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NATIONAL TECHNICAL UNIVERSITY OF ATHENS
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Diploma Thesis

Classification Societies' Regulations on Shaft Alignment

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ABSTRACT

Proper Shaft Alignment ensures the uninterrupted and optimal operation of a vessel. Erroneous shaft alignment has serious consequences, including malfunctions and significant damage to a ship's shafting system. The implications could range from vessel service interruptions and property loss to environmental disasters. As a result, it is critical that all parties responsible for the design, construction, operation, and monitoring of vessels take all required procedures and precautions to eliminate the possibility of the aforementioned undesirable situations occurring.

The Shaft Alignment procedure of a vessel is a complex and multidimensional matter, as the responsible engineers are asked to regulate and fine-tune numerous parameters of the Shafting System, which are dependent to one another, while the entire Shafting System is significantly sensitive to several characteristics of the vessel's structure and operating profile. Furthermore, the Shafting System necessitates careful handling while taking measurements or installing components and mechanisms, as clearances are on the order of a few millimeters.

Classification Societies are responsible for evaluating and verifying compliance to high standards that ensure the vessel’s safety and seaworthiness during its entire lifespan, from design to demolition. Classification Societies ensure that all mechanisms, construction materials, structure components, design characteristics, and safety protocols guarantee the vessel's safe operation throughout its lifecycle, thereby protecting property, the environment, and life at sea. The creation and publication of Shaft Alignment Rules and Regulations falls within the purview of a Classification Society's responsibilities. However, each IACS Class has its own documentation strategies, terminology, and scientific approach to relatively similar Regulations. This phenomenon also applies to Shaft Alignment Regulations, making the process of collecting and comparing Regulations from various Classification Societies for professional, educational or comprehensive purposes challenging. One of the main ideas that stimulated this thesis is that during their studies, Naval Architecture and Marine Engineering students are frequently asked to gather, understand, apply, and compare the Rules and Regulations of various Classification Societies.

This thesis is divided into two major parts: The first part explains and discusses the fundamental principles and characteristics of a shafting system. The rules and regulations of several IACS member Classes (ABS, DNV, LR, BV, and ClassNK) are then documented, explained, and compared. The Rules and Regulations used for this thesis were the most recent updates that each Classification Society publicly provided. The variations and similarities between the regulations released by the aforementioned Classes in terms of technical approach, terminology used, limits set up, suggested tools, recommended practices, and methodologies are then highlighted and analyzed. This work may serve as a guide for educational purposes in the future by providing extensive explanation of the theoretical and operational principles, the main components, and the respective details of rules and regulations regarding shaft alignment in marine vessels, providing students with a comprehensive view of the matter.

Furthermore, Artificial intelligence (AI) is continually evolving, and it is increasingly making its way into our daily lives and important corporate sectors. In terms of document classification and management, AI is commonly used to handle, search, and analyze enormous amounts of data for educational or business objectives. Additionally, with the increased digitization of documents, the use of AI technology is becoming increasingly necessary. Large Language Models (LLMs) are thought to be a promising option because they can interpret and handle text, as well as generate human-like text and conduct language-related tasks.

The Second Part of this thesis is focused on integrating LLMs into a comprehensive document management system capable of understanding and responding correctly to specific technical queries regarding the marine engineer’s profession. In the context of this thesis, experimentation was conducted with CHAT-GPT 3.5 to acquire a better understanding of how this advanced LLM, replied to a series of technical questions about

“Shaft Alignment” and the applicable Regulations provided by Classification Societies. Studying and categorizing the answers provided by CHAT-GPT 3.5 focusing mainly on the answers that the AI model answered incorrectly or inadequately, a pool of questions of interest was created. These questions were then addressed to other opensource LLMs (like Llama 3, Nous Hermes 2 Mistral DPO and GPT4ALL Falcon), which had local document access to the rules, or their comparative summary presented in Part A. All of the questions, which were eventually incorporated in the final set utilized within this thesis’, represent a specific “Quality” Category regarding the performance of each LLM that was tested. Finally, the best performing open source LLM can be further enhanced via Fine-Tuning and be utilized for Academic, Educational or professional purposes, in the context of providing an AI model capable of answering technical questions specified in Shaft Alignment and the respective Classification Societies’ Regulations. Finally, the outcomes of this process may indicate that LLMs that are provided access in appropriate (Local) Documents or fine-tuned with targeted Datasets can be used for more advanced applications.

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List of Figures and Tables

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PART A

A. Introduction of Part A

A single 2-stroke diesel engine and a shaft system which transfers the power to the propeller, are the backbone of a modern commercial ship's propulsion system. In order for the ship to be able to move, the "shaft system" must transfer the necessary power from the main engine to the propeller. The shaft typically consists of three parts: the crankshaft, the intermediate shaft and the tail shaft. A reduction gearbox is a necessity when installing a 4-stroke main engine, since it makes the control over the exact rotational speed in which the power is transmitted to the propeller possible.

However, it is of great significance to determine the exact conditions under which the installation and the functioning of the propulsion system are going to take place. More specifically, this study concentrates on the shaft alignment issue. The weight of the shaft, as well as the radial forces which are excited due to the separate weights during the propulsion system's operation, need to be adequately supported. The propeller of the vessel is attached to the tail shaft, which is mainly housed inside the stern tube. The propeller shaft bearings, aim to absorb and support a significant proportion of the propeller's weight. On certain vessels, there may only be a single stern tube installed; however, this situation necessitates further investigation. Additionally, the intermediate shaft is supported by the intermediate bearings. The number of cylinders in the main engine determines the precise number of bearings on the crankshaft.

Modern construction projects often prioritize the development of vessels with powerful primary propulsion systems, exceptionally efficient propellers and minimal fuel consumption. This strategic approach serves to reduce the deadweight of the vessel while simultaneously maximizing its cargo capacity. Moreover, it aligns with environmental consciousness, fostering the creation of more eco-friendly maritime solutions. More specifically, new ship designs tend to focus on the following:

- The tendency to more efficient propellers and lower engine fuel consumption, which leads to slow speed installations. Thus, the need for large and heavy propellers in order to provide the necessary thrust arouses.
- As the Main Engines tend to aim at higher power, a larger diameter shafting is necessary, to ensure that the torque is effectively transmitted.
- Optimized, thinner and therefore more flexible hull structures.
- Maximization of the cargo capacity, thus resulting in restrained and limited engine room space, making the shafting structure shorter and more rigid.

The lower shaft speed along with the heavy loads from the larger shafting and propeller, result in significantly more load exerted.

Moreover:

- The weighs which are produced by the Main Engine, the propeller or the shaft produce radial shaft loads, which need to be supported by journal bearings (either stern tube bearings, line bearings or crankshaft bearings)
- At the same time, axial loads are transferred from the main engine thrust bearing to the ship's structure

Due to all of the pre-mentioned facts, the need for an adequate and successful Shaft Alignment Analysis and Design Procedure is highlighted.

The actual meaning of the term "Shaft Alignment Analysis and Procedure", is in general the determination of:

- The optimal number of bearings which are going to support the shaft in critical positions

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- The precise longitudinal position of the above-mentioned bearings
- Their precise vertical offset
- The angle of the support bearings, measured from the established reference line
- The specific dimensions of the bearings and
- The reaction forces on each bearing, measured in several operational conditions.

The way the drastic changes and characteristics of the newly designed vessels affect the shaft alignment analysis, have as follows:

- The increased propeller weight results in substantial cantilever load fatigue of the aft stern tube bearing, making it harder for the bearing to create and preserve an adequate lubrication layer during low revolutions or high-speed maneuvering scenarios.
- The larger diameter and the alterations in the length and rigidness of the shaft result in a mismatch between the flexibility of the propulsion shafting and the hull; the more rigid the shaft, the more vulnerable it is to bearing unloading phenomena.
- Due to the optimized and more flexible hull and the relatively stiff shaft, the propulsion system becomes even more sensitive to hull deflections.
- The increase of the load which is exerted due to the low shaft speed and heavy loads from a big shafting/propeller, lead to more pronounced misalignment on the bearings.
- Nowadays, the most common building method which most of the Shipyards follow, is the "Mega Blocks" method, which makes it hard to control the alignment accuracy.

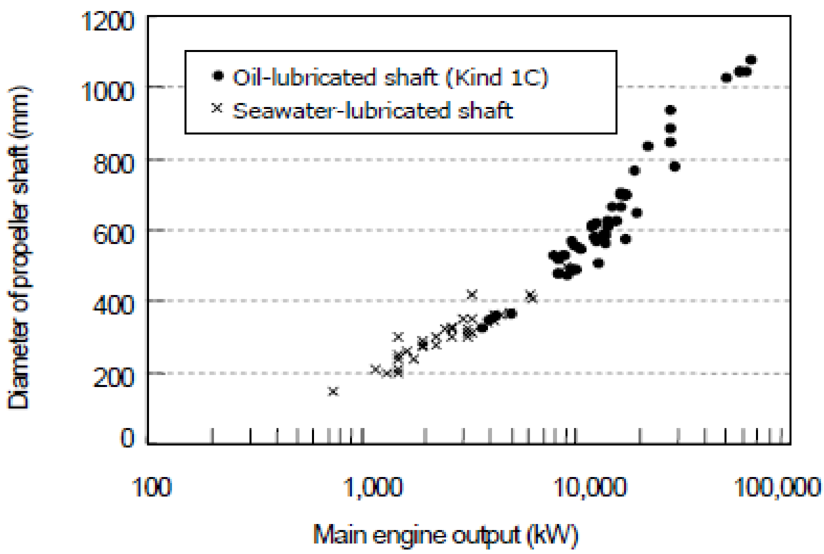

Figure 1: Relationship between propeller and shaft diameter and main engine output, ClassNK [24]

As a result, the alignment of the propulsion system becomes particularly sensitive to small deviations in bearing offsets or angular misalignment. This sensitivity complicates the alignment analysis and the shaft alignment on-board procedure and could lead to severe performance issues and damage.

Over the years, the need for specific, precise and effective regulations regarding Shaft Alignment Procedures was highlighted. As a result, all the Classification Societies have published their own methodology and requirements, in order to ensure the uninterrupted operation of the vessels which are registered to them. The purpose of the first part of this study is the comparative exploration and the respective technical aspects of each Classification Society's "Shaft Alignment" related regulations.

A.1. Classification Societies Chronology

During the 18th century, ship owners, merchants and captains concluded that in order to limit the business risks of sea transportation activities, the needed to pay greater attention to each vessel in a more technical, precise, focused and organized way.

The first Classification Society, Lloyd's Register, was then established in a Coffee House in London. Up until then, the predictions on whether a ship was seaworthy and safe were entirely based on the experience of the seafarers and captains, and not on legitimate documents and practices.

At the beginning of the so called "Coffee Shops", the basic idea, was for the "stakeholders" to share losses with each other, so that they could share the profits as well, if the vessel's journey was completed successfully. Practically, this meant that the names of certain vessels were written on paper and then stuck upon the walls of the Coffee House. If someone thought that this specific ship is safe to sail, he signed his name on the paper, thus committing to all of the prementioned.

As the times changed, so did the designing of ships. As steel superseded sail and timber was replaced with iron and steel, the ships got bigger, more efficient and more complicated, thus resulting in the need for the formulation of more specific guidelines. On around 1860, stricter guidelines, in the form of Rules, started to develop regarding the construction of vessels. In this study, the following IACS Member Classification Societies are of interest:

- American Bureau of Shipping (ABS): American Classification Society, established in 1862 in New York City. ABS Headquarters are currently based in Houston, Texas.
- Det Norkse Veritas (DNV): Norwegian Classification Society, established in 1864. DNV headquarters are currently based in Oslo, Norway.
- Bureau Veritas (BV): French Classification Society, established in 1828 in Antwerp, Belgium. BV headquarters are currently based in Paris, France.
- Lloyd's Register (LR): English Classification Society, established in 1760 in London, UK. LR headquarters are currently based in London, UK.
- ClassNK (Nippon Kaiji Kyokai): Japanese Classification Society, established in 1899. ClassNK headquarters are currently based in Tokyo, Japan.

Figure 2: American Bureau of Shipping; Official Logo

Figure 3: Det Norske Veritas; Official Logo

Figure 4: Bureau Veritas; Official Logo

Figure 5: Lloyd's Register; Official Logo

Figure 6: ClassNK; Official Logo

At this point, further explanation shall be given regarding IACS: The International Association of Classification Societies (IACS), is a non-governmental organization currently consists of 12 Countries-Members. The Classification Societies which make up IACS are covering more than 90% of the world's cargo carrying ship's tonnage. IACS purpose is to establish, develop, review and promote a set of minimum technical requirements regarding design, construction, maintenance and survey of ships and other marine related facilities. Additionally, IACS provides technical support and guidance to the International Maritime Organization (IMO). The Classification Societies which are IACS Members are required to demonstrate continuous compliance with quality standards, as they are determined by periodic audits. Consequently, all IACS Member Classification Societies are considered to be highly trustworthy.

The main responsibility of Classification Societies is to gather information regarding a ship's overall seaworthiness, which is nowadays translated into the respective actions of "regulation" and "inspection". Classification Societies have their main interests in the sectors of Maritime Safety and Environmental Protection. As a result, Societies need to drastically and constantly improve and enhance their regulations and policies, according to the market's movements and expectations. At this point, it is significant to mention the public and political attention, which is drown on the Classification Societies, as during the past years, the protection of human life and property as well as the protection of the environment, especially in the context of Decarbonization, are of great importance. These are considered to be the biggest challenges that Maritime Industry and Classification Societies must deal with.

In conclusion, some Classification Societies differ on the aspect of Regulations from another, however they all have the same goal: Verifying a vessel's seaworthiness, as well as the safety and the protection of human life, property and the environment.

A.2. A Ship's Propulsion System

Modern commercial vessels are thought to be extremely complex structures with state-of-the-art technologies installed, and one crucial requirement: Due to the fact that they must spend many days at sea, they must be completely autonomous, dependable, and self-sufficient. Approximately 80% of goods moving in today's world

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are thought to be shipped. During a voyage, a typical ship experiences multiple continuously variable weights, primarily from the waves in the sea and its own cargo. All of the aforementioned have the potential to impact the ship's substructures, especially the propulsion system.

The propulsion system of a ship has a major impact on its safety and seaworthiness. Therefore, before deciding on the precise parameters of the propulsion system, a great deal of analysis and calculation is done.

The propulsion system of a ship, consists of the following main parts:

- The Main Engine,
- The Propeller, and
- The Shaft

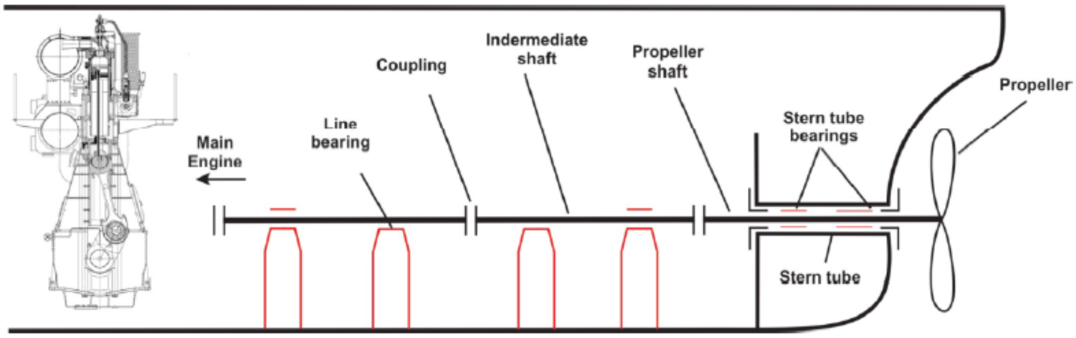

Figure 7: Typical form of a propulsion system for a large vessel

A.2.1. Main Engine

The choice of a vessel's Main Engine initiates the entire process of installing its propulsion system. The primary engine should possess the following fundamental qualities:

- Reliability and safety
- Low fuel consumption
- Enough power in order to adequately provide the propeller with the energy needed
- Minimum running and installment expenses
- Low rate of weight to BHP produced

The Main Engine is selected to usually operate on 80%-90% of its MCR, thus allowing its remaining power to be used in cases of severe hull fouling and/or unfavorable weather conditions.

A typical Main Engine marine installation, consists of the following parts:

1. **Bedplate**

The structure that supports the entire weight of the engine is called the bedplate. It is regarded as one of the engine's most heavily loaded components. Besides the need for the bedplate to be of rigid construction, it is also significant that it simultaneously is a flexible structure. That is, because it has two main roles: To support the Main Engine's entire weight and the forces and tensions which occur during its operation, as well as to absorb and relief the structures from the deflections which take place during the operational state of the vessel. More precisely, it is worth mentioning that the bedplate is placed upon the foundation plate, which is part of the vessel's structure. As a result, if the bedplate's construction is very rigid, under the impact of the hull deformations, this combination of material properties could result in snapping of the bolts that "lock" the engine upon the vessel, and consequently, the bedplate would be severely damaged. The Bedplate must also collect and drain lubricant oil to the sump, support the engine's running parts' dynamic load during operation, and maintain the crankshaft's alignment at all times. A Bedplate is constructed by joining two longitudinal girders which run its entire length, through a number of transverse girders mounted between each throw of the crankshaft and either side of the collar of the thrust. The bedplate is usually made out of cast iron, in case of relatively small engines. For larger vessels, the Main Engines come with their own bedplate from the manufacturer.

2. **A-Frame**

This particular Main Engine component has a visual resemblance to the letter "A" and it is used on large crosshead engines. It is constructed upon each one of the bedplate's transverse girders. It is in contact with the bedplate on the bottom and supports the cylinder block on the top. The jointing compound between the A-block and the Bedplate provides sufficient sealing, and the fitting and tie bolts between the A-Frame and the Engine provide appropriate fastening. Each engine unit (crankcase) has its own contained chamber as a result of the transverse grinders in the space generated between the bedplate and the A-frame.

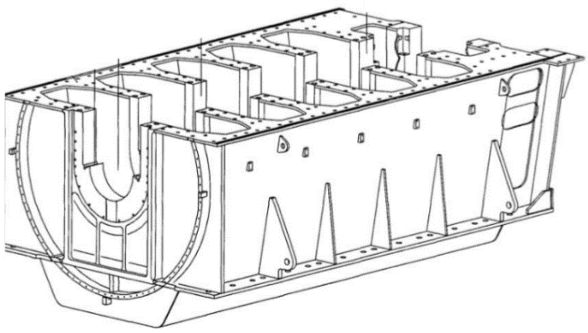

Figure 8: A Main Engine's Bedplate

Figure 9: A Typical A-Frame

3. **Cylinder Block**

Positioned above the A-frame, the cylinder block houses many engine components, including the scavenge area, stuffingboxes, jacket cooling water spaces, and the cavity required for the cylinder liner. The cylinder block's structure needs to be sturdy enough to withstand the combustion forces generated during main engine running without experiencing any issues. While fitted bolts are employed to fulfill location and alignment purposes, the tie bolts installed ensure that the three main engine components—the cylinder block, the A-Frame, and the bedplate-will not separate due to the firing forces during operation.

4. **Crankshaft**

One of the most crucial components of the main engine is the crankshaft, which works with the connecting rod to convert the reciprocating motion of the pistons into rotating motion for the propeller. More specifically, because of the inertial forces which occur because of the rotating masses of the crankshaft and the gas forces which act on each cylinder by the combustion of fuel inside the combustion chamber, the energy and as a result the torque needed for the ship to move is produced. The crankshaft sits on the top of the engine's bedplate. Additionally, it guarantees that the bearings will transfer the axial and rotational forces produced during the operation to the vessel's hull, so that the Main Engine's structure is relieved. The journal, crank webs, and crank pins make up the crankshaft.

Because the whole process of converting the reciprocating movement of the pistons into torque, which is accomplished mainly by the crankshaft, is the adequate thrust produced in order for the vessel to travel.

Due to the firing forces and the dynamic loads of the moving components, the crankshaft is heavily loaded with ranging bending stresses, shear stresses and torsional stresses. For the crankshaft to be capable of managing the continuous cyclic loads which occur during its operation, as well as the static loads which exist when it is at rest, it is usually made out of cast iron of high strength. However, the exact choice of the material is also based on the precise type of crankshaft (e.g.: fully-built, semi-built, welded etc.).

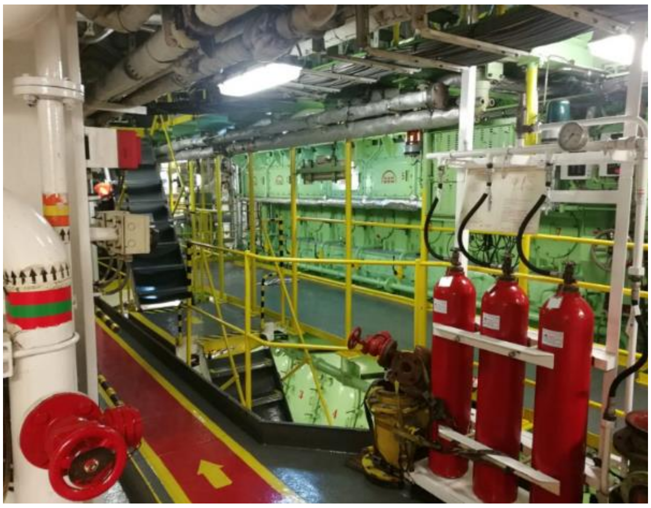

Figure 10: A Typical Cylinder Block

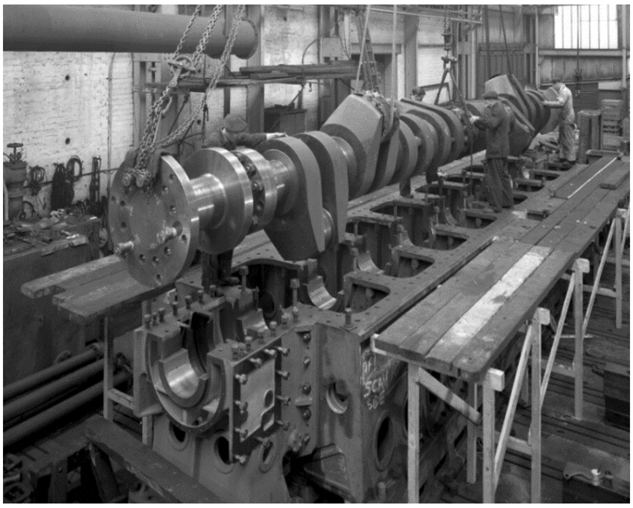

Figure 11: A Main Engine's Crankshaft

5. Connecting Rod

The connecting rod is used in order to connect the crankshaft on each crosshead. The connecting rod and the crosshead both move in tandem with the piston as it moves. The crankshaft along with the connecting rod, converts the reciprocal motion of the piston into rotation, so the crankshaft finally rotates because of this series of transmitted movements. More specifically, the piston transfers the power linearly to the crosshead through a piston rod, and then the crosshead transfers it to the crankshaft through the connecting rod. The "big end" of the rod connects to the crankshaft's crank pin, and the "small end" connects a) in the case of long-stroke reciprocating engines (large marine vessels): to the crosshead, or b) in the case of trunk piston engines (smaller engines such as generators): directly to the piston. The connecting rod also provides the crankpin with oil, which guarantees adequate lubrication and cooling. The connecting rod is thought to be extremely susceptible to fatigue failure since it experiences strong bending and buckling forces as well as continuous cyclic loads when the main engine is operating. The connecting rod rotates at the exact same speed as the crankshaft. However, except for the rotational movement, it also undergoes severe tensile (during the starting and the stopping of the Main Engine), compressive stresses (during the functioning of the Main Engine) and shear/bending stresses (because of the centrifugal forces which occur during its rotation). The overall stress of the connecting rod depends on the compression ratio, change air pressure, ignition timing, engine power and speed, torque and several other factors. The selection of the material which is going to be used for the manufacturing of the connecting rod is of vital significance, due to all of the pre-mentioned conditions. The rod should have high strength in terms of tension and compression, as well as the capability of sufficient resisting fatigue failure. The most common industry practice is for the connecting rods to be constructed out of steel. In case of higher compressive forces, aluminum alloy or cast nodular steel is used. On the other hand, when in need for successful performance under gith tensile forces, cast, forged or fabricated steel is considered to be the preferred choice.

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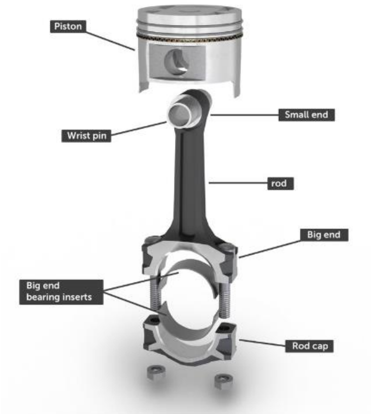

Figure 12: A Typical Connecting Rod

6. **Crosshead**

The rectangular-shaped crosshead serves as a "link" between the connecting rod and the piston. Through a circular pin called a "crosshead journal" or "crosshead pin," the connecting rod is attached to the crosshead. A telescopic pipe provides the lubricating oil needed for the crosshead pin. The lubricating oil then travels to the piston and crank pin through drilled bores. It facilitates the free movement of the connecting rod and transfers side loads from the piston and cylinder liner to the engine structure.

