each plating, in m.

② For double-perimeter, it is to be regarded as two perimeters, a public perimeter is between two single ones, thus, the moment of inertia of torsional pole I_t may be calculated by:

$$I_t = 8 \times \frac{F_1^2 A_2 + 2F_1 F_2 A_{1,2} + F_2^2 A_1}{A_1 A_2 + A_{1,2}^2} \qquad (11.2.31)$$

where: F_1, F_2— half of the area surrounded by each single-perimeter, in m^2;

A_1, A_2— sum obtained according to 1st perimeter and 2nd perimeter(including public perimeter) respectively, however, half section is to be used for calculation:

$$A_1 = \sum_n \frac{S_n}{t_n} \qquad (11.2.32)$$

$$A_2 = \sum_m \frac{S_m}{t_m} \qquad (11.2.33)$$

$A_{1,2}$— sum obtained according to the half public perimeter:

$$A_{1,2} = \sum \frac{S_k}{t_k} \qquad (11.2.34)$$

where: $S_m, t_m, S_n, t_n, S_k, t_k$, see Figure 11.2.4.

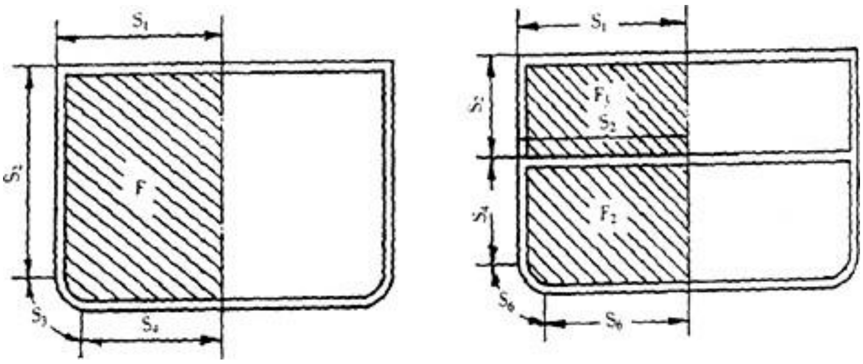


Figure 11.2.4 Single-perimeter and Double-perimeter

11.2.5 Comparative method with parent ships

(1) Based on the information of natural frequency of 1st order vertical or horizontal vibration for the existing parent ships, the natural frequency of hull girders for new design ships is to be calculated. For example, the following relationships are to be referred for the design ships and parent ships:

① same ship type;

② difference of ship length *L* not more than 10%;

③ similar loading condition.

(2) Where the hull structures, such as the frame type, numbers of continuous deck and longitudinal bulkheads are similar, the natural frequency f_n of hull girder vibration for a new design ship is to be calculated by the following formula:

$$f_n = f_o\sqrt{\frac{I_n}{I_o}\frac{\Delta_o L_o^3}{\Delta_n L_n^3}} \quad \text{Hz} \qquad (11.2.35)$$

where: I_f— natural frequency of hull girder vibration for parent ship, in Hz;

I_n, Δ_n, L_n — moment of inertia of midship section, in m^4, gross mass of hull containing entrained water, in t, and length of ship, in m, for new design ships respectively;

I_o, Δ_o, L_o — moment of inertia of midship section, in m^4, gross mass of hull containing entrained water, in t, and length of ship, in m, for the parent ship respectively;the gross mass of ship Δ_1 is to be calculated by:

$$\Delta_1 = (1.2 + \frac{B}{3d})\Delta \quad \text{t} \qquad (11.2.36)$$

where: Δ — ship's displacement, in t;

B— molded width, in m;

d— draught, in m.

11.3 Detailed Calculation for Natural Frequency of Hull Girders Vibration

11.3.1 Introduction

When detailed ship's particulars are available, the overall vibration of hull girder may be calculated by computer procedures with the most common used transfer matrix method and finite element method. The basic principle of transfer matrix method is to disperse the tapered hull girders into stepped ones consisting of several sections of uniformed girders with homogeneously distributed mass, then to carry out the vibration calculation by taking the boundary continuous condition for each section of girder and boundary condition for both ends of girder into account. The finite element method is to disperse the hull into plate or membrane element, girder element, mass element, etc., then to form stiffness matrix and mass matrix so as to carry out the vibration calculation.

11.3.2 Determination of basic parameters for hull girder

(1) The basic parameters of hull girder include the sectional moment of inertia, sectional shearing area, moment of inertia of sectional mass, hull girder mass and entrained water mass.

(2) Sectional moment of inertia and shearing area

The sectional moment of inertia and shearing area of hull girder are to be determined by the same method used for calculation of ship's longitudinal strength. Due to sectional moment of inertia having less influence on the minimum natural frequency, the sectional moment of inertia may be determined for distribution along the direction of ship's length as shown in Figure 11.3.2, at the earlier stage of design.

The effect of shearing area is to be taken into consideration because it has a great influence on high order natural frequency, however, the shearing area *A* of midship section may be taken for the one of all sections in the design.

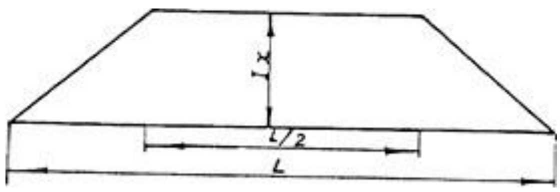


Figure 11.3.2 Sectional Moment of Inertia

(3) Moment of inertia of section mass

In general, the moment of inertia of section mass has less influence on natural frequency of hull girder vibration, and to be calculated by the following formula:

$$J_{iv} = \frac{m_{oi} D^2}{12} \quad \text{tm, for vertical vibration} \qquad (11.3.1)$$

$$J_{ih} = \frac{m_{oi} B^2}{12} \quad \text{for horizontal vibration} \qquad (11.3.2)$$

where: D— molded depth, in m;

B— molded width, in m;

m_{oi}— mass in the i^{th} unit length, in t.

(4) Hull girder mass and entrained water mass

When the hull girder is vibrating, the water around also moves, the movable water is equivalent to the mass of hull girder and is called entrained water mass, the sum of hull girder mass and entrained water mass is called as virtual mass. The entrained water mass has greater influence on vibration frequency of hull girder.

① Hull girder mass: may be calculated according to the ship's longitudinal strength calculation method

② Entrained water mass:

(a) Where the hull girder is vibrating vertically at sea, the entrained water mass $m_{av}(x)$ in unit length in way of each section may be calculated as follows:

$$m_{av}(x) = \frac{1}{2} a_v K_{vi} C_v \rho \pi b_w^2 \quad \text{t/m} \qquad (11.3.3)$$

where: a_v—shallow water correction coefficient, the shallow water effect may increase the entrained water mass of vertical vibration, so the coefficient is greater than 1. It is related to the ratio of H_d / b_w, obtained from Table 11.3.1-1:

Shallow Water Correction Coefficient of a_v and Narrow Channel Correction Coefficient a_h

Table 11.3.1-1

H_d / b_w or H_y / d_w	1.50	2.00	2.50	3.00	3.50	4.00	4.50	5.00	5.50
a_v	1.53	1.36	1.25	1.14	1.07	1.01	1.00	1.00	1.00
a_h	1.38	1.28	1.20	1.14	1.09	1.06	1.03	1.01	1.00

K_v— three dimensional flow coefficient, to be determined by aspect ratio of *L/B* and number of vibration nodes *i*, which is obtained from Table 11.3.1-2;

C_v— the entrained water mass coefficient determined by the shape of calculated section and the ratio between width and draught may be obtained from Table 11.3.1-3.

Three-dimensional flow coefficient of K_{vi},K_{hi} **Table 11.3.1-2**

L/B or L/d	1st order	2nd order	3rd order	4th order	5th order
5.0	0.700	0.624	0.551	0.494	0.477
6.0	0.748	0.678	0.614	0.560	0.515
7.0	0.786	0.719	0.661	0.611	0.568
8.0	0.815	0.756	0.898	0.653	0.611
9.0	0.839	0.784	0.733	0.687	0.647
10.0	0.858	0.808	0.759	0.716	0.677
11.0	0.874	0.828	0.782	0.742	0.706
12.0	0.888	0.845	0.802	0.763	0.730
13.0	0.890	0.859	0.820	0.783	0.751
14.0	0.909	0.870	0.835	0.803	0.770
15.0	0.917	0.883	0.848	0.818	0.788
20.0	0.947	0.920	0.895	0.876	0.857
25.0	0.968	0.944	0.925	0.909	0.894
30.0	0.980	0.958	0.940	0.924	0.910
35.0	0.987	0.967	0.950	0.934	0.922

Coefficient of Entrained Water for Vertical Vibration C_v **Table 11.3.1-3**

b_w / d_w / β	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0

0.0	1.510	1.100	0.935	0.860	0.815	0.785	0.760	0.755	0.750	0.750
0.1	1.250	0.975	0.860	0.800	0.775	0.765	0.755	0.752	0.753	0.753
0.2	1.060	0.880	0.805	0.764	0.750	0.750	0.750	0.750	0.750	0.752
0.3	0.815	0.815	0.760	0.750	0.750	0.755	0.760	0.770	0.775	0.790
0.4	0.800	0.740	0.750	0.750	0.765	0.770	0.775	0.780	0.800	0.801
0.5	0.740	0.760	0.765	0.774	0.785	0.790	0.800	0.816	0.825	0.831
0.6	0.700	0.788	0.802	0.815	0.830	0.842	0.852	0.865	0.875	0.880
0.7	0.860	0.880	0.895	0.905	0.915	0.925	0.925	0.933	0.940	0.942
0.8	1.035	1.035	1.032	1.030	1.025	1.020	1.020	1.018	1.015	1.010
0.9	1.320	1.270	1.240	1.200	1.185	1.162	1.150	1.130	1.120	1.115
1.0	1.980	1.760	1.640	1.570	1.518	1.472	1.434	1.400	1.375	1.355

where: β — area coefficient of the calculated section:

$$\beta = \frac{F_w}{b_w d_w} \qquad (11.3.4)$$

where: F_w — half area under waterline of the calculated section, in m^2;

b_w, d_w — half width and draught in way of waterline of the calculated section, in m.

(b) When the hull girder is vibrating horizontally, the entrained water mass m_{ah} may be calculated by the following formula:

$$m_{ah}(x) = \frac{2}{\pi} a_h K_{hi} C_h \rho d_w^2 \quad \text{t/m} \qquad (11.3.5)$$

where: a_h — narrow channel correction coefficient, obtained from Table 11.3.1-1, however, the values in the first line of the Table are to be determined by the ratio between distance from centerline to shore and draught H_y/d_w;

K_{hi} — three-dimensional flow correction coefficient, to be obtained from Table 11.3.1-2 according to the aspect ratio L/d and order number i;

C_h — entrained water mass coefficient of horizontal vibration, to be obtained from Table 11.3.1-4;

β — see formula 11.3.4.

Entrained Water mass Coefficient of Horizontal Vibration C_h Table 11.3.1-4

β \ b_w/d_w	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
0.0	1.108	1.271	1.406	1.554	1.707	1.863	2.011	2.152	2.295	2.430
0.1	1.083	1.197	1.327	1.440	1.554	1.678	1.791	1.912	2.036	2.134
0.2	1.061	1.160	1.270	1.352	1.431	1.530	1.606	1.683	1.764	1.851
0.3	1.049	1.123	1.184	1.263	1.308	1.387	1.436	1.505	1.554	1.616
0.4	1.024	1.073	1.123	1.172	1.209	1.263	1.295	1.332	1.382	1.419
0.5	1.017	1.049	1.061	1.091	1.123	1.147	1.172	1.191	1.221	1.246
0.6	1.012	1.024	1.036	1.061	1.061	1.073	1.086	1.086	1.098	1.110
0.7	1.007	1.012	1.012	1.036	1.036	1.036	1.036	1.036	1.036	1.036
0.8	0.997	0.997	1.004	1.009	1.009	1.012	1.012	1.012	1.024	1.026
0.9	1.002	1.002	1.012	1.036	1.049	1.049	1.049	1.061	1.061	1.061
1.0	1.049	1.073	1.098	1.110	1.123	1.140	1.160	1.165	1.184	1.197

11.3.3 Natural frequency of hull girder vibration calculated by transfer matrix method

The calculus equation of free vibration for homogeneously distributed girders which have been taken the shearing and section rotary influences into consideration is as follows:

$$EI\frac{\partial^4 W(x,t)}{\partial^4 x}+m\frac{\partial^2 W(x,t)}{\partial^2 t}-(J+\frac{E\,\mathrm{Im}}{GA_s})\frac{\partial^2 W(x,t)}{\partial^2 x\partial^2 t}+\frac{Jm}{GA_s}\frac{\partial^4 W(x,t)}{\partial^4 t}=0 \qquad (11.3.6)$$

Assuming that the free vibration displacement $W(x,t)$ is:

$$W(x,t)=W(x)\sin(\omega t+a) \qquad (11.3.7)$$

where: ω—— natural circular frequency, in rad / s;

a — phase angle, in rad;

I — sectional moment of inertia, in m^4;

m — mass in unit length, in t / m;

J — mass moment of inertia in unit length, in tm;

A_s — shearing area, in m^2.

Substituting the formula (11.3.7) to (11.3.6) to obtain:

$$\frac{d^4W(x)}{dx^4}+\frac{m\omega^2}{EI}(\frac{EI}{GA_s}+\frac{J}{m})\frac{d^2W(x)}{dx^2}-\frac{m\omega^2}{EI}(1-\frac{J\omega^2}{GA_s})W(x)=0 \qquad (11.3.8)$$

Setting the following:

$$\sigma=\frac{m\omega^2}{GA_s}l^2;\quad \tau=\frac{J\omega^2 l^2}{EI};\quad \beta^4=\frac{m\omega^2 l^4}{EI};\quad l\text{ — length of girder;}$$

Formula (11.3.8) may be:

$$\frac{d^4W(x)}{d^4x}-\frac{\sigma+\tau}{l^2}\frac{d^2W(x)}{d^2t}-\frac{\beta^4-\sigma\tau}{l^4}W(x)=0 \qquad (11.3.9)$$

The formula (11.3.9) is a constant coefficient homogeneous equation for mode shape $W(x)$, which may be obtained by:

$$W(x)=A\,ch\lambda_1 x+B\,sh\lambda_1 x+C\cos\lambda_2 x+D\sin\lambda_2 x \qquad (11.3.10)$$

where: $\lambda_{1,2}=\sqrt{\sqrt{\beta^4+\frac{(\sigma+\tau)^2}{4}}\pm\frac{\sigma+\tau}{2}}$

The displacement W, corner θ, bending moment M and shearing force N of girder in any section may be obtained by x differential of formula (11.3.10). The relationship of displacement, corner, bending moment and shearing force at both ends of girder may be expressed by:

$$\begin{bmatrix}W\\\theta\\M\\N\end{bmatrix}_R=\begin{bmatrix}F_{11}&F_{12}&F_{13}&F_{14}\\F_{21}&F_{22}&F_{23}&F_{24}\\F_{31}&F_{32}&F_{33}&F_{34}\\F_{41}&F_{42}&F_{43}&F_{44}\end{bmatrix}\begin{bmatrix}W\\\theta\\M\\N\end{bmatrix}_L \quad \text{or} \quad B_R=FB_L \qquad (11.3.11)$$

where: $B_R = \begin{bmatrix} W \\ \theta \\ M \\ N \end{bmatrix}_R$ —— state vector at the right end;

$B_L = \begin{bmatrix} W \\ \theta \\ M \\ N \end{bmatrix}_L$ —— state vector at the left end;

F —— span matrix, to be expressed by:

$$F = \begin{bmatrix} c_0 - \sigma c_2 & l[c_1 - (\sigma + \tau)c_3] & ac_2 & al[-\sigma c_1 + (\beta^4 + \sigma^2)c_3]/\beta^4 \\ \beta^4 c_3 / l & c_0 - \tau c_2 & a(c_1 - \tau c_3)/l & ac_2 \\ \beta^4 c_2 / a & l/a[-\tau c_1 + (\beta^4 + \sigma^2)c_3] & c_0 - \tau c_2 & l[c_1 - (\sigma + \tau)c_3] \\ \beta^4(c_1 - \sigma c_3)/al & \beta^4 c_2 / a & \beta^4 c_3 / l & c_0 - \sigma c_2 \end{bmatrix}$$

(11.3.12)

where: $c_0 = \lambda[\lambda_2^2 \frac{e^{\lambda_1} + e^{-\lambda_1}}{2} + \lambda_1^2 \cos\lambda_2]$;

$c_1 = \lambda[\frac{\lambda_2^2}{\lambda_1} \frac{e^{\lambda_1} - e^{-\lambda_1}}{2} + \frac{\lambda_1^2}{\lambda_2} \sin\lambda_2]$;

$c_2 = \lambda[\frac{e^{\lambda_1} + e^{-\lambda_1}}{2} - \cos\lambda_2]$;

$c_3 = \lambda[\frac{1}{\lambda_1} \frac{e^{\lambda_1} - e^{-\lambda_1}}{2} - \frac{1}{\lambda_2} \sin\lambda_2]$;

$a = \frac{l^2}{EI}$; $\lambda = \frac{1}{\lambda_1^2 + \lambda_2^2}$.

Due to the hull girder being the one with variable cross-section and the mass in unit length (including masses of hull girder, cargo and entrained water) changeable, it may be similarly regarded that the hull girder consists of several sections with equivalent cross-section and homogeneously distributed mass and the state vectors of each section at both left and right ends have the same relationship as formula (11.3.1). If the hull girder with two free ends is divided into n sections, the state vector at the left free end may be transferred to that at the right free end section by section, their relationship may be obtained as follows:

$$B_n = F_n F_{n-1} \ldots\ldots F_{i\ldots\ldots\ldots} F_2 F_1 B_0 \qquad (11.3.13)$$

where: F_n, F_{n-1}, …, F_i, …, F_1 — indicating the span matrix of each section;

B_n, B_0 — indicating right and left free ends of hull girder and state vectors.

Or changing the formula (11.3.13) to $B_n = \pi B_0$, i.e.:

$$\begin{bmatrix} W \\ \theta \\ M \\ N \end{bmatrix}_n = \begin{bmatrix} \pi_{11} & \pi_{12} & \pi_{13} & \pi_{14} \\ \pi_{21} & \pi_{22} & \pi_{23} & \pi_{24} \\ \pi_{31} & \pi_{32} & \pi_{33} & \pi_{34} \\ \pi_{41} & \pi_{42} & \pi_{43} & \pi_{44} \end{bmatrix} \begin{bmatrix} W \\ \theta \\ M \\ N \end{bmatrix}_0 \tag{11.3.14}$$

According to the boundary conditions of left and right ends of hull girder:

$$M_0 = N_0 = 0\text{；} \quad M_n = N_n = 0 \tag{11.3.15}$$

Substituting formula（11.3.15）in formula（11.3.14）to obtain:

$$\left.\begin{aligned} \pi_{31}W_0 - \pi_{32}\theta_0 = 0 \\ \pi_{41}W_0 + \pi_{42}\theta_0 = 0 \end{aligned}\right\} \tag{11.3.16}$$

In order to obtain the zero solution for W_0, θ_0, it must be:

$$\begin{vmatrix} \pi_{31} & \pi_{32} \\ \pi_{41} & \pi_{42} \end{vmatrix} = 0 \tag{11.3.17}$$

The formula (11.3.17) is the frequency equation. If the root ω_i of the equation is met, the i^{th} order natural frequency of ship hull vibration is obtained. After the ω_i has been got, the state vectors of end for each section of hull girder to the right free end (stem) may be obtained through formula (11.3.12), assuming that the displacement of stern vibration W_0 is the unit displacement, the state vector is to be:

$$B_0 = [1 \quad \frac{-\pi_{31}}{\pi_{32}} \quad 0 \quad 0]^T \tag{11.3.18}$$

Obtaining through formula (11.3.13):

$$\left.\begin{aligned} & B_1 = F_1 B_0 \\ & B_2 = F_2 B_1 = F_2 F_1 B_0 \\ & \cdots\cdots\cdots \\ & B_i = F_i B_{i-1} = F_i F_{i-1} \cdots\cdots F_2 F_1 B_0 \\ & \cdots\cdots\cdots \\ & B_n = F_n B_{n-1} = F_n F_{n-1} \cdots\cdots F_i F_{i-1} \cdots\cdots F_2 F_1 B_0 = \pi B_0 \end{aligned}\right\} \tag{11.3.19}$$

Then, the i^{th} order inherent mode shape curve $W_i(x)$ of the hull girder.

11.3.4 Natural frequency of hull girder vibration calculated by finite element method

(1) Free vibration equation

When the analysis is being carried out by finite element method, it is to disperse the hull structure into several units (such as rod element, girder element, membrane element or plate element), then to form a calculation model through the connection in way of nodes. The unit stiffness matrix is to be calculated on this basis and assembled to compose the stiffness matrix for the whole hull structure, at meanwhile, the corresponding method (such as concentrated mass method or consistent mass method) is to be used to compose the mass matrix. The effect of damping may not be taken into account for the calculation of natural frequency of hull structure, thus to disperse the vibration issue into free vibration equation of multi degree of freedom system:

$$[M]\{\ddot{q}\}+[K]\{q\}=0 \quad (11.3.20)$$

where: $[M]$, $[K]$— mass of hull structure (including mass of entrained water) and stiffness matrix;

$\{q\}$— displacement vector;

$\{\ddot{q}\}$— acceleration vector.

