Engineering Solutions to Mechanics, Marine Structures and Infrastructures

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Review article

Research on Marine Shafting Transmission in China and Global

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Abstract: The continuous enhancement in ship construction has spurred significant research on ships worldwide, with ship power systems being a crucial component. Countries are persistently innovating to enhance the sustainability, safety, and reliability of ship power systems. International studies primarily concentrate on the dynamic characteristics, nonlinear dynamics, shafting alignment, transmission efficiency, and fault diagnosis. Concurrently, numerous methods have been developed, including the finite element method and the Holzer expression method. This paper primarily investigates ship shafting transmission research both domestically and abroad. It delineates the various methods employed in the ship drive system, the subsequent research process and outcomes, the primary research directions within the field, and the significance of these studies. The objective of this paper is to present an overview of the current state of research on ship shafting transmissions worldwide. It will also offer suggestions for addressing the issues of insufficient material strength, shaft lubrication, and shaft alignment that were prevalent in the early years of ship shafting. Furthermore, it will propose future ship shafting systems that address key areas such as wear and tear, noise, and propulsion systems.

Keywords: Mrine transmission system, Dnamic characteristics, Saft alignment technology, Fnite element analysis, Coupling vibration of propeller shaft system hull.

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1. Introduction

Ships play an indispensable role in international transportation strategies, serving as a vital conduit that bridges countries and facilitates economic and trade activities across maritime boundaries. Furthermore, they constitute a fundamental element of national defense. The shafting transmission system, which serves as the fundamental "heart" of a ship, constitutes a pivotal component of the vessel. It bears the critical responsibility of enhancing both performance and safety.

The discipline of mechanics is a broad one, encompassing not only the micro-mechanics of materials and macro-mechanics, but also a number of different areas of study within the broader fields of fluid mechanics and dynamics. In the context of maritime engineering, a multitude of mechanical challenges are prevalent. The advent of modern technology has elevated the axial drive system to a new level of sophistication. The interdisciplinary nature of ship propulsion systems is particularly evident when considering the interplay between science, engineering, and technology[1-3]. Notable scholars at home and abroad have invested significant time and effort into the study of ship shafting transmission. They have witnessed and contributed to its continual evolution and innovation, from the ancient wind-powered transmission to the adoption of steam turbines, diesel engines, and now gas turbines, electric propulsion, and hybrid systems. These developments have directed the evolution of marine shafting transmission systems toward enhanced safety, reliability, and efficiency[4-5]. Similarly, these scholars have conducted in-depth research into the complexities of ship shafting transmission from a variety of perspectives, including theoretical research, design and optimization, materials and manufacturing, experimental and testing methodologies, fault diagnosis, maintenance, green technology, and energy efficiency. Their work provides substantial support for the marine transmission infrastructure.

The objective of this review is to undertake a comprehensive and systematic analysis and synthesis of research on marine transmission systems on a global scale. The objective is to elucidate theoretical frameworks and software applications, summarize findings, identify challenges, and forecast future trends. By synthesizing domestic and international literature, this review aims to facilitate the reader's comprehension of ship shafting transmission.

2. Research status at home and abroad

The fundamental principle of the ship's shafting transmission system is the transfer of power from the main engine to the propeller via the transmission shaft. This enables the propeller to rotate, thereby generating thrust and propelling the vessel forward. The torque generated by the main engine is conveyed through a manifold to the drive shaft. This shaft, supported by bearings, transfers the torque to the shaft, which in turn powers the propeller. In light of this principle, considerable innovations and experiments have been conducted internationally and domestically concerning bearings, drive shafts, and tail shafts. Concurrently, during the process of innovation, it is paramount to employ an integrated approach in mechanical engineering and to utilize the entirety of human knowledge to address issues.

However, for passengers and warships, innovations must be grounded in human nature and prioritize the principle of human safety [6]. The key point of this aspect is to determine the torsional vibration parameters of the shaft system with accuracy, including moment of inertia, damping, stiffness, and so forth. The diagram of shafting rotation is shown in Figure 1.

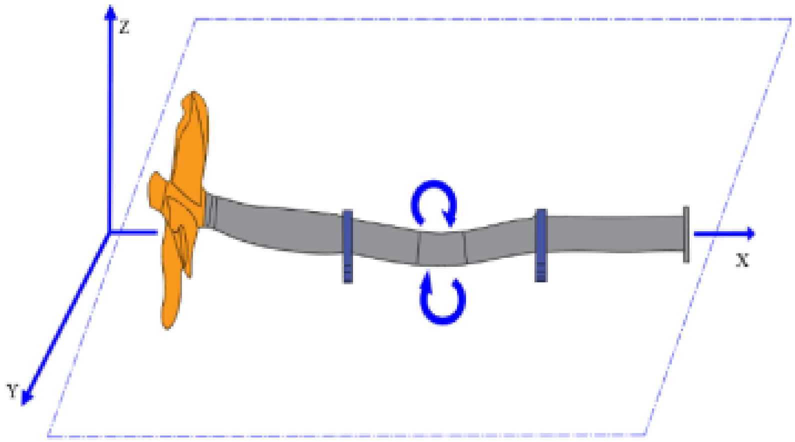


Figure 1 Diagram of shafting rotation[15]

The study of rotating vibration has been comparatively neglected in comparison to the investigation of twisted and longitudinal vibration, a situation that has prompted a growing international focus in recent years. This shift is attributed to a number of factors, including the accumulation of research findings, an increased emphasis on engineering, advancements in computational power, and the proliferation of analytical techniques. The focus of vibrational research in ship transmission systems is on the propeller fluid flow, gear meshing, and the hull structure. This comprehensive approach allows for the fulfillment of the research requirements for managing ship shafting vibration, thereby enhancing control over shafting vibration[7]. Figure 2 presents a review of the literature on cyclotron vibration published over the past 40 years.

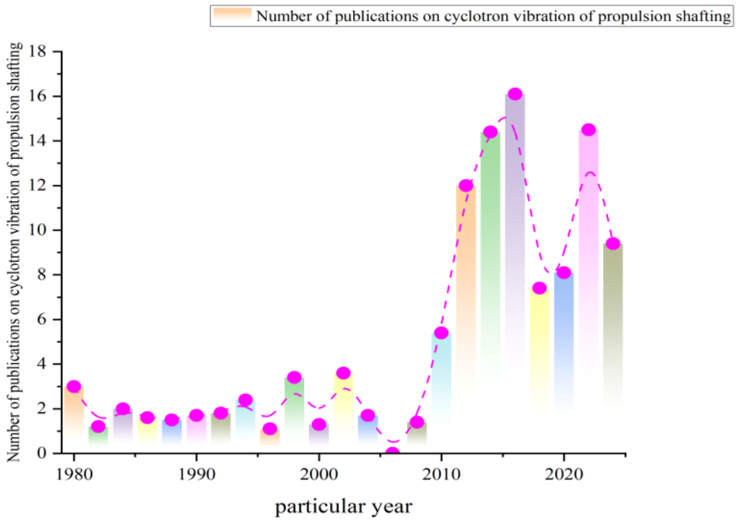


Figure.2 The number of cyclotron vibration literature published in China in recent 40 years[15]

In order to ascertain the natural frequencies of shafting, researchers typically utilise empirical formulae, including those recommended by the Panagopoulos or Jasper methods as outlined in the "Practical Manual for Ship Design" [8]. However, both of these methodologies are subject to limitations in terms of their rigor. The finite element method (FEM) offers a more precise solution to this problem. Zhou Chunliang et al. developed a comprehensive three-dimensional model of the ship's axial system and investigated the vibration phenomenon of the ship's shafting, including the inherent vibration characteristics and the response of the axial system to different forces. Additionally, they analyzed the nonlinear dynamics of a cracked shaft and the coupled vibrations of the shafting system. This study employs the finite element method (FEM) to analyze the vibration response of ship shafting under various excitations. Concurrently, simulation software such as ANSYS, ABAQUS, and NASTRAN are utilized. The FEM allows for the effective analysis of multi-body dynamics and mechanical problems, such as the rotating vibration of propulsion shafting, through the use of Simulink simulations.

In his discussion of a critical issue in the vibration diagnosis of ship propulsion systems, Korczewski [9] focuses on the assessment of the mechanical energy consumed by the propeller shafting vibration from the main engine to the propeller. A simplified calculation model is presented, which enables the estimation of the total energy produced by torsional vibrations based on the axial deflection amplitude measured at the center span between supporting points. To validate this model, a pilot test was conducted on a laboratory rotating machinery system test bed. In the experiment, a uniform (cylindrical) material specimen was subjected to cyclic bending moments to simulate the structural and functional characteristics of an actual propeller shaft at an appropriate scale.

Concurrently, scholars have conducted extensive research on the issue of axial system alignment. The Prince Family and others [10] have established a mapping model for the mean and misalignment values of axial and radial vibration in diesel engines, taking into account the variable load characteristics and compact structure. Utilizing the shaft's shape characteristics, they have subsequently developed a monitoring scheme and a quantitative misalignment detection method. Experimental data indicate that this new method achieves detection accuracy above 90% and is not influenced by operating conditions. Additionally, this method elucidates the relationship between the degree of static misalignment and the fluctuations of the Average Cylinder Axis Position (ACAP), providing guidance for static installation and the analysis of misalignment causes. Donglin zou et al. [11] develop a prediction model for the vibration excitation force of the shafting by utilizing the boundary element method (BEM) and rotor dynamics theory. The accuracy of this model is validated through both numerical and experimental procedures. The study reveals that the nonlinear effect of the wake vortex diminishes at lower vibration amplitudes and frequencies, and that the exciting force is linearly proportional to the vibration amplitude and exhibits a quadratic relationship with the vibration frequency. Consequently, the real and imaginary parts of the exciting force can be approximated as equivalent to constant added mass and additional damping, respectively. Through a series of calculations involving different propellers, a simplified formula for estimating the axial exciting force in linear conditions is proposed and verified. Wang Xianfeng [12] Investigates the calculation of propeller hydrodynamic force and incorporates this factor into shafting alignment calculations. Ultimately, the method for modifying static alignment models under the influence of dynamic effects is established. Zhu Chunlei [13] Delves into the correct installation of shafting post-alignment, discusses the "zigzag offset method" in conjunction with real-world production applications, and introduces the jack lifting technique to assess shafting alignment.

The issue of shaft system vibration and noise reduction has been the subject of considerable research by numerous scholars. Xuwei Ze et al. [14] employ an integrated shaft isolation system to mitigate vibration transmission from the propeller to the hull, supporting bearings, propulsion devices, and auxiliary engines. To achieve this goal of vibration and noise reduction, the vibration energy is deliberately dissipated by altering the vibration transmission pathway within the coupled system. Employing the displacement response and force transfer rate as measures, the vibration isolation efficacy of both traditional and elastic supports is compared and dissected. The outcomes demonstrate that the bearing displacement response within a full elastic bearing system is notably less than that observed with the conventional bearing setup.

Modelling calculations are an invaluable tool for addressing challenges in ship shafting. This article presents a synthesis of insights from scholars at home and abroad on the use of modelling calculations in this field. Design and Implementation of Software for Predicting Longitudinal Vibration Characteristics of Ship Shafting [15] Building upon the strengths of the Holtz method and leveraging the QT platform along with MATLAB software, a comprehensive software tool is developed to predict the longitudinal vibration characteristics of ship shafting. The efficacy of this method is confirmed through finite element analysis and the use of an existing ship shafting vibration calculation software. Furthermore, this work delves into the influence of pivotal shafting parameters on the longitudinal vibration properties.

Sheng Chenxing and colleagues [16] Fabricated specimens with varied strip groove depths using a 3D printer and subject them to testing with the CBZ-1 marine shaft friction and abrasion testing machine. The study reveals that water-lubricated bearings enhance transmission efficiency. Additionally, air-lubricated bearing technology for drag reduction is categorized into three primary types: bubble drag reduction, air layer drag reduction, and partial cavity drag reduction.[17]

Wu Hao et al. [18] employed three distinct multiphase flow models to simulate drag reduction effects achieved through air injection. The study demonstrates that the drag reduction rate is positively correlated with the increasing gas flow rate. Upon reaching saturation levels, the drag reduction rate stabilizes, with a drag reduction rate of 32.78% for the flat groove model. Furthermore, it was determined that the Mixture model is best suited for gas-liquid mixing flows, while the VOF model is more appropriate for gas-liquid separation flows. Current domestic research into air lubrication drag reduction technology is predominantly theoretical. While numerical simulations and model tests have yielded some insights, there is a need for further advancements to enhance product performance and quality.

Bhushan Taskar and colleagues [19] utilized a realistic model of an engine and propeller to simulate responses such as power fluctuations, propeller speed variations, and torque fluctuations. Consequently, the coupled system framework they present can be employed to examine engine load dynamics, propeller wave-induced loads, shaft vibrations, and engine control systems. This framework is also suitable for analyzing the performance and safety of the control systems used in engine regulation.

Kocimirella [20] conducted a study of marine vessels transiting the Otranto Strait, during which the significant role of composite materials in ship shafting and ship construction, along with emission reductions, was validated. Moreover, extensive research overseas on the performance attributes of ship shafting materials, encompassing mechanical, fatigue, and corrosion resistance properties, has notably enhanced the propulsion efficiency of ship shafting.

Vulic Nenad [21] employed simulation software to model the marine propulsion system of a two-stroke diesel engine. Utilizing various models for engine system representation, the simulation outcomes were then contrasted against analytical predictions and measurement results. The selected simulation model underwent verification and validation during the normal operation of the diesel engine.