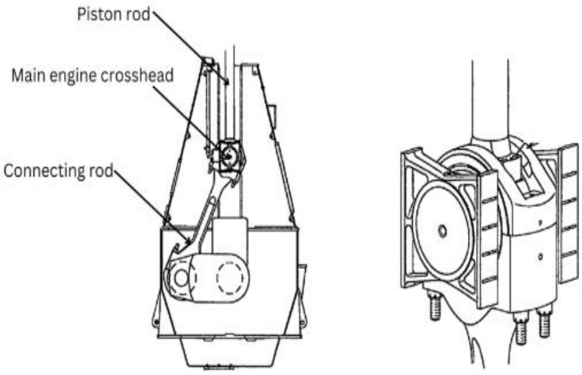

Figure 13: A Typical Crosshead

7. **Piston**

A composite part, of cylindrical shape, which reciprocates within one of the engine's cylinders. It moves as a result of fluid pressure in an engine and converts combustion forces into mechanical forces for the engine. The piston forms the lower part of the combustion chamber, as it seals the cylinder. More specifically, it compresses the air-fuel mixture during the suction stroke. The piston consists of piston crown and piston skirt screwed together, with a significant number of bolts (up to 16 or more). A piston generally suffers a great amount of shock and thermal loadings. The piston suffers from high temperatures which result in the surface of the piston being vulnerable in terms of erosion and burning. It is obviously crucial for its lifespan and sufficiency to be constructed from materials which are high temperature and corrosion proof. In some cases, steel alloyed with chromium and molybdenum is used, and in some exceptional cases, some pistons have a special alloy welded onto the part of the crown which is most exposed to high temperatures. A piston is exposed to compressive

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and tensile stresses which occur due to gas pressure and constant movement -inertia effect. It is also susceptible to thermal stresses

8. Cylinder Liner

This main engine component functions as the combustion chamber by fitting inside the engine's cylinder. Because it comes into direct contact with the air-fuel mixture during the suction stroke and actively contributes to combustion during the power stroke, it needs to be replaced on a regular basis.

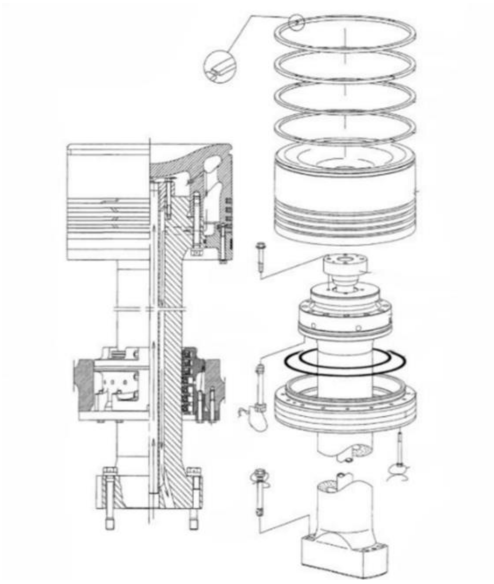

Figure 14: A Main Engine's Piston

Figure 15: A Main Engine's Cylinder Liner

9. Cylinder Head/Cover

Numerous monitoring devices, including the fuel valve, starting air valve, relief valve, indicator valve, exhaust valve, etc., are located inside its area. Usually, it's a metal casting that is placed over a cylinder to seal and guard the upper combustion chamber. Because it has water passages for water coming from the cylinder liner outlet, the cylinder head is also a component of the jacket cooling system.

10. Exhaust Valve

It fulfills the function of providing the exhaust gases with a "route" out of the engine. It is made up of a spindle and a valve housing. The exhaust valve opens to release the engine's internal gases when combustion is finished. The closing and opening of the valve are controlled by hydraulic means. Every Main Engine unit has a separate exhaust valve. The gases exit the vessel's structure through the funnel after passing via the manifolds and spool the turbocharger. A typical exhaust valve of a marine Main Engine should be constructed using materials with the following characteristics: High tensile strength, high-temperature proofness and resistance to high temperature corrosion.

Figure 16: A Cylinder Head/Cover

Figure 17: Exhaust Valve

11. Camshaft

The Camshaft is positioned near the top part of the engine, and it moves due to the crankshaft, via a timing chain. Operating the inlet and exhaust valves (for 4-stroke engines) or just the exhaust valves (for 2-stroke engines) is the camshaft's primary function. It transforms the crankshaft's rotational motion into reciprocating motion by passing it through the cams. It is also capable of controlling the starting air distributor and the fuel injection pumps. It is a piece of machinery that rotates at half the speed of the crankshaft (for 4-stroke engines) or at the same speed as the crankshaft (for 2-stroke engines), with several fixed cams positioned at various angles.

12. Turbocharger

A Turbocharger is a turbine wheel that runs by using the main engine's exhaust gases as "fuel". It has a centrifugal air compressor attached to it rigidly. Their primary function is to supply the combustion chamber with pressurized air, increasing the power output of the main engine. Large modern vessels frequently have two turbochargers. It is thought to be extremely helpful since it is a highly effective method of utilizing exhaust gases to enhance the Main Engine's overall performance. The fundamental idea is that more power can be produced by increasing the air mass in proportion to the fuel quantity. The air temperature rises considerably

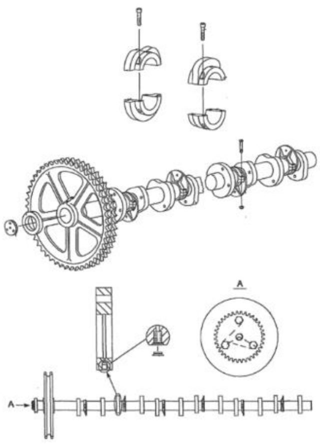

Figure 18: A Camshaft

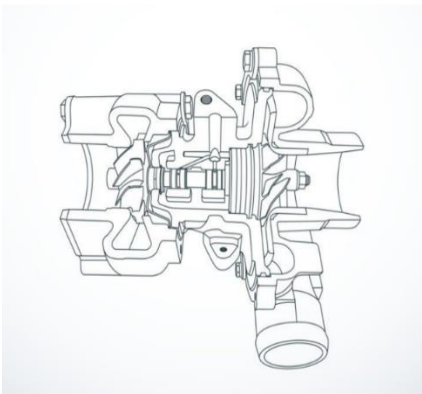

Figure 19: Turbocharger

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during the process, so air coolers are used to bring the temperature down to a desirable level so that the air can be directed to the scavenge air receiver.

A.2.2. The Propeller

One of the most important components of a ship's propulsion system is the propeller. In order to create the thrust required to move the vessel, it makes use of the power produced by the main engine, which is subsequently transformed into rotational motion by the crankshaft and transferred to the intermediate shaft before arriving at the propeller through the stern tube shaft. The thrust that is created gives the water momentum, which causes a force to be generated which acts on the ship and "pushes" it forward. The theoretical background on the Propeller's function, consists of the Bernoulli's Principle and Newton's Third Law: Between the aft and the forward side of the blades, a difference in pressure is created, thus accelerating the water from behind the blades.

A propeller typically consists of a shaft that multiple concentrically positioned helicoidal-shaped blades are arranged around. The Main Engine's Mechanical Energy is transformed into Kinetic Energy as the Propeller revolves. The water's increased momentum in the spaces between the propeller's blades causes forces to be applied to the blades while the propeller is submerged and operating. These forces can be divided into two categories: circumferential and axial forces. The total torque of the propeller, which must be exceeded by the Main Engine for the vessel to accelerate, is equal to the sum of the applied circumferential forces of each blade.

The speed and maneuverability requirements of a ship determine whether it will have one, two, or—in rare cases—three propellers. Typically, corrosion-resistant materials like stainless steel and aluminum alloys are used to build propellers. Alloys consisting of nickel, aluminum, and bronze are employed occasionally.

There are two ways to categorize propellers: by number of blades or by pitch. Below is an explanation of the pitch-based classification:

The two Types of Propellers based on their pitch are the Fixed Pitch Propeller and the Controllable Pitch Propeller.

A.2.2.1. Fixed Pitch Propeller (FPP)

Since the propeller's blades in this instance are fixed to the hub permanently, neither the pitch nor the blade positions can be changed while the propeller is in motion. Despite having a simpler structure and being much more durable and dependable than Controllable Pitch Propellers, these propellers are less maneuverable. More precisely, in the case of FPP, the Main Engine must engage the reversing mechanism in order for the vessel to move backwards.

Figure 20: Fixed Pitch Propeller

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A.2.2.2. Controllable Pitch Propeller (CPP)

With a CPP, the vessel can adjust its pitch by using mechanical and hydraulic arrangements to rotate the blades around their vertical axis. The overall efficiency of the propulsion system and the vessel's maneuverability are greatly enhanced in this condition. Compared to controlled pitch propellers, the change from "ahead" to "reverse" occurs more smoothly and without dead time, minimizing the amount of time and distance the vessel must travel to stop.

Figure 21: Controllable Pitch Propeller

Generally speaking, two factors are used to optimize a propeller's efficiency: its size and speed. Larger propellers with higher RPMs are known to be more susceptible to cavitation. The size of the vessel and the desired speed to be attained determine the propeller's overall size and number of blades. The following factors impact the propeller's performance when the power it absorbs is fixed:

- The Propeller's and Hull's condition in terms of fouling,
- The vessel's Draft and the water's shallowness,
- The speed and magnitude of wind,
- The weather conditions, and
- The state in which the Propeller's blades are (they ought to be kept in good condition and should sustain no damage).

A.2.3. The Shaft

As previously mentioned, one of the three parts of the shaft system is the crankshaft. Below, the intermediate shaft and the tail shaft are presented:

A.2.3.1. Intermediate Shaft:

As the "link between the propeller and the Main Engine," the Intermediate Shaft serves as a connecting element between the tail shaft and the crankshaft. The number of intermediate shafts is determined by the size and requirements of the vessel. However, when the system has more than two shafts, significant catenary forces act on the tail shaft, potentially causing serious damage, making it too difficult to analyze, operate, handle, and maintain. Practically, two intermediate shafts would be preferred in cases where the Main Engine is located far away from the Propeller. To connect the intermediate shaft to the crankshaft and tail shaft, built-in flanges are installed at the intermediate shaft's aft and forward ends. It is customary to have a slightly larger diameter in the specific locations where the intermediate shaft is supported by the intermediate bearings.

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Figure 22: Image of a typical Intermediate Shaft

A.2.3.2. Tail Shaft:

The majority of the tail shaft's part is housed inside the stern tube, and it is directly attached to the propeller. The aft end of the tail shaft is conical in shape and ends in a specific thread that matches the propeller's thread, while the forward end has an integrated flange to connect with the intermediate shaft. Next, while the vessel is moving forward, the propeller is adjusted on the spindle and fastened with a nut that has a tightening torque that is opposite to the propeller's rotational direction. Enough friction is created by this tightening torque to allow the torque to be transferred from the tail shaft to the propeller.

A.2.4. Bearings

The parts of the shaft system of a vessel are required to be supported on bearings. The several types of bearings, have as follows:

A.2.4.1. Thrust Bearing:

Transferring thrust to the hull (astern or forward) and preserving the shaft's axial movement within acceptable bounds are the two primary objectives of the Thrust Bearing's installation. To effectively carry out its function and endure both normal and shock loads, the thrust bearing needs to be built and installed upon a sturdy seating. A suitable foundation is provided by the double bottom structure of the hull section where the thrust bearing is located. Nevertheless, in order for the thrust bearing to function well in operating conditions that often result in a forward tilt, its structure needs to be exceptionally sturdy. Shaft system axial vibrations are another feature that needs to be taken into consideration. These vibrations can get greater due to a slackening of the propeller blade load or the splay of diesel engine crankshaft webs, which are the parts of the crank that are between a crankpin and the shaft or between neighboring crankpins. This usually results in a) a lift of the aft journal of the block, and b) misalignment angle of the shaft. The upcoming chapters will cover these methods in great detail. A thrust bearing may be a standalone unit or an integral part of the main engine.

A unique arrangement is present in the section of the shaft that adapts to the thrust bearing. It is made up of rings coated in special metals, which are attached to the shaft and oriented vertically to the axes of symmetry. The thrust bearing is equipped with ring-shaped grooves and within them, the shaft moves. There is a small gap between the thrust bearing and the shaft where the lubricant moves. Water circulating in the thrust block's shell serves as a means of treating the excess heat generated during the operation of the propulsion mechanism.

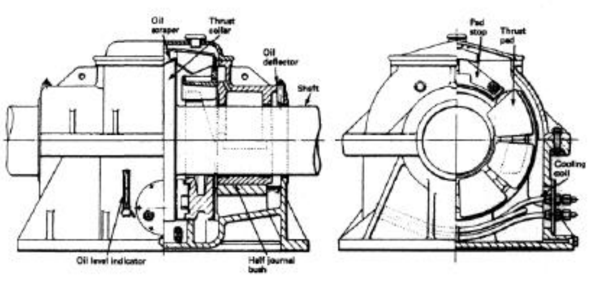

Figure 23: Typical Trust Bearing

A.2.4.2. Stern Tube Bearings:

Stern tube bearings, typically serve two purposes: The first is to support the weight of the propeller shaft and a significant amount of the tail shaft's weight. Sealing the vessel from seawater and preventing lubricating oil leaks into the marine environment serves the second purpose. White metals are used in their construction, along with special lubrication systems. Dependent on the specific design of each vessel, there may be one or two Stern Tube Bearings. However, a design with just one Stern Tube Bearing should be thoroughly investigated and analyzed, in order for the engineer to ensure the safe and uninterrupted operation of the propulsion system. Further analysis of the pre-mentioned will take place in the next chapters.

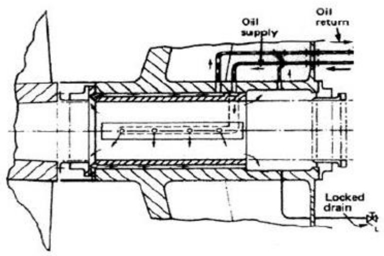

Figure 24: Stern Tube bearing

A.2.4.3. Intermediate – Line Bearings:

Intermediate Bearings are radial bearings, which aim to support the shaft's and all its components' weights. They must be able to withstand a variety of varying loads. It is standard procedure for intermediate bearings in 2-stroke diesel-powered vessels to be made up of thick shell and white metal. There are two parts to a line bearing: the upper part and the lower part. The Lower Part is set on the double-bottom of the vessel, and the Upper Part is fastened with screws or nuts. If the surface inside the line bearing is not properly lathed and aligned, the shaft will be in danger of failing, due to severe bending.

A.3. Sealing Mechanisms of Aft Stern Tube:

The major applications of an aft stern tube seal are:

• Preventing lubricating oil leakage from the aft stern tube into the marine environment, hence preventing water contamination of the lubrication system; and

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• Preventing seawater from entering the vessel's area.

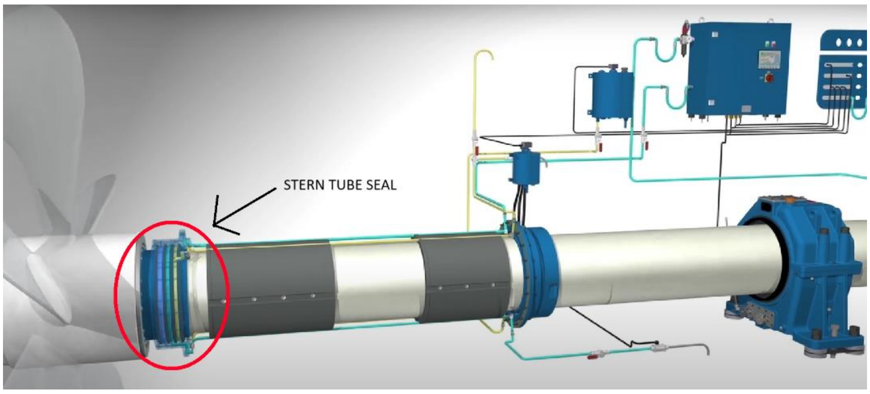

Figure 25: A Typical Stern Tube Seal Arrangement

Oil-lubricated stern tubes necessitate two types of seals: one for the exterior arrangement (aft seal) and one for the inboard end (forward seal). Modern stern tube seals are highly developed devices that are intended and produced to survive a harsh operating environment for extended periods of time.

It is common for vessels to have the lubrication oil tanks positioned around 2 to 3 meters above the water line, so that the pressure differential ensures no water contamination in the lubrication oil tanks.

The following are the primary sealing arrangements utilized to prevent the aforementioned undesired conditions:

- “Stuffing Box” arrangement,
- “Lip Seals” arrangement, and
- “Radial Face Seals” arrangement.

Below, these three types of Sealing Arrangements will be explained:

Stuffing Box: It is one of the oldest forms of sealing arrangements, being available for more than half a century, and was extensively utilized on earlier ships that used timber bearings with sea water lubricating systems. This technique is based on compressed packing material that is crammed around the propeller, making leaks (in the form of water entering the vessel or lubricating oil leaking into sea water) impossible. The compressed material is frequently treated with wax and tallow to prevent abrasion of the shaft's surface. In modern containers, the aforementioned is accomplished using materials like Teflon and/or graphite. A high-strength hose connects the entire setup to the shaft tube. It is critical that the temperature in the StuffingBox arrangement maintains within acceptable ranges (about 140°) to avoid melting of the wax, which can cause serious issues regarding the shafting mechanism.

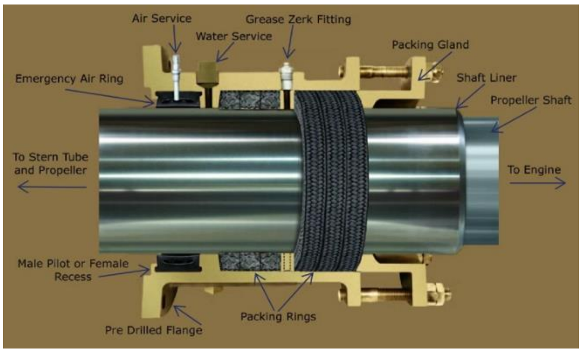

Figure 26: Typical Stuffing Box Arrangement

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Lip Seals: This kind of sealing arrangement comprises of a ring-shaped portion made of elastic materials (elastomers such as nitrile rubber, Carboxylated Nitrile, Fluorinated Ethylene-Propylene) to provide enough grasping and successful sealing. The material chosen is also related to the type of lubricant that will be used, the operating temperatures and environment, as well as the speed and main characteristics of the rotating component of the machinery. It is crucial to note at this point that the temperature in these arrangements must be carefully monitored to be within permitted limits (which vary depending on the individual material of construction), or the rubber may lose its flexibility. The entire sealing arrangement is made up of several rings made of elastic material that are inserted between metal rings. Each lip is in contact with a sustainable sleeve attached to the propeller shaft and a garter spring. To keep the sealing arrangement in good working order, it is critical to check the temperature and quality of the oil on a regular basis. This is to minimize contamination of the lubrication oil (due to dirt or water), which could lead to shaft or elastomer deterioration. Furthermore, as the oil heats up, the elastomer becomes more prone to wear or damage. Finally, paint coatings or dirt buildup might cause excessive heating of the sealing arrangement, thus the installation's condition should be checked on a regular basis.

This particular sealing arrangement is frequently paired with the so-called "Labyrinth Seal." The primary idea behind this arrangement is to build a complex trail of obstacles that try to gradually diminish the pressure applied to the final barrier. The long journey of the barriers eventually leads to a more traditional kind of sealing, such as the Lip Seal.

It should be noted that rotating lip seals are not suitable for separating two liquids. As a result, even if a succession of lip seals is used, some of the lubricant could eventually end up in the water.

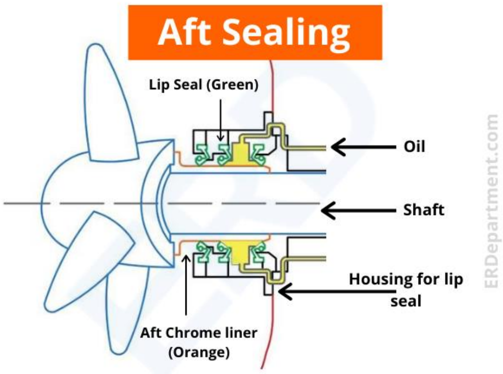

Figure 27: Lip Seal Arrangement

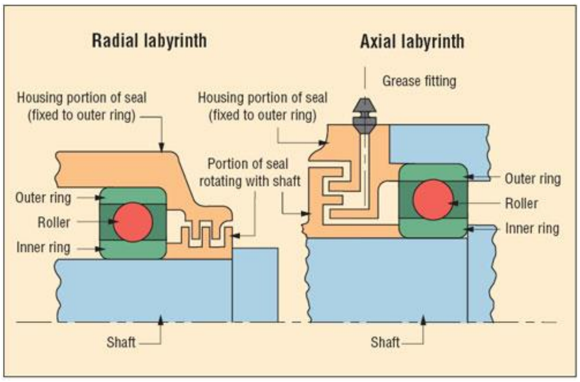

Figure 28: Cross Section of Radial and Axial Labyrinth Sealing Arrangements

Face Seals: A radial face seal arrangement comprises of a flat wear-resistant surface in contact with the spinning section of the equipment to ensure effective leakage protection. Through the application of special springs, the Faces maintain contact with the shaft. The split construction of all its components is a significant advantage of this form of sealing arrangement. This feature makes installation of the arrangement simple, and it also makes inspection and maintenance of the arrangement more uncomplicated. The installation of the arrangement achieves anti-leakage by making perfect mating contact between the opposite faces of the seal's seat, which rotates along with the shaft. The seal faces (which are hydraulically balanced) establish the appropriate contact by utilizing the pressure of the springs and by flexibly mounting the face on the main seal unit (which is immovable). This flexible element is built of robust, but pliable, reinforced bellows to withstand the hull deflections and vibrations.

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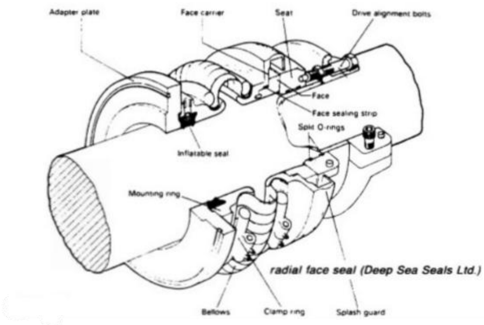

Figure 29: Typical Face Seal Arrangement

B. Shaft Alignment

B.1. Why is it so important?

At this point, it may be of interest to emphasize on some differences on the terminology used:

According to ABS (2019), there is a distinction between the terms Propulsion Shafting and Propulsion Shafting Alignment, as well as between the alignment process and the alignment procedure.

- Propulsion Shafting: "A system of revolving rods that transmit torque and motion from the prime mover to the propeller. The shafting is supported by bearings, whose number and position is determined based on allowable bearing loads and lateral vibration (whirling) requirements. Static shafting alignment analysis criteria defines the acceptable load distribution and contact condition between shafts and bearings."
- Alignment Process: "The alignment process encompasses all shafting alignment activity starting with analyses and reviews, to installation procedures, condition verification and trials."
- Alignment Procedure: Refers only to shafting alignment production work activities on site.
- Propulsion Shafting Alignment: "A static condition observed at the bearings supporting the propulsion shafts." To suitably define propulsion shafting alignment, a minimum set of parameters are defined in the design stage and subsequently confirmed acceptable on board:

The whole procedure, consists of three stages:

1. The Shaft Align Design and Analysis,
2. The Implementation of the alignment procedure and
3. Verification via measurements and constant monitoring of the parameters of interest.

According to ABS (2019), the process of Propulsion Shafting Alignment should at minimum define during the Design Stage, and consequently verify as acceptable on board:

- The vertical offsets of the bearings,
- The reactions of the bearings,
- The bearing to shaft condition (i.e. misalignment angle)
- The crankshaft's web deflections
- The Gear Mesh misalignment.

According to the Classification Societies, the criteria for the application of the guidelines, are obligatory for:

ABS (2019): Provides Guidance which mainly focuses on alignment-sensitive propulsion systems, such as VLCC and ULCC large bulk carriers and large container vessels (more than 9000 TEU). The following designs are specifically addressed:

- Direct drive propeller installations

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- ➢ Low speed diesel installations
- ➢ Systems with relatively short and rigid shafting
- ➢ Installations with no forward stern tube bearing
- ➢ Vessels with a relatively flexible hull structure and
- ➢ Twin screw propulsion.