Assuming that $q=\varphi \sin \omega t$, the free vibration issue is to come down the one to solve the generalized characteristic value of the following equation:

$$[K]\{\varphi\}=\omega^2[M]\{\varphi\} \quad (11.3.21)$$

The natural frequency of vibration is ω_i (i=1, 2, …, n), the corresponding mode shape is:

$$\{\varphi\}=\{\varphi\}_i \ (i=1,2,\ldots,n) \quad (11.3.22)$$

where: ω_i is the natural frequency of hull girder vibration, $\{\varphi\}_i$ is the corresponding inherent mode shape, *n* is the freedom degree of discrete hull structure. The finite element vibration analysis of hull structure is to be realized by special or common computer procedures.

(2) Calculation model

The three-dimensional finite element model of ship hull vibration is to include all of the structural members within the whole range of length, width and depth, the node of finite element model, choice of element type and element properties are to reflect the structural stiffness and mass characteristics. The membrane (or plate), rod and girder are to be used to stimulate the whole hull structure while the node mass or mass element is to be used to stimulate the weight of equipment or effect of entrained water. Stiffness provided by element on three linear displacement directions of each node is to be ensured in order to prevent the low-frequency local mode shape. The three-dimensional model may be used to calculate the vertical, horizontal and torsional vibrations and analyze the local vibration (such as double bottom, deckhouse, etc.), and also to analyze the coupling effects of various mode shapes.

The mass of entrained water may be either determined according to 11.3.2(4) or calculated according to the flow finite element or the other method for calculating the coupling effect of flow and structure, but the working load of data preparation may be greater.

The lightship weight distribution has greater influence on vibration analysis results. Generally, mass element is to be used to stimulate the large equipment onboard ships, to be distributed on the corresponding nodes to ensure the gross weight of equipment and the center of gravity are kept consistent with the actual structure. In general, the error between the center of gravity of finite element model weight and stability information data is to be within the rage of 0.5% of the ship’s length. The longitudinal, transverse and vertical weight distributions of deckhouses and superstructures are to be accurately described. The cargo, ballast water and fuel oil within the compartments/tanks may also be indicated by mass element on the boundaries.

(3) Loading condition

The purpose of hull vibration analysis is to investigate the vibration characteristics of scheduled loading conditions, thus the ship’s service conditions such as full-loaded or ballasted conditions are to be evaluated. The other different loading conditions may be required to calculate by the designers or shipowners.

(4) Evaluation of calculation

For a free hull vibration analysis, the calculation results are to be checked from the natural frequency and inherent mode shape:

① The mode shapes of fore 1st to 6th order vibrations are to be rigidity displacement for pitching, rolling, stem swaying, surging, heaving and swaying, respectively. The vibration frequency of rigidity displacement is usually lower than the 1st order vibration frequency. Normally, the local vibration mode of merchant ships is not lower than 4 Hz. For the excessive low local vibration mode, the correctness of finite element model is to be checked.

② The correctness of finite element vibration analysis results may be evaluated according to the evaluation results of vibration frequency for hull girders carried out at the different design phases.

(5) Calculation cases

The three-dimensional finite element model of a certain large bulk carrier is shown in Figure 11.3.3, the fore 1st and 2nd order vertical mode shapes of hull girder under homogeneous loading condition are shown in Figure 11.3.4 and Figure 11.3.5, the fore 1st and 2nd order horizontal mode shapes are shown in Figure 11.3.6 and Figure 11.3.7.

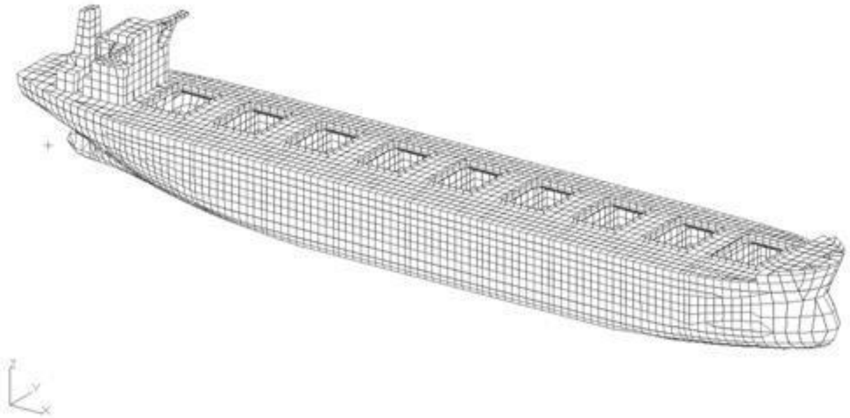


Figure 11.3.3 Three-dimensional Finite Element Model

Figure 11.3.4 Mode Shape of 1st Order Vertical Vibration for a Certain Large Bulk Carrier (f=0.462 Hz)

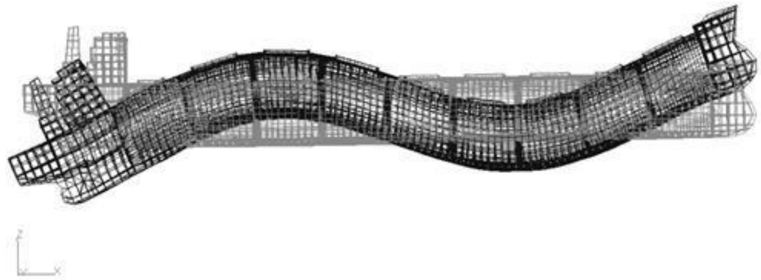


Figure 11.3.5 Mode Shape of 2nd Order Vertical Vibration for a Certain Large Bulk Carrier (f=0.943 Hz)

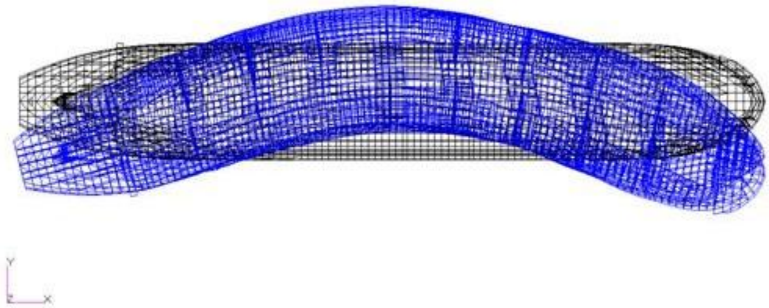


Figure 11.3.6 Mode Shape of 1st Order Horizontal Vibration for a Certain Large Bulk Carrier (f=0.729 Hz)

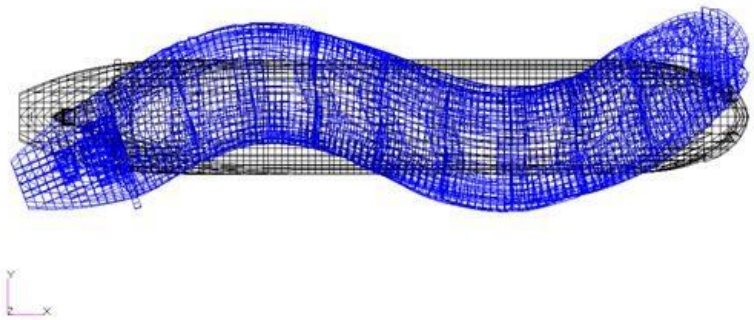


Figure 11.3.7 Mode Shape of 2nd Order Horizontal Vibration for a Certain Large Bulk Carrier (f=1.444 Hz)

11.4 Response Calculation for Overall Vibration of Hull Girders

11.4.1 Purpose

The calculation of response calculation for overall vibration of hull girders is to be carried out for the purpose to obtain the vibration response of hull under the main exciting force so as to understand the ship's vibration characteristics in the design, and if necessary, to take the corresponding vibration damping measures. The vibration response evaluation may be carried out either to compare the calculation results with ISO 6954(2000) or to determine the weighted effective values within the whole frequency range based on the vibration amplitudes of different frequencies according to ISO 6954(1984), see Chapter 15 of the Guidelines for details.

11.4.2 Response calculation for overall vibration of hull girders

After the natural frequency ω_i of hull girder vibration and its corresponding inherent mode shape are obtained, the vibration response of hull girder to the external excitation may be obtained by mode superposition method.

As mentioned above, hull girder is dispensed into homogeneous sections with limited numbers as to obtain several low order natural frequency and inherent mode shape, due to damping of hull girder being increased with the increase of order number, only sufficient precision is taken into consideration for the contribution of first few orders mode shapes in the calculation of forced vibration response. To ensure the orthogonality of mode shape, the entrained water mass of corresponding orders for exciting frequency to be approximate to the natural frequency is to be taken as the common entrained water mass for various orders, however, its precision for calculation is maintained, because the main contribution of vibration response comes from the response of the closest order. The response of hull girder to harmonic excitation may be as follows:

$$W(x,t)=\sum_{i=1}^{n}W(x)_i\frac{F_{e(i)}}{N_{e(i)}}\frac{1}{\sqrt{(1-r_i^{\,2})^2+4r_i^{\,2}\zeta_i^2}}\sin(\omega t-\varphi_i) \qquad (11.4.1)$$

where: $W(x)_i$ — mode shape of i^{th} order vibration;

$r_i=\dfrac{\omega}{\omega_i}$;

$\zeta_i=\dfrac{C_i}{2M_{e(i)}\omega_i}$;

ω — exciting frequency, in rad/s;

ω_i — natural circular frequency, in rad/s;

$\varphi_i=tg^{-1}\dfrac{2\zeta_i r_i}{1-r_i^2}$;

C_i — damping coefficient corresponding to i^{th} order vibration is to be generally determined by full-scale ship, to be calculated according to test data of the same ship type or the following formula at the design phase:

$$C_i=0.5M_{e(i)}\omega_i^{2.45}10^{-3} \qquad (11.4.2)$$

$M_{e(i)}, N_{e(i)}, F_{e(i)}$ are the equivalent mass, equivalent stiffness and equivalent force of i^{th} order vibration, respectively, may be calculated by:

$$M_{e(i)}=\int_o^l m(x)W_i^2(x)dx \qquad (11.4.3)$$

$$N_{e(i)}=\int_o^l EI(x)W_i''^2(x)dx \qquad (11.4.4)$$

If the excitation in way of $x=c$ is the harmonic force $P(t)=P_0\sin\omega t$,

$$F_{e(i)}=P_0W_i(c)\sin\omega t \qquad (11.4.5)$$

If the excitation in way of $x=c$ is the harmonic moment $M(t)=M_0 \sin \omega t$,

$$F_{e(i)}=M_0 W_i'(c)\sin \omega t \tag{11.4.6}$$

As seen from the formula (11.4.5) and formula (11.4.6), where the exciting force or exciting moment is located in way of node or wave loop of harmonic mode shape of i^{th} order vibration, i.e. $W_i(c)=0$ or $W_i'(c)=0$, the i^{th} item of vibration response in formula (11.4.1) is equal to 0. Where the exciting frequency resonates with i^{th} order natural frequency, the proper position of excitation source, such as main engine to be arranged according to the above-mentioned principles, may greatly reduce the vibration.

11.4.3 Response calculation of overall vibration for hull girder three-dimensional finite element

(1) Calculation method

For an equation of motion of forced vibration calculation for multi degree freedom system, the difference between calculations is time domain or frequency domain. In the hull structure, most of the time domain method is limited to use for vibration attenuation analysis under transient excitation, such as slamming flutter.

Mode shape superposition method is usually used for forced vibration response calculation. The first step of the method is to determine the natural frequency and inherent mode shape of vibration within a certain frequency range. The inherent mode shape is to be converted to orthogonal coordinate system, which the process involves the decoupling of all freedom degree in the equation of motion. Due to the deduction of workload for equation solving, the vibration response of large structural system may be calculated within the broadband range.

In addition to the formula of ship hull vibration response induced by hull girder theory and main coordinate method in 11.4.2, the response of ship hull vibration may be calculated for multi-degree of freedom system by finite element method.

$$[M]\{\ddot{q}\}+[C]\{\dot{q}\}+[K]\{q\}=\{F(t)\} \tag{11.4.7}$$

where: $[M],[C],[K]$ — mass, damping and stiffness matrix for hull structure;

$\{q\},\{\dot{q}\},\{\ddot{q}\}$ — vectors of displacement, velocity and acceleration;

$\{F(t)\}$ — vector of hull exciting force.

(2) Excitation source

The main excitation source of ship's steady state vibration comes from propeller and main diesel engine. The fluctuation pressure of propeller may be calculated according to the requirements of Chapter 3 or may be obtained from model testing. The unbalanced moment may be calculated according to the requirements of Chapter 4. In general, the ship hull vibration is mainly induced by the unbalanced moment (1st order, 2nd order and 4th order) of diesel engine.

The exciting frequency of unbalanced moment = order × speed /60(Hz), the unbalanced moment is altered with the speed and it reaches to the maximum at the rated speed.

(3) Damping

The effect of damping is to be taken into consideration for the calculation of response of ship hull vibration. The damping includes the external damping, i.e. dynamic damping and air damping, and also the internal damping, i.e. frication between hull structures, material retardation, cargo damping, etc.

At present, it is not sufficient to understand the hull vibration damping, the effect of damping is to be uniformly considered and determined by full-scale testing to a great extent. In the forced vibration response calculation for hull, the damping coefficient is to be taken approximate to 0.015, if possible, the detailed damping coefficient related to frequency may be used.

(2) Position of calculation

For the hull vibration response analysis under periodical excitation of propeller or main engine, the vibration response may be calculated in time domain or frequency domain. The particular positions of hull vibration response are to be evaluated based on the shipowner's demands or need of certification, chosen in the crew's working, rest areas and essential instruments and equipment areas, such as the middle point of fore end of bridge deck, edge of bridge deck, aft end of main deck, etc.

11.4.4 Response calculation for vertical vibration of hull girders of a certain bulk carrier

(1) Frequency calculation for vertical vibration of hull girders

The frequency of vertical vibration of hull girders and resonance speed for a certain bulk carrier are shown in Table 11.4.1. As seen from the Table, 1st order to 5th order hull girder vertical resonances occur and are induced by exciting frequency of 2nd order unbalanced moment of diesel engine within the rated speed of main engine of 91 r/min, 5th order resonance occurs at the speed of 66 r/min, the frequency ratio is 0.73, which is not within the normal speed range of main engine and meets the requirements of design criteria.

Natural Frequency of Hull Girder Vertical Vibration and Resolution of Resonance

Table 11.4.1

Order / Frequency and velocity	1st order	2nd order	3rd order	4th order	5th order
Natural frequency of vibration (Hz)	0.46	0.94	1.4	1.8	2.2
Natural frequency of vibration (1/min)	28	57	84	108	132
Resonance speed under 1st order excitation (r/min）	28	57	84	108	132
Resonance speed under 2nd order excitation (r/min)	14	29	42	54	66

(2) Calculation condition of vibration response

Although the frequency of vertical vibration for hull girders meets the requirements of design criteria, in order to further investigate the 5th order resonance response, such as consideration of calculation error, the 5th order resonance may occur within the common range of main engine speed, the vibration resonance is to be calculated as the resonance speed of 91 r/min, also to be calculated as the resonance speed of 66 r/min.

(3) Excitation source

Where the rated speed of main diesel engine is 91r/min, the unbalanced moments are as follows:

1st order unbalanced moment = 0;

2nd order unbalanced moment = 2098 kNm;

4th order unbalanced moment = 141 kNm.

Due to the 4th order unbalanced moment being smaller, calculation is not necessary and only the excitation of 2nd order unbalanced moment is to be calculated.

(4) Damping coefficient: is to be taken as 0.015.

(5) Calculated position: stern end.

(6) Evaluation of calculation results.

Adopting of mode shape superposition method, the MSC NASTRAN software is to be used for vibration response calculation.

When the speed is at 91 r/min, 5th order vertical resonance of hull girder will be induced by 2nd unbalanced moment of 2098 kNm, the vibration speed at the stern end is about 4 mm/s; when the sped is at 66 r/min, the 2nd order unbalanced moment is reduced to 53%, (1111.94 kNm), the vibration speed of stern end is about 2.1 mm/s. Based on ISO 6954-1984 criteria, such vibration is acceptable.

11.5 Design Criteria

11.5.1 The effective means to prevent or reduce the resonance response of ship hull vibration is to avoid the resonance caused by low order vibration frequency and main exciting frequency within the range of 85% to 100% maximum running speed.

11.5.2 The correctness of natural frequency calculation for hull vertical vibration is related to the methods used, hence, the error range of natural frequency calculation is to be taken into consideration in the design.

Chapter 12 VIBRATION OF SUPERSTRUCTURES

12.1 Introduction

The superstructure vibration means the global superstructure longitudinal vibration and local structural member vibration.

The displacement of longitudinal vibration of superstructure consists of:

(1) the shearing displacement of superstructure, see Figure12.1.1(1);

(2) the displacement induced by elastic support of hull beneath the superstructure, see Figure 12.1.1(2);

(3) the displacement induced by hull vertical vibration, see Figure 12.1.1(3);

(4) the displacement induced by hull longitudinal vibration, see Figure 12.1.1(4).

Of which (1) and (2) are the primary ones.

The local structural vibration calculation for superstructure may be carried out according to the requirements of Chapter 13.

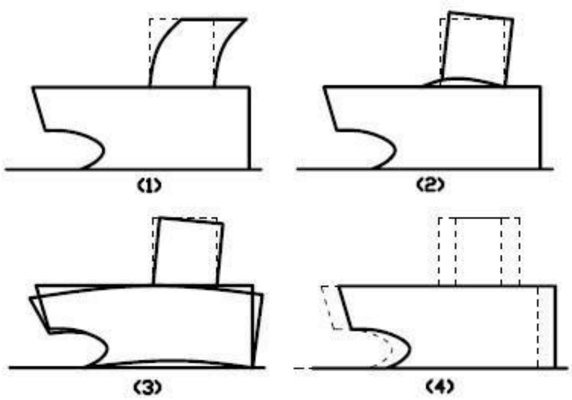


Figure 12.1.1 Superstructure Longitudinal Vibration Displacement

12.2 Excitation and Transmission

12.2.1 Main excitation inducing superstructure longitudinal vibration

(1) The fluctuation pressure induced by propeller is transferred to superstructure through hull structure, the primary harmonic order is the number of blades.

(2) The alternating thrust induced by propeller or frame longitudinal vibration induced by 2nd excitation of shafting longitudinal vibration and torsional vibration is transferred to superstructure through the double bottom, the primary harmonic order is the number of blades or cylinders.

(3) The exciting force induced by poor shafting alignment is transferred to superstructure through hull structure, the exciting harmonic order is of the 1st order.

(4) The shafting longitudinal vibration induced by radial force on the crankshaft caused by diesel engine is transferred to superstructure through thrust bearing and main hull. The primary harmonic order of radial force for two-stroke cycle diesel engine is referred to Table 12.2.1. Where the damper of shafting longitudinal vibration has been installed, consideration of the effect of shafting longitudinal vibration is not necessary.

Primary Harmonic Order of Crankshaft Radial Force Table 12.2.1

Number of cylinder	5	6	7	8	9	10	12
Primary excitation order	5	9, 6	7	8, 5	6, 5, 9	6, 5, 10	6, 5

(5) The shafting torsional – longitudinal vibration induced by crankshaft torsional vibration which is caused by tangential force on the crankshaft induced by diesel engine is transferred to superstructures through thrust bearing and main hull, the primary harmonic order of excitation is the number of cylinders for two-stroke cycle diesel engine, and the number or half number of cylinders for four-stroke cycle diesel engine.

12.3 Simplified Calculation for Natural Frequency of Global Vibration of Superstructures

12.3.1 Model of Simplified Calculation for Natural Frequency of Global longitudinal Vibration of Superstructures

As mentioned above, the superstructure longitudinal vibration mainly consists of two parts, i.e. one is the shearing vibration of the superstructure itself, the other is the vibration induced by elastic support of lower hull of superstructure, and may be described as a whirling vibration around a rotary axis of demarcation line between fore bulkhead and main deck, see Figure 12.3.1.

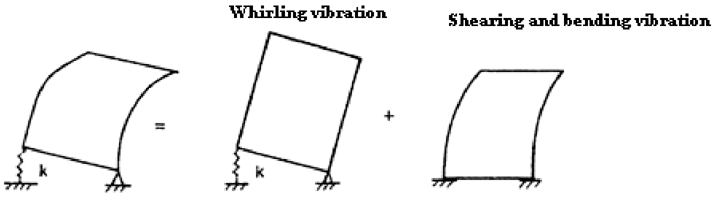


Figure 12.3.1 Simplified Calculation Model of Superstructure Longitudinal Vibration

12.3.2 Structural types of superstructures

In the simplified calculation of global vibration for superstructures, the following five superstructure types are to be divided as Figure 12.3.2:

Type *A* — the superstructure above the main deck is completely separated from the funnel root;

Type A_1 — the superstructure above the main deck is completely separated from the funnel root, but there is connection between upper deckhouse and funnel;

Type *B* — the 3rd or 4th layer deckhouse above the 2nd layer long deckhouse is arranged parallel to funnel independently;

Type *C* — the 4th or 5th layer deckhouse above the 1st layer long deckhouse is arranged parallel to funnel independently;

Type *D*— the superstructure and funnel are arranged as an integral above the upper deck or long deckhouse.