3. Transmission system theory and experiment

In the context of system matrix method, the approach primarily focuses on the dynamic characteristics of ship shafting. This method utilizes free vibration holzer expressions to construct finite element models for examining the longitudinal vibration characteristics of ship shafting. In estimating the natural frequency of the shafting, researchers generally employ empirical formula methods. For shaft alignment calculations, the method involves using bending moment and transfer matrix techniques, incorporating the finite element approach. It is applied under the principle of straight-line alignment, with sensitivity to stiffness adjustments. The system employs a network topology based on fault diagnosis, integrating wide area network technology. This technology enables the formation of a local area network interconnected through wireless means. Methods such as matrix analysis and bending moment techniques are fundamental to the system's engineering approach.

3.1 System matrix method

The ship propulsion system is a complex assembly; however, the calculation of vibration issues should be simplified as much as feasible. Consequently, the matrix method is most suitable for this purpose. As explained in reference [22], the system can be decomposed into simpler dynamic components through the application of the transfer matrix method. This approach allows the vibration state of the system to be represented by the state vector at the endpoints of each component. The relationship between the state vectors of the two endpoints of an element describes the dynamic behavior of that element. This is encapsulated in a matrix known as the element's transfer matrix. Specifically, in the case of a typical straight chain propulsion system, the inertia matrix is diagonal :

$$[J]=\begin{bmatrix} J_1 & 0 & 0 & 0 & 0 & 0 \\ 0 & J_2 & 0 & 0 & 0 & 0 \\ 0 & 0 & M & 0 & 0 & 0 \\ 0 & 0 & 0 & M & 0 & 0 \\ 0 & 0 & 0 & 0 & M & 0 \\ 0 & 0 & 0 & 0 & 0 & J_n \end{bmatrix} \tag{1}$$

The stiffness matrix is a ribbon matrix:

$$[K]=\begin{bmatrix} K_{1,2} & -K_{1,2} & 0 & L & 0 & 0 & 0 \\ -K_{1,2} & K_{1,2}+K_{2,3} & -K_{2,3} & L & 0 & 0 & 0 \\ 0 & -K_{2,3} & K_{2,3}+K_{3,4} & L & 0 & 0 & 0 \\ L & L & L & L & L & L & L \\ 0 & 0 & 0 & L & K_{n-3,n-2}+K_{n-2,n-1} & -K_{n-2,n-1} & 0 \\ 0 & 0 & 0 & L & -K_{n-2,n-1} & K_{n-2,n-1}+K_{n-1,n} & -K_{n-1,n} \\ 0 & 0 & 0 & L & 0 & -K_{n-1,n} & K_{n-1,n} \end{bmatrix} \tag{2}$$

In the context of a free vibration system, since the direct propulsion shafting of a diesel engine falls under the category of weak damping systems, its impact on calculations of free vibration modes can be disregarded:

$$\begin{cases} u(x,y,z,t) = \sum_{i=1}^{n} N_{\mathrm{i}}(x,y,z)u_{\mathrm{i}}(t) \\ v(x,y,z,t) = \sum_{i=1}^{n} N_{\mathrm{i}}(x,y,z)v_{\mathrm{i}}(t) \\ w(x,y,z,t) = \sum_{i=1}^{n} N_{\mathrm{i}}(x,y,z)w_{\mathrm{i}}(t) \end{cases} \tag{3}$$

$$[J]\{\ddot{\varphi}\} + [K]\{\varphi\} = 0$$

(4)

Let the general solution of formula (4) be$\{\varphi\} = \{x\}\cos w_n t$ Subtract (4) from:

$$[K]\{\chi\} = \omega_{\mathrm{n}}^2 [J]\{\chi\} \tag{5}$$

Write the inertia matrix to the left of the equation:

$$[J]^{-1}[K]\{\chi\} = \omega_{\mathrm{n}}^2 \{\chi\} \tag{6}$$

Here, w_n represents the intrinsic angular frequency, and the vector $\{x\}$ constitutes the mode vector for the system's vibration. This equation can be mathematically solved provided the eigenvalues and characteristic vectors of the matrices are computable. Consequently, upon the formation of the system's stiffness and inertia matrices, the determination of all eigenvalues and characteristic vectors becomes straightforward. The matrix method enables precise prediction of the system's dynamic response, thereby serving as a foundation for structural design and optimization.

3.2 Finite element analysis

Through the application of discretization, the entirety of the ship hull's solution domain can be subdivided into a series of finite segments. Consequently, the system of differential equations presented in the initial query is restructured into its equivalent variational formulation. As a result, the initial differential equation system is reduced to a system of simultaneous algebraic equations, simplifying the process of solving the linear algebraic system. The finite element equation is given by Liu niuniu[23].

Firstly, the structure is discretized into numerous small elements, including triangular elements for two-dimensional problems and tetrahedral elements for three-dimensional problems. For each element, a displacement function is assumed, primarily representing displacements within the domain of structural mechanics. The displacement within the discrete element can then be expressed as:

$$\begin{cases} u(x,y,z,t) = \sum_{i=1}^{n} N_i(x,y,z) u_i(t) \\ v(x,y,z,t) = \sum_{i=1}^{n} N_i(x,y,z) v_i(t) \\ w(x,y,z,t) = \sum_{i=1}^{n} N_i(x,y,z) w_i(t) \end{cases} \tag{7}$$

Form of the matrix:

$$u = Na^e \tag{8}$$

$$u = \begin{bmatrix} u(x,y,z,t) \\ v(x,y,z,t) \\ w(x,y,z,t) \end{bmatrix}, \quad N = \begin{bmatrix} N_1 & N_2 \cdots & N_n \end{bmatrix}, \quad a^e = \begin{bmatrix} a_1 \\ a_2 \\ \cdots \\ a_n \end{bmatrix}, \quad a_i = \begin{bmatrix} u_i \\ v_i \\ w_i \end{bmatrix} \tag{9}$$

Where u_i, v_i, w_i represent the internal displacements along three mutually orthogonal coordinates I of the cell, and ni (x, y, z, t) are the shape functions of the I the cell. Denote the shape functions specific to the I the cell. For each element, the Hamiltonian principle is employed to integrate the momentum fromt1 to t2 to ensure adherence to the boundary conditions:

$$\int_{t_1}^{t_2} \left[\delta(T - U) + \delta W_{nc} \right] dt = 0 \tag{10}$$

Where T is the kinetic energy of the system represented by displacement, U is the potential energy associated with the displacement, andδdenotes the virtual work done by the non-conservative forces. The variational sign is denoted by δ .

The kinetic energy of the system can be written as:

$$T = \iiint_v \frac{p}{2} \left[\left(\frac{\partial u}{\partial t} \right)^2 + \left(\frac{\partial v}{\partial t} \right)^2 + \left(\frac{\partial w}{\partial t} \right)^2 \right] (dV) \tag{11}$$

The deformation energy of the system can be expressed as:

$$U = \iiint_v \frac{1}{2} \left[\sigma_x \varepsilon_x + \sigma_y \varepsilon_y + \sigma_z \varepsilon_z + \tau_{xy} \gamma_{xy} + \tau_{xz} \gamma_{xz} + \tau_{yz} \gamma_{yz} \right] dV \tag{12}$$

Where“ρ” is the structural density,σx、σy、σz、τ_x, τ_y, τ_z is the normal stress and shear stress in three directions of each element, $\varepsilon_x, \varepsilon_y, \varepsilon_z, \gamma_{xy}, \gamma_{xz}, \gamma_{yz}$ is the normal strain and shear strain in three directions of each element, respectively. A and V correspond to the area and volume of each element, respectively.

If the nodal force acting on the unit is fi, then the virtual work of the external force can be obtained as:

$$\delta W_f = \int_v \delta u f_i dV \tag{13}$$

Let the viscous damping of the system be μ then the virtual work done by the damping force is:

$$\delta W_{\mathrm{d}} = -u\int_{\mathrm{v}} \delta u u dV \tag{14}$$

The differential equation of vibration can be obtained by substituting Eq. (11), (12), (13), and (14) into Eq. (10) by geometric and physical equations:

$$[M]\{\ddot{u}\} + [C]\{\dot{u}\} + [K]\{u\} = \{F\} \tag{15}$$

Where: $[M]$, $[c]$, $[k]$ represent the mass matrix, damping matrix, and stiffness matrix of the system, and $\{F\}$ represent the external forces acting on the nodes of the system.
There are also:

$$[M] = \sum_{\mathrm{e}} M^{\mathrm{e}}, \quad [C] = \sum_{\mathrm{e}} C^{\mathrm{e}}, \quad [K] = \sum_{\mathrm{e}} K^{\mathrm{e}}, \quad \{F\} = \sum_{e} F^{\mathrm{e}}$$

$$M^{\mathrm{e}} = \int_{\mathrm{v}} \rho N^{\mathrm{T}} N dV, C^{\mathrm{e}} = \int_{\mathrm{v}} \mu N^{\mathrm{T}} N dV, K^{\mathrm{e}} = \int_{\mathrm{v}} B^{\mathrm{T}} D B dV, F^{\mathrm{e}} = \int_{\mathrm{v}} N^{\mathrm{T}} f dV, \tag{16}$$

Where: $B = \partial N$, D is an elastic matrix.
Generally speaking, the damping matrix [c] is not derived from the above integral. But for convenience, mathematically express it in the form of proportional damping:

$$[C] = \alpha_1 [M] + \alpha_2 [K] \tag{17}$$

Where in α_1 and α_2 are coefficients related to the lowest and highest resonance frequencies in the frequency range considered by the system. The displacement response, velocity response, and acceleration response at any point on the structure can be obtained by solving equation (15) directly[24] .

3.3 Holzer method of calculation

The Holzer method is a conventional approach for determining the intrinsic frequency and vibration type of an axially rotating system. The obtained intrinsic frequencies are considered approximate. The analysis of the rotational vibration properties of the axial system, including the intrinsic frequencies and vibration types, is predominantly conducted through the construction of convergence matrices. In comparison to other methods, such as the energy method or the magnification factor method, the Holzer method[25] provides a more accurate reflection of the actual vibration characteristics of shafting.

3.4 Empirical formula method

The empirical formula method [26], employing mathematical formulas to delineate variable relationships, is applicable for predicting and estimating a range of engineering parameters. This includes estimating the natural frequency of shafting, conducting statistical analyses of known data, and deriving empirical formulas. To some extent, it can foresee future trends or outcomes with a degree of reliability and accuracy. The method is advantageous in that it is simple and straightforward, and can resolve issues expeditiously to a certain extent. However, it is also disadvantageous in that it is constrained by the limitations inherent in being based on existing experience and data. Furthermore, its accuracy may be compromised when new situations arise or when it is beyond the scope of the data.

3.5 Equivalent equilibrium force system method

The Equivalent Equilibrium Force System method (referenced in [27]) is an analytical and design approach utilized in mechanical engineering to study and establish the balance state of ships. This method hinges on the principle of force system balance, which incorporates external factors such as gravity and buoyancy to ensure stability and safety across diverse operational conditions. The fundamental principle of the equivalent balance force system method for ships revolves around achieving force system balance. Through the computation and analysis of forces and moments under varying operational conditions, the vessel's stability and safety can be assessed. Specific steps involve plotting the hydrostatic and stability curves of the ship, calculating the variance between buoyancy and gravity at various waterline heights, as well as the restoring moment at different tilt angles, thereby enabling an evaluation of the vessel's stability and safety.

3.6 Three-moment method

The three-moment method (refer to Figure 3) is predominantly employed in numerical analysis and engineering computations within the field of mechanical engineering, utilizing holistic modeling.

This method constructs a three-torque equation model, where the center of gravity's interface change in the axial system is treated as a virtual rigid articulated bracket. The rotation of this bracket and the bending deformation are then calculated. The three-moment method is known for its reliability and efficiency in shaft alignment for small ships. By incorporating longitudinal and rotational vibrations of the shafting system, an influence coefficient and the three-moment equation model are formulated[28]. In this approach, critical points of the shafting are subjected to constraints and loading, with the actual forces and displacements being calculated. Nevertheless, there are inherent limitations to this approach, including the complexity of solving the equations, the dependence of the boundary conditions, and the difficulty of considering complex factors.