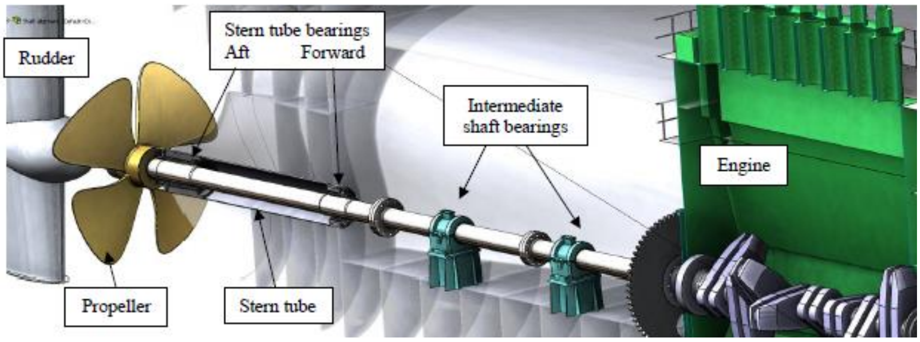

Figure 30: Directly coupled propulsion system, ABS [12]

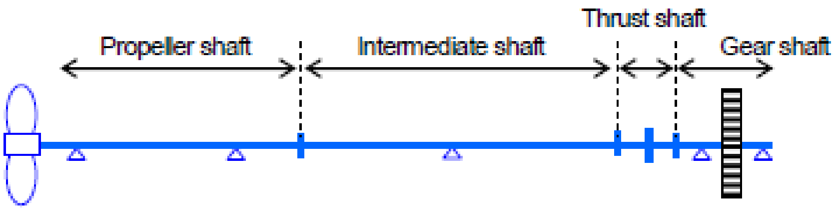

Figure 31: A typical geared propulsion system, ClassNK [24]

DNV (2021): According to DNV, the following systems require calculation reports, whilst for any plant that does not belong in the stated categories, a shaft alignment specification is enough.

- ➢ Plants with minimum diameters (slow speed side) of 400mm or greater for single screw and 300 mm for twin screw,
- ➢ Plants with gear transmissions with more than one pinion driving the output gear wheel, even if there is only one single input shaft as for dual split paths,
- ➢ Plants with shaft generator or electrical motor as an integral part of the low-speed shaft in diesel engine propulsion and
- ➢ Plants with single stern tube bearing arrangement.

Unconventional hull forms (asymmetrical aft ship, and/or novel propulsion arrangements) or plants which are considered to be sensitive to alignment, also require alignment calculations. If requested, supporting analysis of propeller loads in normal and/or transient operation will be needed.

For the Plants which only require Alignment Specification (applications which do not fall under any of the previously mentioned cases), the following are to be submitted:

- Bearing offsets from the defined reference line,
- Bearing slope relative to the defined reference line if different from zero and
- Installation procedure and verification data with tolerances e.g. gap and sag and jacking loads (including jack correction factors and jack positions) and verification conditions (cold or hot, propeller submersion, etc.).

Lloyds Register (LR) (2023): According to Lloyd's Register, the calculations for the Shaft Alignment are to be carried out for main propulsion shafting at propeller speed, including the crankshaft of direct drive systems or

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the final reduction gear wheel when on geared installations. The following Shafting Systems should submit Shaft Alignment calculations for approval:

- All geared installations, with propeller shaft of diameter 300 mm or greater in way of the aftmost bearing,
- All geared installations with multiple input/single output, regardless of shaft diameter,
- All direct drive installations which incorporate three or fewer bearings support in the intermediate and propeller shaft aft of the prime mover,
- Shafting Systems where prime movers in a direct drive installation of shaftline bearings are installed on resilient mountings,
- Shafting Systems where all systems where the screws haft have a diameter of 800 mm or greater in the way of aftmost bearing.

Bureau Veritas (2015): Elastic Shaft Alignment Class Notation was published from BV in 2015. It may be applicable to new ships falling into one of the following categories:

- Ships of all types having propulsion power per shaft line greater than or equal to 30 MW,
- Container Ships having propulsion power per shaft line greater than or equal to 15 MW,
- Liquefied gas carriers having propulsion power per shaft line greater than or equal to 10 MW.

ClassNK (2006) states that, the shaft alignment guidance is applicable to:

- Vessels with propulsion type of 2-Stroke Diesel Engine,
- Vessels with propulsion type of 4_Stroke Diesel Engine,
- Vessels with propulsion type of Steam Turbine (in this case, modifications in the required calculations and methods used are made).

The desirable calculations, which will be defined later in this paper, shall be conducted for the following conditions: Light Draught Condition (Cold Condition), Light Draught Condition (Hot Condition), and Full Draught Condition (Hot Condition). However, according to ClassNK, calculations concerning the Full draught condition (hot Main Engine) shall only be carried out in cases of Oil Tankers, vessels carrying dangerous chemicals in bulk form, Bulk Carriers and General dry cargo ships.

The significance of proper shaft alignment is being emphasized these days. As hull structures get more complex and flexible, the shaft becomes shorter and stiffer for optimization purposes in order to lower a vessel's deadweight and optimize cargo capacity. This combination must be thoroughly analyzed to ensure a vessel's proper operation, as if it is not adequately addressed, it may have a negative impact on propulsion shafting alignment and may also make the shaft system extremely sensitive to small changes in bearing offsets, causing severe damage to the vessel's machinery. Failure to properly construct a shaft alignment plan may result in:

- Increased friction at the shafting system, and as a result loss of power,
- Supporting bearings being unloaded, thus not contributing to the supporting of the loads,
- Increased bending moments on the shaft, which in case they exceed the allowable limits, could lead to failure due to fatigue,
- Overloading of bearings which leads to severe wear and damages both for the bearings and the shaft, which shortens the system’s life, increases the costs of maintenance and operation and puts the vessel’s seaworthiness and safety under consideration and
- If a reduction gear is installed, bad coupling of gears could appear, which could cause serious ware and damage and could possibly lead to gearbox failure.

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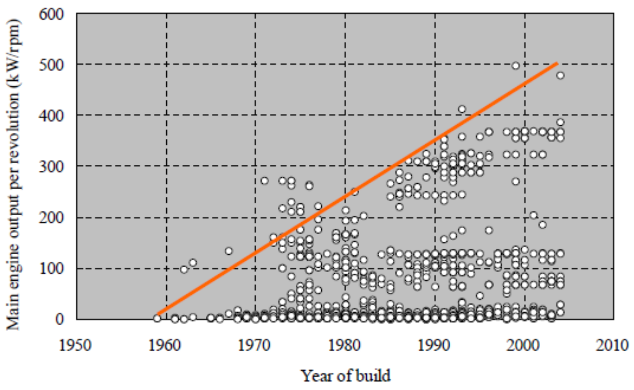

Fig. 1.1 Evolution of main engine output per revolution for tankers.

Figure 32: Evolution of main engine output per revolution for tankers, ClassNK [24]

On the other hand, if a vessel is equipped with a successful, properly designed propulsion shaft system, its shaft stresses will be significantly reduced, and its bearings will be evenly loaded and function within the nominal loading capacity limits, extending its life and lowering operation and maintenance/service costs.

The previously mentioned sensitivity causes issues in precisely computing the alignment and ensuring that the alignment is carried out as planned. As a result, the shaft alignment analysis should offer all of the necessary data to support each stage of shaft alignment implementation. Furthermore, the calculations and on-board verifications of the alignment process (which are performed in static state) must result in the vessel operating uninterrupted during its voyages. According to ABS (2022), the following are the Shaft Alignment process challenges:

- High sensitivity of the shaft alignment to small disturbances in the bearing vertical position.
- Disparity between highly flexible hull girder structure and the rigid propulsion shafting
- Appropriately evaluating hull girder deflections
- Difficulties in obtaining critical service propeller loads during maneuvering
- Problems in maintaining the desired accuracy of the shaft alignment analysis
- Inconsistencies and inaccuracies in the shaft alignment production process

However, it is worth noting that all of the shaft alignment calculations are based on major simplifications and assumptions. As a result, certain aspects of the shaft alignment analysis require special attention. The following are key variables to consider on the shaft alignment issue:

B.1.1. Hull Deflections

Shaft alignment is negatively impacted by hull deformations. Nevertheless, until recently, they were not given much thought, and all propulsion shaft alignment calculations were done solely for the circumstances under which the alignment design would be put into practice. According to Lech Murawski in "Shaft line alignment analysis taking ship construction flexibility and deformations into consideration" (2005), when performing shaft alignment calculations, a basic beam model that is isolated from the ship's hull is typically used, which carries several risks.

The ship actually operates in a variety of loading conditions and sea-wave situations, which equates to a variety of hull behavior scenarios with respect to deformations. The effect of hull deformations on Shaft Alignment could be more severe for very long ships, with relatively flexible hulls and stiff shafts (Korbetis et al, 2014). According to D. Sverko (2003), the design procedures used in contemporary vessels (which were covered in earlier chapters) lead to a wider difference in the structural flexibility of the shafting and the hull. The

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propulsion shaft alignment consequently becomes more susceptible to hull deflections. The period when the design of modern vessels began to change was marked by a significant increase in the frequency of bearing damage related to shaft alignment. This damage may arise from shipbuilding procedures, a lack of regulations, the design of the vessel, or insufficient analysis.

It is important to note at this point that each shipyard's methods will determine how the shaft alignment procedure is implemented. For instance, there are a number of shipyards that would prefer to depend on their ability to resolve issues as they arise rather than properly design and analyze the shaft alignment issue.

The problem lies in the fact that the analyzing methods and calculations do not represent the exact state and behavior of the propulsion systems, and "may not always provide sufficient information to ensure "error free" alignment procedure". More precisely, the shaft alignment process is a static procedure, where all of the forces, moments and loads applied are static. However, the ship operates under severe dynamic and transient conditions, which are not traditionally taken into account during the design and calculation stage. In the paper "Effects of Hull Deformations on the Static Shaft Alignment Characteristics of VLCCs: A Case Study" (G. Korbetis, O. Vlachos, A.G. Charitopoulos, C. I. Papadopoulos, 2014), it is implied that: "The successful application of a static shaft alignment plan is essential for trouble-free dynamic operation of the propulsion system, aiding in decreasing bearing wear, increasing bearing expected lifetime and decreasing maintenance and replacement costs."

The "mismatch" between the analysis and implementation phases is another barrier to the shaft alignment process about the hull deflections becoming entangled in the calculations. More precisely, because the ship is on an even keel, it is relatively simple to establish a reference line when it is in dry dock, which leads to highly accurate sighting-through and bearing pre-positioning. When the vessel is operating and afloat, however, the procedure of checking the shaft alignment can become extremely difficult as all control over the measurement precision may be lost. In conclusion, it would be ideal if the responsible engineer or engineers could exactly forecast or calculate the hull deflections and the bearings' offset changes of the ship in its afloat condition in order for the alignment procedure to be confirmed with adequate accuracy (Sverko, 2003). Sverko (2003) suggests that carrying out the shaft alignment process fully on an even keel (Dry Dock) should be able to address the discrepancy between the analysis and execution phases. This would guarantee the most accurate validation of responses, etc. However, for the above to be possible, the engineers must be capable of:

- Estimating hull deflections, as the alignment needs to be satisfactory in vessel afloat condition,
- Defining the optimal set of prescribed bearing displacements to ensure a robust alignment which is relatively unaffected by hull deflections when vessel is afloat,
- Conducting the whole alignment procedure in the dry dock.

The most widely used industry practice is the use of detailed Finite Element Analysis (FEA) to precisely determine the deformations of the vessel's structure during various operation and weather conditions, with the goal of eliminating, or at least minimizing, the negative effect of hull deflections on the shaft alignment procedure. In fact, according to the most current published regulations of several Classification Societies, it is necessary to undertake a full and in-depth assessment of the shaft alignment process. However, in some cases, measurement of the Hull Deflections on board using special measurement apparatus could be acceptable. This subject falls within the context of the chapters which will be discussed and explained later in this thesis.

However, each Classification Society has its own methodology for assessing and/or precisely calculating the Hull Deflections of a vessel. These methodologies will be thoroughly discussed later. Nevertheless, it is worth mentioning, that ABS (2019) proposes a methodology for estimating the Hull Deflections' impact on the shafting alignment for numerous ship categories (tankers, bulk carriers and container vessels) by creating a software which, when utilized by the engineer responsible, can "estimate the Hull Deflections of a ship with

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relatively high confidence”. More precisely, ABS (2019) suggests that if the Hull Deflection measurements of a vessel are not available, the Influence Coefficients can be used, in order to gather information regarding the susceptibility of the shafting system to changes which occur due to Hull Deflections. More on this subject will be presented in the “Influence Coefficients” chapter of this study.

Furthermore, ClassNK (2006) mentions the importance of the difference between the Shafting Installation Condition and the Typical Service Condition, which is directly correlated to the hull deflections. More specifically, the installation of the shaft is usually conducted under the so called “Launched Condition”, which is characterized by light draft and a cold main engine. However, as will be thoroughly explained later, during operation conditions, the vessel’s draft will significantly increase, and alterations in the temperature in the areas close to the Main Engine will occur. The aforementioned factors will subsequently lead to deformations in the hull and the Main Engine’s structure, which will directly affect the bearings’ offsets.

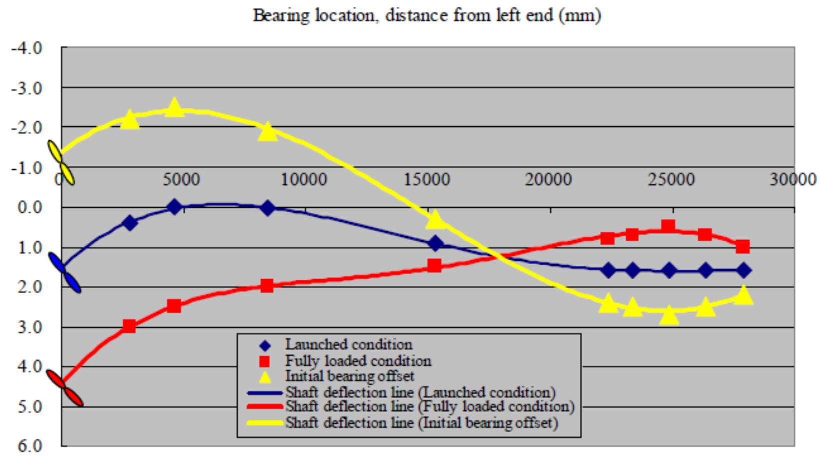

Figure 33: Bearing offsets and shaft deflection lined under different conditions of the vessel, ClassNK [24]

Consequently, ClassNK suggests the following as a preventive measure: The initial offsets of the bearings should be compensated taking into consideration the differences represented in the diagram above, in cases where the initial offsets calculated are not satisfactory. This statement, can also be expressed as follows:

Initial bearing offset= Launched condition - (Fully Loaded Condition-Launched Condition)

In this equation, the term which represents the correction factor is the term: (Fully Loaded Condition-Launched Condition). However, “If the shafting alignment is predicted to be unsatisfactory under other conceivable conditions such as light ballast, then the compensation may need to be reduced to an extent by which all operating conditions can be accommodated” (ClassNK, 2006)

B.1.2. Stiffness of the bearing Foundation:

Another significant factor in the shaft alignment analysis procedure is the rigidity of the bearings' base. Traditionally, the shaft is represented as a series of simple beams that are fully isolated from the rest of the vessel's structure and are simply supported on the locations of the bearings. In actuality, the shaft's supporting points are stiff but not totally rigid. Murawski (2005) provided a new parameter to be addressed while using a thorough FEM model of a big containership: the stiffness characteristics of the bearing foundations. He found that bearing stiffness and oil film properties should be included in the design stage of a comprehensive approach to the shaft alignment problem. (Korbetis et al, 2014). The foundation of the bearings is supposed to deflect elastically under the weight of the bearings and the rest of the system's components.

During the Finite Element Analysis, the stiffness characteristics of the vessel’s hull analyzed together with the main engine body, are of great significance for the propulsion system’s boundary conditions (Murawski, 2005).

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Murawski suggests a new parameter, which presents some similarities with the influence coefficient. The coupled stiffness $k_{ij} = \frac{R_j}{\Delta_{ij}}$, where R_j: is the reaction of the bearing j, and Δ_{ij}: is the bearing's i dislocation under reaction's acting on bearing j. Coupled stiffness represents the ratio of the force acting on bearing j to the i bearing's displacement caused by this force (Murawski, 2005). The coupled stiffnesses of all the bearings can then be utilized in order to calculate the substitute stiffnesses which are then applied to the Finite Element Analysis model as independent boundary conditions. The substitute stiffness of a bearing, can be calculated by the formula:

$$ks_i = \frac{R_i}{\Delta_{s_i}} = \frac{1}{\sum_{j=1}^{n} \frac{1}{R_i/R_j} k_{ij}},$$ where:

- ks_i: the substitute stiffness of bearing i,
- Δ_{s_i}: the summary of the bearing's i deflection, and
- n: the number of bearings

The shaft alignment and lateral vibrations calculations shall be made before calculating the substitute stiffnesses, because the static & dynamic load distributions of the bearings' reactions must be known. By utilizing the substitute stiffnesses, the intermediate shaft and the crankshaft deflections will represent a full hull model analysis. Regarding dynamic analysis, it is worth mentioning that the harmonic frequencies of the propeller must be known as well.

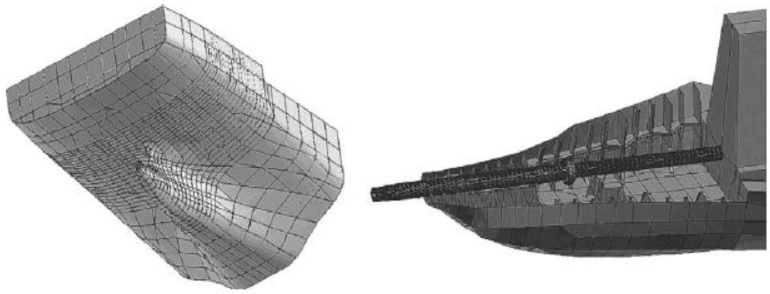

Figure 34: Model of a container 2000 TEU stern part with a power plant compartment, Murawski [9]

Murawski (2005), in the same study, draws attention to the intermediate bearing frame's stiffness. According to this analysis, the intermediate shaft's bearings frames are characterized by increased weight and rigidness. Murawski states that the dynamic calculations in this case could be overlooked, and the distribution of the loading forces shall be determined based on the oil film's pressure distribution. In conclusion, Murawski suggests that the stiffness characteristics of the intermediate bearing's foundation stiffness should be considered and calculated.

At the same paper, Murawski subsumes the importance of different boundary conditions on the shaft alignment analysis. In this example, the power transmission system was modeled with linear beam elements with non-linear boundary conditions (e.g. lube oil film stiffness). When the shaft analysis process is completed and it is verified that all the reaction forces on the bearings are acceptable, Murawski continues with the study concerning the influence of different boundary conditions. More precisely, five different variants of calculations have been conducted, and the difference in the calculated reaction forces are presented. According to the author's opinion, the aft stern tube bearing should be modeled as an elastic continuous support.

The cases studied have as follows:

1. A classical model where all the bearings are modeled as pointwise, and the support is assumed to be ideally stiff.
2. A model similar to the classical, where the aft stern tube bearing is continuous but continues to be assumed as ideally stiff.
3. All bearings are modeled as pointwise supports with elasticity.
4. The stern tube bearing is considered as an elastic continuous support with the rest of the assumptions same as no.3 case.
5. The hull structure is modeled with substitute stiffnesses, and the rest of the assumptions are same as in case no.4.

The bearing's reactions as calculated in all the different assumptions, have as follows:

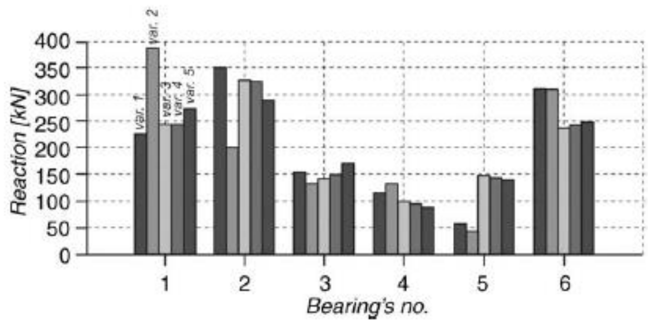

Figure 35: Bearings' reactions in different analysis assumptions, Murawski [9]

ABS (2022) in Enhanced Shaft Alignment Guide, requires that the stiffness of the bearing foundation is taken under consideration in the calculations regarding whirling phenomena.

B.1.3. Thermal Deformations

Temperature changes while the vessel is in operation have a substantial impact on the shaft alignment design. When the main engine is running, the engine parts near the combustion chamber normally have greater temperatures (about 90 °C), but the engine parts near the bedplate, or engine foundation, frequently experience temperatures of around 40 °C. Temperature changes cause the engine's components to expand in three dimensions. However, because the vertical component has a direct influence on shaft alignment, engineers should focus on it during the shaft alignment operation. The problem in this situation, is that, while hull deflections may affect all the propulsion system and its bearings at the same time, and the deflections can be represented by a smooth and continuous curve, thermal deviations cause deflections which are local and therefore cannot be studied and analyzed with the same methodology as Hull Deflections are. ABS (2019) proposes that the Influence Coefficient Matrix is used in order to estimate the change in a bearings' force reactions due to thermal expansions.

In the guidelines on shaft alignment which ClassNK published in 2006, it is stated that vertical displacements caused by thermal aberrations during Main Engine operation can be separated into a parallel rise and a hogging deformation.

- The rise of the temperature on the engine's bedplate causes a parallel rise in the height of all of the bearing positions.
- The relatively larger expansion in the longitudinal direction of the upper parts of the engine structure (cylinder block), due to the difference in the temperature between the upper parts compared to the lower ones, cause a hogging deformation.

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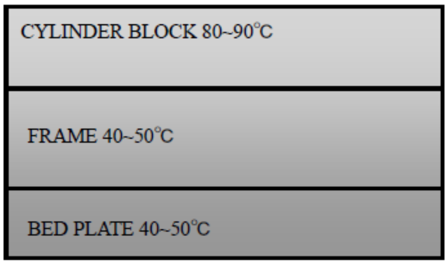

Figure 36: Typical Temperatures for compartments near the Main Engine, ClassNK [24]

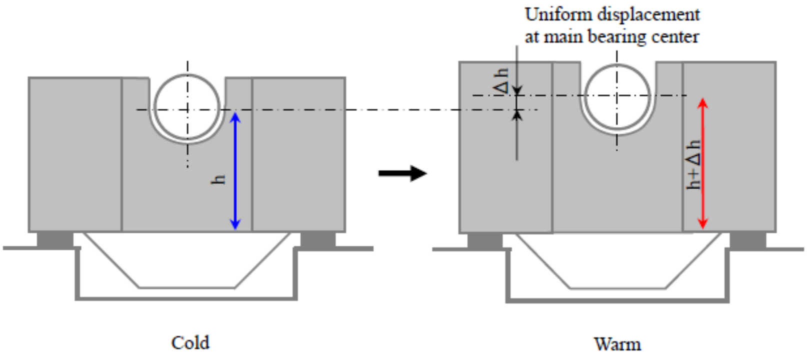

Figure 37: Parallel Rise of bearing offsets due to thermal deviations, ClassNK [24]

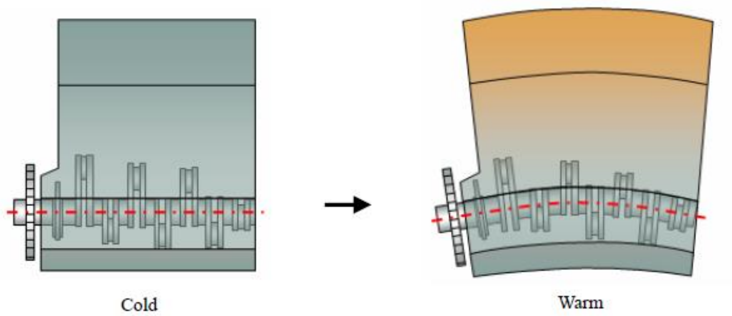

Figure 38: Hogging Deformation due to Thermal Deviations, ClassNK [24]

Sometimes, it is wrongfully assumed that all of the engine bearings are affected equally by thermal deviations. This assumption differs a lot from reality, as the midsection of a typical Diesel Engine expands more “due to less heat dissipation than the forward and the aft engine structure” (ABS 2019). A common industry method , is the bedplate’s pre-sagging, in order to make the effect of thermal deviations on the cranksaft bearings more mediocre.

For the shaft alignment plan to be successful under all vessel operating conditions, the bearings' vertical offsets resulting from thermal deviations must be taken into account in the calculations.

B.1.3.1. Parallel rise of bearing offsets

The parallel rise of the bearing offsets can be calculated if the distribution of the temperature in the bedplate is known. The parallel rise in height of each bearing: Δh can be calculated using the formula:

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$$\Delta h = a * h * (\frac{\Delta T1 + \Delta T2}{2})$$

Where:

- a is the thermal expansion coefficient of the material which is used for the construction of the bedplate,
- h is the height from the engine seating plane to the bearing's center and
- ΔT1, ΔT2 represent the increases in temperature at the engine seating plane and bearing center, respectively.

B.1.3.2. Hogging deformation

The main cause of the hogging deformation is the unequal distribution of temperature within the main engine's structure. According to ClassNK, the most accurate method for precisely describing the hogging deformation caused by heat alterations to the main engine is to quantify the relative deflections that are seen while the engine is running from cold to hot under specific draught (loading) circumstances while on board.