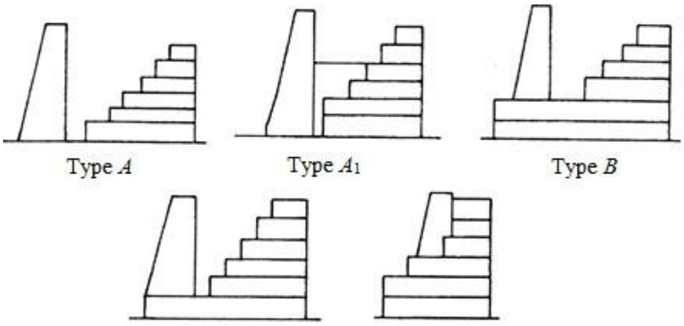


Figure 12.3.2 Structural Types of Superstructures

12.3.3 Calculation for natural frequency of superstructure longitudinal vibration

The natural frequency of superstructure longitudinal vibration f_c is to be calculated by:

$$f_c = 1.07K_1 \frac{f_s}{\sqrt{1+(\frac{f_s}{f_r})}} + K_2 \qquad \text{Hz} \qquad (12.3.1)$$

where: f_c — natural frequency of superstructure longitudinal vibration, in Hz;

f_s — natural frequency of superstructure only as shearing vibration, in Hz; to be calculated according to formula (12.3.2);

f_r — natural frequency of superstructure whirling vibration, in Hz; to be calculated according to formula (12.3.3);

K_1, K_2 — correction coefficient of superstructure type, see Table 12.3.1.

Coefficients of K_1 and K_2 Table 12.3.1

Structural type	K_1	K_2
Type *A*	0.9	2.5
Type A_1	0.72	4.92
Type *B*, Type *C* and Type *D*	1.0	0.0

12.3.4 Calculation for natural frequency of superstructure only as shearing vibration

The natural frequency of superstructure only as shearing vibration f_s is to be calculated by:

$$f_s = 165.5K_3 \frac{n}{h}\sqrt{\frac{K_s \sum l_i \varphi_i}{\sum l_i b_i \varphi_i}} \qquad \text{Hz} \qquad (12.3.2)$$

where: n— total number of layers of superstructure above the main deck;

h— total height of superstructure, in m;

i— number of layers from the lower to upper of deckhouse, $i = 1 \sim n$;

l_i — length of i^{th} layer of superstructure, in m;

$K_s = \frac{Z}{4}$ — correction coefficient of longitudinal bulkheads (including walls and internal continuous longitudinal bulkheads);

Z— number of longitudinal bulkheads;

b_i — width of i^{th} layer of superstructure, in m;

$\psi_i = 30(\sqrt[3]{7} - \sqrt[3]{i}) + 1$ — effective factor of shearing stiffness of i^{th} layer of deckhouse; $\psi_i = 1.0$ for $i > 7$;

$\varphi_i = 3i(i-1) + 1$ — effective factor of deckhouse mass on each layer;

K_3 — correction coefficient of number of layers, see Table 12.3.2.

Correction Coefficient of Number of Layers K_3 Table 12.3.2

Type	Number of layer	K_3
Type *A* and Type A_1	—	1.0
Type *B*, Type *C* and Type *D*	5	0.93
	6	1.0
	7	1.08

12.3.5 Calculation for natural frequency of rigid body rotation

The natural frequency of rigid body rotation f_r is to be calculated by:

$$f_r = 187.7 K_4 \frac{n}{h} \sqrt{\frac{K_5 n l_1^2}{h(\sum l_i b_i \varphi_i + 0.75 \frac{l_1^3 b_1 n^2}{h^2})}} \quad \text{Hz} \qquad (12.3.3)$$

where: K_4— correction coefficient of number of layers, see Table 12.3.3;

Correction Coefficient of Number of Layers K_4 Table 12.3.3

Type	Number of layers	K_4
Type *A* and Type A_1	—	1.0
Type *B*, Type *C* and Type *D*	5	0.97
	6	1.0
	7	1.04

$K_5 = 100 K_6 K_7 K_8$ — elastic coefficient of support;

$$K_6 = \frac{1}{8 - 4(\frac{b_1}{B})^2 + (\frac{b_1}{B})^3} - 0.0862;$$

$$K_7 = \frac{1}{\sin(-0.69\frac{l_1}{l} + 0.845)\pi};$$

K_8 — to be selected as follows:

$K_8 = 2.0$ — where there is a lot of longitudinal and transverse bulkheads under the main deck beneath the superstructure which may involve the layer of platform under the main deck to move together, and $l > 1.2 l_1$;

$K_8 = 1.45$ — where the length of 1st layer of deckhouse for superstructure $l_1 = l$, and the funnel is protruded the 3rd layer and above or similar to Type *D*;

$K_8 = 1.0$ — other cases;

l_1 —— length of 1st layer of superstructure, in m;

b_1 —— width of 1st layer of superstructure, in m;

l —— distance between two transverse bulkheads supporting the lower hull of superstructure, see Figure 12.3.3;

B —— width, in m.

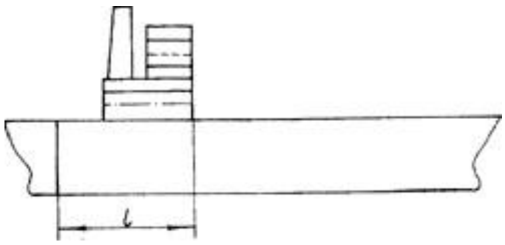


Figure 12.3.3 Determination of *l*

12.4 Finite Element Calculation For Stern - Superstructure Coupled Vibration

12.4.1 Introduction

The superstructures are regarded as an integral and located on the main deck, the main deck and its beneath hull structure are composed of an elastic basis of superstructure. The superstructure vibration includes global vibration and local vibration, the former means the global longitudinal vibration, transverse vibration and torsional vibration of superstructures while the latter means the vibration on each deck structures and bulkhead structures in the superstructures.

In general, engine room and superstructures are arranged at stern onboard modern ships, so that the superstructures are closer to two main vibration sources of propeller and main engine, causing the superstructures in a greater excitation condition; in addition, the superstructures are usually designed higher and shorter along the ship's length for improving the navigation bridge visibility and reducing the crew number; and the superstructures are usually separated from the engine room casing and funnel to reduce the noise pollution in the superstructures, thus, the integral longitudinal stiffness of the superstructures is reduced.

In recent years, for the purpose of reducing the noise in accommodation of superstructures and decrease the integral mass and height of centroid for superstructure, the light steel movable bulkheads containing rock wool are usually used for fixed steel walls inside the superstructures to lead to the increase of structural spans of each deck, and the reduced natural frequency is liable to meeting the blade order frequency of propeller and the longitudinal exciting frequency of main engine frame, leading to resonance and adverse vibration. The coupled vibration of hull-stern-superstructure is dicussed in this Section, for the internal deck structure vibration related to superstructures, see Chapter 13 of the Guidelines.

12.4.2 Finite element model of coupled vibration for hull-stern-superstructure

The global vibration of superstructures depends on bending stiffness, shearing stiffness and torsional stiffness, as well as the supporting of main hull to superstructures.

In order to take the interaction of hull-stern-superstructure into consideration, the following finite element models may be used:

(1) Three-dimensional finite element model, including the whole stern, as shown in Figure 12.4.1, has been considered elastic supporting of main hull to superstructures, but not including the coupling influence between global vibration mode of superstructures and high order vibration mode of main hull.

(2) The stern, engine room and superstructures are three-dimensional finite element model, and the remaining parts are hull girder finite element model, as shown in Figure 12.4.2. Based on Figure 12.4.1, the model is to regard the stern and hull forward of engine room as hull girder, the

connection of three-dimensional model and one-dimensional model is located in way of the transverse bulkhead of main hull, where rigid connecting elements are used to ensure that the sectional corner in way of connection of stern, superstructures and hull is consistent with the displacement.

(3) Three-dimensional finite element model for a whole ship is to be capable of reflecting the elastic supporting of main hull to superstructure and the coupling influence between them. The model is most reasonable but is complicated, and due to a larger data workload, may be selected according to the practical cases.

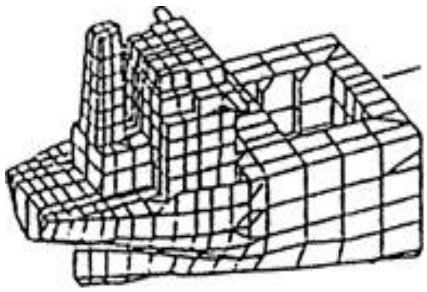


Figure 12.4.1 Three-Dimensional Finite Element Model of Superstructure

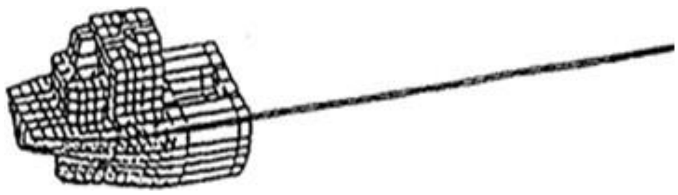


Figure 12.4.2 Three-Dimensional Finite Element Model of Stern, Engine Room and Superstructure

12.4.3 Calculation cases

The coupled vibration calculation model of stern-superstructure for a certain tanker is shown in Figure 12.4.3. The natural frequency of stern-superstructure vibration and resonance speed at ballasting condition are shown in Table 12.4.1. the inherent mode shape is shown in Figure 12.4.4. The rated speed of main engine is 83 r/min, and the number of propeller blades is four. Therefore, means is to be taken to avoid the 4th order 1 times (4 times) blade order resonance at the rated speed of 81 r/min.

Natural Frequency of Stern-Superstructure Vibration and Resonance Speed Table 12.4.1

Natural frequency(Hz)	Natural frequency (1/min)	Resonance speed n (r/min)	
		1st order exciting force	Exciting force of blade order (4 times)
5.4114	325	325	81

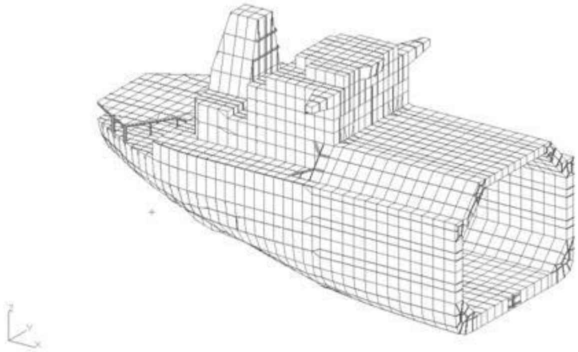


Figure 12.4.3 Three-Dimensional Model of Stern-Superstructure Coupled Vibrationn

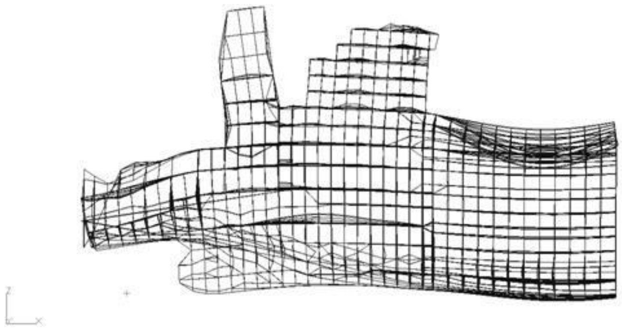


Figure 12.4.4 Mode Shape of Stern-Superstructure Coupled Vibration (f=5.4114 Hz)

12.5 Design Criteria

12.5.1 The effective means to prevent or reduce the resonance response of superstructure vibration is to avoid the resonance caused by natural frequency and main exciting frequency of superstructure within the range of 85% to 100% maximum running speed.

12.5.2 The correctness of natural frequency calculation for superstructure vibration is related to the methods used, hence, the error range of natural frequency calculation is to be taken into consideration in the design.

Chapter 13 LOCAL VIBRATION

13.1 Introduction

Most of the vibration issues onboard ships are those for the local structure vibration. The local structures mean the girder, plate, grillage, shaft rack, shaft bracket, propeller blade, mast, platform, etc. The local structure vibration control is to ensure that its natural frequency does not coincide with exciting frequency and maintain a certain reservation.

The overall vibration of hull girders and superstructure vibration may also be reflected in the local vibration of grillage.

The excitation source leading to local structure vibration may include the related ones inducing hull girder vibration and superstructure vibration.

Therefore, if local structure vibration is induced, causes are to be found and analyzed, i.e. it is the structure vibration induced by the structure it-self or induced by the overall vibration of hull girders or superstructure vibration so as to control the structure vibration issues effectively.

13.2 Natural Frequency of Beam Vibration

13.2.1 The natural frequency of transverse vibration fm for homogeneous straight beam is:

$$f_m = \frac{\lambda_m^2}{2\pi l^2}\sqrt{\frac{EIg}{\rho A}} \quad \text{Hz} \qquad (13.2.1)$$

where: l — length of girder, in cm;

E — elastic modulus of material, in kg / cm^2;

I — sectional moment of inertia, in cm^4;

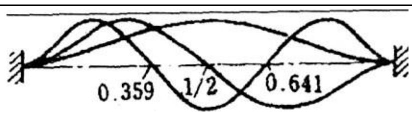
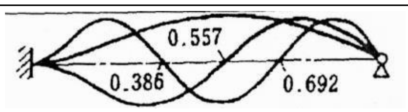
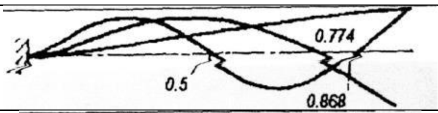
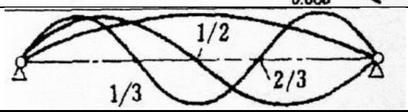
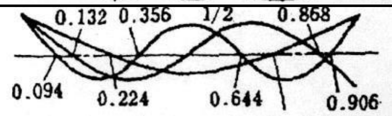
ρ — density of material, in kg / cm^3;

A — cross sectional area, in cm^2;

g — 981cm/s^2;

λ_m — characteristic value, the former 1st to 3rd order to be determined from Table 13.2.1.

Characteristic Value **Table 13.2.1**

Boundary condition	Mode shape	Order	λ_m
(1) rigidly fixed – rigidly fixed		1 2 3	4.73 7.853 10.996
(2) rigidly fixed – simply supported		1 2 3	3.927 7.069 10.21
(3) rigidly fixed - freely		1 2 3	1.875 4.694 7.855
(4) simply supported – simply supported		1 2 3	π 2π 3π
(5) freely – freely		1 2 3	4.73 7.853 10.996

13.2.2 The natural frequency of longitudinal vibration fm for homogeneous straight beam is:

$$f_m = \frac{\lambda_m}{2\pi l}\sqrt{\frac{Eg}{\rho}} \quad \text{Hz} \qquad (13.2.2)$$

where: l — length of girder, in cm;

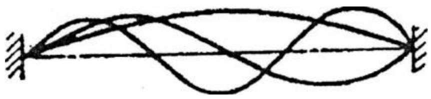
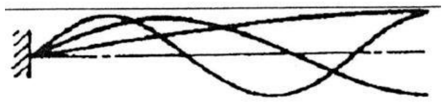
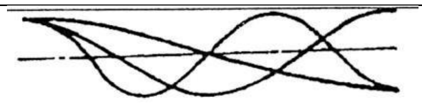
E — elastic modulus of material, in kg/cm^2;

ρ — density of material, in kg/cm^3;

g — 98l cm/s^2;

λ_m — characteristic value, to be determined from Table 13.2.2.

Characteristic Value **Table 13.2.2**

Boundary condition	Mode shape	Order	λ_m
(1) rigidly fixed – rigidly fixed		1 2 3	π 2π 3π
(2) rigidly fixed – freely		1 2 3	$\pi/2$ $3\pi/2$ $5\pi/2$
(3) freely – freely		1 2 3	π 2π 3π

13.2.3 The natural frequency of torsional vibration fm for homogeneous circular or ring cross sectional straight beam is:

$$f_m = \frac{\lambda_m}{2\pi l}\sqrt{\frac{Gg}{\rho}} \quad \text{Hz} \qquad (13.2.3)$$

where: G— shearing elastic modulus of material, in kg/cm^2;

l, λ_m, g, ρ as same as formula (13.2.2).

13.3 Natural Frequency of Plate and Grillage Vibration

13.3.1 Natural frequency of plate vibration

In general, deformation of plating onboard ships under external loads is less, the plating may be treated as the absolute rigid plate.

(1) Natural frequency of rectangular plate vibration in the air

①The natural frequency of rectangular plate with four edges supported by hinges f_{mn}:

$$f_{mn} = \frac{\pi}{2}\left(\frac{m^2}{a^2} + \frac{n^2}{b^2}\right)\sqrt{\frac{Dg}{\rho t}} \quad \text{Hz} \qquad (13.3.1)$$

where: a — length of longer edges of rectangular plate, in cm;

b — length of shorter edges of rectangular plate, in cm;

D — bending stiffness of plate, $D = \frac{Et^3}{12(1-\mu^2)}$, in kg/cm;

E — elastic modulus of plating material, in kg / cm^2;

t — thickness of plating, in cm;

μ — material Poisson's ratio;

g — 981 cm/s^2;

ρ — density of material, in kg/cm^3;

m,n — half wave number along the a and b respectively.

For rectangular plate, the precise natural frequency may be obtained only for one plate with four edges supported by hinges.

② The natural frequency f of rectangular plate with four edges rigidly fixed is:

$$f = \frac{2\pi}{3}\sqrt{\frac{Dg}{\rho t}}\left(\frac{3}{a^4} + \frac{2}{a^2b^2} + \frac{3}{b^4}\right) \quad \text{Hz} \qquad (13.3.2)$$

③ The natural frequency f of rectangular plate with one pair of edges (a) rigidly fixed and the other pair (b) supported by hinges is:

$$f = \frac{2}{\sqrt{3}}\pi\sqrt{\frac{Dg}{\rho t}\left(\frac{3}{16a^4} + \frac{1}{2a^2b^2} + \frac{1}{b^4}\right)} \quad \text{Hz} \qquad (13.3.3)$$

④ The natural frequency f of rectangular plate with one edge (a) rigidly fixed and the other three edges completely supported by hinges is:

$$f = \frac{3.516}{2\pi b^2}\sqrt{\frac{Dg}{\rho t}} \quad \text{Hz} \qquad (13.3.4)$$

⑤ The natural frequency f of rectangular plate with one pair of edges (a) rigidly fixed and the other pair of edges (b) completely supported by hinges is:

$$f = \frac{7.074}{2\pi^2}\sqrt{\frac{Dg}{\rho t}} \quad \text{Hz} \qquad (13.3.5)$$

⑥ The natural frequency f of square plate with four edges completely supported by hinges is:

$$f = \frac{7}{\pi a^2}\sqrt{\frac{Dg}{\rho t}} \quad \text{Hz} \qquad (13.3.6)$$

(2) Natural frequency of rectangular plate vibration in the water

The natural frequency of plate vibration in the air is f_a, the natural frequency of plate vibration in the water is f_w:

$$f_w = \frac{f_a}{\sqrt{1+\mu(\frac{b}{a})\frac{b}{7.85t}}} \quad \text{Hz for single side contacting with water} \qquad (13.3.7)$$

$$f_w = \frac{f_a}{\sqrt{1+2\mu(\frac{b}{a})\frac{b}{7.85t}}} \quad \text{Hz for double sides contacting with water} \qquad (13.3.8)$$

13.3.2 Natural frequency of panel vibration

(1) Natural frequency of panel vibration in the air

The natural frequency of panel is treated according to the longitudinal attached with wing plate, the width of attached wing plate is to be equal to 1/6 longitudinal span, but not more than longitudinal interval *b*,

The natural frequency f_a of panel with both ends supported by hinges is:

$$f_a = 7.92 \times 10^5 \sqrt{I/S} / L^2 \qquad \text{Hz} \qquad (13.3.9)$$

The natural frequency f_a of panel with both ends rigidly fixed is:

$$f_a = 1.8 \times 10^6 \sqrt{I/S} / L^2 \qquad \text{Hz} \qquad (13.3.10)$$

The natural frequency f_a of panel with one end supported by hinges and the other end rigidly fixed is:

$$f_a = 1.3 \times 10^6 \sqrt{I/S} / L^2 \qquad \text{Hz} \qquad (13.3.11)$$

where: L — length of panel, in cm;

I — sectional moment of inertia of longitudinal containing attached wing plates, in cm^4;

S — sectional area of longitudinal containing attached wing plates, in cm^2.