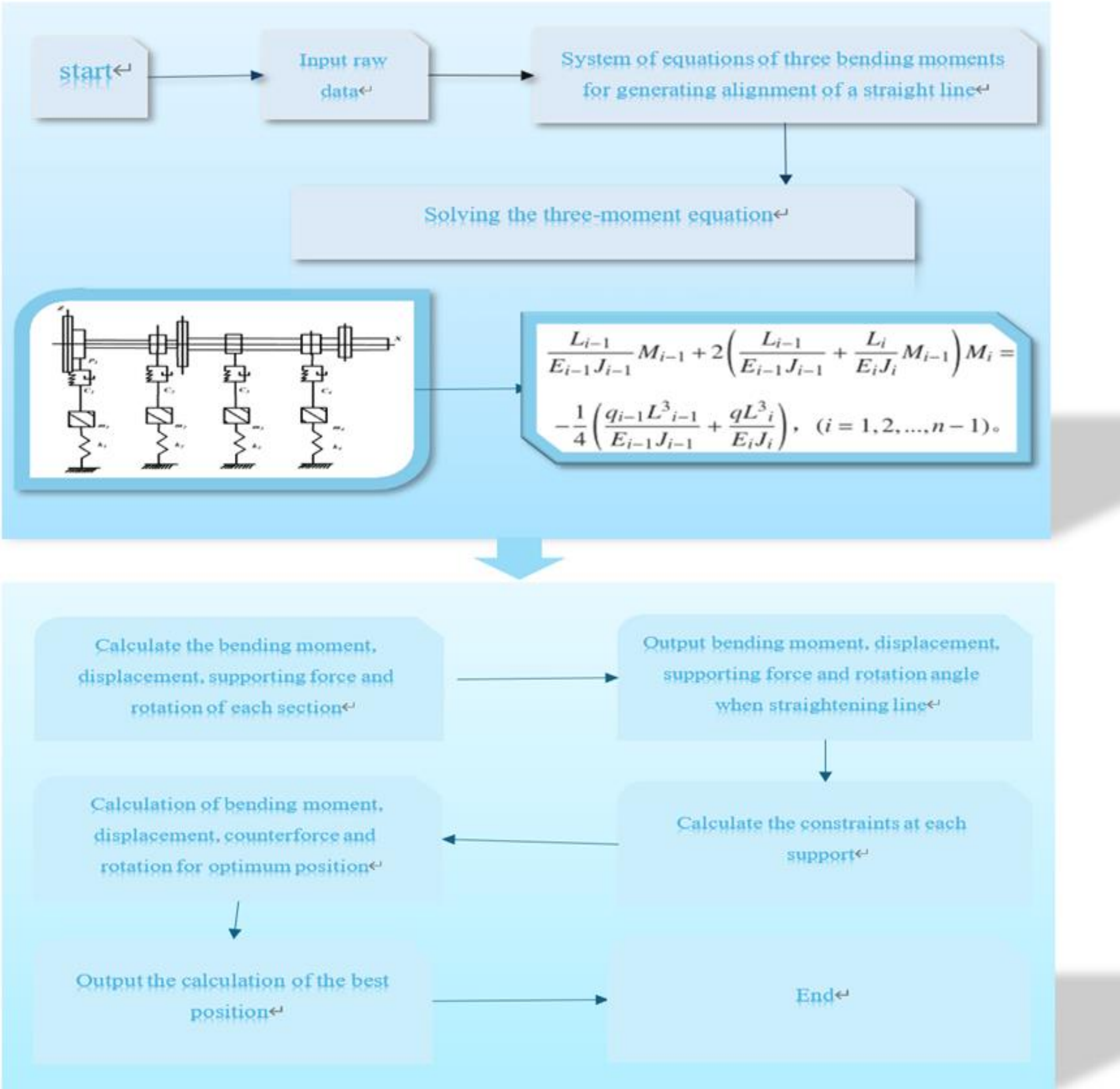


Figure.3 Flow chart of three bending

3.7 Topological optimization method

The essence of topology optimization lies in determining the optimal distribution of materials. The finite element method encompasses the modeling of engineering systems and the decomposition and coordination of complex systems within mechanical engineering (Fig. 5). Currently, the prevalent techniques in continuum topology optimization encompass the homogenization method [29], the Variable Density Method [30], the Progressive Structure Optimization Method (PSO) [31], and the level set method [32]. The fundamental principle of topology optimization involves mathematical modeling and optimizing the material distribution within a defined design domain. Initially, the finite element model of the structure must be constructed; thereafter, the structure is discretized into numerous smaller elements. The overall mechanical properties of the structure are derived through the mechanical analysis of these elements. Subsequently, the topology optimization algorithm computes the optimal structure shape and material distribution. This process may employ various optimization algorithms, including evolutionary algorithms and others. Through iterative refinement and optimization, the optimal structural design is ultimately achieved. Refer to Figure 4 for a flowchart depicting the optimization. process.

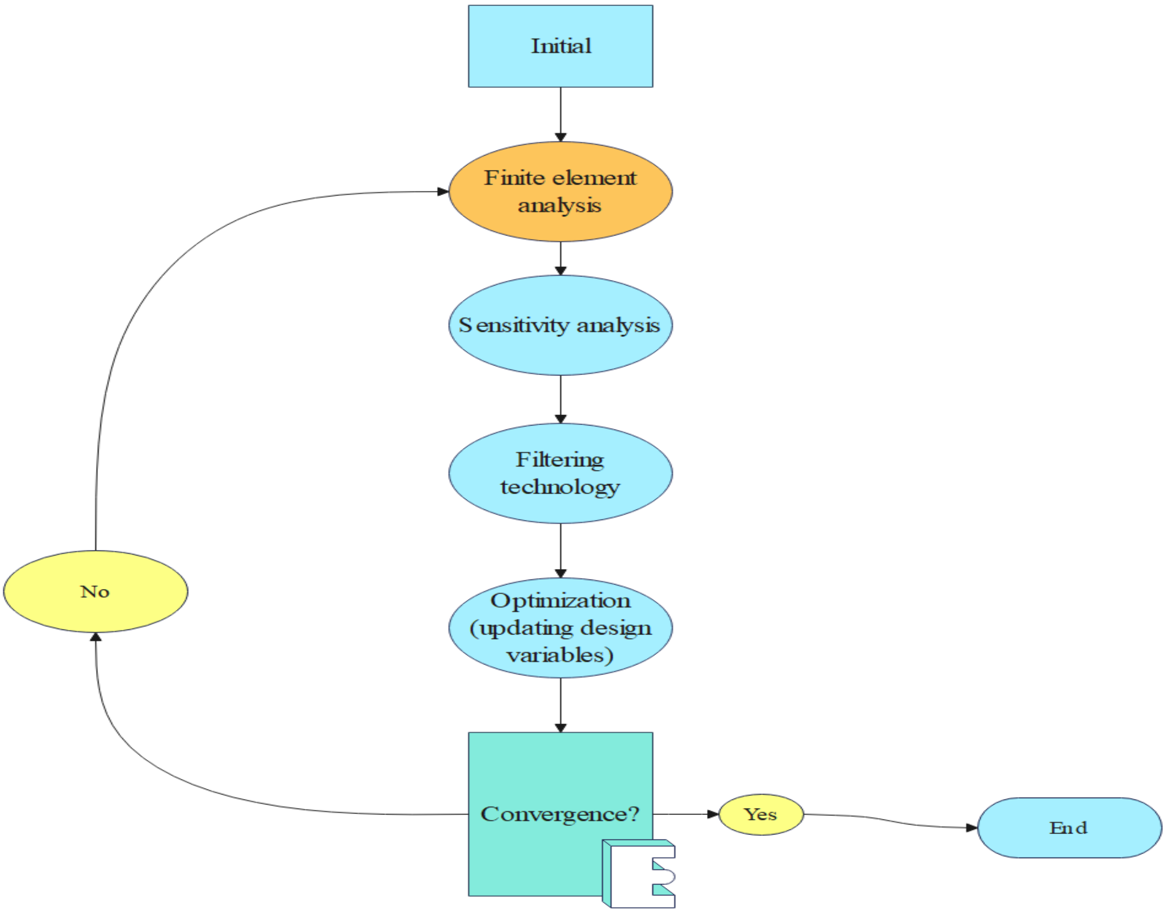


Figure 4 Topology optimization flow

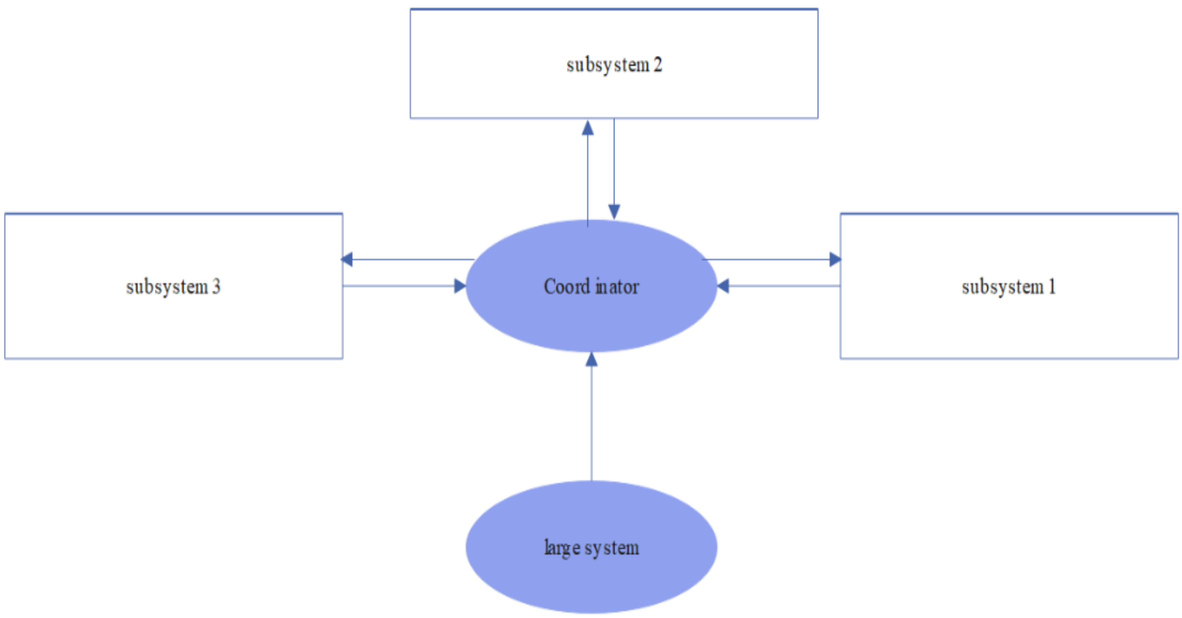


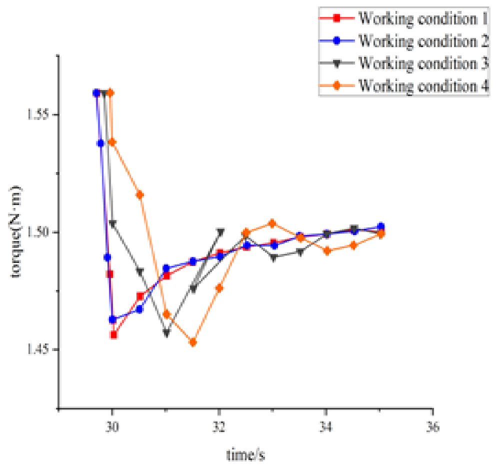
Figure 5 Block diagram of decomposition-coordinated hierarchical control for large systems

3.8 Calculation results of the elastic coupling effect

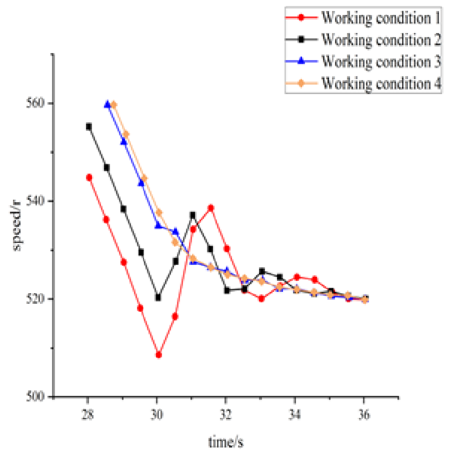
Flexible couplings are extensively employed in marine propulsion shaft systems due to their roles in transmitting torque, compensating for shafting displacement, and mitigating vibration and noise. [33] In shafting systems featuring gear drives, elastic couplings are commonly utilized to intercept fluctuating torque, thereby safeguarding the gears from its impact. Typically, the integration of an elastic coupling within a shafting system can generate a range of torsional vibration resonance frequencies, spanning from low to high frequencies. Notably, in low-frequency torsional vibrations, it is predominantly the elastic coupling that generates the respective vibration waveforms. [34]

3.8.1 Effect of elastic couplings on main engine end

The low rigidity of the elastic coupling diminishes the vibration amplitude at the main engine end. By absorbing impact energy and lessening the transmission of the impact force, the vibration frequency and amplitude of the driving end can be mitigated to some extent. This contributes to enhancing the stability of the driving equipment and reducing wear and fatigue damage to its components. Additionally, the elastic coupling can distribute the impact load, thereby decreasing stress concentration at the main engine end. Absorbing impact energy also lessens the stress on the driving end, thereby enhancing the structural integrity and reliability of the driving equipment. A reduced stress level aids in extending the service life of the driving equipment and minimizing fatigue damage resulting from excessive stress. Conversely, the high rigidity of the elastic coupling can amplify vibration at the driving end, with the stiffness-coupling rapidly transferring the impact force, resulting in a more pronounced vibration response from the main engine. Chronic high vibration levels can impair the performance and accuracy of the driving equipment, potentially leading to failure. The high stiffness of the elastic coupling may increase the stress on the main engine end, as the impact force is directly transmitted to the driving end, potentially exacerbating stress concentration and leading to cracks, deformation, and other types of damage in the equipment's components, thereby compromising the normal operation of the equipment.



Host end torque[35]



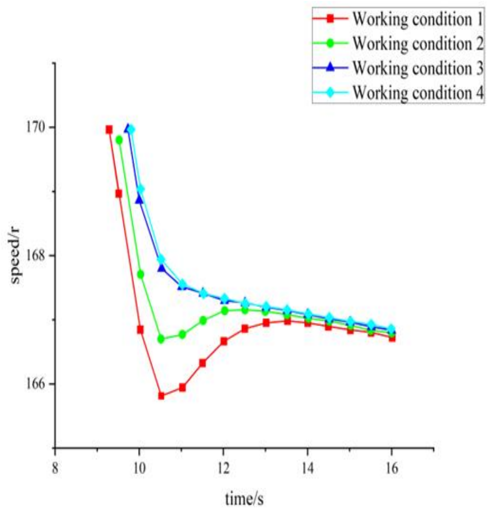
Host end speed[35]

Figure.6 The impact curve of coupling on the main engine load

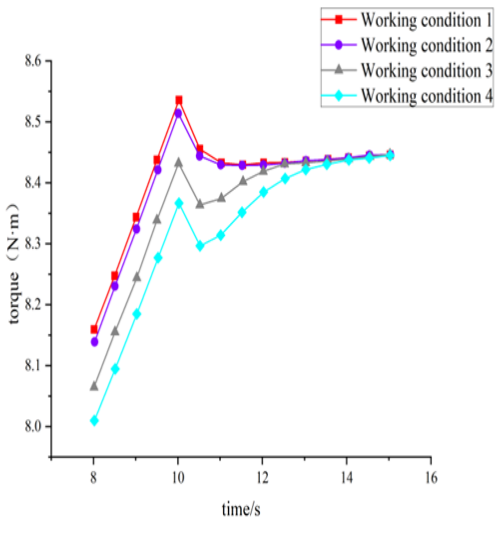
Figure 6 illustrates the influence of the elastic coupling on the main engine. A lower stiffness in the elastic coupling results in a faster speed response from the main engine, but a slower response in terms of torque and propeller matching parameters. This is accompanied by a more pronounced overshoot oscillation at the tail end of the impact. Conversely, a higher stiffness in the elastic coupling brings the response characteristics of the coupling closer to those of a rigid element. As the damping coefficient increases, the system's speed and the amplitude of the torque impact decrease, and the decay rate accelerates, indicating that the damping coefficient can mitigate the torque impact amplitude to a certain degree[35].