B.1.4. Sufficient lubrication of the Journal Bearings

B.1.4.1. A Brief Introduction to Basic Lubrication Theory:

It is of vital significance for the uninterrupted operation of the propulsion shaft, to always ensure the proper operation of the journal bearings, which is based on the hydrodynamic lubrication theory. In the regulations pertaining to the Shaft Alignment, several Classification Societies give particular emphasis to this matter, and that will be covered in detail in the following chapters. The idea underlying the previously discussed focus on journal bearings is that a thick film of oil must form in order for the journal bearing to function properly. The precise thickness of the oil layer is determined by several factors, including the oil's temperature, viscosity, and shaft's rotational speed. Nonetheless, the operational conditions that predominate during the vessel's voyage also have an impact on these parameters. During the design stage of shaft alignment, the following factors are important in determining the exact windup of the oil film's characteristics and should be carefully considered:

- the shaft loads due to the propeller's operation,
- the propeller's immersion state,
- the overall condition of the sealing arrangement,
- the lubricant's distribution systems,
- the quality of the lubrication oil etc.

The Sommerfeld number is a fine quality indicator when it comes to securing the existence of evenly loaded bearings, even if they have different dimensions, as it can be utilized in order to characterize a bearing's loading condition. Both underloaded and overloaded bearings are generally undesirable because they can have a negative impact on their strength behavior and have been shown to be dangerous, particularly for intermediate bearings (Murawski, 2005), which typically have 30–50% higher Sommerfeld numbers than the other bearings (because the deformations of the hull may be the reason for decreasing relatively small reactions). Sommerfeld's number can be calculated from the following formula: $S = \frac{R}{\eta * U} * (\frac{d}{c})^2$, where:

- S: The Sommerfeld number
- R: The bearing's loading unitary force
- η: The lubricating oil absolute viscosity
- U: The peripheral speed

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- d: The diameter of shaft journal and
- c: The bearing slackness

At this point, extra attention shall be focused on the aft stern tube bearing, because of the fact that it is the heaviest loaded bearing and it is also relatively longer than the rest of the bearings. Moreover, due to the propeller's dynamic loadings, it is asymmetrically loaded.

For clarification measures, below the basic lubrication theory concerning a journal bearing is explained:

A journal bearing consists of a rotating circular shaft housed in a circular bush whose inner diameter is one to two parts per thousand larger than the shaft. Maintaining minimal metal-to-metal contact, preventing overheating, and distributing loads uniformly by exploiting the oil layer that forms are the main goals of proper lubrication, which also relieves the bearing. By doing all of the aforementioned, the bearing's wear is reduced, and its lifespan and cost are optimized. Since sliding is the primary movement in a journal bearing, hydrodynamic lubrication is the main theory used to explain the phenomenon.

The presence and amount of continuous lubricant oil flow—which results from the energy transfer from the shaft's moving surface to the lubricant and creates the necessary pressure—as well as the location and size of the minimum oil thickness are the most important factors influencing the viability of the hydrodynamic lubrication theory. (Cameron, 1976; Fundamental Lubrication Theory)

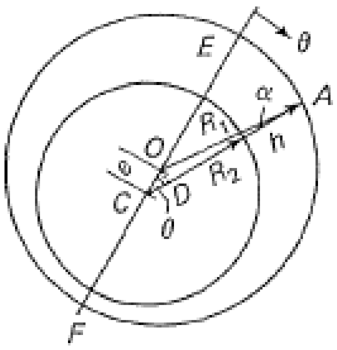

Figure 39: Journal Bearing-Characteristics

For the hydrodynamic lubrication theory to be applicable, the following conditions must be met:

- Relative movement between the two surfaces with a value high enough to create a lubrication film that carries bare loads, and
- Inclination between the sliding surfaces to create the hydrodynamic wedge (if the surfaces are parallel, then the creation of the pressure profile is not possible).

The oil film thickness, is expressed by the following formula (Reynolds equation):

$$\boldsymbol{h = (R_1 - R_2) + e * cos\theta = c + e * cos\theta = c * (1 + \varepsilon * cos\theta),}$$

where: (ε) eccentricity ratio, defined as e/c. When the shaft and bearing are concentric, ε=0, while when they touch ε=1

Based on the above, the minimum oil thickness is described by the equation:

$$\boldsymbol{h_{min} = c * (1 + \varepsilon * cos\pi) = c * (1 - \varepsilon),}$$

as the minimum oil film thickness is encountered at ϑ=π, when cosϑ=-1.

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At this point, it is worth underlining that in general, the value of the minimum oil thickness is affected by the roughness of the surface, the specific characteristics of the lubricant and the bearing's dimensions.

The pressure equation in journal bearings is calculated using the short bearing approximation. Short bearing approximation is assumed when $L/_D < 1/_3$. According to this approach, the pressure distribution equals to:

$$\boldsymbol{p = 3 * U * \eta * \frac{dh/_{dx}}{h^3} * (y^2 - \frac{L^2}{4})},$$ *(Cameron, Basic Lubrication Theory, 1976) where:*

- p: pressure
- U: surface speed of the shaft
- dh/dx: gradient of film oil thickness in the direction of motion
- η: viscosity
- h: film thickness
- L: (axial) bearing length
- y: co-ordinate in axial direction

In the event of a journal bearing, however, the following are applied:

- $x = R * \theta$,
- $h = c * (1 + \varepsilon * cos\theta)$
- $dx = R * d\theta$, so $dh/_{dx} = \frac{-c*\varepsilon*sin\theta*d\theta}{R*d\theta} = -(c * \varepsilon * sin\theta)/R$

Thus, $$\boldsymbol{p = \frac{3*U*\eta*c*\varepsilon*sin\theta}{R*c^3*(1+\varepsilon*cos\theta)^3} * (\frac{L^2}{4} - y^2)}$$

During the vessel's operation, the journal bearings create a hydrodynamic film to carry the load of the shaft. When in transient conditions before a change in the load is made, the bearing reacts in order to carry the "new" load by changing its eccentricity and film thickness.

In hydrodynamic lubrication theory, there are three distinguished operating conditions:

1. Full Film Hydrodynamic Operation: During this condition, there is no metal-to-metal contact. Consequently, no friction or wear is observed. Moreover, the oil film thickness is adequate, and as a matter of fact, much greater than the surface's roughness. This condition describes a "safe operation mode", when everything is functioning as feasibly preferable as it gets. In this case, the friction coefficient is equal to approximately 0.001 to 0.03.
2. Mixed Film Lubrication: During this condition, the oil film's thickness is not adequate in order to entirely prevent metal to metal contact. In this case, the lubricant oil still supports part of the shaft's load. The "Mixed Film Lubrication" refers to the transitional state between the "Full Film Hydrodynamic Operation" and the "Boundary Lubrication". If there is extensive operation under these conditions, the wear is inevitable.
3. Boundary Lubrication: During this condition, there is full contact between the two surfaces (asperity contact). Moreover, the oil film's thickness is less than the surfaces' roughness and the wear is severely damaging.

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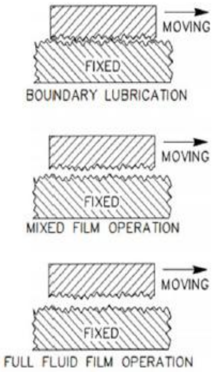

Figure 40: The three different conditions of a lubrication system

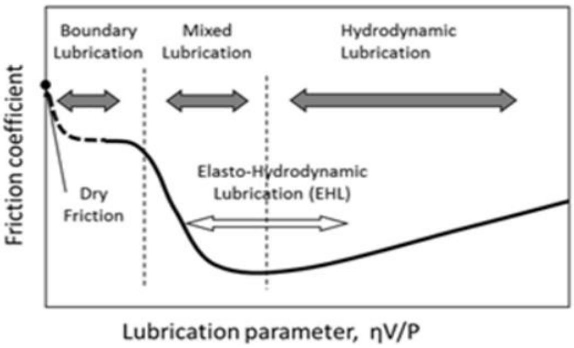

Figure 41: The Stribeck Curve

The Stribeck Curve is a tool in the hands of the engineer who wants to determine the lubrication conditions of a system, as a function of the Hersey Number, which is a dimensionless lubrication parameter, and the friction coefficient. The Hersey Number is calculated based on the formula:

$$\boldsymbol{Hersey\ Number = \frac{\eta * N}{P}, where}:$$

- η: the dynamic viscosity of the fluid,
- N: the entrainment speed of the fluid, and
- P: the normal load per length of the tribological contact.

The Elasto-Hydrodynamic Lubrication region (EHL), which correlates to high-performance and long-lasting lubrication arrangements and exhibits the lowest friction areas on the figure, is the most preferred region. The lubricant's viscosity increases during the Elasto-Hydronamic Lubrication phase, which typically occurs in high pressure operating conditions and causes the metals to undergo elastic deflections. Meanwhile, a rigid oil film of small thickness interferes between the two metal surfaces, preventing contact between them. Gear applications are frequently observed to have this state of lubrication.

When the bearing is lightly loaded (insufficiently loaded bearing) it is susceptible to whirling vibrations. This phenomenon in the lubricant is called “oil whirl”. In this case, the pressure of the lubricant film is much larger than needed to support the shaft’s weight and loads, which results in lifting the shaft up to a certain point and then dropping it down. During this phenomenon, the vibrations caused can lead to damage.

If a state in which the Main Engine’s RPMs and the oil’s temperature are constant is assumed, a rise in a load would be followed by a series of changes in the journal baring’s operational as stated below:

- The eccentricity and the oil film thickness are going to alter, in a change on behalf of the bearing’s assembly to carry the new load
- If the load is too high (overloading), then the temperature will rise
- The increase in the oil’s temperature will lead to failure due to oil overheating.

An overloaded bearing could also suffer from wiping. Another common problem concerning journal bearings, is oil contamination, which can negatively impact the system’s behavior and cause failure.

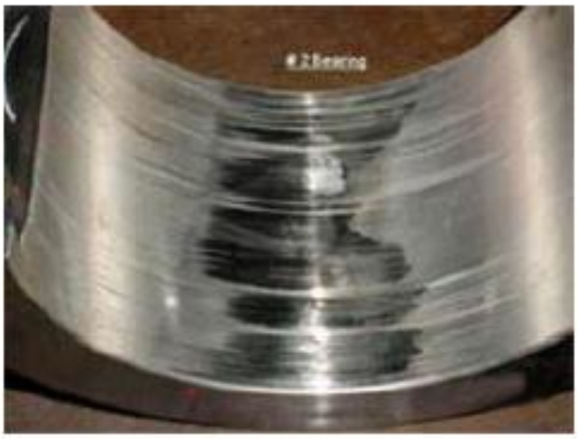

Figure 42: Result of metal to metal contact during to 'oil-whirl' induced vibration

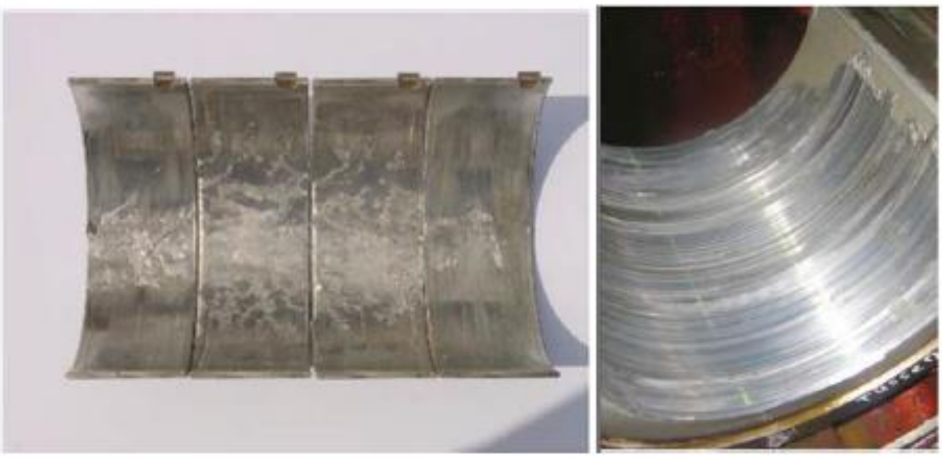

Figure 43: Bearing overloading: Metal Fatigue (left) and excessive wiping (right)

B.1.4.2. Further Explanation on the terms of “Oil-Whirl” and “Wiping”:

At this point, for clarification reasons, some technical terms which were previously mentioned, will be explained in detail, as they very often concern the engineers responsible for the propulsion shaft of a vessel:

- “Oil-Whirl”: One of the most frequent instabilities in hydrodynamically lubricated bearings, such as journal bearings, is thought to be oil whirl. Normally, the oil that is revolving around the shaft accelerates to a velocity that is somewhat slower than the surface speed of the journal. But as this Thesis’ previous chapter noted, a transitory condition takes control when a sudden change is made to the system.
  Under typical circumstances, the shaft rides above an oil gradient, with a particular vertical offset determined by the weight, speed, and pressure of the oil. This compresses the oil into a wedge, creating the pressurized oil film that supports the loads of the shaft. Nevertheless, the system enters a "transient condition" when the forces acting on the shaft change, causing the shaft to deviate from its equilibrium position in terms of eccentricity. Consequently, more oil is added during this stage to fill the space that the shaft left vacant. This condition causes the pressure within the oil film to rise, which creates more forces between the oil film and the shaft. During this stage, the oil may be able to drive the shaft in front of it in a direction that revolves around the bearing and moves in a forward circular motion. The phenomena eventually disappear, and the oil film-shaft system resumes regular

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operation, if the circumstances permit it. If not, the shaft will keep spinning, which could eventually get violent and cause the mechanism to break or wear down severely.

The whirling phenomenon, could occur because of the following reasons:

1. Because of internal characteristics of the system:
   - Insufficient load bearing and/or light dynamic and preload forces
   - Alterations in oil's characteristics, such as: viscosity and/or temperature
   - Excessive bearing wear
   - Excessive bearing clearance
   - Gyroscopic effects (especially on overhung shafts)
   - Changes in internal damping
   - Improper bearing design
   - Changes in oil film's pressure
   - Fluid leakages (in shaft labyrinth seals)
2. Because of vibrations transmitted into the bearing from external units (through structures, floors and/or foundations), or from parts of the same machinery as well. If the external vibrations' frequency has the same frequency as the oil whirl frequency of this specific bearing, they could meet all the requirements (magnitude and correctly tuned frequency) to result in oil whirling of the shaft.

An easy way to recognize the oil whirling phenomenon, is the unusual vibration frequency, which equals approximately 40% to 48% of the shaft's RPM. If the vibration's amplitude reaches 40% to 50% of the normal bearing clearance, it should be considered as dangerous, and corrective actions should be taken in order to avoid future problems and damage. The corrective measures, separated in temporary and permanent, have as follows:

I. Temporary Measures: Deliberately increasing the loading, by changing the oil's viscosity (via changing its temperature). Another way could be the heating or cooling of the support legs in order to alter the alignment condition, as well as grooving the bearing's surface in order to disrupt the lubricant's wedge. Moreover, the oil film's pressure could be modified.

II. Permanent measures: A permanent corrective action could be the installation of a new bearing with proper clearances or the pre-loading of the bearing. If the aforementioned actions prove insufficient, the whole installation could be modified to change the bearing's type. In this case, axial-groove bearings, lobed bearings or tilting pad bearings could be a good alternative.

At this point, it is important to highlight that ABS (2022) in Enhanced Shaft Alignment Guide, requires calculations regarding whirling to be conducted. More specifically, the bearing stiffness as well as the oil film's characteristics should be taken into account, and the calculations should be conducted for the following cases:

1. All bearing positively loaded as per alignment calculation report results and with the appropriate bearing stiffness calculation for each bearing.
2. As per 1. But with unloaded forward stern tube bearing.

The calculations should be made for a number of critical speeds, which should not lie between -20% to +20% of the main engine's MCR. ABS (2022) also suggests that the decision regarding the critical speeds is made by utilizing a Campbell Diagram, with the X-axis representing the propeller shaft's speed and the Y-axis representing the natural frequencies variation with shaft speed.

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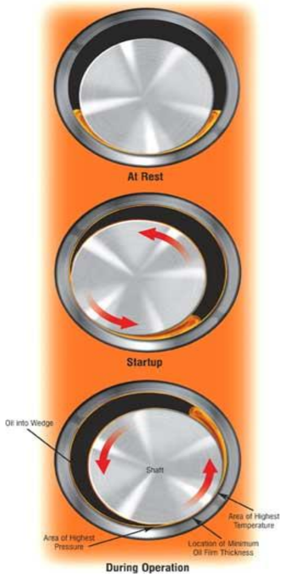

Figure 44: Stages during the operation of a journal bearing

- Wiping: Severe overheating may arise from insufficient bearing clearance (and hence running clearance). This might also happen if there is not enough lubricating oil available. These defects could result in wiping, which is the rubbing, melting, or smearing of a bearing's surface. Extreme loading (particularly during startup or run-down) may cause "wiping". Excessive imbalance or journal instability may also cause a "wiping" phenomenon by disrupting the lubricating oil coating and causing unwanted shaft vibrations. Avoiding operational overloading will reduce the likelihood that the "wiping" phenomenon may occur. Another corrective measure might be to use jacking lubrication systems, which inject high-pressure oil into specific bearing grooves, to lubricate the system more effectively, particularly during the start-up and turn-off phases. Installing backup pumps could also be an option to prevent the lubrication system from being interrupted in the event of a power outage. To support the intended high loads, the complete system might also be changed by adding additional, larger, or stronger bearings. An engineer with experience in bearings should be consulted if insufficient bearing clearance is the source of "wiping".

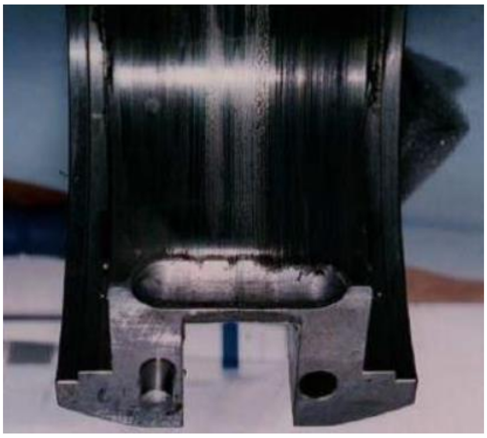

Figure 45: Ring-oiled white metal bearing, wiped due to lack of lubricating oil

Figure 46: Wiped white metal bearing of a turbine

B.1.4.3. A Brief Reference to Lubricants used in the Maritime Industry:

At this point, it would be helpful to provide some fundamental conceptual information about lubricants and, by extension, lubricants that are used in the maritime industry.

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The majority of traditional marine 2-stroke diesel engines use paraffinic base stocks and SAE30/40/50 monograde lubricants. Detergents/dispersants, corrosion inhibitors, antioxidants, wear-depressants, and anti-foam chemicals are the primary additions. Because the variety of marine engines and service applications is limited, there is no system in place for classifying marine lubricants as there is in the automotive sector.

Lubricants are regarded as a vital aspect of the petroleum sector. On the other hand, lubricant usage worldwide amounts to just 1% of total fuel consumption. However, in the absence of lubricants, the machinery's general financial performance and vitality are jeopardized. One of the few studies on the financial aspects of lubrication systems, the Jost Report study, was carried out in Great Britain in 1966 and reports that the additional costs incurred by the country's industry because of improper lubrication of machinery reached up to 600.000.000£/year. Therefore, the importance of appropriate lubrication has an impact on both the financial and operational aspects of the relevant industry.

As lubricant technology stabilized throughout time, the term "lubricants" came to refer to either grease (solid) or oil lubricants (liquid). Lubricants are primarily used to change solid friction between two metal surfaces into liquid friction, which is often of smaller magnitude. An appropriate lubricating system ensures smooth and uninterrupted operation of the machinery,

I. The expansion of the machinery's life span,
II. The minimization of the energy losses thanks to the decreased friction forces, and
III. Reducing maintenance and operation costs of the machinery.

Bibliography states that the energy consumption of the global industry might be reduced by about 4.5% if adequate lubrication was used in all of its machinery. To be more precise, this percentage equals 720.000 tons of petroleum equivalent annually in the Greek industry.

Today's increasingly complicated machinery and technology around the world have brought attention to how important proper lubrication is. Modern applications necessitate more intricate and demanding lubricating system designs because, in addition to having intricate layouts and working concepts, they also frequently run under higher loads and stresses.

The classification of the lubricants, has as follows:

1. Gas Lubricants: Practically refers to air. Gas lubricants find applications in high speed-low load installations, like gas circulators in fusion reactors.
2. Liquid Lubricants: This category incorporates a wide range of substances, from liquified gases to synthetically manufactured oils. Liquid lubricants are the ones which mainly concern the propulsion shaft mechanism of a vessel and they will be more thoroughly explained later. The most common liquid lubricants are:
    i. Water,
    ii. Mineral Oils,
    iii. Synthetic Oils.
3. Semi-solid Lubricants: This category consists of greases of natural derivation (extracted from plants or animals). The most common semi-solid lubricants are Vaseline, wax and other compounds of oils and fats.
4. Solid Lubricants: These lubricants do not melt during their operation and act as a solid surface between the moving metallic parts of the machinery which needs to be lubricated. The most common solid lubricants are Graphite, talc and Molybdenum Sulfide (MoS_2).

All the widely used lubricants are made from Base Oils/Stocks, carefully mixed with additives, in order for the final lubricant to be produced. As their name suggests, the Base Oils are the base of the final mixture and as a

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result they tend to occupy a higher percentage of the final product. Base stocks also significantly affect the properties and the characteristics of the lubricant to be produced. In the context of Maritime Industry applications, and more specific, applications which concern the propulsion shaft of a vessel, more clarifications will be given explicitly on Liquid Mineral or Synthetic Oils, which make up the main Base Oils/Stocks for marine applications.

Mineral Oils:

Mineral Oils are mixtures of hydrocarbons, which also contain solutions of small amounts of solid or liquid additive ingredients. The so called “additives” aim to contribute some specific characteristics on the lubricant, which depend on the application it is going to meet. Mineral Oils have the advantage of being suitable for a wide range of applications regarding the operating temperature, and thanks to the additives, their characteristics can be easily modified to serve a better treatment of the metal surfaces incorporated in the concerned machinery.

The number of carbon atoms found in the structure of the hydrocarbons which make up the most common Mineral Oils, have a range of about C_{20} to C_{70}. The main chemical structures one meets regarding Mineral Oils are: Paraffins, Naphthenes and Aromatic Hydrocarbons. Small concentrations of hydrocarbons with heteroatoms such as sulfur, nitrogen or oxygen added to their molecules can also be found. The most common discrimination amongst Mineral Oils is the one incorporating the type of the hydrocarbons which make up the oil, and has as follows:

1. Paraffinic Base Oils: They make up approximately 85% of the global Mineral Oil market, as they are the preferable solution to the majority of the applications. They are used in the producing of engine oils, transmission fluids, gear oils, metalworking fluids and greases. Their main disadvantage is their behavior in low temperature operating conditions.
2. Naphthenic Base Oils: They are mainly used for the production of turbine oils, hydraulic oils, cylinder oils and greases. Their applications mostly concern low operational temperature arrangements, as they are characterized by low performance regarding their viscosity index and their oxidation stability, which means that their pour point is remarkably low as well.
3. Bright Stock: Bright Stocks are produced from the processing of the remains of the production of Paraffinic Base Oils and the Naphthenic Base oils. They are characterized by high concentrations of aromatic compounds, and they are generally not preferred because of their low oxidation stability characteristics as well as due to their tendency of forming deposits in high operational temperatures. However, they are widely used as constituent in mixtures for the production of engine oils, gear oils, hydraulic oils and greases, thanks to their high viscosity.

Mineral Oils are the most preferable constituent for Base Stocks, and the reasons why this happens have as follows:

- Significantly lower cost of purchasing,
- Availability,
- Suitability for a wide range of applications.

Synthetic Oils:

This type of Base Oils/Stocks are manufactured via chemical synthesis, which could specifically be the process of Polymerization, of Oligomerization, or of Chemical Reaction between two chemical compounds, such as the one between acid and alcohol in order for ester to be produced. By these means, the quality of the lubricant is guaranteed to remain adequate under any circumstances, and the process of the lubricant’s produce can be independent from the starting materials’ availability. Some of the most common Synthetic Oils are Polyesters, Diesters, Alkylated Aromatics, PAGs and silicones.

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The tendency nowadays in the areas of modern cutting-edge engine designs and machinery arrangements and installations in general, leans towards functioning under more intense conditions, higher energy efficiency, maximization of the lifespan and minimization of the maintenance requirements. Consequently, these contemporary requirements oppose that the manufacturing of lubricants evolves towards optimization, so that the modern lubricants answer to the industry's needs for maximum efficiency.

The process of manufacturing Synthetic Oils is very accurate and controllable, and as a result the products are characterized by even chemical structure, less impurities and minimization of undesirable by-products. In general, the main properties of lubricants that Synthetic Oils optimize, are (Technology of Fuels and Lubricants, E. Lois, F. Zannikos, D. Karonis, Athens 2014):

- Viscosity Index,
- Pour Point and behavior in low operational temperatures,
- Oxidation and thermal stability, and
- Volatility.