(2) Natural frequency of panel vibration in the water

The natural frequency of panel vibration in the air is f_a, and the natural frequency of panel vibration in the water is f_w,

$$f_w = \frac{f_a}{\sqrt{1 + \mu(\frac{B}{L}) \frac{B}{7.85 t_m}}} \qquad \text{Hz for single side contacting with water} \qquad (13.3.12)$$

$$f_w = \frac{f_a}{\sqrt{1 + 2\mu\left(\frac{B}{L}\right) \frac{B}{7.85 t_m}}} \qquad \text{Hz for double side contacting with water} \qquad (13.3.13)$$

where: B — width of panel, in cm;

t_m — equivalent thickness of panel containing longitudinal, in cm;

L — length of panel, in cm;

$\mu(B/L)$— depend on side ratio of panel and dimensionless factor of fixed boundary, to be determined by Figure 13.3.2.

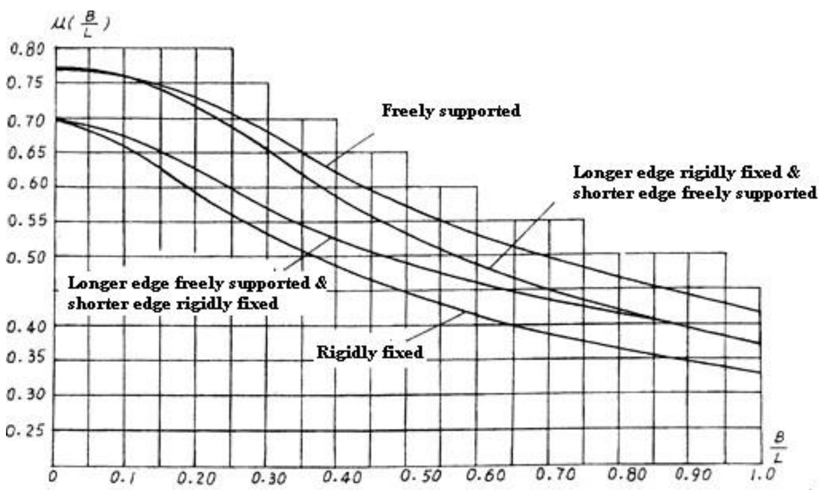


Figure 13.3.2 $\mu(B/L)$ **Coefficient**

13.3.3 Natural frequency of panel vibration
13.3.4

(1) Natural frequency of any panel vibration

In general, the natural frequency of any panel vibration may be calculated by equivalent method. The panel is same as other elastic structures, its free vibration may be regarded as the sum of infinite numbers of main vibration, which is corresponding to one natural frequency and one inherent mode shape, therefore, each main vibration may be converted to single degree of freedom system vibration. The natural frequency of panel f_i corresponding to i^{th} main vibration is:

$$f_i = \frac{1}{2\pi}\sqrt{\frac{K_e}{M_e}} \quad \text{Hz} \qquad (13.3.14)$$

where: K_e— equivalent stiffness of panel corresponding to i^{th} main vibration, in kg/cm;

M_e— equivalent mass of panel corresponding to i^{th} main vibration, in kgs²/cm.

(2) Equivalent stiffness

① Where the mode shape $f(x, y)$ of i^{th} main vibration of panel as shown in Figure 13.3.3 has been known, the corresponding equivalent stiffness K_e is:

$$K_e = \sum\int_0^L EI_i f''^2_{xx}(x, y_i)dx + \sum\int_0^B EL_j f''^2_{yy}(x_j, y)dy \qquad (13.3.15)$$

where: E — elastic modulus of material, in kg/cm²;

I_i — sectional moment of inertia of i^{th} piece of crossed structural member, in cm⁴'

L — span of crossed structural member, in cm;

y_i — coordinate of i^{th} piece of crossed structural member in the direction of y axis, in cm;

I_j — sectional moment of inertia of j^{th} piece of main beam, in cm⁴;

x_j — coordinate of j^{th} piece of main beam in the direction of x axis, in cm;

B — span of main beam, in cm.

Where the sectional moment of inertia of crossed structural members or main beam is changed along the direction of its length, the integration of formula (13.3.15) is to be carried out section by section.

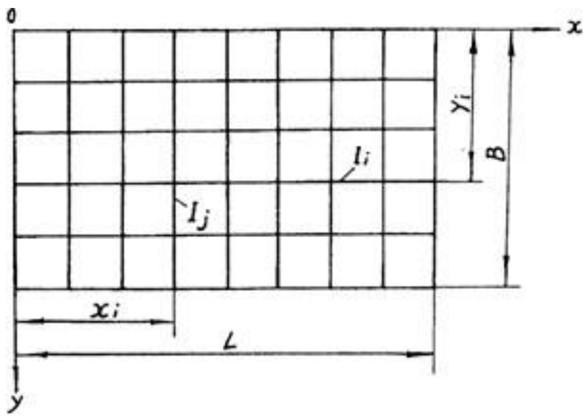


Figure 13.3.3 Panels

② For panel with both ends of main beam and crossed structural members elastically fixed, its equivalent stiffness K_e is:

$$K_e = \frac{K_1}{2}\frac{E\pi^4}{L^3}\sum_{i=1}^{m} I_i \sin^2\frac{\pi y_i}{B} + \frac{K_2}{2}\frac{E\pi^4}{B^3}\sum_{i=1}^{n} I_i \sin^2\frac{\pi x_i}{l} \qquad (13.3.16)$$

where: K_1 and K_2— coefficients related to couple coefficient of support ε , to be calculated by:

$$\left.\begin{aligned} K_1 &= 1+3\varepsilon_1 \\ K_2 &= 1+3\varepsilon_2 \end{aligned}\right\} \qquad (13.3.17)$$

where: ε_1 and ε_2 — couple coefficients of supports at both ends of crossed structural member and main beam,

$\varepsilon_1 = \varepsilon_2 = 0$ for freely supported;

$\varepsilon_1 = \varepsilon_2 = 1$ for rigidly fixed.

(1) Equivalent mass

The equivalent mass M_e is:

$$M_e = \sum_{i=1}^{q} M_i f^2(x_i, y_i) + \sum_{i=1}^{q} J_{ix} f'^2{}_x(x_i, y_i) + \sum_{i=1}^{q} J_{iy} f'^2{}_y(x_i, y_i) + J_0 \int_0^L \int_0^B f'^2{}_x(x, y) dxdy$$

$$+ J_0 \int_0^L \int_0^B f'^2{}_y(x, y) dxdy + \int_0^L \int_0^B (m_0 + m_w) f^2(x, y) dxdy \qquad (13.3.18)$$

where: M_i— concentrated mass in way of (x_i, y_i) coordinate, in kg;

q— number of concentrated mass;

J_{ix}— moment of inertia of concentrated mass relative to rotation axis parallel to x axis, in kg/cm^2;

J_{iy}— moment of inertia of concentrated mass relative to rotation axis parallel to y axis, in kg/cm^2;

m_w— entrained water mass in unit area of panel, to be determined by Figure 13.3.1, in kg/cm^2;

J_0— moment of inertia of mass in unit area of panel, in cm^2;

m_0— mean mass in unit area of panel (including oil and water in double bottom), to be determined by formula (13.3.17):

$$m_0 = \frac{\rho}{g} t_m + m_{0w} \qquad (13.3.19)$$

where: g = 981 cm/s^2;

m_{0w}— mass of oil and water in unit area, in kg/cm^2;

$$t_m = t_0 + \frac{\sum_{i=1}^{m} s_i L + \sum_{j=1}^{n} s_j B}{LB} \qquad (13.3.20)$$

ρ — density of material, in kg/cm^3;

t_0— thickness of panel plating, it is equal to the sum of thickness of bottom plating and inner bottom plating if the double bottom is provided, in cm;

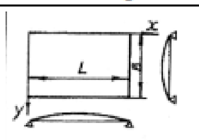
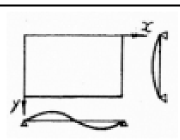
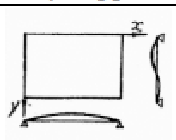
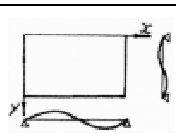
s_i— sectional area of crossed structural members (excluding attached wing plates), in cm^2;

s_j— sectional area of main beam (excluding attached wing plates), in cm^2.

(4) Natural frequency of panel vibration with different supporting panels

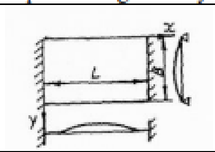
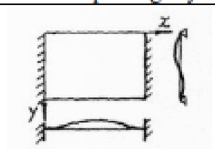
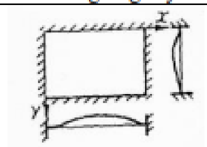
The natural frequency of panel vibration may be determined by the above-mentioned formula provided that the size of mode shape $f(x,y)$ has been known, however, due to the fact that the mode shape of panel vibration is generally unknown in advanced, the mode shape as appropriate, according to the fixed condition of both ends for main beam and crossed structural members is usually selected when the equivalent method is used. For the panel with surroundings freely supported, the low order mode shape may be taken from Table 13.3.1; for the panel with one pair of edges freely supported and the other pair rigidly fixed and the panel with surroundings rigidly supported, its 1st order mode shape may be taken from Table 13.3.2, in which the corners of $f'_x(x,y)$, $f'_y(x,y)$ and $f''_{xx}(x,y)$, $f''_{yy}(x,y)$ are listed.

Mode Shape of Panel with Surroundings Freely Supported　　Table 13.3.1

				
$f(x,y)$	$\sin\frac{\pi x}{L}\sin\frac{\pi y}{B}$	$\sin\frac{2\pi x}{L}\sin\frac{\pi y}{B}$	$\sin\frac{\pi x}{L}\sin\frac{2\pi y}{B}$	$\sin\frac{2\pi x}{L}\sin\frac{2\pi y}{L}$
$f'_x(x,y)$	$\frac{\pi}{L}\cos\frac{\pi x}{L}\sin\frac{\pi y}{B}$	$\frac{2\pi}{L}\cos\frac{2\pi x}{L}\sin\frac{\pi y}{B}$	$\frac{\pi}{L}\cos\frac{\pi x}{L}\sin\frac{2\pi y}{B}$	$\frac{2\pi}{L}\cos\frac{2\pi x}{L}\sin\frac{2\pi y}{B}$
$f'_y(x,y)$	$\frac{\pi}{B}\sin\frac{\pi x}{L}\cos\frac{\pi y}{B}$	$\frac{\pi}{B}\sin\frac{2\pi x}{L}\cos\frac{\pi y}{B}$	$\frac{2\pi}{B}\sin\frac{\pi x}{L}\cos\frac{2\pi y}{B}$	$\frac{2\pi}{B}\sin\frac{2\pi x}{L}\cos\frac{2\pi y}{B}$
$f''_{xx}(x,y)$	$-\left(\frac{\pi}{L}\right)^2\sin\frac{\pi x}{L}\sin\frac{\pi y}{B}$	$-\left(\frac{2\pi}{L}\right)^2\sin\frac{2\pi x}{L}\sin\frac{\pi y}{B}$	$-\left(\frac{\pi}{L}\right)^2\sin\frac{\pi x}{L}\times\sin\frac{2\pi y}{B}$	$-\left(\frac{2\pi}{L}\right)^2\sin\frac{2\pi x}{L}\sin\frac{2\pi y}{B}$
$f''_{yy}(x,y)$	$-\left(\frac{\pi}{B}\right)^2\sin\frac{\pi x}{L}\sin\frac{\pi y}{B}$	$-\left(\frac{\pi}{B}\right)^2\sin\frac{2\pi x}{L}\sin\frac{\pi y}{B}$	$-\left(\frac{2\pi}{B}\right)^2\sin\frac{\pi x}{L}\sin\frac{2\pi y}{B}$	$-\left(\frac{2\pi}{B}\right)^2\sin\frac{2\pi x}{L}\times\sin\frac{2\pi y}{B}$

Mode Shape of Panel With One Pair of Edges Freely Supported and the Other Pair Rigidly Fixed

Table 13.3.2

	One pair of edges freely supported and the other pair rigidly fixed		Surroundings rigidly fixed
			
$f(x,y)$	$\frac{1}{2}\left(1-\cos\frac{2\pi x}{L}\right)\sin\frac{\pi y}{B}$	$\frac{1}{2}\left(1-\cos\frac{2\pi x}{L}\right)\sin\frac{2\pi y}{B}$	$\frac{1}{4}\left(1-\cos\frac{2\pi x}{L}\right)\left(1-\cos\frac{2\pi y}{B}\right)$
$f'_x(x,y)$	$\frac{\pi}{L}\sin\frac{2\pi x}{L}\sin\frac{\pi y}{B}$	$\frac{\pi}{L}\sin\frac{2\pi x}{L}\sin\frac{2\pi y}{B}$	$\frac{1}{2}\frac{\pi}{L}\sin\frac{2\pi x}{L}\left(1-\cos\frac{2\pi y}{B}\right)$
$f'_y(x,y)$	$\frac{1}{2}\frac{\pi}{B}\left(1-\cos\frac{2\pi x}{L}\right)\cos\frac{\pi y}{B}$	$\frac{\pi}{B}\left(1-\cos\frac{2\pi x}{L}\right)\cos\frac{2\pi y}{B}$	$\frac{1}{2}\frac{\pi}{B}\left(1-\cos\frac{2\pi x}{L}\right)\sin\frac{2\pi y}{B}$
$f''_{xx}(x,y)$	$2\left(\frac{\pi}{L}\right)^2\cos\frac{2\pi x}{L}\sin\frac{\pi y}{B}$	$2\left(\frac{\pi}{L}\right)^2\cos\frac{2\pi x}{L}\sin\frac{2\pi y}{B}$	$\left(\frac{\pi}{L}\right)^2\cos\frac{2\pi x}{L}\cos\frac{2\pi y}{B}$
$f''_{yy}(x,y)$	$-\frac{1}{2}\left(\frac{\pi}{B}\right)^2\left(1-\cos\frac{2\pi x}{L}\right)\sin\frac{\pi y}{B}$	$-2\left(\frac{\pi}{B}\right)^2\left(1-\cos\frac{2\pi x}{L}\right)\sin\frac{2\pi y}{B}$	$\left(\frac{\pi}{B}\right)^2\left(1-\cos\frac{2\pi x}{L}\right)\cos\frac{2\pi y}{B}$

13.4　Natural Frequency of Shaft Bracket, Shaft Rack and Propeller Blade Vibration

13.4.1 Natural frequency of shaft bracket

The shaft bracket and shaft rack are located in the propeller-induced alternating pressure field, so they directly withstand the excitation changing periodically. In order to avoid the resonance of these structures, the natural frequency is to be calculated so as to take effective measures and keep it far away from the exciting frequency.

(1) The natural frequency f of single-armed bracket vibration shown in Figure 13.4.1 is calculated by:

$$f = \frac{1}{2\pi}\sqrt{\frac{K}{M}} \quad \text{Hz} \qquad (13.4.1)$$

where: $M = \frac{1}{g}(P_1 + P_2 + P_3 + P_4)$, in kgs²/cm;

$g = 981$ cm/s²;

P_1— weight of propeller, in kg;

P_2— weight of screwshaft with the length of $2S$ (see Figure 13.4.1 for S), in kg;

P_3— weight of hub and bush of shaft bracket, in kg;

P_4— mass of entrained water of propeller, in kg, to be determined by:

$$P_4 = \left[\psi\left(1 + 1.66\frac{H}{D}\right) + 0.083\frac{H}{D}\right]\frac{7.85 - Z}{4.85}\frac{7.40\rho_w}{\rho}P_1 \qquad (13.4.2)$$

where: $\frac{H}{D}$—— pitch ratio;

ρ_w — density of water, in kg/cm³;

ρ — density of propeller material, in kg/cm³;

Z— number of propeller blades;

ψ — coefficient determined according to the surface ratio a_p shown in Figure 13.4.2;

K— equivalent stiffness, in kg/cm, to be calculated by:

$$K = \frac{3EI}{l^3} \qquad (13.4.3)$$

where: E — elastic modulus of material, in kg/cm²;

I— moment of inertia of section of shaft bracket relative to axis vertical to plane with minimum stiffness, in cm⁴;

l— length of shaft bracket, in cm.

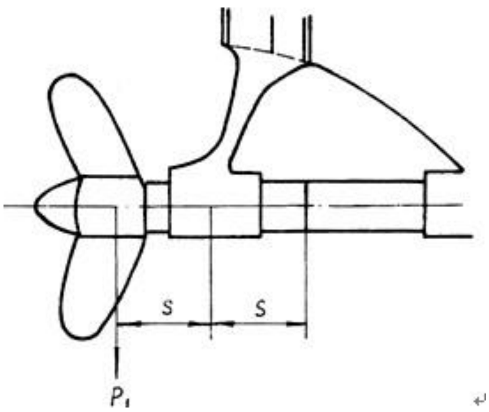


Figure 13.4.1 Shaft Bracket

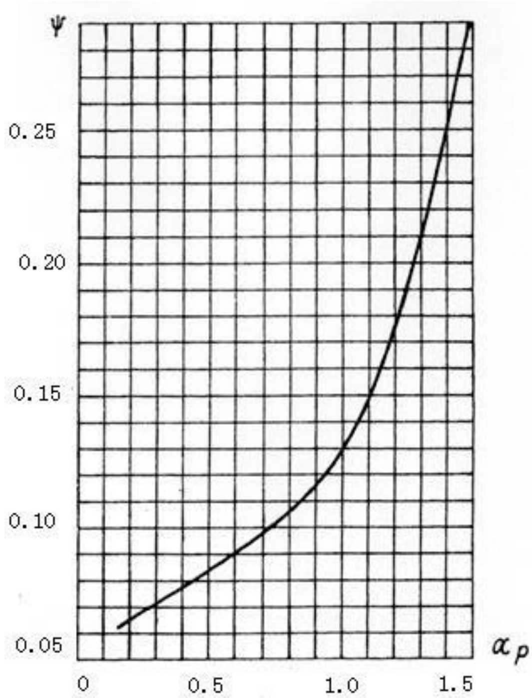


Figure 13.4.2　Relationship Curve of $a_p \sim \psi$

(2) For the double-armed bracket shown in Figure 13.4.3, its natural frequency is still to be calculated according to formula (13.4.1), then main vibrations in three directions are to be calculated. Two main vibrations are located within the plane consisting of two arms and the other one is located within a plane vertical to the two arms.

For the equivalent mass for the former two main vibrations $M_{1,2}$:

$$M_{1,2} = \frac{1}{g}\left[(P_1 + P_2 + P_4)\sin^2 a + P_3\right] \quad \text{kgs}^2/\text{cm} \qquad (13.4.4)$$

For the equivalent mass for the latter main vibration M_3:

$$M_3 = \frac{1}{g}\left[(P_1 + P_2 + P_4)\cos^2 a + P_3\right] \quad \text{kgs}^2/\text{cm} \qquad (13.4.5)$$

where: a — angle between axis and plane consisting of two arms of bracket, as shown in Figure 13.4.3.

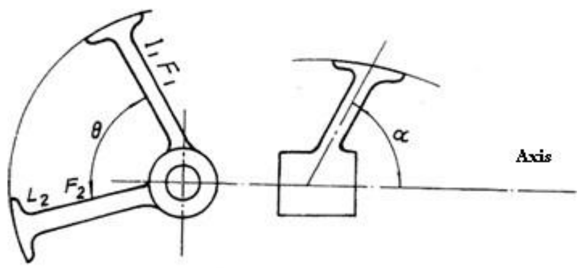


Figure 13.4.3　Double-armed Shaft Bracket

(3) For the former two main vibrations, the stiffness in formula (13.4.1) are:

$$K_1 = K_a \sin^2 \beta + K_b \sin^2(\theta + \beta) \qquad \text{kg/cm} \qquad (13.4.6)$$
$$K_2 = K_a \cos^2 \beta + K_b \cos^2(\theta + \beta) \qquad \text{kg/cm} \qquad (13.4.7)$$

where: $\beta = -\frac{1}{2} tg^{-1} \frac{K_b \sin 2\theta}{K_a + K_b \cos 2\theta}$;

$K_a = \frac{EF_1}{l_1}$ kg/cm;

$K_b = \frac{EF_2}{l_2}$ kg/cm;

E— elastic modulus of material, in kg/cm^2;

F_1 and F_2— cross sectional areas of two arms for shaft bracket respectively, in cm^2;

l_1 and l_2— lengths of two arms of shaft bracket respectively, in cm;

θ — angle between arms of shaft bracket, in °.

(4) For the 3rd main vibration, the stiffness in formula (13.4.1) is:

$$K_3 = 3E \frac{l_1^3 I_2 + l_2^3 I_1}{l_1^3 l_2^3} \qquad \text{kg/cm} \qquad (13.4.8)$$

where: I_1 and I_1——moments of inertia of sections of two arms for shaft bracket relative to axis vertical to plane with minimum stiffness respectively, in cm^4;

(5) In formula (13.4.4) and formula (13.4.5), P_1, P_2, P_3, P_4 are to be calculated according to the formula (13.4.1).