3.8.2 Effect of elastic coupling on propeller end

The elastic coupling facilitates the reduction of vibration, thereby enhancing the propeller's running stability, minimizing fatigue and loosening in the propeller blades, and decreasing damage to the shaft system and supporting structure due to vibration. Adjusting the stiffness and damping characteristics of the elastic coupling allows for altering the vibration frequency at the propeller end to prevent resonance with the main engine or other system elements. This modification is also effective in mitigating stress on the propeller, shafting, and connecting parts, thereby improving their fatigue resistance and damage resistance, and extending service life. Furthermore, the elastic coupling enhances the propeller's start and acceleration performance, improving the smoothness of acceleration and reducing the impact on the power system. Additionally, it contributes to lowering vibration and noise levels during the ship's start and acceleration phases.



Propeller speed[35]



Propeller torque[35]

Figure.7 Impact curve of coupling on propeller

Figure 7 demonstrates the impact of the elastic coupling on the propeller. The figure reveals that the elastic coupling parameters have minimal influence on ship speed. A higher stiffness results in the system's matching response approaching that of a full rigid connection. At low damping, the system exhibits an elastic response. As the damping coefficient increases, the propeller end speed's impulse response can be further controlled. Nonetheless, an excessively high damping coefficient may lead to the system transitioning into a fully rigid response, thereby compromising system stability[36].

3.8.3 New coupling

The double stiffness coupling (Figure 8) employs a cylinder coupling cavity, which incorporates multiple sets of buffers. In this configuration, the cylinder coupling operates independently, generating primary stiffness. Under substantial loads, the buffers activate to create secondary stiffness. The silicone oil rubber coupling involves installing a friction ring within the cylindrical coupling cavity and filling it with silicone oil to achieve high damping. To accommodate the axis offset, the cavity is also fitted with a double universal coupling. Characterized by high elasticity and damping, the silicone oil rubber coupling exhibits superior vibration reduction and vibration avoidance properties. Due to the rubber cylinder's significant elasticity in both the circumferential and axial directions, the friction ring can slide relatively within these axes to generate damping. This characteristic not only makes it effective against torsional vibration but also against longitudinal vibration, positioning it as a comprehensive damping element.

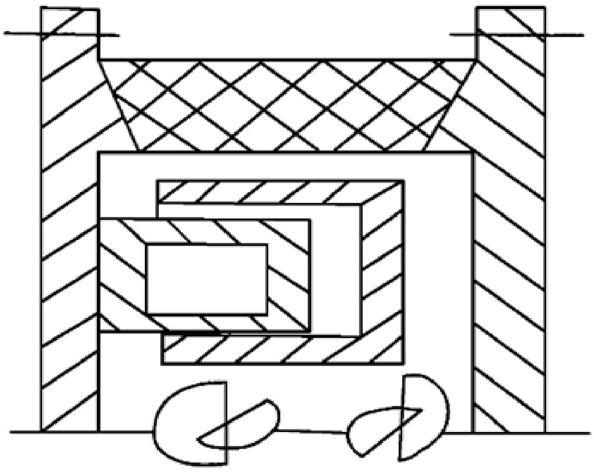


Figure.8 Double rigidity coupling [36]

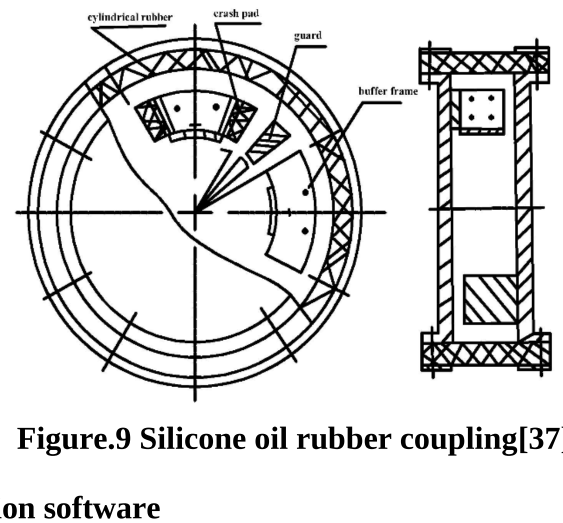


Figure.9 Silicone oil rubber coupling[37]

4 Shafting simulation software

At present, the international commonly used ship shafting simulation software are Shaft Designer, ANSYS, Romex, Adams, Matlab, and Abquas.

ShaftDesigner provides comprehensive shaft and bearing load analysis, calculating shaft calibrations and determining optimal bearing preload values for vessels in various load conditions. It ensures shaft accuracy and operational stability while mitigating bearing wear and seal leakage resulting from improper calibration. The software also calculates torsional, longitudinal, and gyratory vibrations and can predict the vibration characteristics of shafts under various operating conditions. It is designed to facilitate the creation of shafting systems with minimal vibration and stable operation. ShaftDesigner features a common model library for ship shafting, enabling automatic alignment optimization, reverse alignment calculations, and determination of the optimal shaft axis shape. However, the complexity of the software can pose challenges for users to learn. (See Figure 10 for a flowchart.) The modeling and calculation processes include direct alignment computations (represented in Figure 11), direct tolerance analysis, and torque testing.

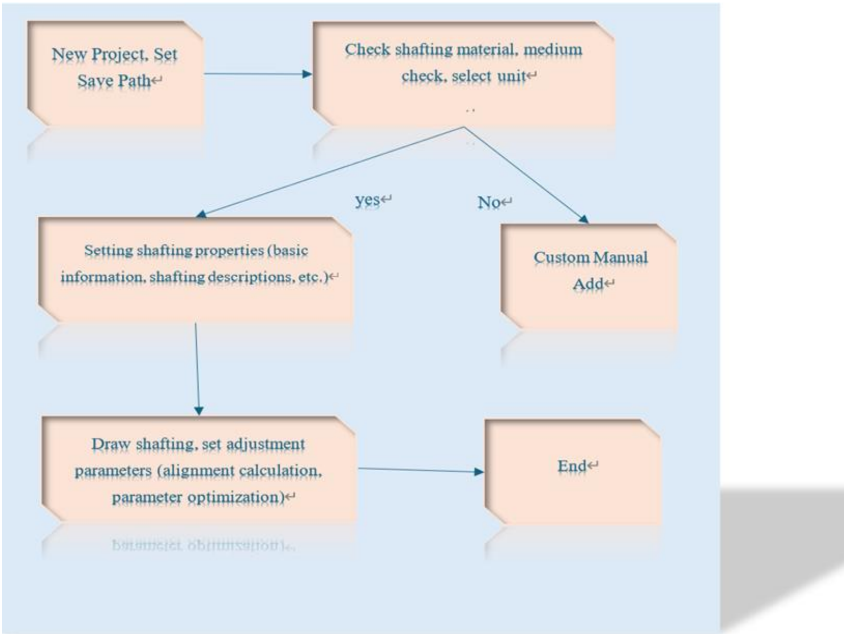


Figure 10 ShaftDesigner flow diagram

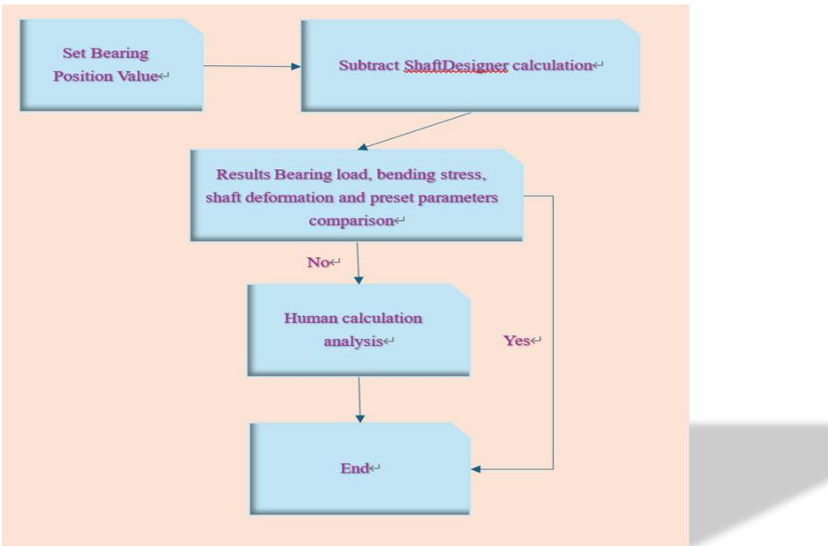


Figure 11 Flow chart of direct

ANSYS[38] is a widely utilized finite element analysis software in both military and civil construction industries. The software is proficient in accurately assessing the dynamic forces encountered by engineering structures subjected to rapid impact loads, such as aircraft collisions and explosions. When compared to other finite element analysis tools, ANSYS offers distinct advantages in this aspect, leading to its extensive application in engineering scenarios involving dynamic impact. It enables structural static analysis for various components of ship shafting systems, including the shaft itself, bearing blocks, and couplings. It determines stress distributions and deformations under various loading conditions and assesses whether the components' strength and stiffness comply with design specifications. The software also analyzes the vibration characteristics of the shaft, calculating inherent frequencies and identifying vibration types to determine resonance risks within the operating speed range. This analysis aids in designing to avoid resonance and investigates thermal issues in shaft transmission systems, such as bearing friction heat generation and gear transmission heating. These evaluations consider the impact of heat on the performance and longevity of shaft system components. ANSYS innovation lies in its ability to transcend the limitations of single-physical field simulation, offering hardware flexibility and integrating AI for a new era of simulation advancements. Its high precision is a notable advantage, though the software's complexity may present challenges in learning and computation. In summary, ANSYS is a reliable fluid analysis tool. (Refer to Figure 12 for the flow chart.) The model is computed using finite element analysis and Fluent calculations.

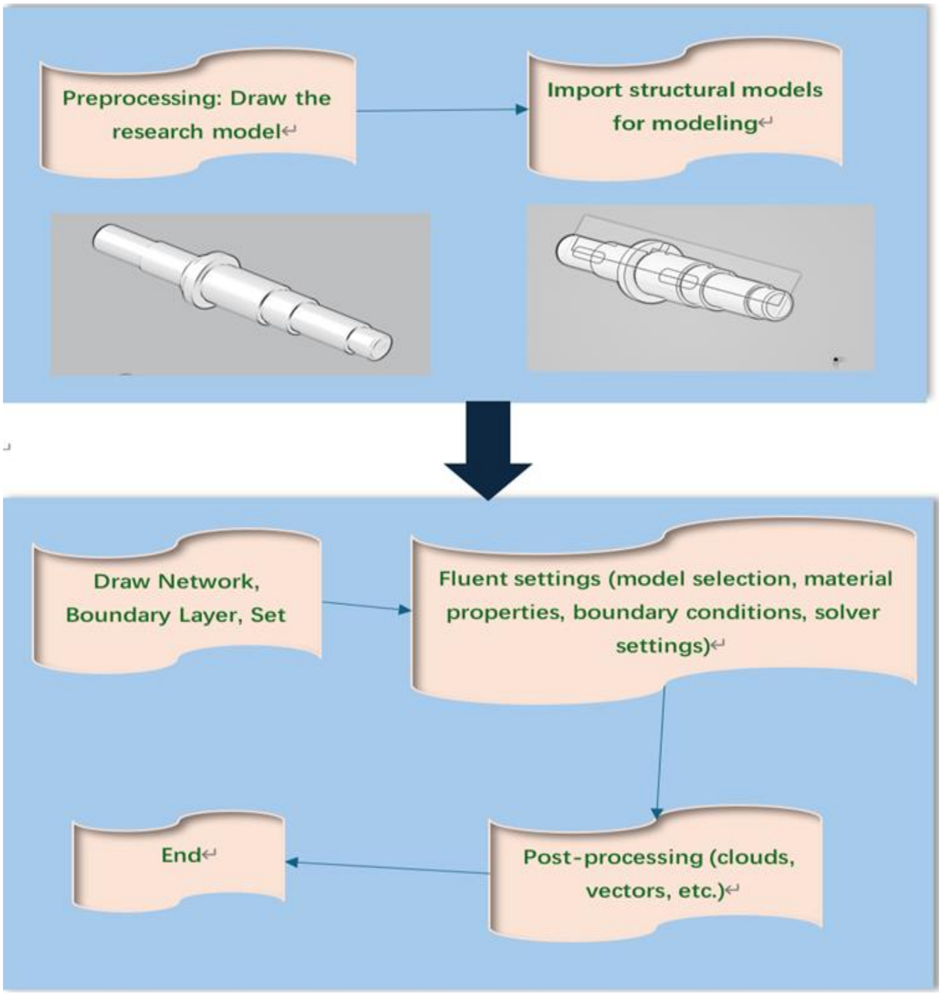


Figure 12 Ansys flow

Romax[39], a key design and simulation analysis software for transmission systems, offers crucial guidance in expediting design processes, reducing testing expenses, enhancing structural optimization, boosting transmission efficiency, and mitigating vibration and noise levels. Romax is employed to design and validate the strength of gears. It evaluates the tooth surface contact stress and the bending stress at the root of the gear, thereby confirming the gear's reliability and durability. The software can also forecast the bearing life within the shaft system, analyzing how strain conditions, lubrication, and other factors influence bearing life under varying operational conditions. This aids in the selection and maintenance of bearings and considers the interactions between gears, bearings, shafts, and other system components. The efficiency, noise, and vibration of the entire transmission system are assessed and optimized to enhance system performance. It is renowned for its high professionalism and efficiency, delivering intuitive results. However, it has limitations in the modeling of the axial system's dynamics, poor data compatibility, and may require additional processing. (See Figure 13 for further details.)