Synthetic Oils are divided in two main categories:

- Synthetic hydrocarbons and
- Sundry synthetic compounds.

The most common synthetic hydrocarbons one meets in Synthetic Oils, are:

i. Poly-alpha-olefins (PAOs),
ii. Poly-isobutylenes
iii. Di-alkylbenzenes
iv. Di-esters
v. Polyesters
vi. Poly-alkylene-glycols
vii. Phosphate esters

The precise decision on the synthetic hydrocarbon, which is going to be used, depends on its advantages and disadvantages, as well as on the installation in which it is going to be used. Synthetic oils are often preferred, because of their improved capabilities in terms of volatility and viscosity. However, they are usually accompanied by significantly higher prices in the market.

According to the American Petroleum Institute (API), the main categorization of lubricants is based either on their origin (in terms of Mineral Oils or Synthetic Oils), or on some of their characteristics (API 1509). This categorization serves as a helpful tool for the responsible engineer to choose the right lubricant for a specific application. For practical reasons, a table has been published from API, containing five groups, which will be explained as follows:

I. Group I: Generally, groups I to III contain Paraffinic Base Oils. However, the lubricants which fall into Group I have as their classification criterion their Sulfur content. The lubricants which belong to Group I have a viscosity index of about 80-120. At this point, it would be useful to mention that the most found viscosity index of lubricants in the market is approximately 95-105. In general, a value of viscosity index equal to 95 is the minimum one can find on the lubricants' market.
II. Group II: As pre-mentioned, this group contains Paraffinic Base Oils as well, with the difference from Group I being that the criterion in this case is their percentage on hydrocarbons. In this Group, the lubricants usually have a viscosity index approximately equal to 95-105, similar to typical viscosity index values of Group I.

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III. Group III: As aforementioned, this group contains Paraffinic Base Oils. In this case, the lubricants have similar Sulfur content and hydrocarbon percentage values with the lubricants of Groups I & II, however they are characterized by a significantly high viscosity index.
IV. Group IV: This Group entirely concerns Poly-alpha-olefines (PAOs).
V. Group V: This Group contains the lubricants which do not fall into one of the previous categories, such as Naphthenic Base Oils, esters, Poly-glycols and others.

Due to the constant changes in the industry of Lubricants, two additional un-official sub-groups have been created (II+, III+), aiming to describe and include more contemporary, newly designed, optimized lubricants, which have their main characteristics (which led to them falling into the Groups II or III) slightly improved, with values on the upper limit areas of the Group. More accurately, the lubricants of Group II+ have viscosity indexes approximately equal to 110-120, as well as improved behavior in terms of cold pour and low volatility. Moreover, lubricants of Group III+ are characterized by viscosity index values which in specific cases reaches up to 130, and they are usually manufactured in gas-to-liquid (GTL) units. For clarification reasons, the classification table of API is presented below:

Table 1: Classification Table of API, Source; API 1509

Group	*Sulfur, % by weight*		*Saturated Hydrocarbons, % by weight*	*Viscosity Inder (VI)*	
I	>0.03	and/or	<90	$\geq 80\ to < 120$	
II	≤ 0.03	and	≥ 90	$\geq 80\ to < 120$	
II+	≤0.03	and	≥ 90	$\geq 110\ to < 119$	Un-official sub-category
III	≤ 0.03	and	≥ 90	≥ 120	
III+	≤0.03	and	≥ 90	$\geq 130\ to < 150$	Un-official sub-category
IV	All of the Poly-alpha-olefins (PAOs)				
V	All of the categories not included in the Groups I-IV				

Lubricant Additives:

As previously mentioned, in order for a lubricant to be produced, Base Stocks/Oils are mixed with Additives, which are usually made from synthetic compounds or mixtures. The spectrum of the market's additives is very wide. The main scope of using Additives to produce the final lubricant, have as follows:

- Rendering to the final product new characteristics,
- Improving some of the existing characteristics of the lubricant,
- Reducing the rate of quality dropping as the years of the lubricated part's functioning increase.

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All of the lubricants one may come across in the market these days have a number of additives in them. The lubricants typically contain additives in the amount of approximately 1% to 25% by weight, depending on the specific installation the lubricant is intended to be used. The diagram below indicates the precise percentage of additives added in Lubricants (in 2006), as a function of the specific usage the lubricant is intended to have. [3].

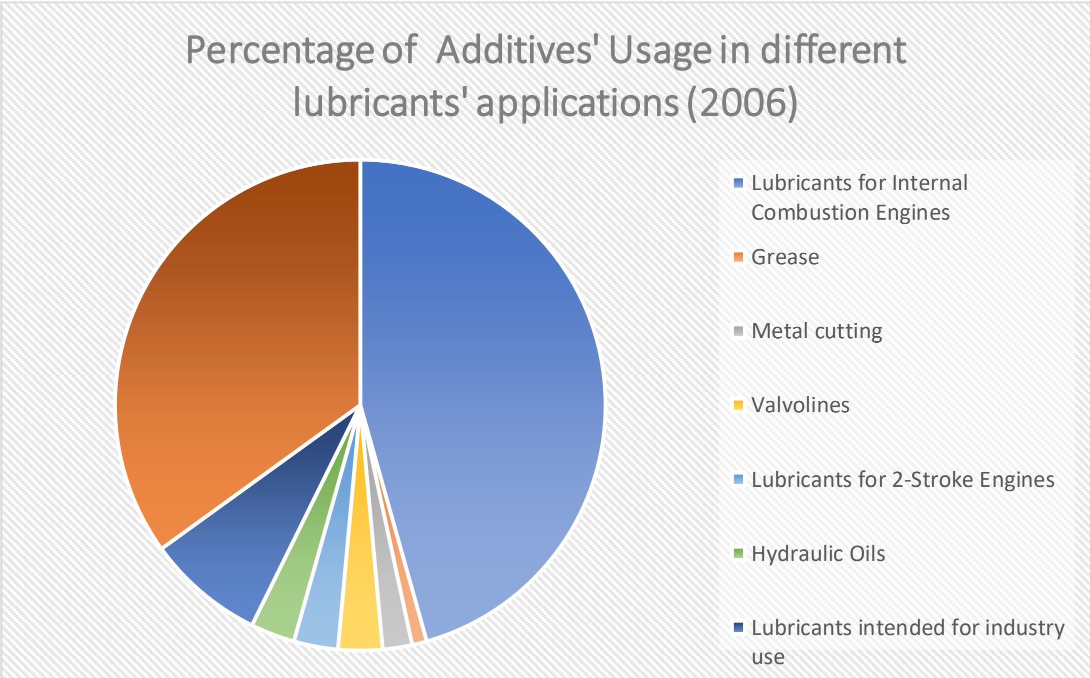

Figure 47: Diagram; Percentage of Additives' Usage in different lubricants' applications (2006)

The Additives can be classified in the following categories, based on the purpose they serve in the final product:

- Stabilizers, which are additives aiming to regulate the number of deposits in the lubricants,
- Ingredients which serve the purpose of creating a lubricating membrane, aiming to protect the lubricating parts from wear and corrosion,
- Polymers, which serve the purpose of control over the rheological properties of the lubricant (such as viscosity or pour point), and
- Other Additives which offer lubricant properties and characteristics which are not included in the previous categories.

Stabilizers:

The most common Stabilizer Additives are presented below:

I. Oxidation Inhibitors & Metal Deactivators:

These Additives aim to regulate the quality drop of the lubricant because of oxidation, as if the lubricant gets oxidated, its viscosity is altered. The Additives which tend to decrease the lubricant's vulnerability to oxidation are called Antioxidants, and due to their acid composition, they generally protect the lubricated metal parts from corrosion. The reason why the pre-mentioned is important lies in the fact that the commonly known Base Oils are susceptible to oxidation, due to their organic nature. More precisely, Synthetic Oils are the ones with the highest Oxidation Stability, while Mineral Oils have a mediocre behavior. The main factors which are widely known for their contribution to Oxidation, are:

- The presence of NO_x,
- The presence of metals along with their ions,

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- Large amounts of Oxygen, and
- High operational Temperatures.

Internal Combustion Engines, unfortunately meet all of the above, thus being the “Top Applicants” for Additives such as Antioxidants to be present in their lubricants.

II. Dispersants: For machinery which operates in a wide range of loads, it is very common for insufficient combustion to take place. The remaining fuel along with traces of water, creates a sludge which is capable of increasing the lubricant’s viscosity, blocking the filters of the lubrication system and/or making the cooling of the Engine significantly harder. The Dispersants are usually compounds of Nitrogen or Oxygen and are absorbed by the particles in the sludge’s surface, thus preventing the creation of aggregations of colloid particles and the imminent creation of sludge.

III. Detergents: Detergents’ action is similar to the one of Dispersants’. They usually are compounds of Barium, Magnesium or Asbestos. Their purpose is to eliminate the acid properties of the products of oxidation and combustion, thus preventing the creation of rust, corrosion and the deposition of resinous constituents in the Engine.

Ingredients which serve the purpose of creating a lubricating membrane:

I. Friction Modifiers: During the Engine’s cold start, as well as during each pause and restart of the Engine, the friction is remarkably higher, as the RPM increase, and the lubrication system goes from a Boundary Lubrication Condition to a Mixed Film Lubrication Condition and finally to a Full Film Lubrication condition. The Friction Modifiers create a thin film of oil on the surfaces of the lubricated metal parts which suffer friction, thus reducing the friction concerning the pre-mentioned vulnerable phases of the Engine’s operation. By these means, the energy losses are also minimized, and the fuel consumption of the engine is optimized. Friction Modifiers are usually combined with Extreme Pressure (EP) Additives, which will be discussed later.
A very interesting aspect of all of the above is a form of the Stribeck Diagram which indicates the preferable Additives of the Lubricants as a function of the Hersey’s Number and the Friction coefficient, for all Lubrication System’s conditions.

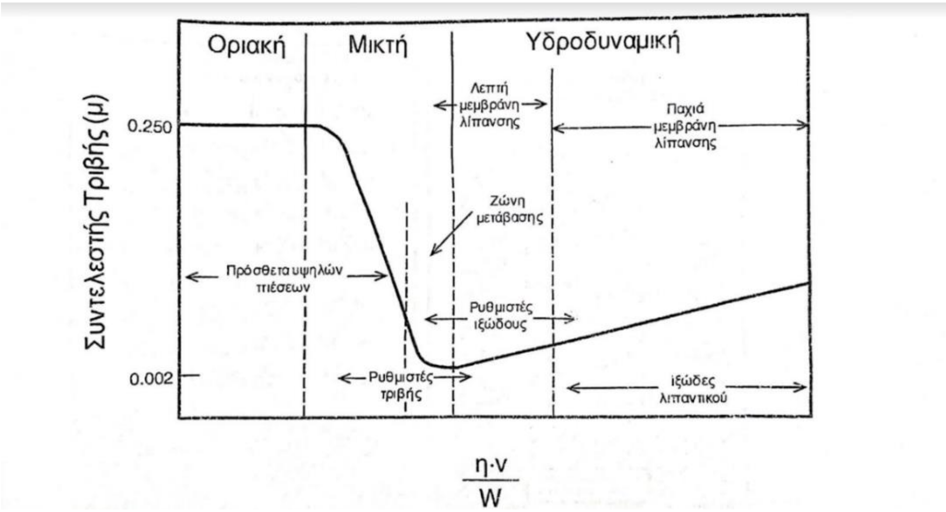

Figure 48: Stribeck Diagram: The role of Additives in different Lubrication Conditions, Technology of Fuels and Lubricants [33]

Extreme Pressure Additives – Additives aiming to prevent Wear

In machinery which contains rotating parts, wear is always present. There are four different conditions, which can lead to wearing of a machinery:

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1. Contact between two surfaces (frictional or adhesive wear),
2. Contact of a surface with a foreign matter (abrasive wear),
3. Corrosion (corrosion wear), and
4. Contact between two surfaces, which additionally suffer repeatedly loads for long time periods (fatigue wear).

The Additives of this category, aim to regulate and control the Abrasive Wear, by using Anti-wear and Extreme Pressure (EP) agents. During time periods where significantly high loads are developed, the EP agents take the place of Friction Modifiers, while serving the purpose of protecting the metal surfaces from Abrasive Wear. As the loads are reduced, the Friction Modifiers take over again, serving their own purpose, which is already discussed.
The most common EP agents contain Alkyl and Aryl Disulfides and Polysulfides, Di-thiocarbamates, Chlorinated and Sulfochlorinated Hydrocarbons, Di-alkyl Hydrogen Phosphites and Salts of Alkyl Phosphoric Acids, while the most common anti-wear agent is Zinc Di-alkyl Di-thiophosphates.

II. Anti-Corrosion Additives: The term corrosion is used in order to describe the phenomenon of the destructive alteration of a metal due to chemical or electrochemical reactions, caused by its environment. Several types of corrosion exist, however, in the context of lubricants, the interest leans onto corrosion with the presence of electrolytes (electrochemical corrosion) and onto corrosion in the absence of electrolytes (chemical corrosion). The most common electrolytes which cause electrochemical corrosion are water, acids or salts. Chemical compounds which are known to lead to chemical corrosion are acids, sulfur or sulfur compounds. In metal alloys, the corrosion may be selective or non-selective. When a corrosion is selective, it means that one specific metal corrodes more preferably than another. A non-selective corrosion means that all of the metals which the alloy contains corrode at the same rate.
A metal is more susceptible to corrosion when both acids and bases are present, as the combination of the two may accelerate the corrosion process. Speaking in the context of lubricants, acids are produced by the oxidation of sulfur, which is contained in fuels, as well as in the Base Oils/Stocks. Moreover, some of the Additives contain small amounts of compounds (organic or inorganic) which are mediocrely bases. These bases, if examined alone, do not cause any kind of problem. However, when combined with the acids pre-mentioned, they create electrolytes, thus creating an environment which meets the criteria for accelerated corrosion to take place. Especially in the context of Engines, where the temperatures are high, corrosion is significantly facilitated. If the corrosion problem is not properly treated, significant amounts of material may be lost, and the operational integrity of the machinery may be compromised.
The Additives which are used in the industry in order to treat the aforementioned issues are the Corrosion Inhibitors. Their function is based either on the idea of deactivating/neutralizing the acids, either on creating an impenetrable film of oil or facilitating the creation of a film oil from the lubricant itself, thus protecting the metal's surface.

Polymers:

I. Anti-Foaming Agents: The amount of air which is contained in the lubricant plays a significant role in its performance. More specifically, gases have a constrained solubility regarding their dissolving in lubricants, which depends on the temperature of the lubricant and the characteristics of the gas. For instance, an amount of approximately 8%-9% by weight of air is soluble in Mineral Oil, in room temperature. The amount of dissolved air in a lubricant affects its viscosity, its compressibility, the heat transfer, the oxidation, the boundary lubrication condition and its foam tendency. When the amount of gas in a lubricant exceeds the permissible limits, the entrainment phenomenon occurs, when bubbles of gas come out of the lubricant at a relatively slow rate and make the lubricant blurry for a specific time period. The creation of foam is happening due to large amounts of gases trapped inside the lubricant in liquid phase and is undesirable in lubrication applications.

Anti-foaming agents serve the purpose of controlling the foam forming phenomenon by modifying the surface tension of the lubricant, thus allowing the separation of the gas and the liquid to take place more easily.

II. Viscosity Modifiers: The aim in controlling and minimizing the changes in the viscosity of the lubricant, which occur due to temperature alterations. The Viscosity Index of a lubricant serves as a tool of connecting the response of the lubricant to temperature modifications. Practically, Viscosity Modifiers tend to minimize the alterations in the viscosity of the lubricant which occur due to temperature changes. As a result, the range of the operational temperatures of the arrangement widens towards higher temperature areas, without affecting the behavior of the lubricant in low temperature areas. Viscosity Modifiers are Polymers, and they are often used in lubricants of low viscosity to improve their rheological characteristics in higher temperatures. While Viscosity Modifiers increase the viscosity of the lubricant in all temperatures, they also result in less thick oil films in low temperatures. A Diagram which shows the effect of a polymer used as a Viscosity Modifier on the relationship between the viscosity and the temperature, compared to the same lubricant without this specific additive, is shown below:

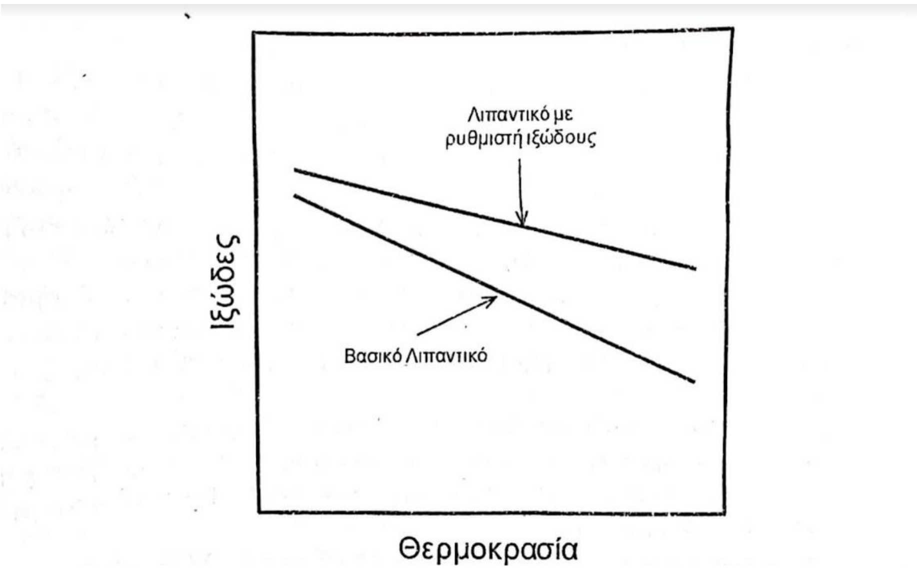

Figure 49: Diagram; Effect of Viscosity Modifier in the relationship between viscosity and temperature, Technology of Fuels and Lubricants [33]

It is obvious, that the decreasing of the lubricant's viscosity as the temperature rises, is significantly less intense and smoother in the case of the Lubricant with Viscosity Modifier as Additive.

Viscosity Modifiers also affect some other characteristics of the lubricant, such as Pour Point, the functioning of the Dispersants, the fuel consumption and implicitly the performance of EP Additives.

III. Pour Point Depressants:

By the term Pour Point, the lowest temperature in which a fuel or a lubricant can pour, if cooled under precise circumstances, is indicated. The Pour Point of a lubricant significantly affects its performance and behavior in low temperatures. For this reason, Pour Point Depressants are used as additives.

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Other Additives:
Some other types of Additives are usually preferred for the lubricant to gain a characteristic which is not present without the specific additive, or for the lubricant to be capable of adapting successfully under conditions which are not explicitly connected to lubrication.

I: Emulsifiers-Demulsifiers:

Emulsifiers are used to make the mixing of two ingredients with the scope of creating a smooth and uniform final product (emulsion) possible. More precisely, lubricants mixed with water are widely used in some industrial applications, because they are fire-retardant, easy to dismiss and cheaper compared to other lubricants (non-emulsified). However, there are some cases where the opposite process is needed, demulsification. In some cases, the presence of water leads to the formation of emulsions which are not desirable at all. With the help of Demulsifiers, the water can be separated from the lubricant and the foam creation can be significantly controlled as well.

II: Seal Swell Agents: Elastic seals are widely used in lubrication systems, to: Isolate the lubricant from harmful substances, eliminate the possibility of contamination of the lubricant, preserve the hydraulic pressure and allow the discharge or replacement of damaged parts of the lubrication system without a total dismantling being needed. These elastic seals are usually made of Polymeric compounds, and in combination with some lubricants they may be vulnerable in terms of shrinking, fragility and quality drop. All the pre0mentioned can drastically affect the credibility of the lubrication system. This is the reason why Seal Swell Agents are used as Additives in some lubricants.

III: Dyes: These Additives are used to make the discrimination between numerous lubricants possible. They are also used, so that in case a leakage occurs, it can easily be detected and treated. For example, the Automatic Transmission Fluids (ATFs) are usually dyed red color, and the lubricants of 2-Stroke Diesel Engines are usually dyed blue or purple color.

IV: Biocides: Generally, because of the high operational temperatures of the arrangements in which lubricants can be found, combined with Biocide Additives, lubricants are durable in terms of microorganism infestations. However, in some lubricants which contain larger amounts of water (like cutting fluids or hydraulic fluids), contamination from germs or fungi is characterized by a higher likelihood. The dangers of the prementioned either concern the quality and performance of the lubricant, or they concern sanitary reasons. The purpose the Biocides serve is accomplished though the usage of water-soluble derivatives of Triazine, Morpholine, Imidazoline and Thiazoline.

B.1.5. DNV's Latest Suggestion on the Control of the Risk of Propeller Shaft Bearing Damage:
In an effort to eliminate the dangers related to damage or failure of the aft stern tube bearings, DNV (2023) recently published some news concerning the reducing of risks when it comes to aft stern bearing damages, in which references in the Lubrication Criteria of the aft stern tube bearing is made. As pre-mentioned, the aft stern tube bearing is severely impacted by the lubrication conditions which exist during its operation, and it is extra sensitive when it comes to lubrication matters.

As DNV (2023) states, several difficulties lie on the whole procedure of detecting aft bearing damage in its early stages, which in turn affects the pro-active actions one could take to minimize the operational risks of the propulsion shaft. Some indicative reasons are described below:

- Old vessels which use Environmentally Acceptable Lubricants (EALs) exhibit substandard load capacity in severe transient operating conditions, along with high local oil film pressure in the bearing.

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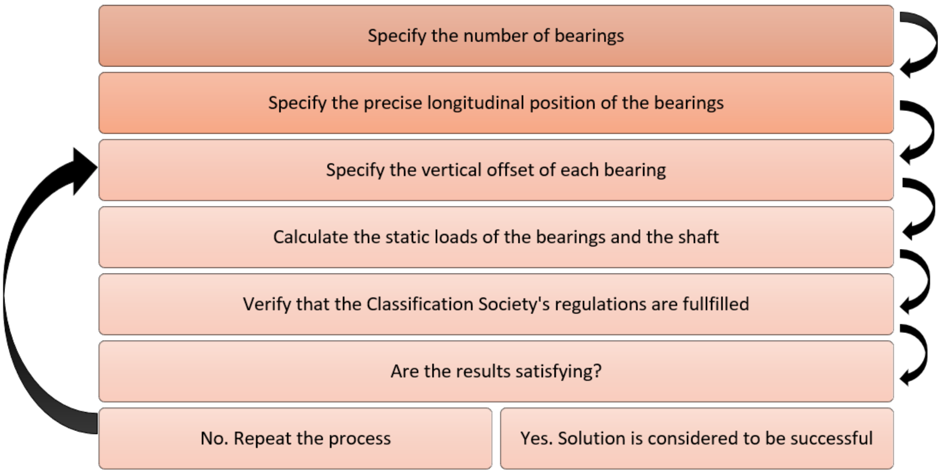
- Aft sealing system lubricant quality which is not monitored to ensure suitability for continued use, especially when EALs is used. The quality of the lubricant oil has a huge impact on the inner shaft sealing ring condition, which in turn risks contamination of the aft stern tube lubricant system.
- Continuous operation with incomplete propeller immersion leads to downward bending of the shaft because of the eccentric propeller thrust forces. This runs the risk of edge loading in the aft part of the bearing and consequential local overheating.
- Abrupt increase in the aft bearing temperature before the high temperature alarm is activated, particularly at lower operating temperatures.

DNV (2023) suggest the following in order to monitor more successfully the aft stern tube bearing's behavior, thus reducing failure and damage risks:

1. Using double-slopped aft bearing designs, to optimize the load distribution and enhance the operating margins in continuous and extreme transient conditions
2. Providing means of warning for the high rate of rise of aft bearing temperature and incomplete propeller immersion, to give the opportunity to the operators for pro-active actions.
3. Increasing the EAL viscosity to the next higher grade above design specification for EAL application on older installations (for vessels with applicable DNV Rules older than July 2019). Vessels with applicable rules as of July 2019 or later will need a new approval if design oil type or viscosity grade is changed (ref. DNV EAL news Oct. 2019).
4. Investigating and undertaking prompt actions if main stern tube system lubricant analysis reveals elevated levels or an increasing trend on water content, Total Acid Number (TAN), bearing and/or shaft material wear elements, etc.

B.2. Basic Parameters in The Shaft Alignment Procedure

The process of studying, analyzing, calculating and verifying a vessel's proper propulsion shaft alignment procedure is considered to be significantly complex and complicated. A lot of parameters interact with each other, thus necessitating a method of circulation of the information gained during the process. For example, the precise number and longitudinal position of the shaft's bearings, is one of the issues which require iterative method calculations. Below, an iterative process plan concerning the number and the longitudinal position of the bearings is presented. For clarification reasons, it should be mentioned, that the parameters which generally need to be verified and optimized in this occasion, are: The vertical offsets of the bearings, the beatings' reactions, the static loads of the bearings and the shaft, and the verification of the Classification Society's requirements.



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However, there are some basic parameters which are going to be particularly active during the entire shaft alignment process. As these terms are going to appear very often, it would be useful to facilitate the readers by describing and explaining these parameters now.