(6) The minimum of the above-mentioned three natural frequencies is the natural frequency needed to calculate, in general, the natural frequency is to be required to more than or equal to the main excitation frequency.

13.4.2 Natural frequency of shaft rack

As shown in Figure 13.4.4, the symmetric vibration regards the shaft rack as a cantilever beam with concentrated mass at the free end, the cross section of cantilever beam also includes area of partial bossing plate (the width is taken as $0.4l$, l — length of span of arm for shaft rack), starting from the natural frequency of cantilever beam, experimental factor is to be used to carry out correction for inhomogeneity of stiffness for span of arm, mass of entrained water, etc.

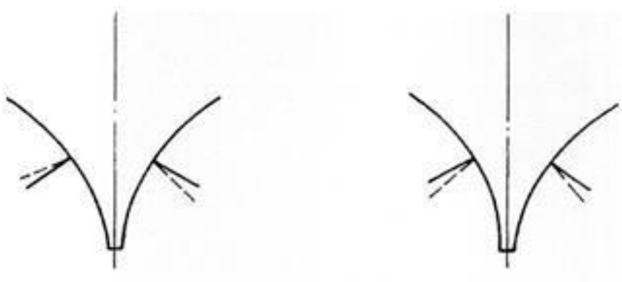


Figure 13.4.4 Mode Shape

The natural frequency f of shaft rack for the above-mentioned symmetric vibration is:

$$f = \frac{\eta}{2\pi} \sqrt{\frac{3E(I_1 + I_2)g}{(P_1 + P_2) l^3}} \qquad \text{Hz} \qquad (13.4.9)$$

where: E—— elastic modulus of material, in kg/cm^2;

P_1— weight of propeller, in kg;

P_2— sum of weights of hub, bush and screwshaft after the hub of span of arm for shaft rack, in kg;

l— length of span of arm for shaft rack, in cm;

η — to be determined by Figure 13.4.5 according to the angle between span of arm for shaft rack and the horizontal plane φ ;

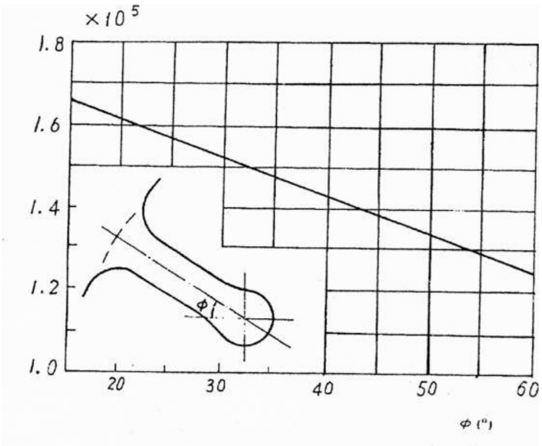


Figure 13.4.5 $\eta \sim \varphi$ Curve

g = 981 cm/s^2;

I_1 and I_2— moment of inertia of cross section of span of arm for shaft rack and moment of inertia of bossing plate to be considered respectively, as shown in Figure 13.4.6. If $I = I_1 + I_2$ changes greatly along the direction of length of span of arm, may be calculated as follows:

$$I_1 + I_2 = \frac{I_R}{\frac{9}{20} + \frac{3}{5}\left(\frac{I_R}{I_m}\right) - \frac{1}{20}\left(\frac{I_R}{I_B}\right)} \quad \text{cm}^4 \qquad (13.4.10)$$

I_R, I_m, I_B in formula (13.4.10) are shown in Figure 13.4.7, and are obtained by extrapolation of I'_R, I'_m, I'_B.

For the anti-symmetric vibration shown in Figure 13.4.4, its natural frequency is to be 0.8 times that of the symmetric vibration.

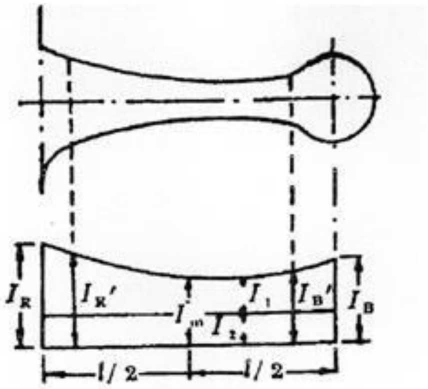


Figure 13.4.6 Moment of Inertia I_1 and I_2

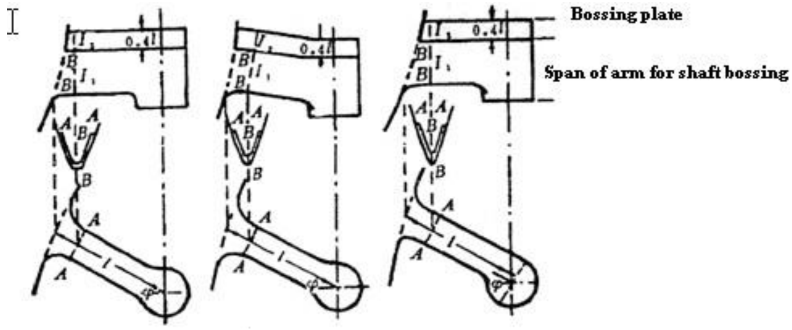


Figure 13.4.7 I_R, I_m and I_B

13.4.3 Natural frequency of propeller blade vibration

When the propellers are operated in an inhomogeneous flow field, the force withstood by blades changes periodically, if the natural frequency of blade vibration is consistent with the frequency of such periodically changed force, resonance of blades may be caused and unfavorable for the strength of blades. The calculation formula of natural frequency for blade vibration may be given by energy method and model testing.

(1) The natural frequency of propeller blade vibration in the air f_{pa} is:

$$f_{pa} = \frac{\lambda}{2\pi l^2}\sqrt{\frac{EI_0}{\rho_m A_0}} \qquad \text{Hz} \qquad (13.4.11)$$

where: l— length of blade, in cm;

ρ_m— density of blade material, in kg/cm^3;

E— elastic modulus of material, in kg/cm^2;

I_0— moment of inertia of section of blade root, in cm^4;

A_0— sectional area of blade root, in cm^2;

λ — characteristic value related to width, thickness and deflection of blade, to be calculated by:

$$\lambda = \sqrt{\frac{\int_0^1 \frac{b}{b_0}\left(\frac{t}{t_0}\right)^3\left(\frac{d^2\eta}{d^2\xi}\right)^2 d\xi}{\int_0^1 \frac{b}{b_0}\left(\frac{t}{t_0}\right)\eta^2 d\xi}} \qquad (13.4.12)$$

where: $\frac{b}{b_0}$ — ratio between blade width and root width, ratios of four blade types are shown in Figure 13.4.8;

$\frac{t}{t_0}$ — ratio between blade thickness ad root thickness;

η — ratio between blade deflection and blade tip deflection;

ξ — dimensionless length relative to blade root, $\xi = 1$ in way of blade tip.

Assuming:

$$\frac{b}{b_0} = (1-\xi)(1+k\xi) + c\xi(1-\xi^{10}) \qquad (13.4.13)$$

If the maximum width of blade is in way of $\xi = 0.6$ (approximately equal to $\frac{r}{R} = 0.68$),

$$k = 5(c-1) \text{ and } c = \frac{\frac{b_1}{b_0} + 0.8}{1.8} \qquad (13.4.14)$$

where: b_1 — maximum width of blade.

$\frac{b}{b_0}$ may be as follows:

$$\frac{b}{b_0} = (1-6\xi+5\xi) + c\xi(6-5\xi-\xi^{10}) \qquad (13.4.15)$$

The intermediate section η of blade with medium pitch is equal to 1/3 of blade tip, assuming:

$$\eta = \xi^2(1.5-0.5\xi) \qquad (13.4.16)$$

Assuming that the blade thickness is distributed in a linear:

$$\frac{t}{t_0} = (1-0.9\xi) \qquad (13.4.17)$$

Obtaining:

$$\lambda = \frac{8.58}{0.55 + \frac{b}{b_0}} \qquad (13.4.18)$$

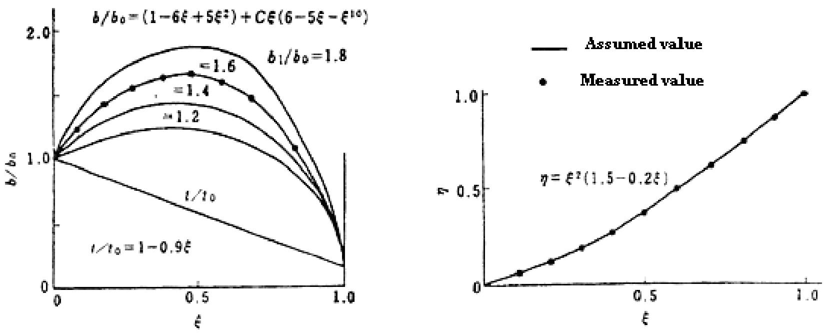


Figure 13.4.8 Curve of Relationship between $\frac{b}{b_0}$ and ξ

The relationship curve between λ and b_1/b_0 is shown in Figure 13.4.9.

Also assuming that $I_0 = c_m b_0 t_0^3$ and $A_0 = c_a b_0 t_0$, I_0, A_0 and formula (13.4.17) is substituted to formula (13.4.11) to obtain:

$$f_{pa} = \frac{\mu}{2\pi}\frac{t_0}{l^2}\sqrt{\frac{E}{\rho_m}} \qquad \text{Hz}$$

where: $\mu = \frac{\lambda}{2\pi}\sqrt{\frac{c_m}{c_a}}$;

c_m— coefficient of sectional moment of inertia;

c_a— coefficient of sectional area.

Obtaining based on the testing:

$$\mu = 0.305\sqrt{\frac{b_0 t_m}{b_m t_0}} \qquad (13.4.19)$$

where: t_m— mean thickness of blade, in cm;

b_m— mean width of blade, in cm.

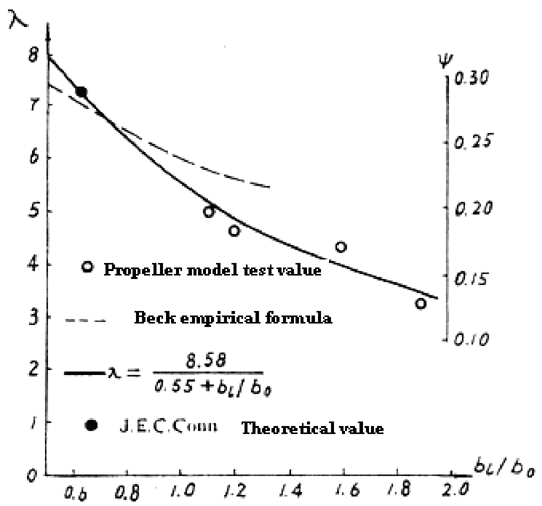


Figure 13.4.9 Curve of Relationship between λ and b_1/b_0

(2) The natural frequency of propeller blade vibration in the water f_{pw} is:

$$f_{pw} = f_{pa}\sqrt{\frac{1}{1+\varepsilon}} \qquad \text{Hz} \tag{13.4.20}$$

where: $\varepsilon = \dfrac{J\pi\rho_w b_0}{4c_a\rho_m t_0} r$ — coefficient of entrained water mass, see Figure 13.4.10;

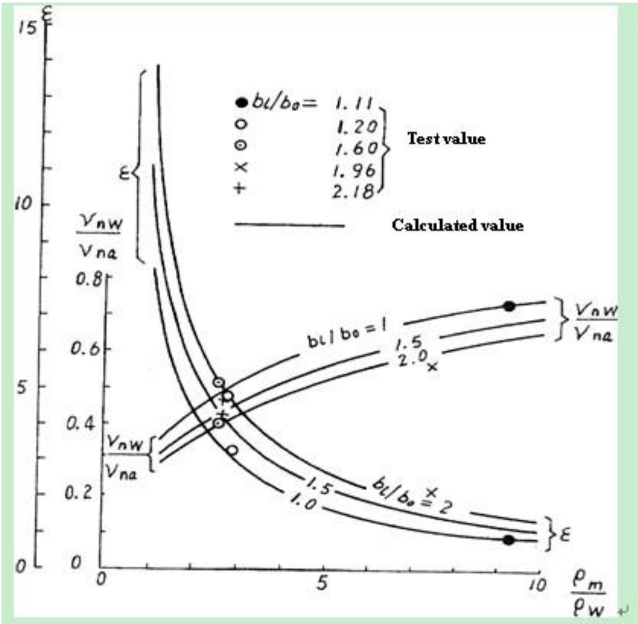


Figure 13.4.10 Coefficient of Entrained Water Mass

ρ_w — density of water, in kg/cm^3;

r— coefficient, obtained from five propeller model test results:

$$r = 2.61\frac{b_1}{b_0}$$

$J = 1 - 0.485\sqrt[3]{\dfrac{b_1}{l}}$ — correction coefficient of three direction flow.

13.5 Natural Frequency of Mast Vibration

13.5.1 In general, the natural frequency of mast structure system calculated by finite element method could adopt the beam model, if necessary, three-dimensional shell model may be adopted for barrel-typed mast structure, particular attention is to be given to correctly simulate the fixture of mast root.

13.6 Finite Element Calculation for Inherent Vibration Characteristics of Local Structures

13.6.1 The local structure vibration analysis include analytical method, empirical formula estimation method and finite element method. For the complicated local structures, it is difficult to use the analytical method, and the empirical formula estimation method is also limited to use for the simple grillages, etc., with homogeneous loads due to representing capability of model and restriction of calculation accuracy.

13.6.2 The local structures onboard ships, such as girder, plate, grillage, shaft rack, shaft bracket, etc., are not isolated, but connected with the other structures with their elastically fixed boundary conditions, so the natural frequency obtained from the above-mentioned calculation is similar, in order to take the consideration of influences of the adjacent structures, it is recommended that the adjacent structures are to be included for calculation in the model by finite element method, then to segregate the natural frequency of the designated structures.

13.6.3 For the influence of entrained water, it may be calculated by the simplified formula, then to distribute homogeneously on the contacting surface of water. For the structures contacting with water, the natural frequency may be calculated by fluidsolid coupling method.

13.6.4 For the large space or large platform above the propeller, multiple order natural frequency of vibration is to be calculated by finite element method.

13.6.5 Where the helicopter platform is located in stern and its fore end is connected with main deck, several columns are provided for supporting beneath the platform. The local grillage model of helicopter platform is to be stimulated according to actual plate-girder combined conditions, the finite element model is shown in Figure 13.6.1, the freedom degree of z direction in way of the above-mentioned columns and side supportings is to be restrained, while the freedom of *x*, *y*, *z*, θ_x, θ_z in way of connection of fore end bulkhead to be restrained.

The natural frequency of helicopter platform and resolution of resonance for a certain tanker are as shown in Table 13.6.1, the inherent mode shape as shown in Figures 13.6.2 to 13.6.5.

The rated speed of main engine is 83 r/min, and the number of propeller blades is four. As seen from Table 13.6.1, 1st order twice (8 times) blade order resonance is induced at the rated speed of 81 r/min, and 4th order four times (16 times) blade order resonance induced at the rated speed of 79 r/min. Therefore, measures are to be taken for prevention.

Natural Frequency of Platform Vibration and Resonance Speed **Table 13.6.1**

Order	Natural frequency(Hz)	Resonance speed induced by propeller exciting force n (r/min)				Mode shape
		1 times blade order	Twice blade order	3 times blade order	4 times blade order	
1	10.832	163	81	54	41	Figure 13.6.2
2	19.014	286	143	95	71	Figure 13.6.3
3	19.823	297	189	99	74	Figure 13.6.4
4	21.081	316	15 8	105	79	Figure 13.6.5

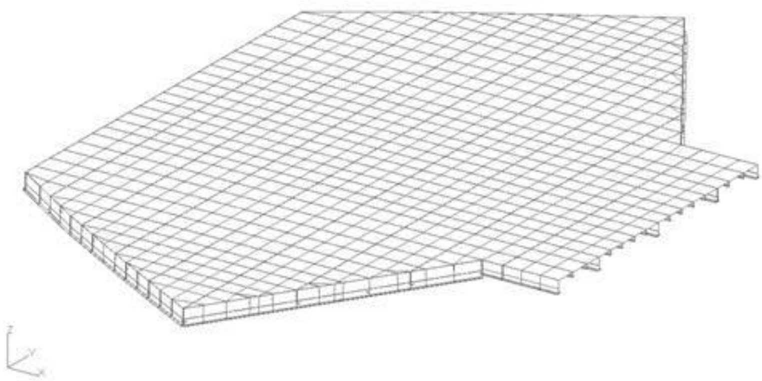


Figure 13.6.1 Helicopter Platform Model

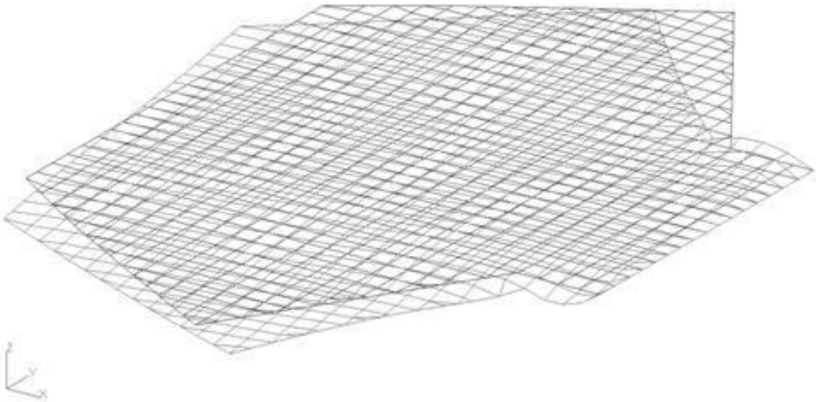


Figure 13.6.2 1st order Mode Shape of Helicopter Platform (f=10.832 Hz)

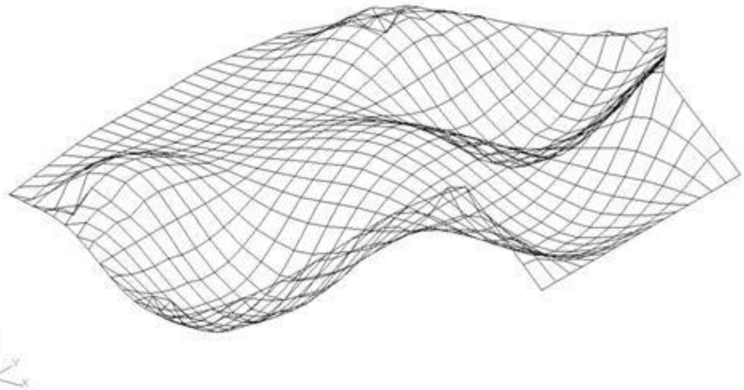


Figure 13.6.3 2nd Order Mode Shape of Helicopter Platform (f=19.014 Hz)

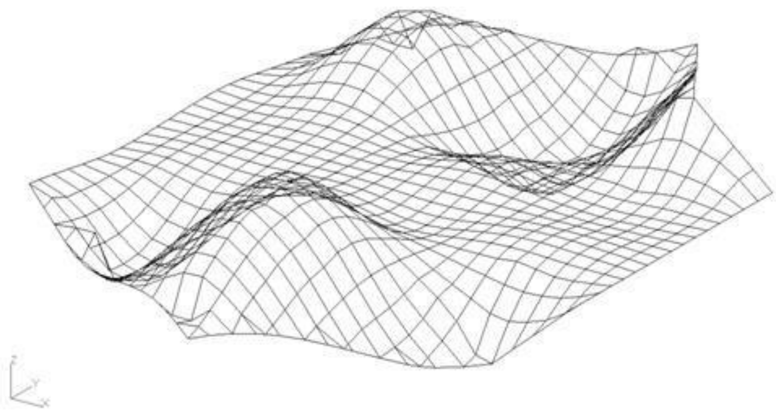


Figure 13.6.4 3rd Order Mode Shape of Helicopter Platform (f=19.823 Hz)

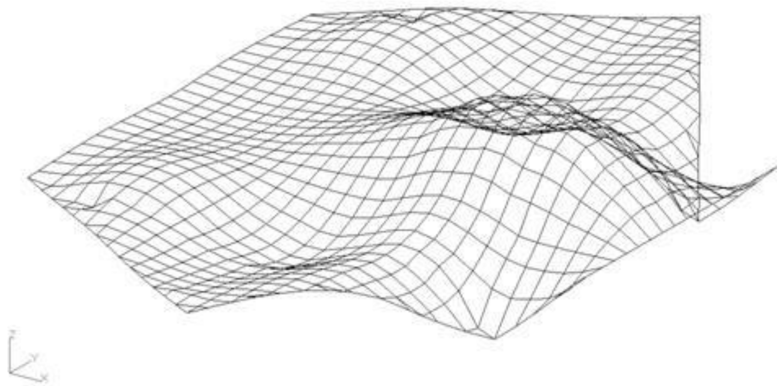


Figure 13.6.5 4th Order Mode Shape of Helicopter Platform (f=21.081 Hz)

13.7 Design Criteria

13.7.1 The effective means to prevent or reduce the resonance response of superstructure vibration is to avoid the resonance caused by the minimum natural frequency and main exciting frequency within the range of 85% to 100% maximum running speed.