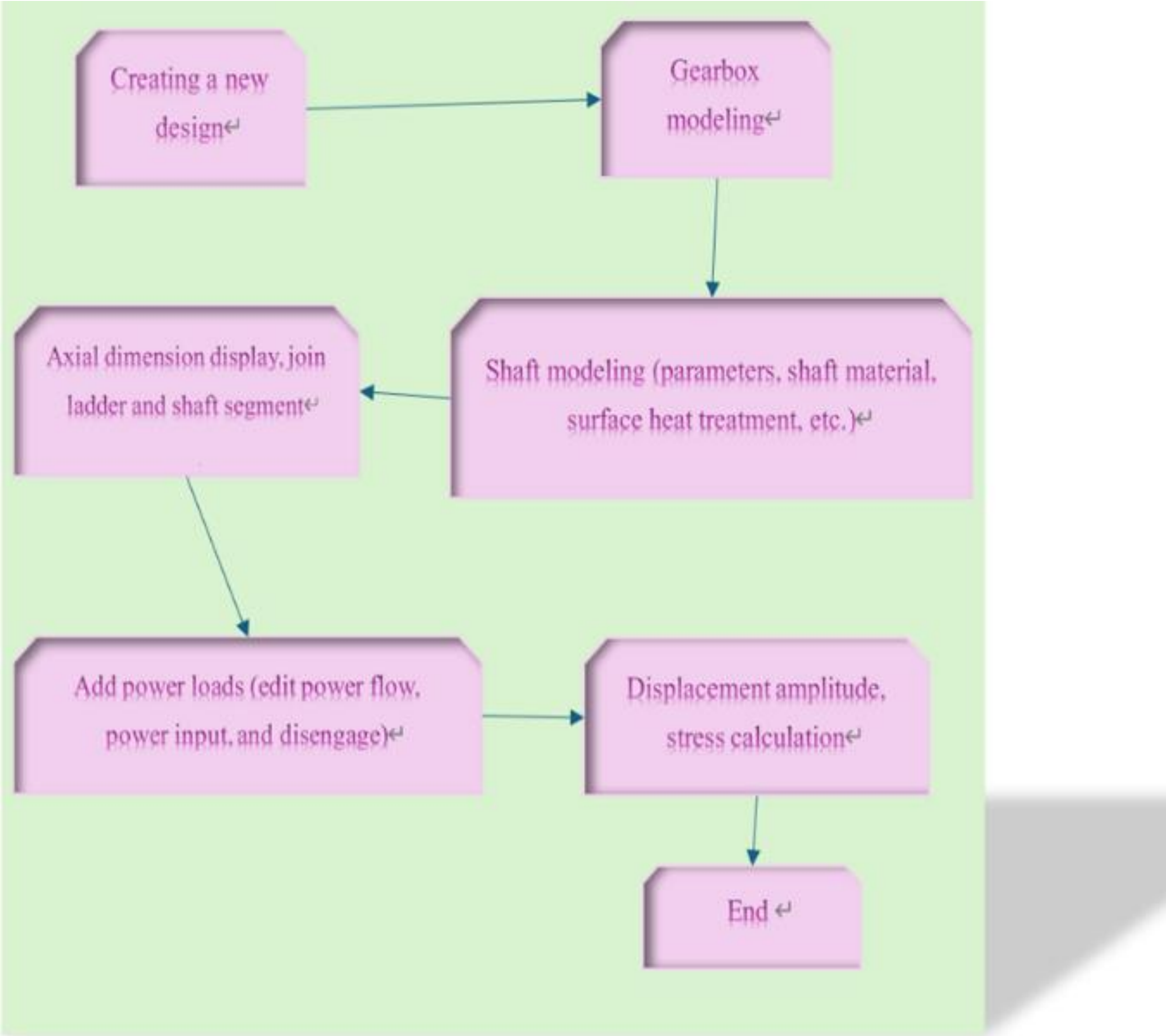


Figure 13 Romax flow chart

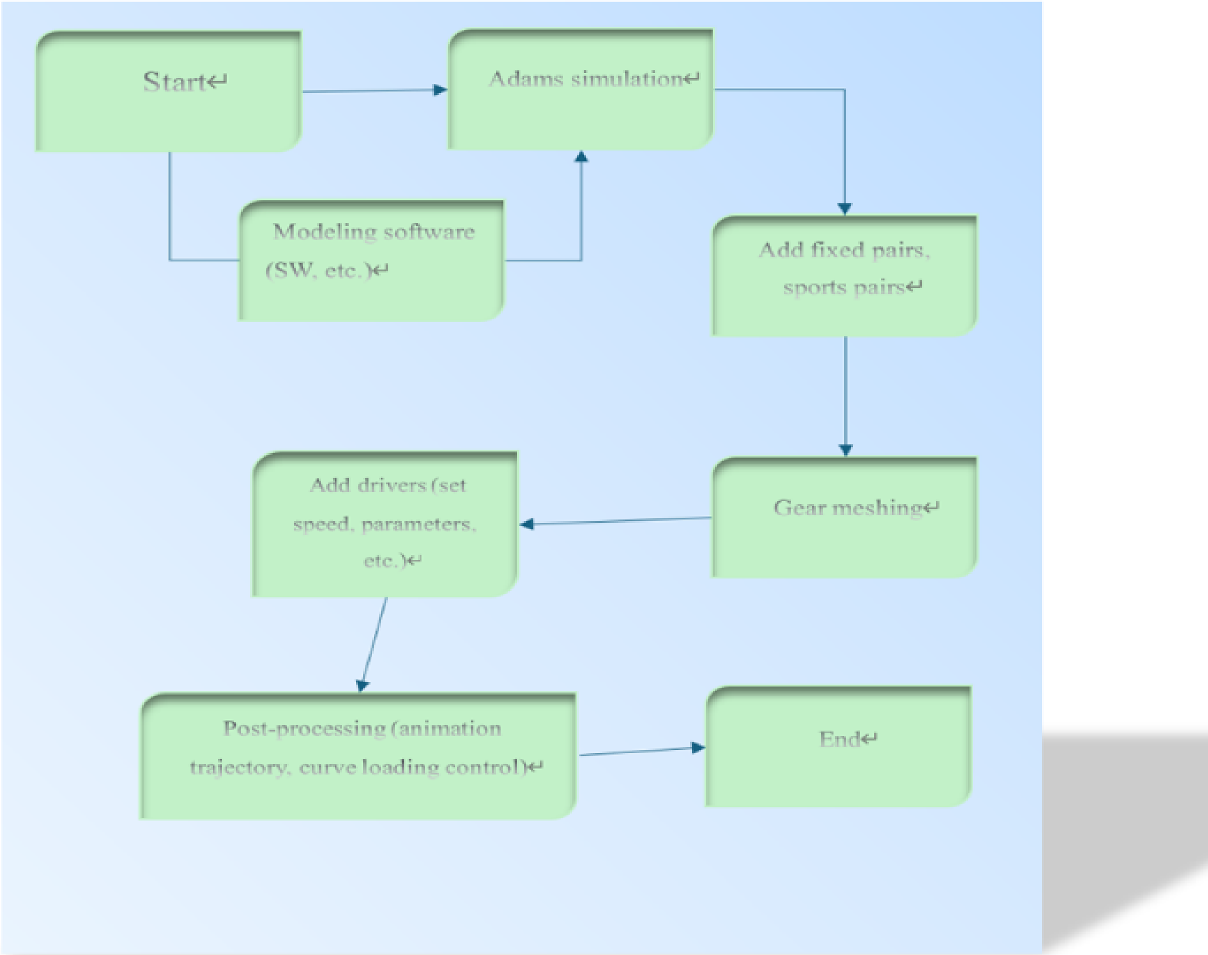


Figure. 14 ADAMS flow chart

ADAMS[40] (Figure 14) is the leading software for mechanical system dynamics simulation. It enables the execution of kinematic, dynamic, and static analysis, with the capability to output simulation results in the form of animations and graphical plots. It is capable of simulating the motion dynamics of the ship shafting transmission system and analyzing the dynamic behavior of the shafting under diverse conditions, such as the elastic deformation of the shaft and the meshing action of gears. The analysis of bearing friction and lubrication contributes to the evaluation of the system's dynamic performance. It can also simulate fault scenarios within the shaft drive system, such as gear wear, bearing failure, and loose joints, by adjusting the parameters of the shaft power system, including the shaft diameter, gear count, and bearing type, thus enhancing system performance and reliability while reducing design costs. The software's visualization capabilities are robust, with high 3D calculation accuracy and efficiency, integration, and a comprehensive software package. However, its geometric model accuracy and limitations in material and thermal properties are noteworthy.

Additionally, Abaqus is employed, primarily for auxiliary calculations involving the ship's axis. For instance, simulations are conducted in conjunction with Adams and MATLAB. This is achieved by utilizing support files located within the Adams installation directory and exporting models from Adams/Controls to generate Adams models in the form of both model files and interface files for MATLAB. Upon executing the file within MATLAB, the fusion of machinery and control systems into a unified model can be accomplished, enabling the resolution of complex engineering problems. This method finds applications in a wide range of engineering and scientific disciplines, encompassing control system design, kinetic analysis, and the development and testing of sophisticated products. ANSYS and MATLAB are also used in conjunction for joint simulations, with MATLAB scripts controlling ANSYS analyses and facilitating real-time data transfer. This technique is adaptable for various engineering analyses and optimization tasks (Figure 15).

Table 1. Comparison of simulation softwares

Strengths, weaknesses, and complementarities	ShaftDesigner	ANSYS	Romax	Matlab
Advantages	Targeted Rich in function; Wide range of Applications; Easy to model; Computationally accurate.	powerful High precision Good openness. Innovative professional.	High computational efficiency, Results Intuitive.	High efficiency, flexibility Tool Box Rich, Algorithm is simple
Disadvantages	High software costs, Difficulty of learning.	operating complexity, Computing Requirements Big.	Functional Limitation Data compatibility,	Calculation Limited 3D Modeling, Low computational efficiency.
Complementarity	In the overall design of the ship shafting transmission system, you can first use ShaftDesigner for shafting initial design and alignment calculation, Finally, the structure of the system is verified and optimized by ANSYS or Adams. Matlab can provide support and supplement for other software in algorithm development, digital processing, and control system design. Co-simulation of ADAMS and Matlab, Co-simulation of Adams and Matlab, Co-simulation of Romax and Abaqus [41] Shaft Designer has the difficulty of learning, but it is convenient to build models. Adams and Matlab are limited to 3D models, which can realize complementary modeling.			

5. Research results of stern shaft propeller

5.1 Tail shaft seal

The mechanical engineering challenge of the ship's stern shaft has been a key research area within ship engineering, and the research outcomes on ship stern shafts, both domestically and internationally, have had significant implications. The stern shaft is situated at the extremity of the ship's axis system and is interfaced with flanges, shaft rings, and mechanical seals. Its tail end has a conical shape to accommodate the propeller. The sealing integrity of the stern shaft is of paramount importance, as it is directly exposed to seawater. Poor sealing can lead to seawater infiltration into the engine system via the stern shaft. [42] In earlier ship designs, oiled asbestos rope was wrapped around the stern shaft and then glands were pressed to achieve a seal, but due to the continuous rotation of the shaft, significant friction losses occurred, leading to overheating and wear on the shaft diameter. Consequently, modern ships predominantly use leather bowls for sealing (Figure 15). Typically, this is divided into two sealing devices: the front seal, situated at one end of the engine room to prevent lubricating oil from the stern shaft tube from leaking into the engine room, and the tail seal, positioned at the propeller end to prevent mutual leakage between the stern shaft tube and seawater. The tail shaft seal is constructed in three parts, progressing from the exterior to the interior: the seawater chamber, the compressed air chamber, and the lubricating oil cavity. If the leather bowl's opening is left ajar, the pressure within the left chamber surpasses that of the right chamber, causing the sealing ring to be pressed firmly against the sealing device, thereby achieving a seal. The lubricated oil also serves as a safety feature, acting as an insurance against the failure of the air seal. Common simplex sealing devices [43] employ an elbow-type sealing design, utilizing the sealing ring's inherent elasticity and the force exerted by the spring to provide an initial pretensioning force. Under seawater pressure and lubricating oil pressure, the sealing ring is made to tighten against the lining pipe.

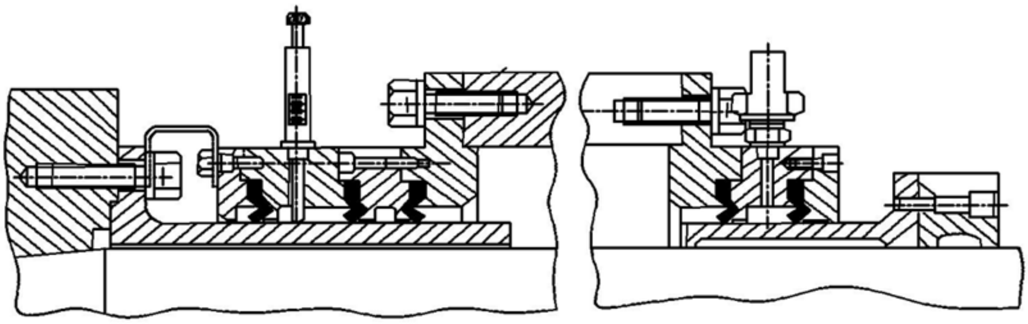


Figure 15 Structure of tail shaft seal

5.2 Coupling vibration test of the propeller, shafting, and hull

When the propeller is functioning, it emits a linear sound. Concurrently, the ship's hull is excited by the shafting system, leading to considerable acoustic radiation. The study of the coupled vibration and acoustic radiation within the propeller-shafting system and ship hull has received growing interest. [44] Research has concentrated on the coupling transfer characteristics of a scaled, inactivated shafting hull model on shore. Dai Mingcheng, et al.