B.2.1. Influence Coefficients:

The Influence Coefficients are present in almost every IACS Class rule. Consequently, it would be useful to introduce the basic theoretical background of their concept:

During the shaft alignment planning process, the longitudinal position and the vertical offset of each bearing is decided. However, even a slight alteration in the vertical offset of a specific bearing could have a significant impact on the forces and the loads applied to the rest of the bearings. This practically describes the results of a change in a bearing's offset in the load distribution among all of the bearings, and it indicates the sensitivity of the system to modifications on the offsets of the bearings. It is also a parameter which correlates the relationship between two specific support points.

More precisely: Each Influence coefficient, measures the change in a specific bearing's reaction force, which occurs due to the modification of one bearing's vertical offset, if the rest of the bearings have a constant vertical offset and that all of the bearings have a constant longitudinal position.

As a result, there are numerous influence coefficients: Precisely, if N is the number of the bearings, then the total number of Influence Coefficients equals to N^2. If these influence coefficients take the form of a matrix, then the Influence Coefficient Matrix is produced, which is a symmetrical and square matrix. Each bearing has an influence coefficient for every single bearing which could be modified in terms of vertical offset. When discussing about the influence coefficient σ_{ij}, one means the effect of a unit change in the vertical offset of bearing i, on the change in the reaction force of bearing j. The mathematical definition of an influence coefficient is described below:

$$\boldsymbol{\sigma_{ij} = \frac{W_{ij} - W_j{}^0}{y_i}}, \text{ where:}$$

- σ_{ij}: The influence coefficient of the bearing i on the bearing j,
- W_{ij}: The reaction force of the bearing j, if the bearing i has changed its vertical offset by y_i,
- y_i: The change in the vertical offset of the bearing i,
- $W_j{}^0$: The reaction force of the bearing j, assuming that all the bearings have zero vertical offsets (straight-line arrangement).

The aforementioned concept could be utilized in terms of reverse engineering, as one can directly calculate the reaction forces of a bearing for any vertical offset value desirable, if the requirement of knowing its influence coefficient and its reaction force in a straight-line installation are met. Then, the reaction force of the bearing will be equal to:

$$\boldsymbol{W_{ij} = W_j{}^0 + \sigma_{ij} * y_i}$$

The value of an influence factor indicates the susceptibility of the shaft system to small changes or external disturbances. As ABS (2019) states, "The influence coefficient matrix can be used to evaluate shafting sensitivity to possible disturbances in bearing offsets and assess the impact of this on the bearing reactions". The disturbances which ABS mentions, could possibly occur due to Hull Deflections, Thermal Deviations or Bearing Offset Adjustments. This is of great significance, as during a vessel's operation, the weather, the loading condition and the loading distribution have an immediate effect on the vertical offsets of the bearings. By calculating the Influence Coefficients of the matrix, the vulnerability of the system to the aforementioned changes can be approached. More precisely, the greater the value of an influence coefficient is, the more

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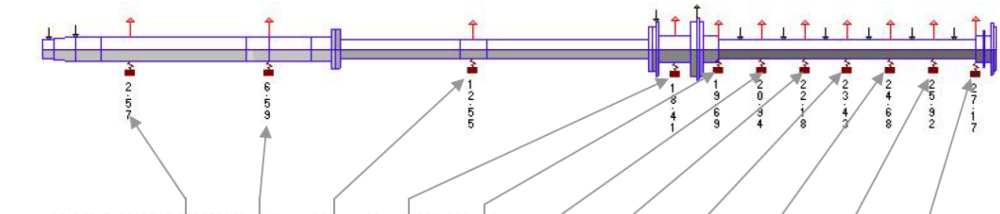
susceptible the system is to these small changes, and consequently greater risks lie in terms of misalignment or shaft alignment correlated failures/damages, while the smaller the influence coefficient is, means that the system is not dangerously affected by these changes.

At this point, it is worth explaining for clarification reasons, that according to ClassNK regulations regarding Shaft Alignment, which were published in 2006, the terminology used to refer to Influence Coefficients is slightly different. The term "Influence Coefficient" is referred to as "Bearing Reaction influential Number" and the term "Influence Coefficient Matrix" is referred to as "Shafting Stiffness Matrix". ClassNK (2006) suggests that by using reverse calculating methods, one can obtain the Bearing Reactions (without taking into consideration the weight of the bearings and any external loads), as pre-mentioned. ABS (2019) also proposes that the Influence Coefficient Matrix can be used to assess the impact of Hull Deflections on the reaction forces of the bearings, as well as to obtain an estimation of "the ratio between the stiffness of the shafting and the hull structural stiffness". This knowledge can be utilized during the design process of the shaft alignment, while determining the vertical offsets of the bearings.

ABS describes three possible scenarios regarding the ratio between the stiffness of the shaft and the flexibility of the hull structure, in which the influence coefficient matrix can be utilized in order to estimate the impact of hull deflections in the shaft alignment process. In these cases, only static condition and structural deflections occurring due to changes in the vessel's draft are considered.

1. Compliant System; Proportionally rigid shafting and hull: This scenario is most commonly present on smaller vessels, for example below 30.000 DWT. In this case, the shafting will have a large influence on coefficient numbers, because of its rigid structure, and consequently, it will be severely affected by small alterations in the vertical offsets of the bearings. Simultaneously, however, the proportionally rigid hull will not severely affect the bearings, as because of its stiffness, hull deflections on the stern part of the vessel "are expected to be of similar magnitude".
2. Noncompliant shafting; Rigid shafting and elastic structure: Although this scenario is not desirable, it is very usual for large modern vessels (commercial). In this case, as pre-mentioned in Scenario 1, the shaft's rigidness results in large influence coefficients, and as a result the shaft becomes really vulnerable to changes in the vertical offsets of the bearings. On the other hand, the flexible construction of the hull equals increased deformations of the hull. As the hull deflections take place, the shaft, due to its proportionally more rigid construction, at some point will not be capable of accommodating the deformations and will eventually lead to unloading of the bearings.
   This scenario is often in large vessels (ULCCs, VLCCs, large bulk carriers or large container vessels) with relatively short and rigid shafts, with a large diameter.
3. Compliant Shafting; Elastic Shafting and rigid structure: This is the most preferred scenario. This specific combination results in a shafting which can completely address and accommodate the hull deflections.

In the example of the bellow figure, one can easily notice that the influence coefficients of the engine bearings are larger than the coefficients of the rest of the bearings. However, in order to determine whether the Main Engine Bearings are actually more sensitive than the other bearings, one must consider the flexibility of the structure which supports the Main Engine Bearings. In this example, ABS (2019) explains that, even if the Main Engine Bearings have larger influence coefficients, the structure which supports the engine block, and the crankshaft is relatively more rigid than the structure which makes up the double bottom under the intermediate shaft. Consequently, the hull deflections concerning the specific area of the Main Engine and its bearings, in not that actively engaged in the Hull Deflection concept, and as a result the Reaction Forces of the crankshaft's bearings will not change dramatically.



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RELATIVE BEARING REACTIONS [kN] -> R[0.1-offset]-R[0-0ffset]
  Due to 0.1[mm] OFFSET relative to the ZERO bearing Offset
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Node	Supp.	< 7> 1	< 14> 2	< 27> 3	< 41> 4	< 46> 5	< 48> 6	< 50> 7	< 52> 8	< 54> 9	< 56> 10	< 58> 11
< 7>	1	4.289	-7.896	4.768	-3.222	2.120	-0.069	0.014	-0.003	0.001	-0.000	0.000
< 14>	2	-7.896	15.425	-11.112	9.951	-6.546	0.215	-0.044	0.009	-0.002	0.000	-0.000
< 27>	3	4.768	-11.112	13.421	-25.421	18.858	-0.618	0.125	-0.025	0.005	-0.001	0.000
< 41>	4	-3.222	9.951	-25.421	166.594	-250.824	123.820	-25.141	5.104	-1.035	0.202	-0.028
< 46>	5	2.120	-6.546	18.858	-250.824	511.825	-379.402	125.081	-25.395	5.148	-1.004	0.141
< 48>	6	-0.069	0.215	-0.618	123.820	-379.402	461.940	-295.733	108.086	-21.910	4.273	-0.599
< 50>	7	0.014	-0.044	0.125	-25.141	125.081	-295.733	389.320	-280.955	104.921	-20.460	2.870
< 52>	8	-0.003	0.009	-0.025	5.104	-25.395	108.086	-280.955	386.156	-279.504	100.648	-14.120
< 54>	9	0.001	-0.002	0.005	-1.035	5.148	-21.910	104.921	-279.504	381.883	-259.045	69.538
< 56>	10	-0.000	0.000	-0.001	0.202	-1.004	4.273	-20.460	100.648	-259.045	281.235	-105.848
< 58>	11	0.000	-0.000	0.000	-0.028	0.141	-0.599	2.870	-14.120	69.538	-105.848	48.046

```
Each column provides bearing reactions change corresponding
to the relating bearing rise of 0.1[mm] at respective bearing
(e.g. column #2 are bearing reactions to 0.1[mm] offset at bearing #2)

Support designation for this particular model:
        #1 Aft S/T brg; #2 Fwd. S/T brg; #3 I/S bearing; #4 to #12 M/E bearing;
```

Figure 50: A sample of an influence coefficient matrix of a sensitive vessel which belongs to Scenario 2, ABS [12]

However, as ABS (2019) states, at this specific example, the pre mentioned is true for all the Main Engine bearings except the three aftmost. The three aftmost main engine bearings are connected to the part of the shaft which lies upon a more flexible hull structure; thus, they are more sensitive to hull deflections. Similarly, in this example, in the cases where the influence coefficients are smaller, the engine's structure is more flexible and as a result the changes in the bearing offsets of these bearings is expected to be bigger.

In this specific example, as ABS (2019) states: "The reason while the engine is still considered very sensitive to hull deflections is not because of the engine itself, but rather because of the discrepancy in stiffness between the engine structure and the double bottom structure aft of the engine, below the shafting."

B.2.2. Static Condition Vs. Running Condition: The Differences

The conditions which apply on a vessel at any given time, can be divided into two categories: The Static Conditions and the Running Conditions. Below, the difference between the two are going to be described:

- Static Condition: This term refers to situations where the vessel is in Dry Dock or anchored. The main characteristics of this condition are:
  - The shaft is stationary, not executing rotational movement, thus hydrodynamic lubrication is not present.
  - The propeller is stationary as well and does not operate in order to produce thrust. As a result, the propeller induced loads are equal to zero, with the weight of the propeller and the buoyancy being excluded. Moreover, no torsional vibrations are present.
  - The Main Engine is not running, in cold conditions.
  - Because of the fact the propulsion shaft is already installed, the clearance of each bearing is established. As a result, vertical movements of the bearings are acceptable, as long as they are between the limits which the clearances oppose.
- Running Condition: This term refers to the real, active condition with the vessel fully-functioning and in sea-going condition and focuses mainly on dynamic phenomena. The main characteristics of this condition are:

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- The shaft is on rotational movement phases, so Hydrodynamic lubrication is present, and an oil film is developed between the shaft and the bearings. This oil film slightly lifts the shaft above the lower half of the bushing.
- Tha main engine is functioning and in hot (running) condition. Because of the deflections the Main Engine suffers due to thermal deviations, the crankshaft bearings are subjected in alteration of their vertical offsets. Moreover, due to the functioning of the Main Engine, it is of great significance to take into consideration the vibrations which are produced.
- The propeller produces eccentric thrust, and as a result bending moment is applied on the propulsion shaft.
- If any misalignment between the shaft and the bearings is present, it will result in alteration of the single point support position which was assumed during the calculations phase.

B.2.3. Reaction Forces:

During the shaft alignment process, one of the most crucial parts is the calculation and verification of the reaction forces of the bearings. As previously stated in this study, the reaction forces of each bearing must be positive, which practically means that the bearings are in contact with the shaft at their bottom half of the geometry. Unloading of a bearing is equally significant as overloading it. The specific requirements concerning the acceptable limits for the reaction forces of the bearings depend on each Classification Society's regulations and will be thoroughly discussed in the next chapters of this study. The most common concept is the limitation of the pressure applied on the bearing (which correlates to the reaction force). In some cases, a minimum oil film thickness is utilized as a control parameter. Furthermore, each Classification Society has its own policy regarding the allowable deviation between the calculated reaction force and the reaction force measured on-board. This matter will also be described later in this Thesis. Now, the main principles concerning the reaction forces and their accepted values will be discussed.

According to L. Murawski (2005), the actual reaction forces during the vessel's operation in the intermediate shaft and the stern tube bearings should neither be too high, nor too small. The limitation regarding "too high reaction forces" focuses on avoiding overloading, while the limitation regarding "too small reaction forces" focuses on avoiding unloading of the bearings, which is almost equally risky regarding the shaft's reliability. Moreover, Murawski on his paper comments that, if a bearing is verified to have small reaction force in static condition, thorough investigation should be made in order to eliminate the possibility of damage or failure due to the influence lateral vibrations may have on the bearing during running condition. It is also important that all of the bearings are unvaryingly loaded. However, due to the fact that each bearing has its own dimensions, in order to compare the magnitude of a bearing's loading, the Sommerfeld Number is used. The formula, which is used to calculate Sommerfeld Number, as presented earlier, is:

$$\boldsymbol{S = \frac{R}{\eta * U} * (\frac{d}{c})^2}$$, where:

- S: The Sommerfeld number
- R: The bearing's loading unitary force
- η: The lubricating oil absolute viscosity
- U: The peripheral speed
- d: The diameter of shaft journal and
- c: The bearing slackness

A low loading condition is dangerous for a bearing, especially if it is an intermediate shaft bearing. The reason for this, lays on the fact, that during the vessel's operation, hull deformations which occur due to the weather or/and the loading condition, may cause small decreasing of the intermediate shaft's bearings reactions. Consequently, if these bearings are characterized by a low Sommerfeld Number, unloading of the bearings is an absolutely possible scenario, which should be avoided at all costs. Thus, "It is reasonable that the

Sommerfeld number of the intermediate shaft bearings are sometimes 30%-50% greater than other bearings." (Murawski, 2005)

When discussing reaction forces, attention should be drawn to the loading distribution of the aft stern tube as well. As previously stated, the aft stern tube bearing is one of the heaviest loaded bearings. That it because, amongst other reasons, it is relatively long compared to the other bearings, and it supports the weight of the propeller, absorbs its vibrations and is highly impacted from its behavior. Consequently, the loading of the aft stern tube bearing is significantly asymmetrical (because of the propeller's forces) and it is susceptible to part seizure (Murawski, 2005). A way of determining whether the aft stern tube bearing's condition in terms of loading is acceptable, is comparing the reaction forces in its forward and aft end. Another way could be the measurement of the relative deflection between the journal's line and the tube axis. In the last method, the deformations concerning the hull structure and the tube's axis shall be considered (Murawski, 2005).

Furthermore, the loads of the crankshaft due to the shaft line in terms of shear force, must be considered and verified to be inside acceptable limits. The maximum acceptable loads for the crankshaft are usually defined by the Main Engine's manufacturer. The bending stresses of the shaft line should also be calculated and compared to the acceptable limits. It is common that the pre-mentioned verifications should be completed for a series of vessel's service conditions, according to the Classification Society in which the owner of the vessel chooses to register it.

B.2.4. Deflection Curve:

At the latest published regulations regarding the Shaft Alignment issue from Classification Societies, the Deflection Curve has played a significant role. ABS (2019) precisely states that:

"The relative misalignment angle of the shaft inside the bearing is defined by the shaft deflection curvature, and the angle is measured from the theoretical zero alignment line, as established in shaft alignment calculation."

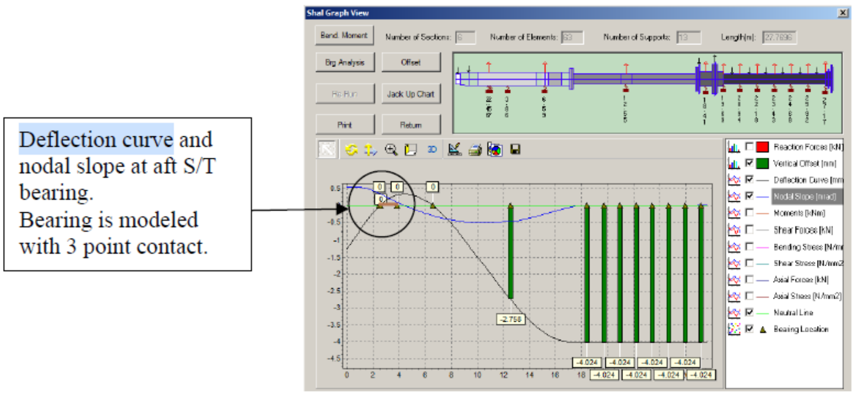

Figure 51: Deflection Curve for the purposes of determining the misalignment angle, ABS [12]

Almost all of the Classification Societies establish a higher limit regarding the misalignment of the shaft inside the stern tube bearing, and most commonly require that in cases where the limit is exceeded, slope boring or bearing inclination methods must be applied. More on the specific requirements will be discussed in the next chapters of this study.

For clarification reasons, explanations shall be given on the term "misalignment":

The stern tube bearing requires special attention during the design and calculation stage of the shaft alignment procedure. That it, because of its bigger length compared to the rest of the shaft's bearings, as well as due to

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the large cantilever loads exerted from the vessel's propeller. Inside the aft stern tube bearing, the shaft presents an inclination, which is named "angular misalignment". The support points which will be used in order to study the aft stern tube bearing are a significant issue, and the different Classification Societies introduce different requirements, which will be thoroughly discussed in the next chapters. However, it is worth mentioning that the limitations regarding the maximum misalignment angle correlate to its dimensional parameters, such as its length. Practically, the misalignment angle shall not exceed the ratio of the radial bearing clearance to the bearings' length. Furthermore, by the term "relative misalignment slope", as ABS (2019) states, the following is described: "The difference between the shaft angle (defined by two points on the shaft's centerline – located above the two end points of the bearing), and the slope of the bearing".

The difference between the methods of bearing inclination and slope boring will also be discussed in the next chapters. Another parameter which will be discussed, is the "dilemma" between the single or double slope design, as in some cases, the two-slope boring design is preferred. In the figure below, a two-slope design is presented.

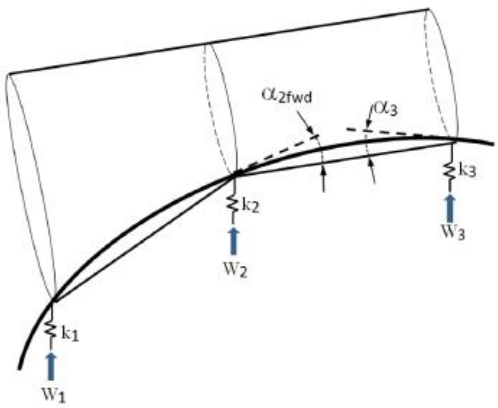

Figure 52: A two slope design stern tube bearing

1.SHAFT ALIGNMENT DESIGN

1.1. Introduction

As previously mentioned, the shaft alignment process, consists of the following stages:

- The design & analysis stage,
- The alignment procedure stage, and
- The measurements and verification stage.

According to ABS (2019) the minimum requirements in terms of parameters which need to be defined during the design stage and subsequently verified by measurements on board, are:

- The vertical offsets of the bearings,
- The reaction forces of the bearings,
- The Crankshaft's deflections,
- The condition of the contact between the bearings and the shaft (misalignment angle etc.), and
- The Gear Mesh misalignment.

At this point, it would be useful to remind the main difference between a 2-Stroke Main Engine installation and a 4-Stroke Main Engine Installation, in terms of the effect it has on the shaft alignment procedure and calculations: In 4-Stroke Main Engine installations, it is mostly preferred for the propulsion system to be equipped with a Reduction Gear, in order to regulate the RPMs which end up to the propeller, due to the fact that this type of engines are characterized by higher RPM values. On the other hand, in the case of 2-Stroke

Main Engines where the RPMs are lower, it is usually not needed. In the table below (BV, 2015), the main types of propulsion types are correlated to the shaft system they usually tend to be combined with:

Table 2:Propulsion types and shaft alignment systems [Source; BV, 2015]

Propulsion Type:	Prime Mover:	Alignment System:
Direct Drive Installation	Low-speed Diesel/gas engine	From propeller to crankshaft
	Electric motor	From propeller to rotor shaft
Geared Drive installation	Medium-speed Diesel/gas installation	From propeller to main gearbox output shaft
	Steam/gas turbine	
	Electric Motor	

The main scope of the shaft alignment procedure is the optimization of the parameters of interest, under all declared operating conditions of the vessel.

Regarding clarification reasons, the definition of the term "Offsets" as given by BV (2015), which is going to be widely used from now on, is given:

"Offsets are the initial Vertical and horizontal positions of a bearing fixed by the alignment procedure and modified by the flexibility of the structure, the loading deformations and the thermal expansion". (BV, 2015)

Moreover, BV states that, the equilibrium of the beam elements which are subjected to external forces and supported by bearings, are investigated in three dimensions, which makes the horizontal and vertical offsets coupled. DNV (2021) on the other hand, states that, while vertical analysis is mandatory, horizontal alignment investigation in conducted upon request, depending on the specific case.

BV also states, that if required as needed by the Classification Society, a whirling investigation should also take place.

At this point, the input data and the parameters which one should take into consideration regarding the shaft alignment procedure will be presented, as each Classification Society requires.

ABS (2019)

As far as regarding the required input parameters, ABS (2019) demands that in order for the calculations to be verified, they should at minimum include:

- The influence coefficient matrix.
- The reactions of the bearings and allowable loads on all bearings.
- The clearance on the stern tube bearing.
- The deflection curvature.
- The requirements of the stern tube bearing slope.
- The angular inclination on the main gear wheel.
- The Shear Forces and Bending Moments.

Furthermore, the calculations shall be conducted and verified for the following conditions of the vessel:

1. Vessel in dry dock with cold engine and gear box.
2. Vessel afloat with cold and hot engine and gearbox.

According to ABS, the general requirements which the shaft alignment calculation stage and verification stage shall fulfill, have as follows:

I. Bearing loads under all operating conditions are within the acceptable limits specified by the bearing manufacturer.

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II. Bearing reactions are always positive, i.e. the bearings support the shaft.
III. Shear forces and bending moments on propulsion equipment and at the crankshaft flange are within the limits specified by the manufacturer.
IV. The designed relative misalignment slope between the shaft and the aft stern tube bearing is to be positive, and not to exceed $0.3*10^{-3}$rad.

Additionally, the analysis conducted should include the following:

1. For geared systems, the calculated misalignment between main gear and pinion is to be less than $0.1*10^{-3}$ rad.
2. The designed relative misalignment slope between the shaft and the aft stern tube bearing is to be positive.
3. The stern tube bearing fitting calculation based on the actual interference fit tolerances, including fit-pressure and push-in distance.
4. A clearance calculation on aft and forward stern tube bearings with alignment model showing only the propeller shaft supported on two stern tube bearings. Installations with no forward stern tube bearing may be subject to specific requirements.
5. Sag and gap values and the locations of the temporary supports.
6. Jack-up locations for the verification of bearing reactions.

According to ABS (2019), the following have to be submitted to the responsible Engineers working for ABS in order for the Shaft Alignment procedure to be verified:

• Shaft alignment model
• Scope of submitted calculation
• Analysis results
• Shaft alignment procedure
• Run-in procedure

Moreover, it is worth highlighting, that ABS (2022) in Enhanced Shaft Alignment states that the calculations regarding Shaft Alignment should be made in the following vessel's conditions:

1. Dry dock or waterborne at very light ballast at cold engine or gearbox and with propeller partially immersed
2. Fully ballasted vessel with cold engine or gear box
3. Fully ballasted vessel with hot engine or gear box
4. Fully laden vessel with cold engine or gearbox, and
5. Fully laden vessel with hot engine or gearbox

Note: Conditions 2 to 5 should be studied considering the propeller fully immersed.

For the purposes of optimization, ABS (2022) in Enhanced Shaft Alignment Guide, requires that the procedure of "Shaft Alignment Calculation Optimization" is conducted. During this procedure, the optimal set of bearing offsets is to be decided, by taking into consideration several operating conditions of the vessel. ABS (2022) states that, for these purposes, the following additional conditions should be studied:

1. Dynamic conditions including propeller loads due to influence from wakefield or bearing fluid film onto shaft (i.e., utilizing fluid structure interaction type of calculations), or
2. As deemed appropriate by ABS with respect to the level of engineering assumptions, analysis and calculation simulation

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Design vs. Review; The difference as ABS entrepreneurs it:

At first, it would be useful to establish the differences between the terminology used in this chapter's title. As ABS (2019) declares:

- Review Process: According to ABS (2019), the term "Review Process" refers to the stage of the shaft alignment procedure, where the verification of the soundness of the design developed takes place. For this reason, a number of requirements should be met, which ae defined in ABS Rules regarding the Shaft Alignment. These rules are established in order to ensure the safety and the prosperity of the designed arrangement, which in this case is the Shaft System and its propulsion, and they are correlated to the performance and the strength parameters of the system. The main scope of the entire procedure is to ensure safety in terms of environment, life at sea, and equipment.
- Design Procedure: According to ABS (2019), the Design phase is a lot more complex (compared to the Design Stage), as the engineers responsible must ensure that a set of parameters comply with the requirements of the Classification Society. The main set of parameters is: The vertical offset of the bearings, their longitudinal position and the slope-boring angle between the shaft and the aft stern tube bearing. The parameters of interest (in their wide form), however, correlate with each other, and in some cases are defined as coupled, thus making the optimization of the system a complicated procedure. Furthermore, the design shall also verify the alignment complies with the shaft geometry and its material's properties, the constraints imposed by the vessel's design, and other qualifications forced by the propulsion shafting interaction with the surrounding systems and machinery.