13.7.2 For large space or large platform structure (with the area exceeding $40m^2$) above the propeller, the minimum natural frequency is generally to be more than 1.2 ×2 ×number of blades ×exciting frequency at the rated speed.

13.7.3 The correctness of natural frequency calculation for local structure vibration is related to the methods used, hence, the error range of natural frequency calculation is to be taken into consideration in the design.

Chapter 14 MEASUREMENT FOR VIBRATION

14.1 Introduction

14.1.1 Purpose of vibration measurement onboard

(1) To confirm the ship's vibration characteristics to be in compliance with the requirements of the Guidelines and the rules for classification through full-scale vibration measurements.

(2) In the case of adverse vibration, or not complying with the requirements of the Guidelines and the rules for classification by the measurements, the causes of such vibration are to be analyzed and necessary vibration damping measures are to be taken.

14.1.2 For ships applying for class notation of HAB (VIB), the vibration measurement onboard is to comply with the requirements of this Chapter.

The main international standards referenced are:

(1) ISO6954 Mechanical vibration — Guidelines for the measurement, reporting and evaluation of vibration with regard to habitability on passenger and merchant ships;

(2) ISO4868 Code for the measurement and reporting of local vibration data of ship structures and equipment.

14.1.3 For ships applying for the class notations of VIB (S) or VIB(M) or VIB, the vibration measurement onboard is to comply with the requirements of this Chapter.

The main referenced international standards are as follows:

(1) ISO4868 Code for the measurement and reporting of local vibration data of ship structures and equipment;

(2) IS0 10816-6 Evaluation Of Machine Vibration by Measurements on Non-rotating Parts - Reciprocating Machines with Power Rating above 100 kW.

14.1.4 For ships classed with ISC, shafting vibration measurements are to comply with the requirements of this Chapter.

14.1.5 For ships applying for ISC class notation of (VIB), the relevant requirements of Chapter 15 of the Guidelines are to be additionally complied with.

14.1.6 The vibration measurements onboard ships are to be carried out according to the agreed or approved Shipboard Vibration Measurement Program. However appropriate adjustment may be made in accordance with the actual cases with the agreement of the party concerned.

14.2 Requirements of Vibration Measurement

14.2.1 Contents of vibration measurement onboard:

(1) ship vibration measurement;

(2) mechanical vibration measurement;

(3) engine frame vibration measurement;

(4) shafting vibration measurement;

(5) habitability vibration measurement.

14.2.2 Methods of vibration measurement onboard

(1) single item measurement;

(2) synchronous measurement of related project;
(3) synchronous measurement of general project.

14.2.3 Ship vibration measurement

(1) Ship vibration measuring instruments:

① to choose the electronic measuring system with multi channels, which is capable of maintaining the records for a long period of time and consists of sensor, amplifier, filter, recorder, etc.;

② to have the sufficient width of frequency range and amplitude linearity to meet the requirements of frequency and amplitude for the measured part and be suitable for the environmental conditions onboard ships, such as temperature, humidity, noise, etc.;

③ metrological verification and calibration for sensitivity, amplitude frequency characteristic and amplitude linearity of the instruments are to be carried out periodically, generally not to exceed two years, as to maintain the accuracy of instruments in a specified range;

④ the sensor is to be installed and secured, and not to be moved throughout the measuring process;

⑤ where the pulse device is installed on main engine or screwshaft, the pulse signal is to be relative to the position of top dead center of the first cylinder of main engine or a blade of propeller;

⑥ under the condition of complying with the measurement requirements, an electronic instrument for single point measurement or handheld mechanical vibration measurement instrument may be used.

(2) Parameters of ship vibration measurement (where applicable):

① amplitude: displacement, in mm; velocity, in mm/s; acceleration, in mm/s^2; strain;

② frequency, in Hz.

14.2.4 Mechanical and shafting vibration measurement

(1) Mechanical and shafting vibration measuring instruments:

① to correctly reflect the amplitude or deformation in way of the measured point;

② the instruments are to be calibrated periodically, generally not to exceed one year, so as to maintain the accuracy of instruments in a specified range;

③ the measuring instrument and system generally consist of sensor, amplifier, recorder, monitoring indicator, etc.;

④ to have wider frequency range, the allowable error of straight section of frequency response is with the range of ±10％ and being suitable for the environmental conditions onboard ships, such as temperature, humidity, noise, etc.;

⑤ the sensor is to be installed and secured, and not to be moved throughout the measuring process.

(2) Parameters of mechanical and shafting vibration measurement (where applicable):

① amplitude: displacement, in mm; velocity, in mm/s; acceleration, in mm/s^2;

② frequency, in Hz.

14.2.5 Measurement conditions

(1) The ship is to be in ballasted or full-loaded condition and the propeller to be fully immersed in

the water.

(2) The water depth is generally not to be less than four times the mean draught, if the ship always navigates at the shallow water, the vibration measurements may also be carried out in its service areas.

(3) The sea condition is not to be more than 3 scale, and the wind is not to be more than Beaufort wind level 3.

(4) The ship is to maintain in a straight course as practicable as possible and the change of rudder angle is to be less than ±2°.

(5) Unless otherwise specified, the main engine is to run at 90% to 100% maximum service speed and be maintained in a stable state, all of the other machineries are to be in normal running state.

(6) For offshore engineering ships with dynamic positioning system, the side thrusters are to be maintained under a working condition as specified in contract or with minimum 40% of power.

(7) If the vibration measurements are not carried out under the above-mentioned conditions, remarks are to be made in the measurement reports.

14.3 Measurement for Hull Vibration

14.3.1 Measuring position and direction

(1) The measuring points are to be considered to arrange in way of the positions with greater amplitude, for large space or large platform (exceeding 40 m^2), as there may be natural frequency of vertical vibration above 2nd order, several measuring points are to be arranged in combination with the corresponding mode shapes.

The 2nd order mode shape of helicopter platform vertical vibration is shown in Figure 13.6.3, the frequency being 19.014 Hz; the 3rd mode shape of helicopter platform vertical vibration is shown in Figure 13.6.4, the frequency being 19.823 Hz; the maximum deformations are not located in the centers of platforms. In the measurement, additional measuring points may be arranged according to the vibration situation, and are to be arranged in way of the position with maximum deformation.

(2) All of the measuring points are to be arranged on the rigid supporting points.

(3) In order to find the relationship between hull vibration characteristic and vibration source, the measuring position and direction are shown in Table 14.3.1-1; the arrangement of measuring points for a certain large oil tanker is shown in Figure 14.3.1.

Vibration Measuring Points and Measuring Directions for Vibration Source

Table 14.3.1-1

No.	Space	Measuring position	Measuring direction
1	Upper (main) deck	Intercept of longitudinal centerplane and front edge bulkhead	Vertical, transverse and longitudinal
2	Stern	In way of aft end of longitudinal centerplane of deck	Vertical, transverse and longitudinal
3	Bridge deck	Intercept of longitudinal centerplane and front edge bulkhead	Vertical, transverse and longitudinal

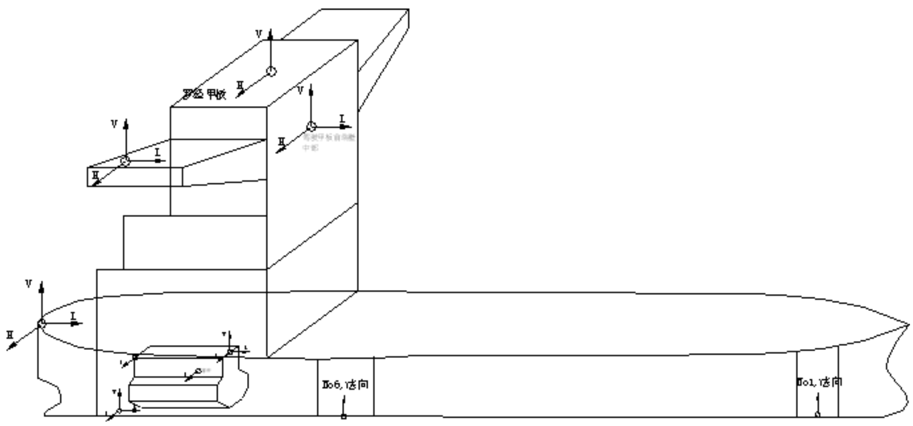


Figure 14.3.1 Arrangement of Vibration Measuring Points for a Certain Large Oil Tanker

(4) Ship's habitability (vibration) criteria come from ISO 6954, the space range required by habitability (vibration) criteria is same as that required by comfort (vibration), including the passenger space, crew space and working space. The measuring spaces with their positions and directions for habitability (vibration) criteria are shown in Table 14.3.1-2.

Measuring Positions and Directions for Habitability Vibration Table 14.3.1-2

No.	Space	Measuring position	Measuring direction
1	Crew's cabin	Central	Vertical, transverse and longitudinal
2	Navigation bridge, radio room, chart room	Central	Vertical, transverse and longitudinal
3	Crew's public space, mess room, meeting room	Central	Vertical, transverse and longitudinal
4	Clinic	Central	Vertical, transverse and longitudinal
5	Office	Central	Vertical, transverse and longitudinal
6	Machinery shop	Central	Vertical
7	Engine room control room	Central	Vertical

(5) The measuring positions and directions for hull structure vibration are shown in Table 14.3.1-3, and the measuring points are to be arranged in accordance with the requirements of Table 14.3.1-2.

Measuring Positions and Directions for Hull Structural Vibration Table 14.3.1-3

No.	Space	Measuring position	Measuring direction
1	Stern	In way of stern of deck longitudinal centerplane	Vertical, transverse and longitudinal
2	Superstructure	Central	Vertical, transverse and longitudinal
3	Local structure	Position may cause vibration	Vertical, transverse and longitudinal
4	Deck girder	Sufficient measuring points	Vertical, transverse and longitudinal
5	Foundation structure	Foundation	Vertical, transverse and longitudinal
6	Steering gear room	Sufficient measuring points	Vertical, transverse and longitudinal
7	Hold/tank structure	Central	Vertical, transverse and longitudinal
8	Funnel	Top end	Vertical, transverse and longitudinal
9	Radar mast	Top end	Vertical, transverse and longitudinal

(6) Notwithstanding the requirements of Table 14.3.1-2 and Table 14.3.1-3, not all vertical, transverse and longitudinal vibrations in all positions are to be measured, transverse and longitudinal vibrations may not be required provided sufficient measuring points are arranged representing vibration characteristics.

(7) For ships of less than 65 m in length, only vertical vibration may be measured.

14.3.2 Measuring methods

(1) The ship vibration may be measured independently, where the vibration source needs to be analyzed, it is to be measured together with the frame vibration, shafting torsional vibration, shafting longitudinal vibration and shafting whirling vibration.

(2) In order to find the relationship between hull vibration characteristic and vibration source, the main engine is to start from the minimum steady speed to the rated speed with measurement according to the various speed grades. The speed interval is to be properly reduced approximate to the resonance speed, as to make amplitude-speed curve, as shown in Figure 14.3.2.

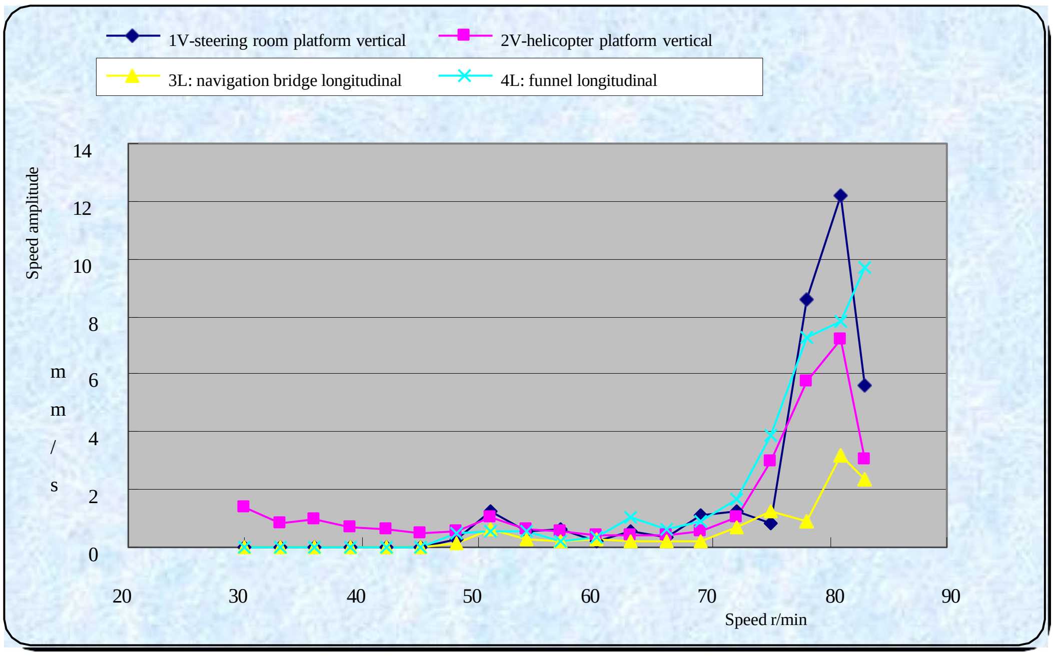


Figure 14.3.2 Amplitude-speed Curve for 4th Order Propeller Vibrations

14.3.3 Measurement reports

(1) Data collection:

① The recording interval of computer signals is not to be less than 60s.

② When the measuring data are recorded, the pulse signals and vibration response signals are to be synchronously recorded.

(2) Data analysis:

① In computer analysis, a random truncation method or truncation of waveform of time domain in the whole period method may be used.

② In DFT spectrum analysis, where the sampling time $\triangle T$ is generally 4s, the frequency resolution $\triangle f$ is to be calculated by:

$$\Delta f = \frac{1}{\Delta T} \quad \text{Hz}$$

where: $\triangle T$— sample time, in s.

③ In FFT spectrum analysis, where the sampling frequency is f_s and the number of sampling points is N, the frequency resolution $\triangle f$ is to be calculated by:

$$\Delta f = \frac{f_s}{N}$$

(3) In manual analysis, a 10% maximum value method is to be used, e.g. 10 maximum amplitude values are selected from the 100 baseband waveforms and the averaged value is treated as the maximum one.

(4) Form of shipboard vibration measurement report is given in Appendix 1.

14.4 Measurement for Mechanical Vibration

14.4.1 Measuring positions and directions

(1) The measuring points are to be arranged in the typical positions with stronger stiffness. The number of measuring points depend on the size of measured object, less measuring points taken for smaller machinery and more measuring points taken for larger one.

(2) The measuring positions and directions for mechanical vibration are shown in Table 14.4.1.

Measuring Positions and Directions for Mechanical Vibration Table 14.4.1

No	Equipment	Measuring position	Measuring direction
1	Low-speed diesel engine	Fore and aft ends of the frame top	Vertical, longitudinal and transverse
2	High and medium-speed diesel engine	Fore and aft ends of the frame top	Transverse
3	Generator driven by diesel engine and motor for propulsion	Fore and aft ends of bearing	Transverse
4	Generator driven by turbine	In way of bearing	Any directions
5	Supercharged engine	On the top of compressing end	
6	Turbine	In way of bearing	Any directions
7	Main driving gear box	In way of base and output bearing	Any directions
8	Shafting bearing	In way of height of shaft centerline	Transverse or vertical
9	Motor, separator, hydraulic driving motor, fan	In way of bearing	Any directions
10	Screw type or centrifugal compressor	In way of bearing	Any directions
11	Reciprocating compressor	In way of bearing	Any directions

14.4.2 Measuring methods

(1) The mechanical vibration may be measured independently, where the ship vibration source needs to be analyzed, a synchronous measurement is to be carried out.

(2) Where vibration source is to be investigated or sought, the main engine is to start from the minimum steady speed to the rated speed with measurement according to the various speed grades. The speed interval is to be properly reduced approximate to the resonance speed.

14.4.3 Measurement reports

(1) The mechanical measurement report is to include harmonic orders of each measuring point at different speeds, vibration displacements, speeds or accelerations in different directions, as weill as vibration criteria, etc., with curves indicated.

(2) Form of mechanical vibration measurement report is given in Appendix 2, and Form of frame vibration measurement report is given in Appendix 3.

14.5 Measurement for Shafting Torsional Vibration

14.5.1 Arrangement of measuring points

(1) For instruments measuring angular displacement, the measuring points are generally to be arranged in way of the free ends of diesel engine.

(2) In full-scale measurement, where the arrangement of measuring points in way of the free ends of crankshaft of diesel engine is difficult, the measuring points may be arranged in way of shafting with the larger relative amplitude.

(3) Where two resonance speeds with different mode shapes are similar and interferes with each other, the measuring points are generally to be arranged in other positions in addition to the free ends, as to separate the amplitudes of various mode shapes.

(4) Where torsional vibration stress is measured directly by strain gauge, the measuring points are generally to be arranged in way of the positions close to the nodes of intermediate shaft or screwshaft.

14.5.2 Measuring methods

(1) The torsional vibration may be measured independently, where the other vibration source needs to be analyzed, a synchronous measurement is to be carried out.

(2) The measuring system is to be completely inspected after installation to ensure the normal measuring working. The stimulation electronic measuring instrument is to be calibrated before measurement for each voyage, and its sensitivity is to be set to correct position. Where a waveform recording device is used, calibration signal is to be recorded for quantitative analysis. The calibration is to be made again after measurement of each voyage.

(3) In the measurement, the main engine is to start from the minimum steady speed to the rated speed with measurement according to the various speed grades. In general, 5 to 10 r/min is for low-speed engine, 10 to 30 r/min for medium-speed engine and 30 to 50 r/min for high-speed engine. The speed interval is to be properly reduced approximate to the resonance speed.

(4) For shafting installed with high elastic coupling or driven by gears, particular attention is to be given to preventing the elastic coupling damage or severe toothstrike on the surface of gears caused by excessive vibration torque at the resonance speed. If the above-mentioned phenomenon occurs in the measurement, it is to be rapidly surpassed.

(5) In order to obtain the stable data, measurement is to be carried out after the speed of main engine is steady.

14.5.3 Measurement reports

(1) When the frequency error between actual test and theoretical calculation is generally less than 5%, the tested amplitude or torsional vibration stress may be used to calculate the torsional vibration stress of each shaft and vibration torque in way of gear box, elastic coupling, etc., by the calculated mode shape.

(2) For shafting with high elastic coupling or damper, the shaft stress and component torque are to be calculated by mode shape obtained by synthesis method. The relative damping parameters may be adjusted to make the calculated amplitude in way of measuring point consistent with the measured one, the vibration characteristics of the system may be evaluated by the forced vibration calculation results.

(3) The measurement report is to include harmonic orders, amplitude or strain, natural frequency of each measuring point at different speeds, torsional vibration stress of each shaft and vibration

torque of each elastic coupling or gear, with curves of stress/torque and speed together with plotted allowable values. Form of torsional vibration measurment report is given in Appendix 4.
(4) The restricted speed range may be determined based on measured results in accordance with the relative rules requirements.
(5) The determined restricted speed range is to be indicated in red on the tachometers in navigation bridge and on main engine control console by shipyards and placards are to be provided in way of the control positions.

14.6 Measurement for Shafting Longitudinal Vibration/Whirling Vibration

14.6.1 Arrangement of measurement
(1) The longitudinal vibration measuring points are to be arranged in way of the free ends of crankshaft of diesel engine.
(2) In whirling vibration measurement, where non-contact sensor is used, vertical and transverse measuring points are generally to be arranged in way of the positions with larger amplitude. Where strain gauge is used, the measuring points are to be arranged in way of the positions with larger bending stress.

14.6.2 Measuring methods
The same as in 14.5.2.

14.6.3 Measurement reports
(1) The measurement report is to include harmonic orders, amplitude, natural frequency of each speed, with curves of amplitude and speed together with plotted allowable values for longitudinal vibration. Form of longitudinal/whirling vibration measurement report is given in Appendix 4.
(2) The restricted speed range may be determined based on measured results in accordance with the relative rules requirements.
(3) The determined restricted speed range is to be indicated in red on the tachometers in navigation bridge and on main engine control console by shipyards and placards are to be provided in way of the control positions.

14.7 Report of Vibration Measurement

14.7.1 Submission of report
(1) After completion of the full-scale vibration measurement, the ship vibration measurement organization is to prepare corresponding measurement report in combination with the Analysis Report of Shipboard Vibration Calculation, and submit it to the parties concerned.
(2) For ships classed with ISC and applying for class notation, the vibration measurement report is to be confirmed and approved by ISC.

14.7.2 Contents of vibration measurement report
In addition to the above-mentioned specific requirements of this Chapter, the vibration measurement report is at least to include:
(1) name and signature of vibration/noise measurement organization;

(2) ship's particulars;
(3) description of environmental condition, ship condition and measuring instruments;
(4) summary (at least including measurement basis, measurement conditions, applicable standards and measurement conclusions);
(5) arrangement of measuring point (equipment and diagram);
(6) analysis results of vibration/noise measurement;
(7) curve of amplitude-speed (typical position, if any);
(8) main original measurement record.