[45] A full-scale test rig designed to analyze the coupling vibrations of propeller, shafting, and hull has been investigated, providing insights into the shafting's vibrational response. The test rig for the coupled vibrations of the propeller, shafting, and hull includes the test section, its bottom base featuring a wooden pier with a predetermined degree of elasticity resting on a beam, a tail static and dynamic loading device, and a bow support device.

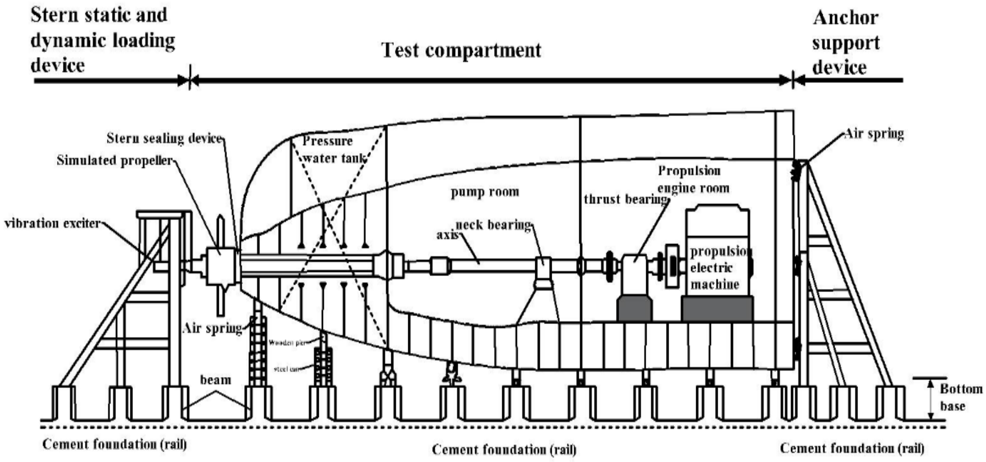


Figure 16 Real scale test bed for coupled vibration of propeller shaft system and hull[45]

5.3 Tug propeller

An innovation in propeller design is particularly evident in tugboats, where the majority use propellers as their primary means of propulsion. These propellers are driven by engines and exert force on the water as they rotate at high speeds, adhering to Newton's third law, which posits that for every action, there is an equal and opposite reaction. In response, water exerts a counterforce on the propeller, propelling the tugboat forward. Tugboat propellers commonly have larger diameters and operate at lower speeds to generate increased thrust. For instance, certain large port tugs are equipped with propellers up to 2-3 meters in diameter, providing substantial towing power at slower speeds. However, some tugboats opt for water jet propulsion systems. This system employs a pump to draw in water from the vessel and expel it at high speeds from the stern, with the water's spray propelling the tug ahead. Enhanced maneuverability makes water jet-powered tugboats ideal for navigating narrow channels and are particularly useful in inland ports or in shallow waters, as the lack of a protruding propeller reduces the risk of damage. The rudder, a critical component in tugboats, is essential for changing the vessel's navigational direction. Upon rotation, the rudder alters the current's direction of impact on the stern, prompting the tugboat to turn. Tugboat rudders typically feature a larger surface area and greater steering efficiency, enabling rapid and precise responses to directional commands during towing operations. Furthermore, many advanced tugboats are equipped with electric or hydraulic steering systems, further enhancing handling capabilities. The rudder's role extends beyond mere maneuverability, playing a pivotal part in practical applications as well[46]. For example, when dealing with large vessels or operating in complex aquatic environments, coordination among multiple tugboats is often necessary. These tugboats utilize wireless communication equipment to synchronize their thrust direction and magnitude. In situations like docking a large container ship, a tug at the bow may take charge of the ship's heading, while another at the stern manages the ship's backward and lateral movements, working together to ensure accurate alignment with the designated docking position.

5.4 Matching characteristics of ship, engine, and propeller

The primary objective of marine engine propeller matching is to optimize the interaction between the ship, marine diesel engine, and propeller, ensuring optimal performance and efficiency. As the propulsion force's origin, the diesel engine transmits this force to the propeller through the shaft system, which is designed to function harmoniously with the propeller's characteristics. The propeller then rotates through the water, overcoming the ship's resistance and maintaining a constant speed[47]. The optimal propeller matching is achieved by analyzing the impact of varying compatibility levels to ascertain the ideal parameters. This process involves considering factors such as the ship's principal speed, desired velocity, and load capacity, with the goal of developing an optimal design that enhances the ship's speed, safety, and economic efficiency. The operation of ship propulsion is essentially an energy conversion process, and the match between the ship, engine, and propeller significantly affects the efficiency of this conversion. Consequently, it also affects the economic aspects of the propulsion unit, which can be evaluated using metrics such as fuel consumption rate, total fuel consumption per voyage, and fuel consumption per kilometer.

The load power-speed characteristic curve and the main engine power-speed characteristic curve for the propeller are typically plotted in the same coordinate system to examine the energy balance relationship between the ship and the medium being propelled, as depicted in Figure 17.

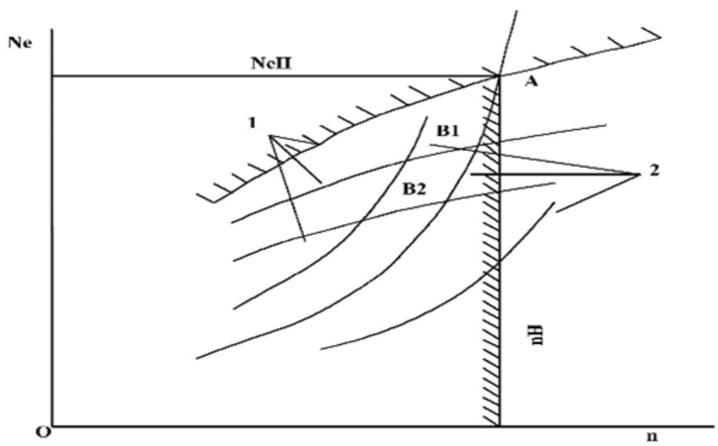


Figure 17 Drive-load balance diagram[47]

Curve1- Main Engine Power - Speed Characteristic Line;

Curve2-propeller load-speed characteristic line;

Points A, B1, and B2 - the equilibrium point. The diesel engine and the screw propeller can only achieve a smooth working state on A, B1, and B2 (curve intersection), and the state of other points is in an unstable working condition. [48].

5.4.1 Ship resistance characteristics

The ship's resistance characteristics (Figure 18), a pivotal determinant of engine and propeller performance, are critical. Generally, ship resistance, denoted by the symbol 'Rt', increases with the vessel's speed, 'vs', typically exhibiting a variation spanning three to six orders of magnitude. This relationship is further modulated by the type of ship, with distinct resistance profiles observed among various ship classes. Under specific ship types and environmental conditions, the relationship between hull resistance and speed, 'vs,' is as complex and nuanced. Follows[47]:

$$Rt = f_1(vs) \tag{18}$$

The power necessary to overcome hull resistance at speed vs is referred to as the Extreme Effective Power (Pe):

$$P_e = Rt \bullet vs \tag{19}$$

Upon activation of the propulsion unit, energy is inevitably lost during the shafting transmission and the conversion of propeller rotation into thrust. As a result, the actual power output is reduced compared to the marine diesel engine's rating. Equation reveals that the effective power, denoted by 'Pe,' is a function of speed, 'υs'. The graph illustrating this relationship between effective power and speed is commonly termed the effective power curve.

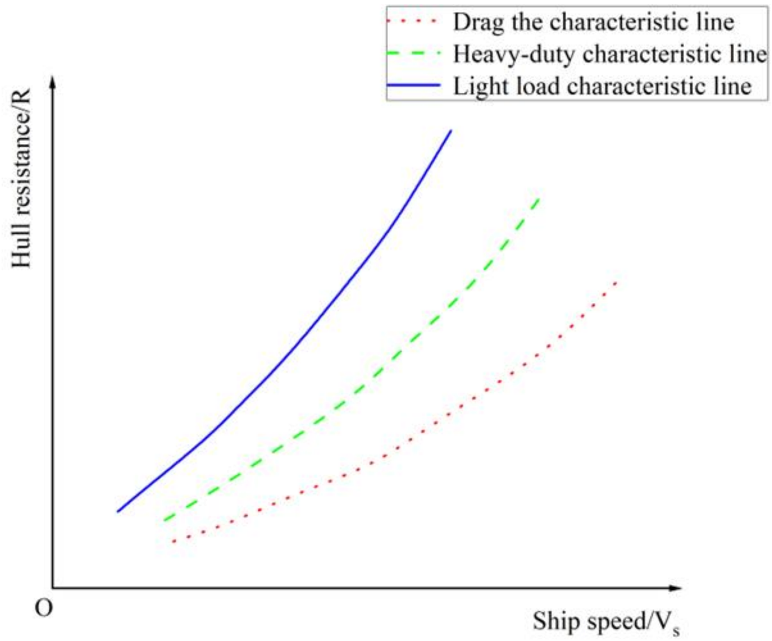


Figure.18 Ship resistance characteristic curve [35]

5.4.2 Ship main engine characteristics

Diesel engine power plants are categorized into low-speed, medium-speed, and high-speed units. Key operational parameters of diesel engines include the maximum combustion pressure, 'pz', the maximum exhaust temperature, 'tr', and the operating speed, 'n'. [49] The external characteristics of marine main engines, often referred to as speed characteristics, depict the relationships between operating speed, 'ne', main engine end torque, 'Me', and power, 'Pe', under varying operating conditions. These relationships are established by measuring a set of performance parameters while maintaining constant injection rates and adjusting the external load. Figure 19 illustrates that, with a defined main engine load (throttle position), the engine speed-torque and speed-power relationships can be established.

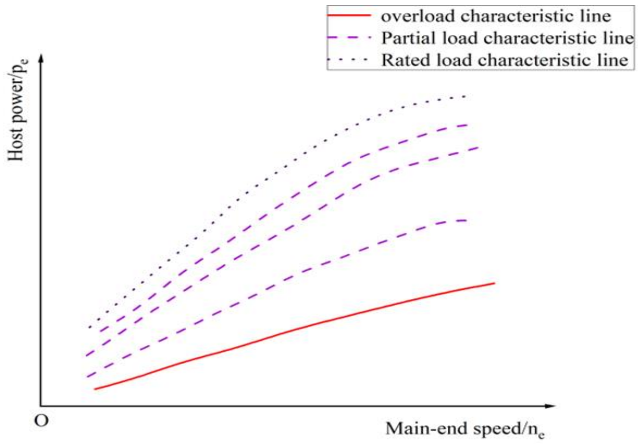


Figure 19 External operating characteristic curve of the main engine[35].

The effective power of a diesel engine is calculated by[48]:

$$P_e = c \bullet p_e \bullet n \bullet i \tag{20}$$

In the formula:

Pe - the effective power of the diesel engine,
Wc- the cylinder constant, m3
pe- the average effective pressure,
pan - the speed of the diesel engine, r / min
I - The number of cylinders
the dimensionless c and i are the structural parameters of the diesel engine, and the c and i are constant for a set of diesel engines. Therefore, the effective power Pe is only determined by the average effective pressure pe and the diesel engine speed.

5.4.3 Propeller characteristics

According to propeller theory [50], the function of the propeller is to absorb the power of the main engine and convert it into the thrust required for the movement of the ship, which itself has a rotational speed n and moves axially with the ship. The axial speed of the propeller relative to the water is called the advance speed, which is expressed by V_A [48]:

$$V_p = (1 - \omega) V_s \tag{21}$$

Form in: V_A ——Propeller speed, m/s

V - the speed of the ship, m / s

ω - the wake fraction of the propeller rotates one week,

the distance forward in its axis is called the process, expressed by h_p , can be calculated by the following formula:

$$h_p = \frac{V_A}{n} \tag{22}$$

Form in: h_p ——Propeller process,m/r

V_A ——Propeller advance, m/s

n——Propeller speed,r/s

Propeller advance coefficient J is equal to the ratio of propeller progress h_p to propeller diameter D, that is:

$$J = \frac{h_p}{D} = \frac{V_A}{n \bullet D} \tag{23}$$

The efficiency of the propeller can be derived from the above formula as follows:

$$\eta_0 = \frac{TV_A}{Q \cdot 2\pi n_p} = \frac{K_T}{K_Q} \cdot \frac{J}{2\pi} \tag{24}$$

The open water characteristic curve for the propeller can be derived (Figure 20). As the advance ratio increases, the propeller's thrust coefficient and torque coefficient decrease. Within a specific range, a decrease in the advance ratio can enhance the propeller's thrust and torque. The efficiency curve indicates that the maximum efficiency is achieved when the advance ratio reaches a critical value. For propellers of varying pitch ratios, the maximum efficiency value differs; higher pitch ratios correspond to higher maximum efficiency values and greater advance ratios, allowing for increased ship speed at a given diameter and speed. [51] In accordance with the propeller blade's working principle (Figure 21), lift, denoted as dy, and drag, as dx, are generated on the blade element's leading edge, with dy being perpendicular to the flow direction and dx aligned with the chord [52].