According to ABS (2019), the Design Stage of the Shaft Alignment aims to address the following:

- Acceptable bearing reaction loads,
- Acceptable bearing misalignment limits,
- Uniform load distribution in all bearings,
- Shaft strength limits,
- Satisfactory crankshaft deflections,
- Acceptable gear contact condition,
- Satisfactory coupling bolt strength,
- Acceptable misalignment tolerances for clutches and flexible couplings.

According to ABS (2019), a plan review is conducted by the Classification Society, prior and after the installation of the system of interest, in order to verify compliance with the relative Rules.

The engineers responsible, should possess the following in order for the review stage to take place:

- Shaft Alignment Model,
- Scope of submitted calculation,
- Analysis results,
- Shaft alignment procedure,
- Run-in procedure.

Moreover, the following are needed in terms of precise Plan and Particular knowledge:

For the Propulsion Shafting:

- Shafting arrangement,
- Rated power of main engine and shaft's RPMs,
- Thrust shafts, line shafts, tube shafts and tail shafts, as applicable,
- Couplings-integral, demountable, keyed or shrink-fit- coupling bolts and keys,

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- Line shaft bearing details,
- Stern tube bearings detailed drawings,
- Main Engine, and if applicable, gearbox bearings,
- Allowable bearing loads,
- Stern tube seal arrangements,
- If applicable, power take-off to shaft generators, propulsion boosters, or similar equipment, rated 100 kW (135 hp) and over,
- Material properties of the shafts and the bearings (modulus of elasticity and density),
- Propeller mass and material properties.

For the Cardan Shafts:

- Dimensions of all torque-transmitting components and their material properties,
- Rated power of main engine and shaft rpm,
- Engineering analysis

For the Calculations and the Procedures:

- Propulsion shaft alignment calculations,
- Detailed shaft alignment procedure,
- Run-in procedure for alignment sensitive installations.

For the Diesel Engine:

- Crankshaft Equivalent Model,
- Location and mass of the following equipment, when applicable: Flywheel, chain drive, tightening load, cam-drive gear, torsional vibration damper,
- Detailed drawings, location and material properties of the main engine bearings,
- Allowable bearing loads.

For the Reduction Gear:

- Detailed drawings of ear shafts and main gears with material properties,
- Mass and location of the gears,
- Location and material properties of the gear shaft bearings,
- Allowable bearing loads.

DNV (2021)

DNV (2021) requires the following as minimum input data of the shaft alignment calculations:

- Propulsion plant particulars, e.g. rated power of main engine and propeller shaft rpm
- Equipment list, i.e. manufacturer and type designation of prime mover, reduction gear (if applicable) and bearings
- Geometry data of shafts, couplings and bearings, including reference to relevant drawings. For direct coupled plants, the crankshaft model shall be according to the engine designer's guidelines
- Propeller data
- Bearing clearances

The calculations shall be conducted for the following conditions:

1. Alignment condition (during the erection of shafting),
2. Cold, static, afloat, fully submerged propeller,

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3. Hot, static, afloat, fully submerged propeller,
4. Hot, running, with hydrodynamic propeller loads.

While for geared shafting systems:

1. Running conditions as required to verify gear acceptance criteria,
2. All relevant combinations of prime mover operation,
3. Horizontal alignment is upon request.

Furthermore, the influence parameters which shall be taken into consideration, are:

1. Buoyancy of propeller,
2. Thermal rise of machinery components (including rise caused by heated tanks in double bottom and other possible heat sources),
3. Gear loads (horizontal and vertical forces and bending moments),
4. Angular working position in gear bearings for gears sensitive to alignment,
5. Bearing wear (for bearings with high wear acceptance e.g. bearings with water or grease lubrication),
6. Bearing stiffness (If substantiated by knowledge or evaluation, otherwise infinite),
7. Hull and structure deflections,
8. Hydrodynamic propeller loads

At this point, some terms mentioned above will be clarified:

- "Sensitive to alignment gears" refers to arrangements such as the ones with large face width or with more than one pinion driving the output wheel. In these cases, even tiny alterations in the bearings' offsets may lead to significant changes on the gear force load distribution.
- The hull deflections of the vessel are influenced by the vessel's design, draught, trim, aft peak filling etc. The precise calculations and methodology DNV suggests will be discussed further in the next chapters.

The results to be submitted regarding the shaft alignment calculations, have as follows:

1. Bearing offsets from the defined reference line,
2. Calculated bearing reaction loads and pressures,
3. Bearing reaction influence numbers,
4. Graphical and tabular presentation of the shaft deflections with respect to the defined reference line,
5. Graphical and tabular presentation of the shaft bending stresses as a result of the alignment,
6. Nominal relative slope between shaft and bearing centerlines in aft most propeller shaft bearing, and if applicable, details of proposed slope bore,
7. Results from aft stern tube bearing lubrication criteria,
8. In cases where a consideration of transient operation is required (upon request), resulting local bearing pressures shall be reported and evaluated by means of a detailed contact analysis (Finite Element Analysis),
9. A shaft alignment procedure with verification method and data tolerances (e.g. aft bearing slope and geometry, reference line, stern tube bearing offsets, calculated sag & gap values and jacking loads including jack correction factors). The procedure shall clearly state at which vessel condition the alignment verification shall be carried out (cold or hot, submersion of propeller etc.) The positions of jacks and temporary supports shall be specified. The procedure shall be possible to use again when in service.

The acceptance criteria of the calculation and analysis process, have as follows:

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- Acceptance criteria defined by manufacture of the prime mover, e.g. limits for bearing loads, bending moment and shear force at flange,
- Acceptance criteria defined by the manufacturer of the reduction gear, e.g. limits for output shaft bearing loads and load distribution between bearings,
- Bearing load limits as defined by bearing manufacturer,
- Zero or very low bearing loads are only acceptable if these have no adverse influence on whirling vibration,
- Tolerances for sag and gap less than 5/100 mm are not accepted.

The acceptance criteria regarding the aft most stern tube bearing, are:

- In hot static and hot running conditions the relative nominal slope between shaft and aft most propeller shaft bearing should not exceed $3*10^{-4}$rad and 50% of minimum diametrical bearing clearance divided by the bearing length, whichever is less. This criterion is only applicable in single slope or no slope designs. A white metal aft stern tube bearing which is either double sloped or has a journal diameter of 500mm or greater, shall fulfill other requirements, which will be discussed later.

LR (2023)

According to LR, the shaft alignment calculations should take into account the following:

- Thermal displacements of the bearings between cold static and hot dynamic machinery conditions.
- Buoyancy effect of the propeller immersion due to the ship's operating draughts.
- Effect on predicted hull deformations over the range of the ship's operating draughts, where known.
- Effect of filling the aft peak ballast tank upon the bearing loads, where known.
- Gear forces, where appropriate, die to prime-mover engagement on multiple input/single output installations. For multiple input systems, consideration is to be given to each possible combination of inputs.
- Propeller offset thrust effects.
- Maximum allowed bearing wear-down, for water or greased-lubricated stern tube bearings, and its effect on the bearing loads.

Consequently, the shaft alignment calculations are to state:

1. Expected bearing loads at light and normal ballast, fully loaded and any other draughts deemed to be part of the ship's operating profile, for the machinery in cold and hot, static and dynamic conditions.
2. Bearing influence coefficients and the deflection, slope, bending moment and shear force along the shaftline.
3. Details on propeller offset thrust.
4. Details of proposed slope-bore of the aftermost stern tube bearing, where applicable.
5. Manufacturer's special limits for bending moment and shear force at the shaft couplings of the gearbox/prime movers.
6. Estimated bearing wear-down rates for water or grease-lubricated stern tube bearings.
7. Expected hull deformation effects and their origin, viz. whether finite element calculations or measured results from sister or similar ships have been used.
8. Anticipated thermal rise of prime movers and gearing units between cold static and hot running conditions.
9. Manufacturer's allowable bearing loads.

The shaft alignment procedure, shall be available for review, containing at minimum:

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- Expected bearing loads at light and normal ballast, fully loaded and other draughts deemed to be part of the ship's operating profile, for the machinery in cold and hot, static and dynamic conditions,
- Maximum permissible loads for the proposed bearing designs,
- Design bearing offsets from the straight line,
- Design gaps and sags,
- Location and loads of the temporary shaft supports,
- Expected relative slope of the shaft and the bearing in the aftermost stern tube bearing,
- Details of slope bore of the aftermost stern tube bearing, where applied,
- Proposed bearing load measurement technique and its estimated accuracy,
- Jack correction factors for each bearing where the bearing load is measured using a specified jacking technique,
- Proposed shaft alignment acceptance criteria, including the tolerances, and
- Flexible coupling alignment criteria.

BV (2015)

BV (2015) demands that the following are taken into consideration:

- Deformations of the ship's structure with respect to the declared loading conditions of the ship: light ship, ballast, full load etc.
- Aft hull structure flexibility matrix calculated in way of each supporting point.
- Propeller hydrodynamic efforts (forces and moments) in vertical and transverse directions in straight course.
- Thermal expansion of supports (seat, sleeve, antifriction material).
- Deformation of prime mover or gearbox foundation: In case of low-speed engine, pre-sag of main bearings in to be considered.

In cases where more precise calculations are desired or deemed as necessary, the following should be accounted for:

- Deformation of the ship structure with respect to the sea swell.
- Propeller hydrodynamic efforts in turning conditions, including rudder effects.
- Temperature effects lubrication, by a calculation of global and local dissipation.

The results to be submitted, should be:

Table 3: Results to be submitted for shaft calculations analysis, BV (2015)

In static Condition:	In running Condition:
Reaction distribution between shaft bearings	
Reaction distribution along effective length of aft bush bearing	
Shaft location inside bearings	
Static contact pressure on anti-friction material	Oil film pressure
Squeezing of anti-friction material (for information)	Oil film thickness

The Documentation which is required to be submitted for approval in order for the ESA Notation to be obtained by the parties interested, a calculation report is demanded. The calculation report should present:

- General Description of the Calculation Method,
- Assumptions,
- List of Investigated Calculation Conditions,
- Input Parameters,

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- Detailed Results,
- Conclusions,
- Shaft Line Model,
- Hull Flexibility Matrix,
- Hull Relative Deformations,
- View of Complete Ship Finite Element Model,
- Detailed Views of Finite Element Model of Aft Part of Ship Structure,
- Detailed Alignment Procedure.

ClassNK (2006)

According to ClassNK (2006), the calculation sheets which will be submitted to the Classification Society for verification and approval regarding Shaft Alignment, shall include:

- Diameter (inner and outer) and length of shafts,
- Length of bearings,
- Concentrated loads and loading points,
- Support points,
- Bearing offsets from reference line,
- Reaction influence numbers,
- Bending moments and bending stresses,
- Bearing loads and nominal nearing pressure,
- Relative inclination of the propeller shaft and aft most stern tube bearing or maximum bearing pressure in the aft most stern tube bearing,
- Deflection curves for the shafting,
- Sags and gaps between coupling flanges,
- Procedures for measuring bearing loads (in cases where such measurement is required).

As previously stated, the calculations should be conducted for the following conditions of the vessel: Light Draught Condition (Hot condition), Full Draught Condition (Hot Condition) and Light Draught Condition (Cold Condition).

1.2. Numerical Shaft Alignment Calculations

Firstly, a preliminary calculation with all of the bearings in straight-line shall take place, in order for the influence coefficients and the shaft's deformation to be approached. It is also crucial, to determine the hull deflections of the vessel at this point, so that they can be used in the design stage for the scope of optimizing the offsets of the bearings and the shafting system's behavior under all loading conditions and possible drafts. That is, because the installation of the shafting system usually takes place when the vessel is in relatively light draft conditions, where the hull deflections do not significantly affect the performance. However, when the vessel begins its operation life cycle, when its range of operational conditions is quite wide, the load distribution amongst the shaft's bearing changes, and precautions shall be taken in order to ensure that under these conditions as well, the requirements are fulfilled.

There is always the choice of not calculating precisely the hull deflections and relying entirely on empirical data or data collected from similar applications to other vessels with identical propulsion systems. However, this approach entails numerous dangers, which according to ABS (2019) are correlated to:

- The hogging and sagging of the Main Engine's bedplate, which is capable of inducing undesirable alterations in the main engine's bearings, thus leading to unloaded main engine bearings or crankshaft damage.

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- Distortion of the sighting line, which is already established in dry dock condition, leads to changes in the distribution of the bearings' loads or even in unloading some of them.

ABS: Hull Deformation Calculation

For these reasons, ABS (2019) requires, that when a detailed calculation procedure regarding the hull deflections is not conducted, "additional bearing load verification through jack-up measurements at selected vessel service drafts and with the engine in a hot static condition" shall take place.

ABS has developed software which serves the purpose of confidently estimating the hull deformations of a vessel, by utilizing data collected by a number of vessels of different types and sizes in order to calibrate the results. In order to produce the results, the Software uses a combination of analytically obtained and measured deformation values.

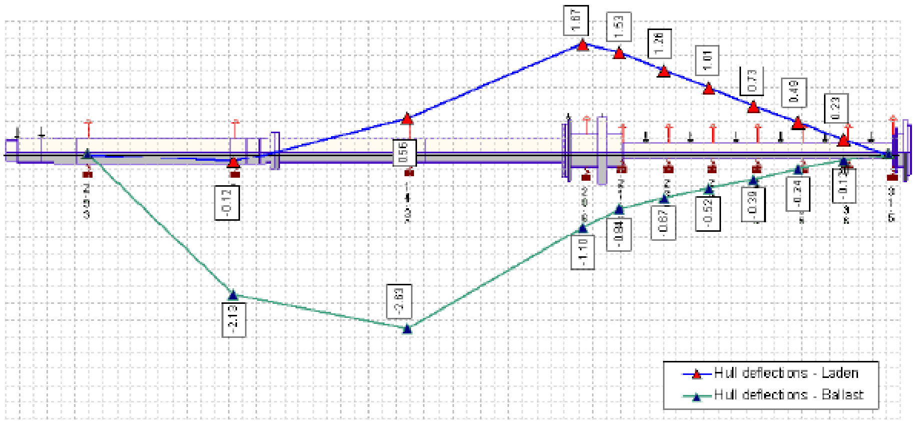

Figure 53: Changes on the bearings' offsets under ballast and laden condition, ABS [12]

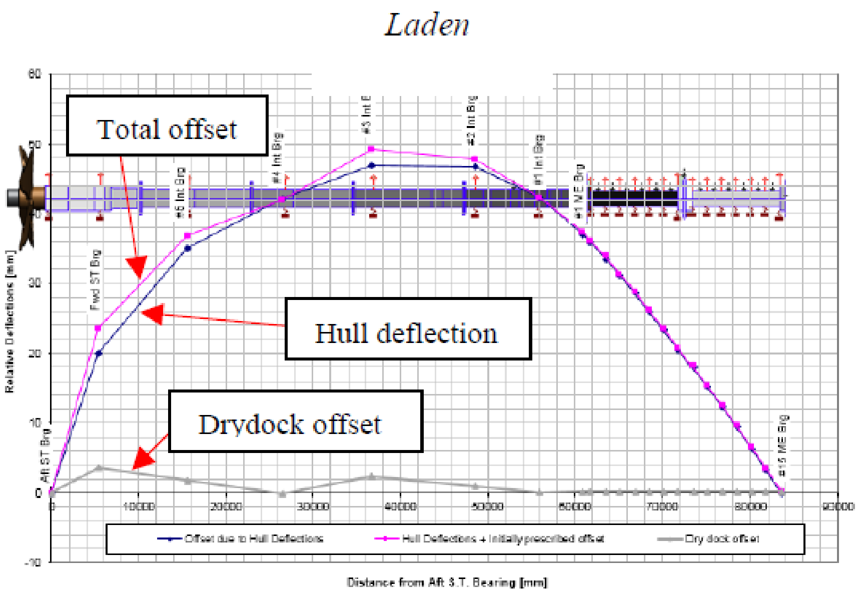

Figure 54: An example of the hull deflections, total offset and dry dock offset diagrams for Laden Condition, ABS [12]

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ABS (2019) proposes, that the total offsets of the bearings shall be calculated by summarizing the Dry Dock offsets of the bearings and the estimated hull deflections for the specific loading condition of the vessel.

The Numerical/Analytical approach methods ABS suggests, are two: A Finite Element Analysis (3-D) or an application of the 1-D Beam Theory. While the aft part of the vessel is the one of interest in calculations regarding shaft deflections, it is more beneficial and common to produce a FEA Model of the whole vessel for the puprose of dynamic loading analysis, and then utilizing the aft stern part only for hull deflection calculations and investigation, as the production of a 3-D FEA is time consuming and expensive. As long as the hull deformations are of interest, the engine room, the engine and the shafting must be approached with high accuracy during the process of constructing the 3-D model of the ship. However, if the information regarding the sectional modulus inertia and shear area is provided, a 1-D Beam theory approach could be accurate as well. Nonetheless, because of the weakness the 1-D Beam model may encounter because of the high stiffness of the stern tube structure, a calibration by utilizing data from the 3-D FEA Model may be required.

Another method which could be used in order to calculate the hull deflections of a vessel, is by measuring them explicitly on the aft stern part of the vessel. The main idea, is to measure the bearings' offsets at two different loading conditions and then subtract them. Then, the result will indicate the hull deflections of the vessel. However, due to the fact that the measurement of the bearings' offsets is not feasible at this specific time, because as ABS (2019) states, "the optical sighting-through line is obstructed by the already-installed shafting". Nevertheless, reverse calculation could be conducted by utilizing the bending of the shafting, in order to obtain the offsets of the bearings', thus enabling the engineer to gain the information needed in the first place: The hull deflections of the vessel. As ABS (2019) requires, in order to accurately define the hull deflections, measurements shall be conducted in all of the following loading conditions:

- Dry-Dock,
- Ligh draft immediately after launching and before any bearing adjustment,
- Ligh draft after bearing adjustment,
- Ballast, and
- Fully Laden.

The bending of the shaft shall be calculated using strain gauges in several different positions for reasons of accuracy. The bending curvature of the shaft shall also be known. ABS (2019) suggests that for shafting systems with one intermediate bearing, at minimum three strain gauges locations are needed.

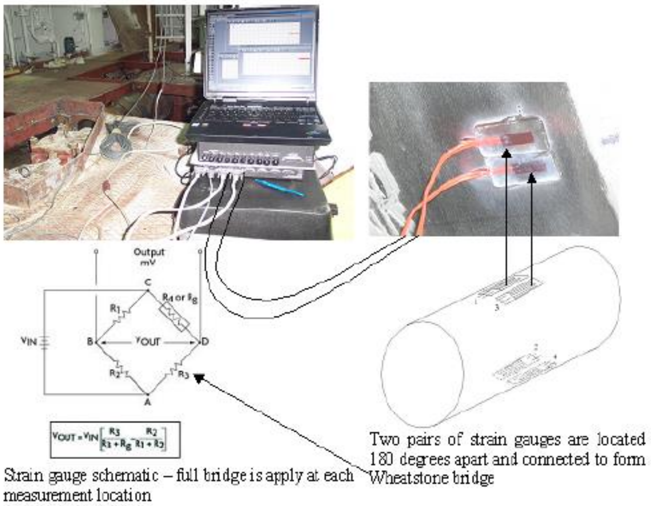

Figure 55: Schematic of a Strain Gauge measurement, ABS [12]

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BV: Calculation of Hull Deformations, Shaftline Stiffness and Hull Flexibility Matrices

In order for the Shaft Alignment analysis and design stage to be feasible, some Preliminary Calculations are required. According to BV (2015), these calculations shall serve the purposes of Hull Deformation calculations and Hull Flexibility Matrix calculations.

For the purpose of Hull Deformations, BV (2015) demands that a Finite Element Model of the whole vessel is produced (3-D). This model shall be utilized in order to gain information for several loading conditions of the vessels and their impact on hull deformations, as well as for calculating the impact of sea swell in terms of Hull Deflections. A view of the ship for every loading condition studied through FEA models of the ship shall be submitted in the final report. The results obtained from the pre-described process should be used in order to accurately determine the relative displacements of the line-shafting supports, as a function of the vessel's operating conditions.

BV (2015) requires calculations regarding the relative deformations between light ship condition and "any relevant operating conditions (ballast and full load in particular)" to be conducted, so that the desirable bearing offsets are obtained. BV specifically states that the reference line for the design process shall be defined by the aftermost support of the aft stern tube bearing and the forwardmost support of the shaft line.

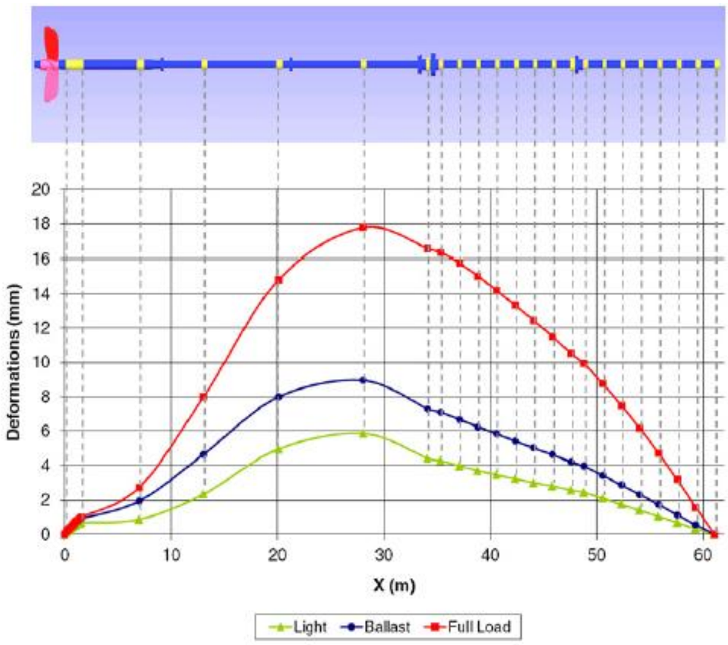

Figure 56: Hull Deformations for three different operating conditions, BV [23]

As far as the deflections of the vessel's hull due to sea swell are of interest, BV makes clear that sea waves originated from local winds shall not be taken into consideration. More precisely, BV (2015) suggests:

- Sea swell characteristics to be defined for maximizing the double bottom relative deformations in way of shaft line supports.
- Wave parameters (direction, height; H and wavelength; λ) are to be chosen in order for the couple (H, λ) to be physically realistic (i.e. the wave should not break with the chosen values), and only head sea conditions are to be investigated.
- Loading due to wave defined by (H, λ) should be applied considering two sinusoidal equivalent profiles: Maximum pressure (wave crest) and low pressure (wave trough) located in the way of the aft peak.

Each couple of relative displacements of shaft supporting points calculated in the previously mentioned methodology, should be taken into consideration as additional bearing offsets during the Alignment Calculations through the Hertz theory method, and they should be given at the same support points where the displacements due to hull deformations are given.

The Hull Flexibility matrix is calculated by utilizing a FEA model of the aft part of the vessel. The precise definition of the Hull Flexibility Matrix is given by ABS (2022) as follows:

"In way of supports, the displacements in transverse and vertical directions induced by a transverse or vertical unit force applied on one support determine a line of flexibility matrix".

The form of the flexibility matrix has as presented below:

$$\begin{bmatrix} d_{1,T1} & d_{1,V1} & \cdots & d_{1,Tj} & d_{1,Vj} & \cdots & d_{1,Tn} & d_{1,Vn} \\ \vdots & \vdots & \ddots & \vdots & \vdots & \ddots & \vdots & \vdots \\ d_{i,T1} & d_{i,V1} & \cdots & d_{i,Tj} & d_{i,Vj} & \cdots & d_{i,Vn} & d_{i,Vn} \\ \vdots & \vdots & \ddots & \vdots & \vdots & \ddots & \vdots & \vdots \\ d_{2n,T1} & d_{2n,V1} & \cdots & d_{2n,Tj} & d_{2n,Vj} & \cdots & d_{2n,Tn} & d_{2n,Vn} \end{bmatrix}$$

Figure 57: Typical form of a Hull Flexibility Matrix, BV [23]

Where:

- d: Displacement in transverse or vertical direction,
- n: Total number of supporting points,
- i: Row index for the load case reference (i ∈ $[1, 2n]$).
- J: Index for the considered support (j ∈ $[1, n]$). For each support j, two columns are built for transverse and vertical displacements (column indexes T_j and V_j)

Each term of the hull flexibility matrix is noted as follows:

$$d_{i,T_j}: Transverse\ displacement\ of\ support\ j\ due\ to\ load\ case\ i.$$

$$d_{i,V_j}: Vertical\ displacement\ of\ support\ j\ due\ to\ load\ case\ i.$$

In order for the model which is going to be utilized for the development of the Hull Flexibility Matrix to be constructed, a FEA Model of the aft part of the ship is required. More precisely, the model shall extend "from the aft end up to the forward watertight bulkhead of the engine room", and the model could be developed for the purposes of hull flexibility matrix only, or it can be extracted from a complete 3-D FEA model of the whole vessel. The exact specification regarding the construction of the FEA model to be used in the calculations of the Hull Flexibility Matrix according to BV (2015), have as follows:

A) Nodes should be restrained in displacement and rotation in the way of the forward transverse section.
B) Longitudinal secondary stiffeners are to be modeled in order to ensure a sufficiently refined mesh of the ship structure. As a consequence, standard size of the finite elements used is to be based on the secondary stiffener spacing.
C) The structural model should be built on the basis of the following criteria:
    - Webs of primary members are to be modeled with at least three elements on their height
    - Plating between two primary supporting members is to be modeled with at least two element stripes

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- The ratio between the longer side and the shorter side of the elements is to be less than three in the areas expected to be highly stressed
- Holes for the passage of ordinary stiffeners may be disregarded.