14.7.3 Confirmation and signature of measurement report

For ships classed with ISC and applying for VIB class notation, the surveyor is to confirm the relative project and sign together with the parties concerned:
(1) to confirm the measurement conditions;
(2) to confirm the measurement project and measuring points;
(3) to confirm the measured values;
(4) to sign in the corresponding recorded documents.

14.7.4 Conservation of original time domain signals

(1) The measurement organization is to keep the original time domain signals of vibration measurement in order to provide for information.
(2) Where check of the original data of vibration measurement is required by ISC, the measurement organization is to submit the original time domain signals.

Appendix 1 REPORT OF VIBRATION MEASUREMENT

Report of Vibration Measurement

1 Basic data

Basic Data

Name of ship		Category		Service area	
Manufacturer		Unit		Delivery year/month	
Hull			**Main engine**		
Length between perpendiculars L_{pp} (m)			Manufacturer		
Molded width B (m)			Type ×Number		
Molded depth D(m)			Rated power (kW)		
Full-loaded draught d(m)			Rated speed (r/min)		
Full-loaded displacement △(t)			Diameter of cylinder × stroke ×number of cylinder		
Dead weight DW (t)			Firing order		
Block coefficient C_b			**Generator**		
Propeller			Manufacturer		
Type			Type ×Number		
Speed (r/min)			Rated power (kW)		
Number of propeller ×Number of blade			Rated speed (r/min)		
Diameter of propeller (m)			Diameter of cylinder × stroke ×number of cylinder		
Expanded area ratio			Firing order		

2 Environmental conditions, measurement conditions and measuring instruments

Environmental conditions, measurement conditions and measuring instruments

Environmental conditions			
Measured water area		Water depth (m)	
Wind power		Wave scale	
Ship conditions at measurement			
Displacement (t)		Fore draught (m)	
Aft draught (m)		Mean draught (m)	

<table>
<tr><td colspan="5">Mechanical conditions at measurement</td></tr>
<tr><td>Main engine</td><td>Number</td><td></td><td>Speed (r/min)</td><td></td></tr>
<tr><td>Generator</td><td>Number</td><td></td><td>Speed (r/min)</td><td></td></tr>
<tr><td>Ventilator in engine room</td><td>Number</td><td></td><td>Speed (r/min)</td><td></td></tr>
<tr><td colspan="5">Measuring instruments and personnel</td></tr>
<tr><td rowspan="3">Measuring instruments</td><td>Sensor</td><td></td><td>Amplifier</td><td></td></tr>
<tr><td>Recorder</td><td></td><td>Analytical instruments</td><td></td></tr>
<tr><td>Calibrated date</td><td></td><td></td><td></td></tr>
<tr><td>Measuring personnel</td><td colspan="2"></td><td>Measuring date</td><td></td></tr>
</table>

3 Summary (at least including: measurement basis, measurement conditions, applicable standards and measurement conclusions)

4 Arrangement of measuring points (name of space and diagram)

5 Analysis results of shipboard vibration measurement

Analysis results of shipboard vibration measurement

No.	Position of measuring point/name of space	Speed of main engine (r/min)	Frequency (Hz)	Harmonic order	Direction (V/T/L)	Speed (mm/s) or displacement (mm) or acceleration (mm/s^2)	Criteria

6 Curve of amplitude-speed (typical position)

7 Main original measurement records

Appendix 2 REPORT OF MECHANICAL VIBRATION MEASUREMENT

Report of Mechanical Vibration Measurement

1 Basic data

Name of ship______________________________ Type of machinery_____________________

Measuring organization or personnel____

Measuring date___________________________ Measuring place________________________

Measuring purpose____

Measuring instrument___________________________ Calibrated date____________________

2 Summary (at least including: measurement basis, measurement conditions, applicable standards and measurement conclusions)

3 Arrangement of measuring points (name of space and diagram)

4 Analysis results of mechanical vibration measurement record

No.	Name of equipment	Measuring position	Speed of main engine (r/min)	Frequency (Hz)	Harmonic order	Displacement (mm)	Speed (mm/s)	Criteria

5 Main original measurement records

Appendix 3 REPORT OF FRAME VIBRATION MEASUREMENT

Report of Frame Vibration Measurement

1 Basic data

Name of ship______________________________ Type of machinery_____________________

Measuring organization or personnel____

Measuring date___________________________ Measuring place________________________

Measuring purpose____

Measuring instrument___________________________ Calibrated date____________________

2 Summary (at least including: measurement basis, measurement conditions, applicable standards and measurement conclusions)

3 Arrangement of measuring points (diagram)

4 Analysis results of frame vibration measurement record

No.	Position of measuring point	Speed of main engine (r/min)	Frequency (Hz)	Harmonic order	Measured amplitude (mm)			Criteria
					Vertical	Transverse	longitudinal	

type L natural frequency:

type X natural frequency:

5 Curve of amplitude-speed

6 Main original measurement records

Appendix 4 REPORT OF SHAFTING VIBRATION MEASUREMENT

Report of Torsional Vibration/Longitudinal Vibration/Whirling Vibration Measurement

1 Basic data

Name of ship______________________________ Type of machinery_____________________

Measuring organization or personnel____

Measuring date___________________________ Measuring place________________________

Measuring purpose____

Measuring instrument___________________________ Calibrated date____________________

2 Summary (at least including: measurement basis, measurement conditions, applicable standards and measurement conclusions)

3 Arrangement of measuring points (diagram)

4 Analysis results of vibration measurement record

Measured natural frequency:

Calculated natural frequency:

5 Curve of torsional vibration stress (torque)/amplitude/stress – speed and allowable value

6 Main original measurement records

Chapter 15 CRITERIA OF VIBRATION

15.1 Introduction

15.1.1 For the purpose of improving the crew's accommodation condition and working condition onboard ships, ISO 6954-1984 Mechanical Vibration and Shock – Guidelines for the Overall evaluation of Vibration in Merchant Ships and ISO 6954-2000 Mechanical Vibration — Guidelines for the Measurement, Reporting and Evaluation of Vibration With Regard to Habitability on Passenger and Merchant Ships are the habitability criteria applicable to vibration acceptable to crew and passengers. From point of view of vibration influence on personnel, the habitability criteria may ensure the comfort and health for personnel, where the vibration limits are not exceeded, a health damage will normally not occur.

15.1.2 Comfort vibration criteria for passenger's and crew's accommodation spaces and working spaces are the description of different grades of comfort environment as required in classification societies' rules based on the habitability criteria.

15.1.3 The vibration habitability and vibration comfort criteria do not reflect the issue of fatigue crack induced by hull structural vibration or the issue of fatigue damage or accelerated wear of movable component caused by mechanical vibration. Therefore, the issues for hull structure vibration criteria and mechanical vibration criteria are put forward.

15.2 Habitability Vibration Criteria

15.2.1 Habitability criteria for crew and passengers by ISO 6954-1984:

ISO 6954-1984 may be used for evaluation of the vibration of normal working and accommodation spaces of hull and superstructures for personnel, as shown in Table 15.2.1-1 and Figure 15.2.1, which, however, is not regarded as the standards for acceptance or survey of machinery and equipment. ISO 6954-1984 takes the peak values of displacement, speed or acceleration of each measuring point within the frequency range of 1 Hz～100 Hz as the evaluation indexes, regardless of vertical, longitudinal or transverse vibration, and the evaluation may be conducted by means of curves.

In the curve, the upper and lower margin lines are the constant acceleration line when ≤5 Hz, it is, or being the constant velocity line when ≥5 Hz. The measured vibration response under the lower margin line is the minor vibration and indicates that the ship's vibration characteristics are in good state; the measured vibration response above the upper margin line is the vibration unacceptable and unsatisfactory; the banded zone between the two margin lines is the vibration which may be generally felt but still acceptable.

Such criteria evaluate the shipboard vibration based on maximum repetitive value, where the measured value is *rms*, it is to be converted to the equivalent maximum repetitive value in accordance with the following formula, as to compare with the curve:

$$\text{The maximum repetitive value} = (C_F\sqrt{2}) \times rms \qquad (15.2.1)$$

where: $C_F\sqrt{2}$ — equivalent to crest factor (C_F = 1.0 means pure steady sinusoidal vibration).

The corresponding conversion coefficient C_F is to be determined by measurement or assuming $C_F = 1.8$.
The evaluation process is different from ISO 2631-2, hence, causing ISO 6954 to be revised.

Habitability Criteria Required by ISO 6954-1984 **Table 15.2.1-1**

Curve	Frequency range	
	1 Hz～5 Hz	5 Hz～100 Hz
Upper margin	Peak acceleration = 285 mm/s^2	Peak acceleration = 9 mm/s
Lower margin	Peak acceleration = 126 mm/s^2	Peak acceleration =4 mm/s

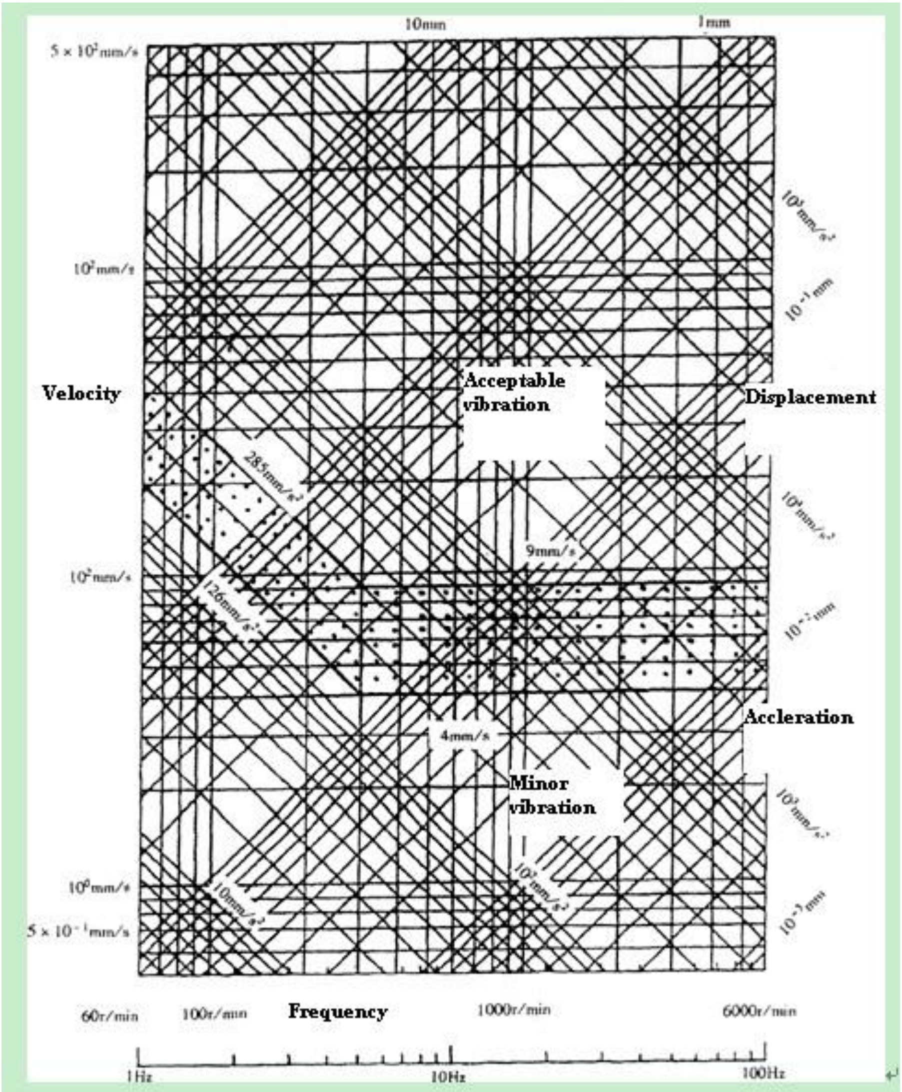


Figure 15.2.1 Vibration Evaluation Criteria Required by ISO 6954-1984

15.2.2 Habitability criteria for crew and passenger required by ISO 6954-2000:
(1) Application
ISO 6954-2000 reflects new research results on the vibration sensitivity of human beings, which is applicable to criteria of vibration evaluation related to habitability onboard passenger ships and

merchant ships. The evaluation of motion sickness caused by low frequency motion may be referred to the other international standards.

(2) Habitability vibration criteria

The vibration levels in ISO 6954-2000 mean the root-mean-square of vibration frequency weighted acceleration or the root-mean-square of velocity within the frequency range of 1 Hz to 80 Hz, which are called full frequency weighted RMS acceleration or velocity. The full frequency weighted RMS value is no longer a peak one.

Three different zones onboard, such as Zone A (passenger space), Zone B (crew space) and Zone C (working space) are divided by ISO 6954-2000, with the habitability vibration criteria of frequency weighted RMS in different zones shown in Table 15.2.2-1.

Habitability Vibration Criteria of Frequency Weighted RMS within the Range of 1 Hz to 80 Hz in Different Zones **Table 15.2.2-1**

Parameter / Evaluation	Category of zone					
	A (passenger space)		B (crew space)		C (working space)	
	Acceleration (mm/s^2)	Velocity (mm/s)	Acceleration (mm/s^2)	velocity(mm/s)	Acceleration(mm/s^2)	Velocity (mm/s)
A problem if greater than the value	143.0	4.0	214.0	6.0	286.0	8.0
No problem if less than the value	71.5	2.0	107.0	3.0	143.0	4.0

(3) Weighting function

The synthetic weighting curve in ISO 2631-2 is to be used for weighting function, as shown in Table 15.2.2-2 or Figure 15.2.2, the weighting function applies to all directions.

Frequency Weighting within 1/3 Times Frequency Range of 1 Hz to 80 Hz Table 15.2.2-2

Band Hz	Velocity weighting Wv	Acceleration weighting Wa	Band Hz	Velocity weighting Wv	Acceleration weighting Wa
0.20	0.00221	0.0629	10.00	0.869	0.494
0.25	0.00439	0.0994	12.50	0.911	0.411
0.32	0.00870	0.156	16.00	0.941	0.337
0.40	0.0170	0.243	20.00	0.961	0.274
0.50	0.0325	0.368	25.00	0.973	0.220
0.63	0.0589	0.530	31.50	0.979	0.176
0.80	0.0979	0.700	40.00	0.978	0.140
1.00	0.147	0.833	50.00	0.964	0.109
1.25	0.201	0.907	63.00	0.925	0.0834
1.60	0.260	0.934	80.00	0.844	0.0604
2.00	0.327	0.932	100.00	0.706	0.0401
2.50	0.402	0.910	125.00	0.533	0.0241
3.15	0.485	0.872	160.00	0.370	0.0133
4.00	0.573	0.818	200.00	0.244	0.00694
5.00	0.661	0.750	250.00	0.156	0.00354
6.30	0.743	0.669	315.00	0.0995	0.00179
8.00	0.813	0.582	400.00	0.0630	0.00899

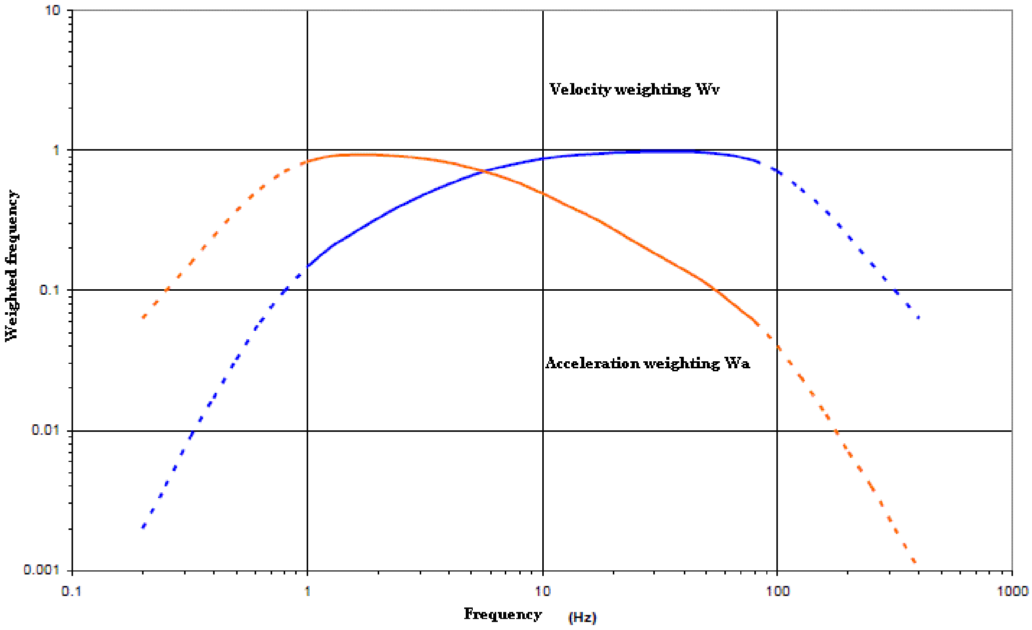


Figure 15-2-2 Weighted Function Curve Required by ISO 6954-2000

(4) Full frequency weighted value

The full frequency weighted value is to be determined by:

$$a_w = \sqrt{\sum_i a_{wi}^2}$$

where: a_w — acceleration or velocity of frequency weighting;

a_{wi} — acceleration or velocity of i times weighting within 1/3 times frequency range.

15.2.3 The general range of above-mentioned zones is as follows:

(1) Passenger space: means the spaces provided for passengers, including public spaces (e.g. mess room, clinic, living room, reading and game room, fitness room, corridor, shop) and leisure zone on weather deck.

(2) Crew space: means the spaces provided for crew, including accommodation space (e.g. cabin, radio room, clinic, office, laundry, mess room, leisure room) and bridge space (including navigation bridge, radar room, monitoring position, bridge wings).

(3) Working space: means the spaces provided for crew to work, including workshop, engine room control cabin.

15.2.4 ISO 6954-2000 may also be applicable to other ships as requested by shipowners.

15.2.5 In application of the habitability criteria of ISO 6954-2000 for crew and passengers, the frequency weighted accelerated velocity RMS is used within the scope of 1 Hz to 5 Hz, and the frequency weighted velocity RMS is used within the scope of 5 Hz to 80 Hz.

15.3 Structural Vibration Criteria

15.3.1 Notwithstanding the compliance with shipboard habitability criteria, the adverse vibration in other zones not covered by the criteria are likely to occur, such as tank structure, or other structural members in stern and engine room, etc., For the purpose of avoiding occurrence of excessive vibration to reduce the risk of structure damage, structural vibration criteria has been provided. The consideration of prevention of structural fatigue cracks induced by vibration is mainly given in structural vibration criteria, while the primary factors affecting the structural fatigue strength are:

(1) vibration mode;

(2) structural details (stress concentration);

(3) material;

(4) welding procedure;

(5) manufacturing processing;

(6) environment (corrosion medium).

15.3.2 The structural vibration parameters apply displacement, velocity or acceleration amplitude (peak values) as characteristics. Displacement or acceleration amplitude is used for vibration frequency within the range of 1 Hz to 5 Hz; velocity amplitude is used for vibration frequency within the range of 5 Hz to 100 Hz.

15.3.3 Unless otherwise agreed, the vertical, transverse or longitudinal vibration displacement or velocity amplitude of each point for the values of hull structural vibration criteria is to be generally controlled within the following range, as shown in Figure 15.3.3:

(1) within the frequency range of 1 Hz to 5 Hz: less than 1.0 mm to be recommended; damage may occur when more than 2.0 mm;

(2) within the frequency range of 5 Hz to 100 Hz: less than 30 mm/s to be recommended; damage may occur when more than 60 mm/s.

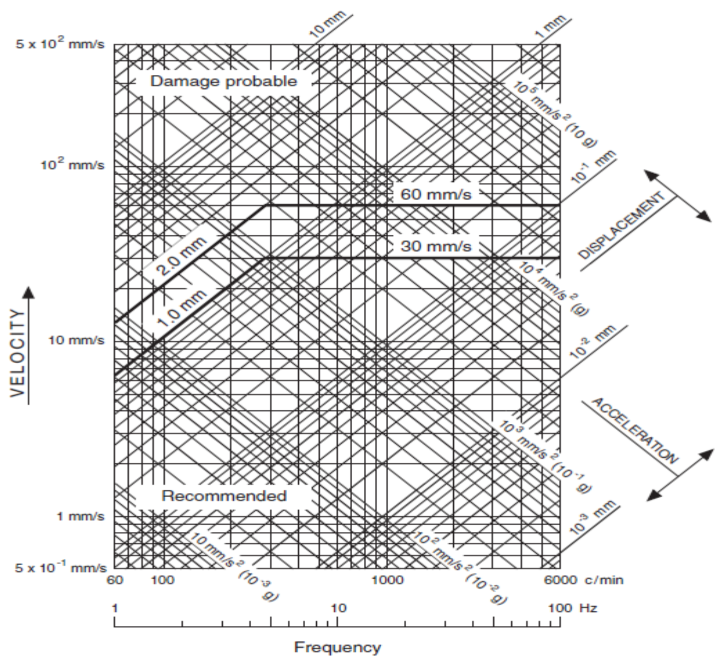


Figure 15.3.3 Structural Vibration Evaluation Benchmark

15.3.4 Unless otherwise agreed, the vertical, transverse or longitudinal vibration parameters of each point for the values of hull structural vibration criteria of high-speed craft, light ships and surface naval ships of more than 35 m in length are to be generally controlled within the range as shown in Table 15.3.4.