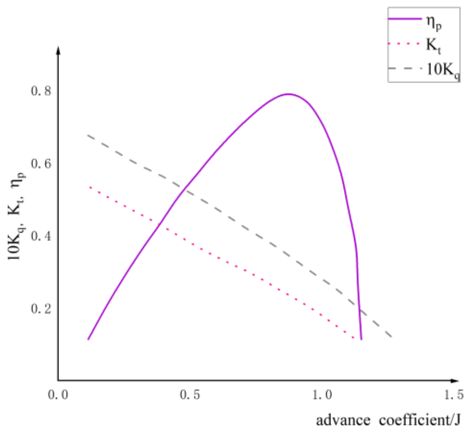


Figure 20 Propeller open water characteristic curve[48]

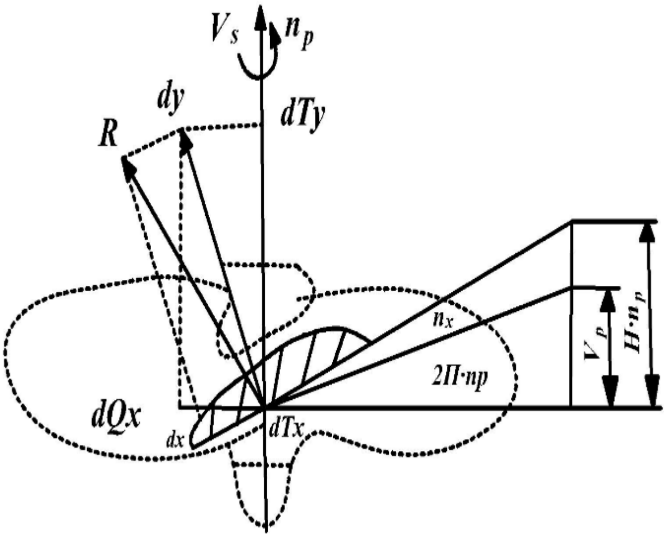


Figure 21 Working principle of propeller blade[48]

Grounded in wing theory and considering the open water characteristics of the propeller, the thrust and torque coefficients are derived, subsequently enabling the calculation of the thrust, 'T', and torque, 'Mp', for the current state of the propeller.

$$T = k_t \rho n_p^2 D^4 \tag{25}$$

$$M_p = k_q \rho n_p^2 D^5 \tag{26}$$

Form in:ρ——Density of water

k_t ——Thrust coefficient

k_q ——Torque coefficient

D - Diameter of Screw Rotor

k_t and k_q are both functions of the propeller advance coefficient J. For a propeller whose propeller diameter D is determined, the advance coefficient J varies with the advance speed V_A and the rotational speed n. J is an important parameter to indicate the motion state of the propeller [53] Because of the sudden change of propeller load, it is necessary to add the impact of torque load shock M_{pi} to the torque , namely:

$$V_p = (1 - \omega) V_s \tag{27}$$

Form in: M_{p1} ——Propeller torque considering load impact

Due to the interaction between the hull and the propeller, there occurs a reduction in thrust and an accompanying flow, which on the one hand diminishes the actual departure thrust, 'T,' and on the other, because the water surrounding the hull is influenced by the hull's movements, the resulting flow leads to the propeller moving at a speed relative to the surrounding water that is slower than the ship's speed. The true effective thrust is[48]:

$$T_e = T + \Delta T \tag{28}$$

Form in: ΔT ——Thrust reduction

$$V_p = (1-\omega)V_s \tag{29}$$

Form in: ω ——Wake coefficient

Figure 22 illustrates the propeller characteristic curve, where each point corresponds to a specific advance coefficient. A separate propulsion characteristic curve exists, with various values correlating to distinct thrust characteristics and efficiencies. At a constant speed, a decrease in the speed coefficient results in an increase in torque and a rise in the required power. Consequently, a lower speed coefficient corresponds to a steeper propulsion characteristic curve[54].

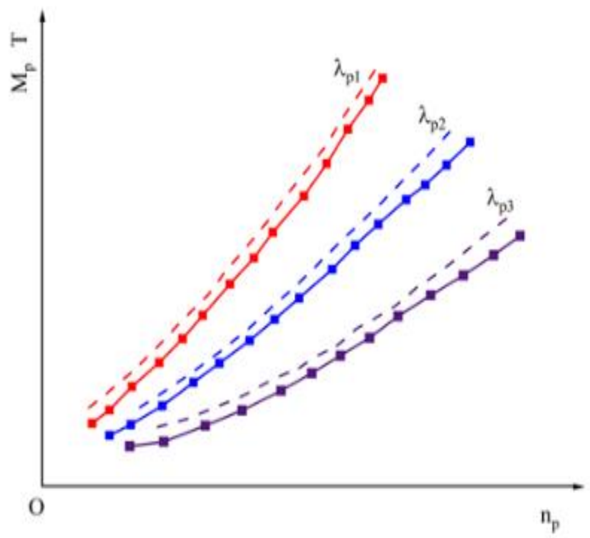


Figure 22 Propeller propulsion characteristic curve[48]

6. Shafting alignment

The marine propulsion shafting system, a key component of the power plant, relies on the shafting's alignment characteristics (primarily concerning the load conditions of its components) for its safety, stability, and sustained operation. Historically, the shipbuilding industry has witnessed numerous cases of abnormal wear on shafting bearings due to misalignment, as well as shaft fractures, both of which pose significant threats to marine navigation safety. To guarantee the alignment quality of ship propulsion shafts, numerous research institutions and scholars both domestically and internationally have extensively researched the methods for shafting alignment and its calculation techniques, amassing a wealth of shafting alignment theory and design standards [55-57]. The simplified mathematical model of propulsion shafting is depicted in Figure 23 [58].

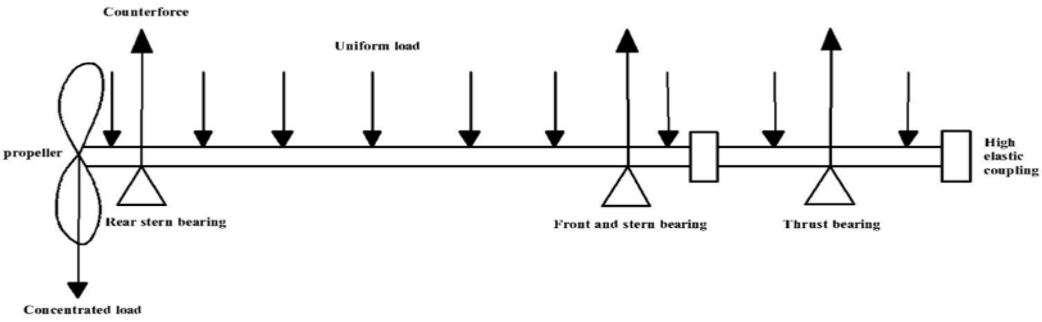


Figure.23 Simplified mathematical model of shafting alignment[58]

Shafting alignment techniques primarily encompass static and dynamic alignment methods. In static shafting alignment, the optimal pivot position and support load are determined through the computation of bearing conditions in linear states. Dynamic alignment more closely mimics the actual operating conditions of the shafting system. Beyond the static external forces affecting the shafting, the impact of the bearing liquid film in dynamic conditions is also accounted for. The technical approaches to static and dynamic shafting alignment are depicted in Figures 24 and 25[59].

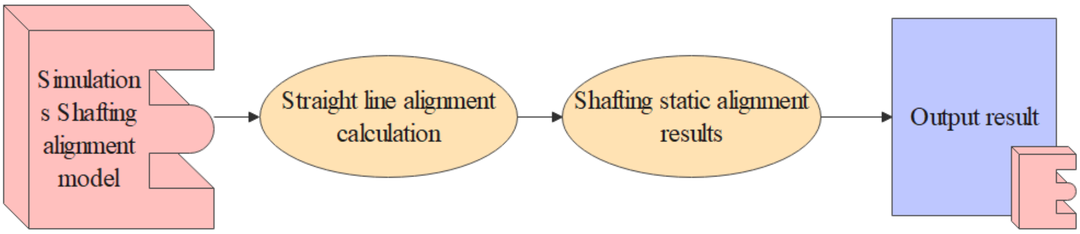


Figure 24 Shafting static alignment scheme

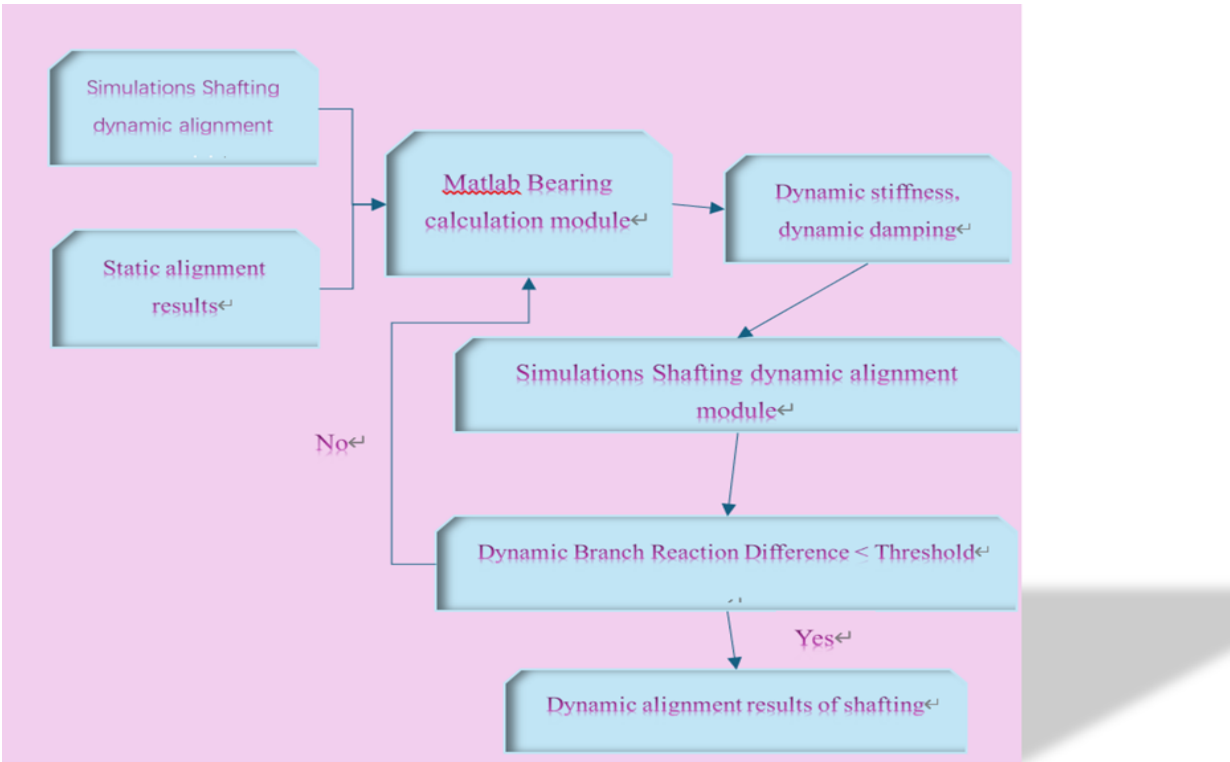


Figure 25 Shafting dynamic alignment

Additionally, the combined effects of the ship's cargo, wave loading, environmental temperature, and other coupled phenomena can lead to significant hull deformation. This deformation may result in the displacement of bearings, uneven load distribution due to bearing pad burning, poor gear meshing, and other forms of failure. Consequently, it is crucial to investigate algorithms that account for hull deformation in shafting alignment[60]. There are four primary methods to obtain hull deformation data: hull finite element modeling analysis, real ship measurements, referencing established hull deformation databases, and employing empirical formulas. The hull finite element modeling analysis involves calculating the relative deformation under full load and ballast conditions, as referenced to the baseline line. This is achieved by creating a three-dimensional finite element model of the ship's "mast machinery-cargo space" structure, considering the gravity and buoyancy distribution loads under the main load conditions. This method is capable of predicting the deformation of the "stern, engine room, and cargo hold" as well as the vertical deformation of the shafting's center line[61].

DNV has indicated that hull deformation is directly proportional to the square of the captain's load. During ship launching operations, the impact of hull deformation on shafting alignment is predominantly manifested through the relative difference in displacement between the ship's light-load and full-load states (Figure 26). To facilitate this analysis, NK Classification designates the main engine rear end bearing as the origin of the coordinate system (with the main engine end serving as the origin in the presence of a gearbox) and defines the height of the main engine rear bearing as corresponding to the relative displacement of the hull's structural components. After extensive data validation, a calculation method for the relative displacement of each bearing height due to hull deformation has been established[60].

$$\delta = \begin{cases} \delta_{\mathrm{b}}\left(X / L\right)^{1.5} & \left(X \le L\right) \\ \delta_{\mathrm{b}}\left\{1.5\left(X / L\right)\text{-}0.5\right\} & \left(X \le L\right) \end{cases} \tag{30}$$

Where δ_b is the relative displacement at the rear wall of the engine room; L is the distance from the rear wall of the engine room to the main terminal (or the output terminal of the main terminal); X is the distance to the host (or host output).