Values of Hull Structural Vibration Criteria for High-Speed Craft, Light Ships and Surface Naval Ships **Table 15.3.4**

Zone		Frequency range of 1 Hz to 5 Hz		Frequency range of 5 Hz to 100 Hz
		Acceleration (mm/s^2)	Displacement (mm)	Velocity (mm/s)
Main zone: deck which may be generally accessible by personnel and the structure installed with equipment		160	—	5
Stern zone: deck which may be generally accessible by personnel and the structure installed with equipment		220	—	7
Mast top zone		—	—	15
Other structures not influencing the comfort or skilled operation by personnel and not installing with essential equipment, such as hold/tank, void space, etc.	Steel	—	1.0	30
	Aluminium alloy	—	0.33	10

15.4 Mechanical Vibration Criteria

15.4.1 In order to avoid the fatigue damage of machineries or accelerated wear of moving parts, the mechanical vibration parameters are to be controlled.

15.4.2 The mechanical vibration parameters apply displacement amplitude, velocity amplitude or acceleration amplitude (peak values) as characteristics.

15.4.3 Unless otherwise agreed, the mechanical vibration is not to exceed the values of vibration criteria as shown in Table 15.4.3.

Mechanical Vibration Criteria Table 15.4.3

Equipment	Frequency (Hz) and displacement (mm)	Frequency (Hz) and velocity (mm/s)	Remark
Low-speed diesel engine	1 Hz to 2.4 Hz 1.5 mm for vertical or longitudinal direction; 1.0 mm for transverse direction	2.4 Hz to 100 Hz 10 mm/s for vertical or longitudinal direction; 25 mm/s for transverse direction	See 5.5.3 of the Guidelines
Medium and high-speed diesel engine	—	4.8 Hz to 100 Hz 15 mm/s for fixedly installed; 25 mm/s for elastically installed	
Generator driven by diesel engine and motor for propulsion	—	4 Hz to 100 Hz 18 mm/s	Applying for fixedly and elastically installed
Generator driven by turbine	—	4 Hz to 1000 Hz 7 mm/s	Applying to fixedly installed and elastically installed
Supercharged engine	3 Hz to 4.8 Hz 1.0 mm	4.8 Hz to 26.5 Hz 30 mm/s	
Turbine	—	5 Hz to 1000 Hz 5 mm/s	Applying to fixedly installed and elastically installed
Main driving gear box	—	4 Hz to 1000 Hz 7 mm/s	
Shafting bearing	1 Hz to 2 Hz 0.4 mm	2 Hz to 100 Hz 5 mm/s	
Motor, separator, hydraulic driving motor, fan	—	4 Hz to 200 Hz 7 mm/s	
Screw type or centrifugal compressor	—	4 Hz to 200 Hz 7 mm/s for fixedly installed; 10 mm/s for elastically installed	
Reciprocating compressor	—	4 Hz to 200 Hz 30 mm/s	Applying to fixedly installed and elastically

			installed

15.4.4 Unless otherwise agreed, the mechanical vibration of high-speed craft, light ships and surface naval ships of more than 35 m in length is not to exceed the values of vibration criteria as shown in Table 15.4.4.

Mechanical Vibration Criteria of High-Speed Craft, Light Ship and Surface Naval Ship

Table 15.4.4

Equipment	Frequency and displacement	Frequency and velocity	Remark
Medium and high-speed diesel engine	1 Hz to 4.8 Hz 0.5 mm	4.8 Hz to 100 Hz 15 mm/s	
Generator driven by diesel engine	1 Hz to 3.2 Hz 0.5 mm	3.2 Hz to 100 Hz 10 mm/s	
Generator driven by turbine	1 Hz to 2 Hz 0.4 mm for rigidly installed; 0.8 mm for elastically installed	2 Hz to 100 Hz 5 mm/s for rigidly installed; 10 mm/s for elastically installed	
Supercharged engine	3 Hz to 4.8 Hz 1.0 mm	4.8 Hz to 26.5 Hz 30 mm/s	26.5 to300 Hz 5g for acceleration
Turbine	——	5 Hz to 1000 Hz 5 mm/s	
Gear	——	5 Hz to 1000 Hz 5 mm/s	
Shafting bearing	1 Hz to 2 Hz 0.4 mm	2 Hz to 100 Hz 5 mm/s	
Motor, separator, hydraulic driving motor, fan	1 Hz to 2 Hz 0.4 mm	2 Hz to 100 Hz 5 mm/s	
Screw type or centrifugal compressor	——	4 Hz to 200 Hz 7 mm/s for fixedly installed; 10mm/s for elastically installed	
Reciprocating compressor	1 Hz to 3.2 Hz 0.5 mm	3.2 Hz to 100 Hz 10 mm/s	

Appendix 5 Identification of Vibration Issues and Recommended Remedial Measures on Ships

1. Introduction

The most efficient way to meet vibration criteria is to undertake a vibration analysis and apply appropriate controls during the design stage. After the vessel is built there are less options readily available to rectify vibration problems in hull structures and the cost of fixing such problems is much more expensive in-service than if incorporated into the design from the preliminary design stage. Therefore, careful consideration should be given to incorporating vibration reduction elements during the design stage by either using experience with satisfactory service history or by employing analytical methods.

Having mentioned the above, this appendix is limited to identifying vibration problems in hull structures on newly built or in-service vessels and lists a few common remedial actions to make improvements to address those problems. It is recommended that consideration be given to employing experts in the measurement, evaluation and resolution of issues should vibration problems in hull structures be present.

This Recommendation may be utilized to supplement the IMO GBS requirements, i.e. SOLAS II-1/Reg. 3-10, the functional requirement 3.2.1.11 in Resolution MSC 296(87) GBS verification guidelines, and ISC RULES FOR CLASSIFICATION OF SEA-GOING STEEL SHIPS, Vol. 6, Part 9-1 Ch 10 Sec 2 [3.1.1].

2. Terminology

Damping: The dissipation of energy with time or distance. In this document, damping generally refers to dissipation of vibrating energy in structures.

Hard Mounting: Rigid attachment of a machine to its sub-base or foundation.

Isolation: The coupling of a vibrating structure (e.g., machine) to another structure (e.g., foundation or hull) by means of resilient or compliant supports that prevent the transmission of the vibration from the vibrating structure to the coupled structure.

Resonance: (1) The phenomenon of amplification of a free wave or oscillation of a system by a forced wave or oscillation of exactly equal period. The forced wave may arise from an impressed force upon the system or from a boundary condition. The growth of the resonant amplitude is characteristically linear in time. (2) Of a system in forced oscillation, the condition that exists when any change, however small, in the frequency of excitation causes a decrease in the response of the system.

Vibration: The variation with time of the magnitude of a quantity which is descriptive of the motion or position of a mechanical system, when the magnitude is alternately greater and smaller than some average value. Also referred to as structure-borne noise.

Vibration Level: Measure of the amplitude of vibration. Most commonly given as acceleration in mm/s2 or velocity in mm/s.

3. Sources of Vibration

The most basic forces of interest are those generated by engines, propellers, and shafts. Other common sources of vibration include: gears, screws, thrusters, fans, compressors, pumps, pipes, and valves. Flow generated vibrations may also occur.

3.1 Machinery Excitation – Unbalanced or misaligned machinery, particularly propulsion machinery such as the main propulsion engines, is the major source of excessive vibration and can develop excitation forces in the frequency range of interest for both the equipment and structure in the vicinity of the machinery. These frequencies of excitation match either the rotation rate, or twice the rotation rate.

3.1.1 Diesel Engines – Diesel engines, whether low, medium or high speed – two or four stroke, generate significant force and moment coupled vibration excitation. The diesel firing forces, which depend on rotation rate and the number of cylinders, are impulsive in nature, which causes components of vibration at many harmonics of the firing frequency. Low-speed propulsion diesel engines are typically more of a concern than high-speed diesel engines because their low excitation frequencies are more likely to lie in the range of a hull natural frequency. As a result, diesel engines, and other reciprocating machinery, may develop forces and moments, which are sufficient to create excessive ship vibration. These forces/moments may excite the machinery itself, the ship foundation on which it is attached, the hull girder, or other structures within the vessel.

The excitation of a reciprocating engine can be divided into two parts:

- Unbalanced inertial forces
- Firing forces/moments

The unbalanced inertial forces are associated with the rotating or reciprocal masses. The frequencies of resulting forces are multiples of shaft rotation rate.

A 4- stroke engine has twice as many ‘firing’ harmonics as a 2- stroke engine. Analytical calculations of diesel engine generated forces are not easily undertaken and should be left to the diesel engine vendor.

3.2 Propulsion and Shafting Excitation – The basic design purpose of the propulsor is to generate steady thrust to the vessel. In addition, this propulsor generates undesired fluctuating dynamic forces and moments due to its operation in a non-uniform wake and due to passage of the blades close to hull and appendages.

3.2.1 Hull Pressure vs. Bearing Forces – Propellers generate two types of fluctuating forces. These are hull pressure forces and bearing forces. The hull pressure originates from the hydro-acoustic pressure variations caused by the passage of a propeller through non-uniform inflow or wake. These hull pressure forces are affected by propeller-hull clearance in the vertical and horizontal directions, by blade loading, and by changes in the local pressure field around the blades.

Bearing forces are caused by fluctuating forces on the blade during a propeller rotation which generate both vertical and horizontal forces on the shaft. These vertical and horizontal forces produce lateral and axial forces and moments on the support bearings and thrust bearing. The frequencies of these forces and moments are the same as for the hull pressure, shaft rate, blade rate, and multiples.

The thrust bearing forces provide an excitation to the propulsion system in the longitudinal direction; the fluctuating torque produces shaft torsional vibration. The blade frequency vertical bearing force, when combined with same frequency vertical hull pressures, provides the overall vertical excitation of the hull in the vertical direction. Similarly, the combined horizontal forces excite the hull in the horizontal direction. Dynamic unbalance of any component of the system (propeller, shaft, coupling, gear, and engine) will increase the resulting hull vibration levels.

3.2.2 Cavitation – Where there is the occurrence of cavitation, (the sudden formation and collapse of low-pressure bubbles in liquids by means of mechanical forces, such as those resulting from rotation of a propeller), there is a significant increase in the hull pressure forces. Again, the frequencies of these propeller impacts are shaft rate, blade rate, and 2× and 3× multiples of the blade rate.

3.2.3 Thrusters – Another significant vibration source may be the bow or stern thruster(s). These units can be located relatively close to accommodations and can cause high vibration. As discussed above for the prime propulsor, inflow and propeller design control the induced vibration levels.

4. Structural Response

Generally, it is important to ensure that the natural frequencies of the fundamental hull girder vibration modes do not coincide with the excitation frequencies from any of the major excitation sources (propellers/main engines). This is normally ensured by global vibration calculations during design stage. However, increased vibration levels caused by hull girder modes excited by machinery or propellers can sometimes be experienced during sea trial or operation. The natural frequencies of the hull girder modes are influenced by the draft, trim and load distribution along the hull girder.

Substructures, like decks, machinery platforms, or the deckhouse, of sufficient mass and flexibility also need to be considered in a vibration-related survey. These substructures have their own natural frequencies, which should not coincide with primary excitation frequencies. The natural frequencies of these substructures are usually higher than that of the whole hull. However local structural elements like plates, bulkheads, parts of the deck, rudders, and machinery platforms may have natural frequencies which are close to the excitation frequencies. Massive equipment installed on platforms may have a large influence on the resulting natural frequency.

5. Vibration Assessment

If observations regarding vibration are made, it is helpful to remember that vibration levels experienced as “uncomfortable” are normally well below vibration levels necessary to create structural problems.

If found necessary based on vibrations experienced by the surveyor or personnel during the vessel trials or operation, vibration measurements should be considered. Changes in the text plan might be necessary or a test plan should be developed which includes evaluation of relevant measurement locations and vessel operating conditions. This may include measurement positions suitable for describing global vibration behavior of hull girder and deck house, as well as local positions in accommodation and machinery areas. Locations should include the potentially ‘worst locations’ (e.g., closest to sources or places expected to have maximum vibration).

Vibrations can in all cases be expected during maneuvering or during operation far outside of normal operation range for the vessel. Such temporarily experienced vibrations are normal and should not be considered problematic.

There are several international standards which address vibration, measurement procedures and evaluation. It is recommended to use the following standards, although not limited to them, as guidelines:

- ISO 20283-5:2016, Mechanical Vibration – Measurement of Vibration on Ships – Part 5: Guidelines for Measurement, Evaluation and Reporting of Vibration with Regard to Habitability on Passenger and Merchant Ships
- ISO 8041:2005, Human Response to Vibration – Measuring Instrumentation
- ISO 5348:1998, Mechanical Vibration and Shock – Mechanical Mounting of Accelerometers
- ISO 20283-2:2008, Mechanical Vibration – Measurement of Vibration on Ships – Part 2: Measurement of Structural Vibration

6. Measurements

The following summarizes some typical measurement types that may be relevant in case of severe vibrations. For all vibration related issues where measurements are considered, it is important that a proper measurement procedure is established by personnel with vibration expertise.

6.1 Global Structures – If vibrations are of a global nature, e.g. longitudinal vibrations in the deck house, it may be relevant to carry out measurements to establish the vibration behavior of the hull girder or deck house. Reference is made to ISO 20283-2.

6.2 Local Structures – Local structure includes plates and stiffened panels in deck or bulkheads. For local deck areas in the accommodation vibrations are normally only a problem for human comfort. This is further discussed in Recommendation 132. Reference is also made to ISO 20283-5. Vibrations in plates or panels in way of bulkheads, e.g. tank boundaries, can be a cause of structural damage.

If severe or critical local vibration exists, vertical, transverse, and/or longitudinal measurements are taken at the suspect location in order to determine the need for corrective measures. If the problem is established as highly localized resonant vibration of plating panels, piping, etc., then the vibration survey likely needs to go no further. It is usually obvious in such cases how natural frequency changes, through local stiffening, can be effectively accomplished to eliminate the local resonant conditions.

6.3 Semi-global structure – If vibrations are of a semi global nature, e.g. including an entire deck or coupling between decks, vertical and transverse vibration bending shape measurements can be taken using a significant number of points necessary to determine the mode shapes at low frequencies while avoiding local resonances. These types of measurements are made by use of a reference transducer at the stern along with a portable transducer. Torsional modes may require phased deck-edge measurements.

6.4 Vibration source measurements – Some of the following measurements may be relevant investigating the source of any excessive structural vibrations in hull or accommodation areas. Often measurements of machinery vibration or propeller pressure pulses are taken simultaneously with measurements of relevant structural positions. In cases with variable rpm on relevant machinery or propellers, such measurements are normally carried out as a run-up.

6.4.1 Local Machinery and Equipment – Vertical, transverse and/or longitudinal measurements are taken on non-rotating parts of the machinery where there is evidence of large vibration amplitudes. Types of machinery and equipment include diesel generator, electric motors, air compressors, ballast pumps, etc. Vibration measurements are normally taken at the bearing position and /or foundation.

6.4.2 Shell Near Propeller – If necessary, the measurements of hull surface pressure are taken in order to confirm design estimates, to obtain design data or to investigate hull pressure forces or potential cavitation problems.

To minimize the effect of plate vibration, all transducers in the hull plating are located as close as possible to adjacent frames or partial bulkheads.

6.4.3 Main Engine and Thrust Bearing –, Vertical, transverse, and longitudinal measurements are taken in order to determine the need for corrective measures. In some cases, measurements to establish an operational deflection shape (ODS) of the engine can be relevant. ODS measurements are carried out either with simultaneous measurement with numerous fixed sensors or more commonly with one fixed sensor and one or more roaming sensors. Such measurements are time consuming and with high demand to equipment and postprocessing tools. However, they may be a helpful tool for troubleshooting cases.

6.4.4 Lateral Shaft Vibration – If this measurement is conducted, vertical and transverse vibration measurements are made on the shaft. Additional measurement points may be taken. These measurements are made throughout the normal operational range of the ship.

In order to eliminate possible error, shaft run-out is checked by rotating the shaft by the turning gear and recording the first-order signal. This signal is phased, and the shaft vibration measurement corrected accordingly.

6.4.5 Torsional Shaft Vibration – If this measurement is conducted, torsional vibration measurements are taken either at the free end of the propulsion machinery, using a suitable torsional vibration transducer, and/or on the main shafting, using strain gauges. Alternatively, depending on the system characteristics, a mechanical torsiograph, driven from a suitable position along the shafting or free end, may be used for this purpose.

7. Remedial Measures

A foundation for critical vibration sources (e.g., large reciprocating machinery) typically is fitted with a thick foundation top plate, stiff floors, and local gussets between these two members in way of attachment points in order to provide damping of the vibratory forces. This is true whether the machinery is isolation mounted or not.

7.1 General Approach – The test report should describe the operating conditions for measurements, including weather, sea state, speed, rpm, water depth under keel, and other pertinent data. The report should show the locations of measurements and overall rms value in mm/s or mm/s2. For additional information the spectrum and/or single peak level with corresponding frequency could also be recorded. As the vibration control treatment designs evolve, drawings and other design information should be reviewed by one familiar with the design of treatments to verify that the treatment design adequately reflects vibrational considerations mentioned in the report.

Just as in developing a vibration-sufficient ship design, there exist three possibilities for correcting a vibration-deficient one in normal practice:

i) Reduce vibratory excitation
ii) Change natural frequencies to avoid resonance
iii) Change exciting frequencies to avoid resonance.

Most of the excessive diesel engine excited hull vibration can usually be corrected by the following provisions:

i) Mechanical or electrical moment compensators against engine external moments, or
ii) Engine lateral stays of weld, hydraulic or friction type against engine lateral vibration, or
iii) Axial damper against crank shaft vibration, or
iv) Dynamic absorber against fore-and-aft vibration of deckhouse.

The detail information necessary for the installations of above provisions is typically provided by engine or equipment manufacturers.

7.2 Structural Modifications – The most cost-effective approach for eliminating structural resonances is usually to shift natural frequencies through structural modifications such as adding panel stiffeners.

7.3 Propeller or wake modifications – For cases where the pressure pulses or shaft forces from the propellers are the source of excessive vibrations, it may be a solution to modify the propeller design or to manipulate the wake in order to reduce the excitation forces.

7.4 Machinery modifications – For vibration caused by unbalanced or misaligned machinery, the 'treatment' in these cases is to correct the mechanical problem. This represents another basic design function often carried out by the equipment vendor, who usually takes responsibility for the proper selection of vibration dampers, torsional couplings, and flexible shaft couplings.

7.4.1 Top stays - For vibration caused by e.g. main engine H-moment, the solution may be to add (or in some cases remove) top stays. Active top-stays can in some special cases be relevant.

7.4.2 Moment compensator – The excitation by main engine forces can in many cases be limited by installation of a moment compensator. This can be a solution for both 1st order, 2nd order, H-moment and X-moment excitation.

7.4.3 Resilient Mounts – If there is no inherent mechanical problem, resilient mounts may provide some abatement. However, isolation mounting provides no vibration reduction below the systems natural frequency and may amplify the vibration in the vicinity of the system resonance. Resilient mounts need careful design for two reasons. Vibration levels of the machinery itself are actually higher on resilient mounts than that if the machinery is hard mounted. Another concern is that the excitation frequencies of the machinery should not coincide with isolation system's natural frequencies. To avoid such a coincidence, the selection of the resilient mounts should be made with consultation of experts or the equipment manufacturer accompanied with calculations of natural frequencies of a 'machinery-resilient mount' system (six degrees of freedom calculations). The lower the natural frequency (softer resilient mounts), the lower the vibratory effect will be on the foundation and adjacent structures.

7.4.4 Foundation – Designing a stiff foundation structure is the most important approach to prevent excessive machinery induced vibration. The incorporation of a stiff egg-crate like framing and flooring system is preferable and should be a relatively light system of floors in either the longitudinal or transverse direction only. Transverse plating of a foundation should be a continuation of rigid floors; longitudinal plating should be incorporated into longitudinal stiff girders. The structure of thrust bearing foundations is also important, especially for systems with a Cardan shaft (universal joint).

8. Verification of Countermeasures

It is important that proper verifications are carried out to determine the effectiveness of the vibration treatment measures. This can include re-measurement of affected areas and/or calculations or documentation of the effect of the modifications carried out. The extent of necessary follow-up needs to be evaluated from case by case. The performance of vibration control treatments is ultimately dependent on the quality of the implementation. Seemingly trivial deviations from the detailed design, or inadvertent errors due to unfamiliarity with vibration control treatment materials and constructions, may compromise vibrational performance. Hence re-measurements will often be requested.