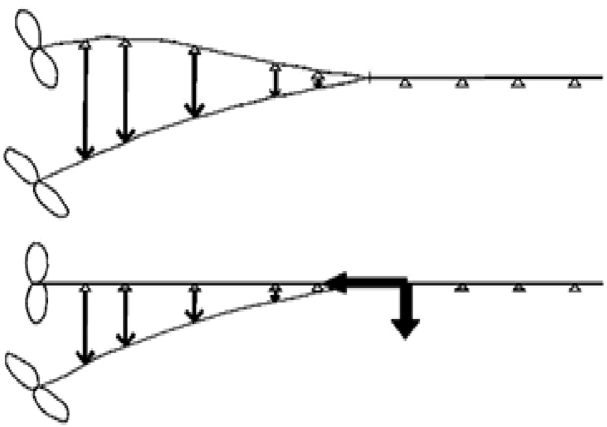


Figure.26 Relative displacement of bearing caused by hull deformation[60]

Consequently, hull deformation exerts the most significant influence on the middle shaft in the shafting of large ships. The factors associated with hull deformation are imperative to consider during shafting alignment.

7. The main challenges to be overcome for ship propeller shafts in the future.

In modern ship power systems, the podded electric propulsion system is widely used due to its high efficiency, energy saving, and low noise benefits. However, during operation, its shaft system often encounters issues such as wear and tear, noise, and a decrease in transmission efficiency. These problems not only affect the ship's navigation performance and comfort but may also have adverse effects on the long-term reliability and life of the equipment.

7.1 Shaft wear

Currently, the recommended calculation model for single-point support [62] incorporates those that reduce the shaft system to multi-point elastic supports, continuously distributed elastic supports, or contact supports [63-64]. Typically, multi-point and distributed supports are determined by selecting several points or sections of the shaft to replace equivalent supports of equal stiffness [65]. Guo Yuke et al. [66] investigated how different propeller vibration forces affect both the vibration characteristics of the shaft system and the wear on the stern bearing. Cao Hongrui et al. [67] developed a dynamic mechanics model to simulate bearing wear faults in intermediate bearings, and analyzed how changes in bearing surface and clearance affect shaft system vibration.

For stern tube wear, there are two types: open water lubrication and closed water lubrication[68]. The open water lubricated stern tube and tube assembly is suitable for vessels navigating in waters with clean water quality, free of mud and sand. This type of stern tube and tube assembly has only a forward sealing device, designed to prevent seawater from entering the ship. The closed water lubricated stern tube and tube assembly, on the other hand, consists of systems such as the stern tube, white metal bushing, fore and aft sealing devices, oil tank, and its pipelines, using lubricating oil as a coolant and lubricant. Due to its closed structure, it effectively isolates mud and sand, avoiding their wear on the shaft and bearings, and extends the service life of the stern tube and shaft [69-70].

Another aspect of wear is material selection; some materials have low friction properties and can also lead to wear, such as nitrile rubber (NBR). However, when used in blends with other materials, there may be an improvement. Zhou et al. obtained good low-speed friction performance by blending and modifying nitrile rubber with molybdenum disulfide nanoparticles [71], ultra-high molecular weight polyethylene and graphene powder [72], SiO_2 [73], or nano-SiO_2 [74]. Qu et al. [75] incorporated polyurethane (PU) and epoxy resin (EP) into nitrile rubber to prepare a novel ternary NBR/PU/EP interpenetrating polymer network (ipn), achieving a minimum friction coefficient 75.4% lower than that of pure nitrile rubber. It has been shown that blending with appropriate materials can enhance the tribological effects of nitrile rubber. An increasing number of high polymer materials are being used for ship fasteners to improve their performance. To reduce the friction of these materials, researchers have begun to focus on the study of material modification.

7.2 Shafting Noise

Ship noise is divided into two parts: air noise and structural noise. The sources of noise on a ship include main engines, auxiliary machinery, fans, and other mechanical equipment, which are basically located centrally in the machinery compartment. As shown in Figure 27.

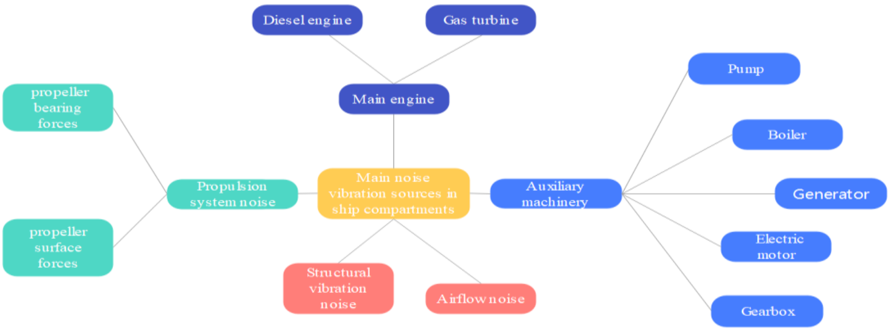


Figure.27 Illustration of ship noise sources

Noise is generated by vibration. The selection of the ship's propeller is a major influencing factor on hull vibration. The surface forces induced by the propeller and cavitation can cause the ship to experience severe stern swaying. Therefore, during the ship design phase, it is particularly important to choose appropriate propeller blade pitch, diameter ratio, thickness distribution, and number of blades. Propellers with more blades and a larger diameter ratio can significantly reduce cavitation, lower surface forces, and weaken vibration. Therefore, it is essential to ensure the quality of the propeller construction, for example, by reducing pitch errors and ensuring that balancing tests meet regulatory requirements. In the design phase, propellers should be chosen and arranged rationally to improve the stern profile, allowing the incoming and outgoing flows around the propeller to be smooth. Additionally, measures such as laying concrete above the propeller within the ship's hull and creating vibration relief holes in the ship's bottom plate in that area can effectively avoid excessive hull vibration. Large side-shrouded propellers can be used to reduce the impact of wake vortices in the stern propeller area and to prevent propeller tones[76].

Many scholars have also made significant contributions to the reduction of noise in ships. Peng et al. [77] compared the spectrum characteristics of WLRBs when frictional vibration occurred and during normal operation, which provided a reference for the identification of abnormal friction noise in stern bearings. Dong et al. revealed the relationship between the lubrication properties of polymers and friction noise performance, which contributed to the material selection in bearing design [78].

The vibration and noise problems of WLBs should be solved by reducing bearing vibration and improving the vibration isolation capacity of bearing. The generation of friction noise is related to the self-excited vibration caused by the stick–slip motion and lubrication state change between friction pairs. The friction coefficient [79] and the ratio of static and dynamic friction coefficients [80] could also have an impact on friction noise. The prevention and control of friction noise need to be approached from both structure and material to improve the lubrication performance and reduce the friction coefficient fluctuations[81].

7.3 Propulsion system

The design concept of the azimuth thruster comes from icebreakers and was first proposed by the Finnish shipyard Kvaerner Masa-Yards (KMY) and ABB Company in 1989. Due to its unique advantages, it has gradually garnered attention. The thruster housing propulsion system boasts advantages such as high efficiency, energy saving, environmental friendliness, ease of operation, low vibration and noise, and high space utilization inside the cabin [82-86].

The azimuth thruster breaks through the conventional design paradigm of prime mover plus an open drive shafting, eliminating the traditional shafting accessories and rudder. Therefore, compared to conventional propulsion methods, it has many characteristics in terms of ship design, performance, manufacturing, and maintenance [87-91]. For example: flexible space configuration, saving cargo volume; improved ship maneuverability; enhanced hydrodynamic performance, fuel savings, and good economic efficiency; reduced noise and vibration; high modularity for easy installation and maintenance. Simultaneously, the podded propulsion vessel is more maneuverable and has better safety performance than traditional mechanical propulsion vessels.[92]

Foreign research on podded propellers has been more in-depth, but China's research on podded propellers started late and is still in the early stages. Domestic institutions are also accelerating their research by absorbing successful foreign experiences. Researchers such as Ma Pin, Ji Luming, and Chen Xianggang attach great importance to the development of podded propellers and have conducted a series of studies. These studies have analyzed and demonstrated aspects such as the reliability of podded propellers, hydrodynamic performance analysis and calculation, experimental methods for podded propulsion devices [93]. Shen Xingrong and others have analyzed and introduced the model testing methods, characteristics of testing, issues to be noted in open-water and self-propelled tests, as well as several methods for numerically simulating and forecasting the hydrodynamic performance of podded propellers [94]. Dalian Maritime University has conducted detailed research and analysis on the structural characteristics and development trends of mainstream podded propellers internationally, as well as the main systems of the Azipod podded propulsion system, and has proposed a management and control scheme based on configuration software and PLC for systems that require routine maintenance checks by engineers [95]. Shanghai Jiao Tong University has conducted numerical research on the hydrodynamic problem of podded propellers using CFD methods, and validated it using a single-screw tug podded propeller as the experimental object [96].

Through integrated propulsors such as stern thrusters, bow thrusters, and lift thrusters, the propulsion efficiency is optimized to reduce energy consumption.

The development of integrated deep-sea electrical propulsion systems in foreign countries started early, and a complete design methodology has been established from propellers, propulsion motors to drive systems. These deep-sea propulsion systems exhibit strong reliability and excellent comprehensive performance. In contrast, China's deep-sea propulsor industry has made rapid progress after the 21st century, going through stages of independent exploration, direct procurement, import and absorption, and independent research and development. Currently, independently developed deep-sea propulsors have been applied to the manned submersible "Jiaolong," which operates at full oceanic depths, and has demonstrated excellent comprehensive performance[97].

In integrated electric motor propellers, the water-lubricated radial bearings mainly bear the gravity of the impeller shaft. During the design process of their water-lubricated radial bearings, in addition to considering the radial bearing capacity, it is also necessary to meet the requirements for shafting stability and to try to maximize the water film stiffness of the radial bearings[98]. This can greatly enhance the propulsive efficiency of the vessel.

The other is ship electric propulsion, with some scholars focusing mainly on variable-frequency speed regulation devices in ship electric propulsion systems, and using simulation software (such as EMTP) to conduct simulation studies of ship electric propulsion systems [999-100]. There is also a group of scholars whose research emphasizes the relationship between the motors and generators in the electric propulsion system [101-102]. However, the drawback of the existing research is that it is too isolated and concentrated on the characteristics of motor and generator control devices, lacking comprehensive and systematic modeling and simulation of the entire ship electric propulsion system [103]. Therefore, it is necessary to rely on Matlab/Simulink software to build an overall model of the ship electric propulsion system[104]. Electric propulsion is still an important research direction for ship shaft systems.

8. Conclusion

1) In the realm of theoretical research and experimentation on drive shaft systems, scholars have developed numerous theoretical models for torsional vibration, including the lumped mass model and the distributed mass model, which are utilized to mimic the circumferential periodic motion of the shaft system subjected to torque excitation. Standard calculation techniques encompass the Holzer method and the transfer matrix method, among others. These techniques are adept at determining the vibration characteristics of shafting, including natural frequencies and vibration modes, forming the foundation for shafting design and analysis. With respect to shafting alignment, various calculation methods have emerged, such as the three-moment equation method, the transfer matrix method, and the finite element method. Each method boasts unique strengths and limitations, allowing for a tailored selection based on the specific shafting structure and operational conditions. For instance, while the finite element method adeptly captures complex shafting structures and boundary conditions, it demands considerable computational resources. Conversely, the transfer matrix method, while expedient, is less suitable for complex configurations.

2) The research on stern shafts and propellers reveals a significant modal coupling between the propeller-shaft system and the hull. This implies that the vibration modes of the two systems are interdependent, with the coupled system exhibiting distinct modal frequencies and mode shapes compared to those of the individual propeller, shaft, or hull. A complex interaction exists between the hull and the propeller shafting. Enhancing hull line designs and the arrangement of the propeller shafting can significantly improve both the

propulsion efficiency and the navigational performance of vessels. The first-order unsteady force constitutes the majority of the energy consumed by the unsteady excitation of the propeller. Varying the grain fullness has minimal impact on the experimental objectives and satisfies the required engineering precision. The numerical method used to predict the coupled vibration and noise between the propeller shafting and hull is subject to modification and refinement through experimentation.

3) The axial cohort aspect involves both static coherent computation and dynamic coherent modeling. The combination of simulation and MATLAB facilitates the intuitive modification and testing of axial coefficient data, enhancing the efficiency of modeling and analysis in the study of axial cohesion issues. An optimization algorithm is applied in shafting alignment calculations, significantly reducing computation time and increasing efficiency without compromising reliability. Moreover, shafting alignment is a complex physical and mathematical problem. The integration of the enhanced optimization algorithm with shafting design calculations can enhance working efficiency， particularly in ship design. As a critical factor in shafting alignment, hull deformation influences the load distribution on bearings and is imperative to consider, particularly for large ship shafting alignment.

4) The principal challenges confronting marine shafting drive systems can be broadly classified as follows: enhancing transmission efficiency, reducing energy loss, and improving system reliability and endurance. A review of the literature on ship shafting transmission, both domestic and international, reveals that this technology has been evolving for many years, with varying degrees of maturity and application between countries. Foreign nations have been at the vanguard of the adoption of shaft washer and braking technologies, while China is gradually narrowing the gap, particularly in the application of shaft washing technology. The country has initiated the implementation of sophisticated washer technologies and the optimization of bearing load distribution. Presently, China is placing a premium on dependability and longevity of marine shafting, in addition to the localization of alternatives. This is being achieved by way of increased R&D investments and the enhancement of the self-controlled transmission capability of marine shafting systems. Moreover, fault detection technologies are being utilized. In foreign countries, there is a greater emphasis on technological innovation, the development of ship shafting intelligence, and automation, as well as the utilisation of high-performance materials to enhance the efficiency and performance of shafting transmission. It seems inevitable that shafting drive systems internationally will evolve towards intelligent control and remote monitoring in the near future. The integration of the Internet of Things (IoT), big data, and artificial intelligence makes real-time monitoring and intelligent diagnostics of vessel shaft drive systems a realistic possibility. Concurrently, the utilisation of advanced composite materials, smart sensors, and adaptive control systems aims to achieve more efficient energy conversion and precise control. Moreover, environmental sustainability and energy efficiency remain critical avenues for future development, with a focus on adopting materials and technologies that consume less energy to mitigate environmental impact.

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