D) Cast part of bossing as well as forward stern tube bush steelwork should be modeled with solid elements (8 nodes bricks), as shown in the figure below

E) Longitudinal position of the equivalent supporting points is to be exactly the same on the line shafting and on the structure.

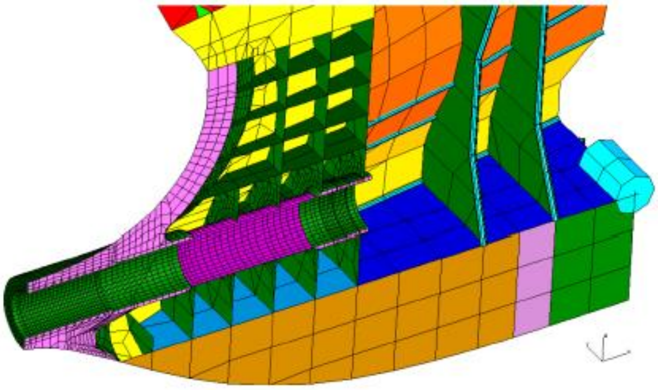

Figure 58: Stern bossing details on a FEA 3-D model of a vessel, BV [23]

As long as the Line Shafting stiffness matrix is of interest, BV (2015) requests that it is calculated, and that the influence coefficients are computed by utilizing it. The influence coefficients in BV's case, are to be calculated "for both transverse and vertical variations of the reactions on the supporting points when a unit displacement is successfully applied to each point in vertical and transverse directions".

Regarding the Line Shafting stiffness matrix, BV (2015) demands that it is "computed, and reduced, if necessary, in way of supporting points, for vertical and transverse directions, with suitable calculation method which should be specified in the submitted report". It is important to note, that the supporting points taken into consideration during the calculation of the Line Shafting stiffness matrix shall be the same supporting points considered in the Hull Flexibility matrix.

At this point, a remarkable difference shall be highlighted: Opposingly to ABS's (2019) suggestions, BV requests that the analysis and investigation should be conducted in both vertical and transverse directions.

ClassNK: Measurements and Hull Deflection Predictions

The accurate calculation of the Hull Deflections of the vessel is an issue of high interest, as it directly affects the functionality of the shafting mechanism, and consequently, the whole vessel as well. According to ClassNK (2006), for these reasons, on-board calculations on specific locations (as it will be later discussed) regarding the Hull Deformations of the vessel are necessary. However, if a Finite Element Analysis is feasible and proven to provide accurate predictions, it is considered to be a satisfactory substitute of the measurement procedure. It is important to note, that ClassNK composed the regulations and Guidelines regarding Shaft Alignment in 2006, therefore a possibility exists that at the time, FEA method was not as established, trusted, developed and validated as it is today. In conclusion, ClassNK (2006) proposes two equally accepted methods regarding Hull Deflections Calculations and Predictions: Either measuring the deflections in specific locations on-board or estimating them by utilizing a FEA Model.

Measurement of Hull Deflection:

While measuring the Hull Deflections of a vessel, an issue arises: The locations where the measurement is feasible, are limited to the red dots as shown in the figure below, because of several restrictions which occur

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due to the structure of the hull, the space arrangement, the engine and the shafting line. What this practically means, is that the locations suitable for measurements are limited to the spaces: Of the tank top beneath the shafting line, from the aft end of the engine to the fore stern tube bearing and the space alongside the engine.

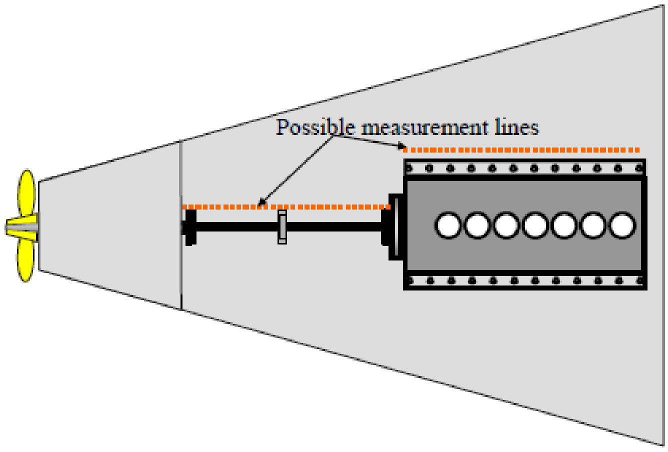

Figure 59: Possible Measurement Lines for Hull Deflection measurements, ClassNK [24]

The items of interest during the measurement procedure shall be the relative deflection of the tank top from one draft condition to another, with respect to a reference line connecting the two ends of the measured line. Moreover, ClassNK proposes a method of measurement, where the relative deformations are measured at one end of the measurement line from one draft condition to another with respect to a reference point which is produced by a laser beam from the other end of the measurement line. The two pre described methods are depicted in the figure below:

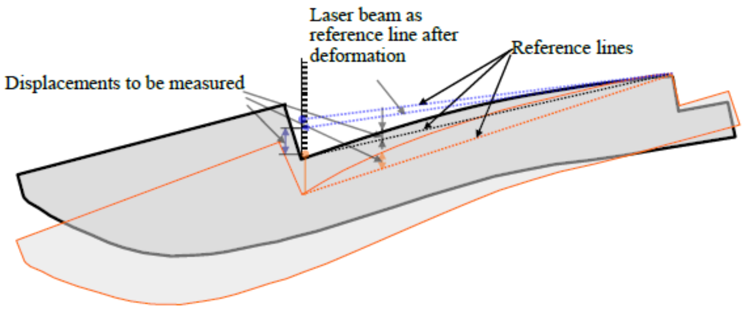

Figure 60: Depiction of the displacements which need to be measured. In blue; Laser Beam Reference Point method, ClassNK [24]

The method which uses a Laser Beam originated Point as e measurement line reference, has some advantages and disadvantages as well. The main benefits of applying this method have to do with the fact that it provides a more detailed and total understanding of the entire profile of the displacements which take place along the entire length of the measurement line. However, on the other hand, more precise and careful measurements shall take place, as the displacements which occur could be characterized by an extremely small value, and moreover, the measurements of the middle points can be difficult.

The exact guidelines regarding the apparatus used and the measurement method itself, will be developed in the next Chapters of this study.

Estimations on Hull Deflections by utilizing FEA Models:

As previously stated, ClassNK (2006) refers to the methods of Measuring the Hull Deflections or Predicting them via FEA Models equal, only if the FEA method is approved to be of high accuracy and acceptable. As at

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the time of the composition of the Guidelines on Shaft Alignment from ClassNK in 2006, the FEA process in general, and more precisely the isolation of the aft part of the vessel, were not fully evolved in the industry and as much investigated as they are these days, the part of the Guidelines which refers to the FEA approach almost entirely focuses on the Boundary Conditions, the modelling conditions and the results extracted from a case study which was conducted by ClassNK (2006). However, ClassNK (2006) does make a reference on the Dynamic Components of Hull Deflections related to the vessel's motions and the Hull Deflections induced by the Thrust.

In conclusion, it is estimated, that during a case study investigation which was escorted by relevant measurements, the Hull deflections caused by the Ship's motions during a typhoon phenomenon, reached a maximum of 0.3mm in the shafting portion of the measurement line and a maximum of 0.2mm in the engine side of the measurement line. These displacements are equal to half the displacement caused by a 10m increase in the vessel's draught (which is approximately equal to 0.6mm). ClassNK (2006) bases on these measurements, states that the dynamic components of Hull Deflections play an important role in the shaft alignment process.

As far as the Thrust induced Hull deflections are of interest, the measured deformations reached a maximum value of 0.2mm along the entire span from the propeller to the aft most part of the engine. Consequently, it is advised for the Thrust induced Hull Deformations to be ignored during the design stage.

According to Lloyd's Register (2023), for main propulsion installations, the shaft alignment design must verify that under all circumstances regarding the vessel's loading condition and machinery operation, the distribution of the bearing's loads must function in a way that eliminates as much as possible the effects of hull deflections, thus optimizing the load distribution amongst the bearings.

DNV: Hull Deflection Calculations with FEM

DNV (2021) proposes that hull deflections shall be investigating by utilizing FEM Models as well, but they should be investigated upon request from the Classification Society, during the process of receiving the Enhanced Shaft Alignment; Shaft Align (2) notation process.

1.3. Design Stage

During the Design Stage of the Shaft Alignment, the shaft is usually considered as a continuous beam supported to specific points which represent the locations of the shaft's bearings. Consequently, the bearings' reactions, the shear stress and the bending moments which each section of the shaft suffers are calculated, and later verified to be in compliance with the Class of choice regulations. However, while executing these calculations, there is a significant number of parameters to be considered and which, if not taken under consideration, may lead to significant inconsistencies, errors, and consequently damages in the propulsion shafting. Each Classification Society has established its own regulations, and there is a possibility that some specific parameters, methodologies, proposed limits or sequence of steps differs from one Classification Society to another.

ABS indicates, that the following topics should be addressed carefully on the Design phase of a Shaft Alignment Procedure:

- Equivalent model of the propulsion system,
- Application of static and dynamic loads,
- Slope boring design,
- Consequence of the intermediate shaft bearing offset adjustment,
- Alignment design with no forward stern tube bearing,
- Main engine bedplate pre-sagging application,

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- Sag and gap procedures,
- Bearing clearance,
- Bearing elasticity,
- Bearing material,
- Gear meshes misalignment.

ABS (2019) imposes the following in relation to the Design Stage of the shaft alignment:

The influence coefficient matrix, as it is explained previously on this thesis, can be used in order to obtain information regarding the Hull Deflections of the vessel, the Thermal deviations which occur during the vessel's operation, and the adjustment of the bearings' offsets. However, ABS has developed a methodology for estimating accurately several parameters of the shaft alignment process, in which the Hull Deflections are included. In terms of Hull Deflections, ABS's software can be used for evaluating the behavior of the following ship types: Tankers, Bulk Carriers and Container Vessels.

Additionally, ABS (2022) in Enhanced Shaft Alignment Guide states that the calculations regarding the reaction forces of the bearings should be made for the following conditions:

1. Dry dock or waterborne at very light ballast at cold engine or gearbox and with propeller

partially immersed and aft peak tank empty

2. Fully ballasted vessel with cold engine or gear box, and aft peak tank empty
3. Fully ballasted vessel with hot engine or gear box and aft peak tank full or as filled up as

permitted by the vessels' loading manual

4. Fully laden vessel with cold engine or gearbox and aft peak tank empty
5. Fully laden vessel with hot engine or gearbox and aft peak tank full or as filled up as permitted

by the vessels' loading manual

Note: Conditions 2 to 5 should be studied considering fully immersed propeller.

1.3.1. Bearing Reactions

ABS (2019)

As far as the Bearings' Reactions are of interest, ABS states that "ABS Rules stipulate that the primary criterion for acceptance of alignment is satisfactory bearing reactions. An alignment condition is acceptable as long as the bearing reactions remain positive under all service drafts, and no bearing is unloaded". However, if the measured value of the Bearing Reaction deviates from the calculated value more than 20%, the shaft alignment conditions shall be re-examined in order for the issue to be addressed. The following is preferred, with respect to avowing unloading or overloading of the bearings:

1. At least 10% of the allowable bearing load is desired on each bearing, and
2. Measured bearing reactions may not exceed 80% of the manufacturer's maximum allowable load limit.

Utilizing Simulated Jack-Up Diagrams

Although the precise methods for measuring the bearing reactions according to will be discussed in the next chapters, at this point, it is important to highlight the utilization of simulated jack-up diagrams in order to assess the accuracy of the actual bearing load confirmation. However, the calculation of the reaction forces of the bearings, are most commonly extracted by Shaft Alignment software (ABS, 2019). These calculated values shall be compared to the measured ones.

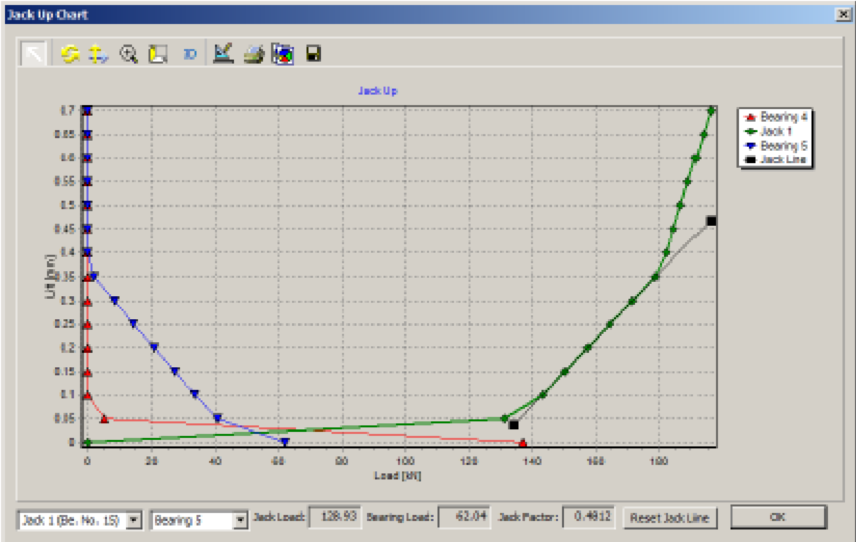

Figure 61: Simulated Jack-up Diagram, ABS [12]

ClassNK (2006)

ClassNK (2006) on the Guidelines on Shaft Alignment, investigates the issue of determining the initial Bearing Offsets. More specifically, while the most common industry method is to determine them based on empirical knowledge, ClassNK proposes a method for estimating properly the initial bearing offsets by utilizing the Shafting Stiffness Matrix. As mentioned earlier, the Shafting Stiffness Matrix is the terminology ClassNK uses on this specific publish for the Influence Coefficient Matrix, which is also explained in the previous chapters of this Thesis. Moreover, for clarification reasons, it should be mentioned that ClassNK refers to the influence coefficients by the term “Bearing Reaction Influential Number”.

Offset (mm) / Reaction (kgf)	δ 1 (1mm)	δ 2 (1mm)	δ 3 (1mm)	δ 4 (1mm)	δ 5 (1mm)	δ 6 (1mm)	δ 7 (1mm)	δ 8 (1mm)	δ 9 (1mm)
R1	139295.220	-214398.670	81956.652	-9075.998	8643.455	-6589.810	213.664	-53.416	8.903
R2	-214398.670	342737.790	-147868.080	25863.061	-24630.482	18778.394	-608.860	152.215	-25.369
R3	81956.652	-147868.080	86843.213	-29740.647	34253.719	-26115.195	846.745	-211.686	35.281
R4	-9075.998	25863.061	-29740.647	25755.066	-64621.265	53184.964	-1724.440	431.110	-71.852
R5	8643.455	-24630.482	34253.719	-64621.265	487261.830	-602809.510	204508.110	-51127.028	8521.171
R6	-6589.810	18778.394	-26115.195	53184.964	-602809.510	895929.320	-459651.520	152728.030	-25454.671
R7	213.664	-608.860	846.745	-1724.440	204508.110	-459651.520	451271.610	-264078.490	69223.181
R8	-53.416	152.215	-211.686	431.110	-51127.028	152728.030	-264078.490	255095.360	-92936.091
R9	8.903	-25.369	35.281	-71.852	8521.171	-25454.671	69223.181	-92936.091	40699.447

Figure 62: An example of a Shaft Stiffness Matrix, ClassNK [24]

The Bearing Reactions which participate in the Shaft Stiffness Matrix shall explicitly refer to the forces which develop due to the change in the bearing offset of interest, and the (self-) weight or other external forces should not be accounted for.

Consequently, the bearing reactions which occur because of the bearings’ offsets can be calculated (without the (self-) weight and the external forces). The matrix shown above can be written in the following form:

$R = A * \delta$, *where:*

$R = \{R1, R2, \dots R9\}^T$,

`

$$A = \begin{bmatrix} 139295.220 & -214398.670 & 81956.652 & -9075.998 & 8643.455 & -6589.810 & 213.664 & -53.416 & 8.903 \\ -214398.670 & 342737.790 & -147868.080 & 25863.061 & -24630.48 & 18778.394 & -608.860 & 152.215 & -25.369 \\ 81956.652 & -147868.080 & 86843.213 & -29740.647 & 34253.719 & -26115.195 & 846.745 & -211.686 & 35.281 \\ -9075.998 & 25863.061 & -29740.647 & 25755.066 & -64621.265 & 53184.964 & -1724.440 & 431.110 & -71.852 \\ 8643.455 & -24630.482 & 34253.719 & -64621.265 & 487261.830 & -602809.510 & 204508.110 & -51127.028 & 8521.171 \\ -6589.810 & 18778.394 & -26115.195 & 53184.964 & -602809.510 & 895929.320 & -459651.520 & 152728.030 & -25454.671 \\ 213.664 & -608.860 & 846.75 & -1724.440 & 204508.110 & -459651.520 & 451271.610 & -264078.490 & 69223.181 \\ -53.416 & 152.215 & -211.686 & 431.110 & -51127.028 & 152728.030 & -264078.490 & 255095.360 & -92936.091 \\ 8.903 & -25.369 & 35.281 & -71.852 & 8521.171 & -25454.671 & 69223.181 & -92936.091 & 40699.447 \end{bmatrix}$$

$\delta = \{\delta 1, \delta 2, \ldots, \delta 9\}^T$.

The Methodology ClassNK (2006) proposes, has as follows:

- Firstly, a straight-line approach is applied, where the bearings' reactions, the deflection curve, the shear force and the bending moment of the shaft are calculated.
- From the calculated values of the reaction forces of the bearings, the desirable reaction forces of certain bearings can be estimated.
- Assuming that the target bearing reactions are known, the additional Reaction force (ΔR) can also be calculated.
- By applying reverse calculations, the bearing offsets in order for the additional force to be generated can be calculated form the Shaft Stiffness Matrix. (Note: If in straight line the bearing of interest is overloaded, ΔR<0, while if it is underloaded, ΔR>0)

The desired bearing offsets are calculated by using the formula:

$$\delta = A^{-1} * \Delta R,$$

However, it is important to note, that the system described above has infinite solutions. In order for the engineer responsible to calculate the offsets of interest, as submatrix of A^{-1} should be generated, by erasing the first and last rows and the first and last columns. This approach means that "the constraints of the rigid motions do not affect the bearings, including the translation and rotation of the shafting line", which practically means that the offsets of bearings Number 1 and Number 9 are set equal to zero. The vertical offsets of the bearings Number 2 to Number 8 can then be calculated. The assumptions described above are proved not to have a remarkable effect on the accuracy of the calculated bearing offsets.

Furthermore, another parameter which ClassNK engages in the Guidelines on Shaft Alignment, is the optimization of the precise location of the intermediate bearing.

As was previously discussed, the span between the bearings has a strong effect on the reaction forces which apply on them. Moreover, when the vessel is operating, changes in drafts or hull deflections which may occur due to sea swell or the loading condition of the ship, also have an effect on the bearing reactions of the bearings. These two parameters, if combined, indicate that the longitudinal position of the intermediate bearing strongly affects the behavior of the aftmost main engine bearing.

In terms of settling the small span between the intermediate bearing and the aft most engine bearing issue, which may cause severe problems, the logical solution would be adjusting the intermediate bearing's position further away from the aft most main engine bearing. However, by doing so, the influence coefficient of the intermediate bearing itself increases, as it approaches the stern tube bearing. It is obvious, that the prementioned represents an optimization problem, regarding the influence coefficients of the intermediate and aft most main engine bearings. In order to estimate and investigate the effect on the sensitivity of the rest of the bearings, which occurs due the adjustment of the longitudinal position of the intermediate bearing, ClassNK (2006) proposes the following formula which represents a Sensitivity Index:

$$Sensitivity\ Index = \sum_{i=1}^{n} (R_{mi})^2, \text{ where:}$$

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- n: The total number of bearings,
- R_{mi}: The influence coefficient of the intermediate bearing to the ith bearing,
- m: The number which represent the intermediate bearing.

By calculating the Sensitivity Index for a series of longitudinal positions of the intermediate bearing, "a unique optimal location for the intermediate bearing" is determined.

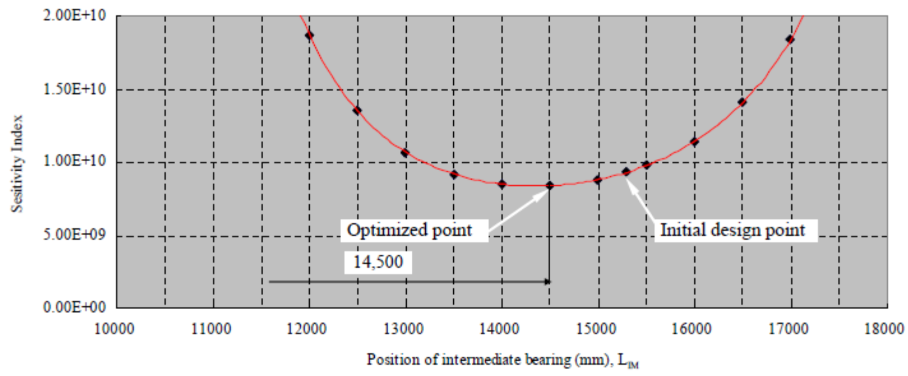

Figure 63: An example of plotting the Sensitivity Index as a function of the longitudinal position of the intermediate bearing, ClassNK [24]

ClassNK (2006) requires that all of the bearings except the aft stern tube one, must be investigated using only one supporting point, located at the middle of the bearing. In the case of modeling the aft stern tube bearing using one supporting point, this point should be placed either at D/3 or L/4 (where: L; the length of the aft stern tube bearing and D; The diameter of the shaft) from the aft end of the bearing. However, if the aft stern tube bearing is modelled using two supporting points, these points must be at the two ends of the bearing. If the supporting points are three or more, the engineer responsible can make the decision regarding the exact locations of the supporting points. The bearings shall be modeled assuming either rigid or elastic support. ClassNK (2006) also requires, that in cases where the thrust shaft is integrated with the crankshaft, the minimum number of main engine bearings that participate in the calculations regarding shaft alignment, should be five main engine bearings.

Last but not least, ClassNK (2006) states that, while all the bearing reactions are required to be positive (which practically means that they are supporting the shaft), the bearing reaction of the aft stern tube bearing may be equal to zero, if the manufacturer agrees.

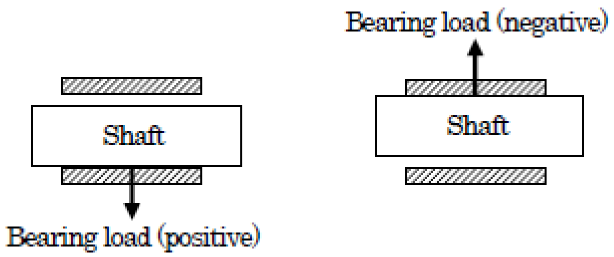

Figure 64: Direction of bearing reaction forces as ClassNK assumes them, ClassNK [24]

However, in light draught condition with hot engine, ClassNK requires that: All of the bearings' reaction forces are positive, the propeller must be assumed to be fully immersed in all of the calculations, the bending moment which occurs due to eccentricity thrust of the propeller may be taken into account during the calculations and, for shafting with reduction gear, the difference between the bearing loads of the forward and aft bearings of the gear wheel in hot condition must comply with the limits set by the manufacturer.

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In cases of Full Draught conditions with hot (running) engine, all Tankers, Chemical Carriers, Ships carrying dangerous chemicals in bulk, Bulk Carriers and General Dry Cargo ships must comply with the following:

All bearings should be evenly loaded, even when hull deformations are present in Full Draught Condition. However, the relative displacement which occurs due to the difference between the hull deflections in Light Draught condition and the hull deflections in Full Draught condition (which are calculated as δ_{B2}, δ_{B3} respectively) may cause the second or third aft most main engine bearing to unload, which is highly undesirable. For this reason, δ_{B2} *and* δ_{B3} must always be greater than the lower limit which is set to be equal to δ_{BM}, as shown in the figure below:

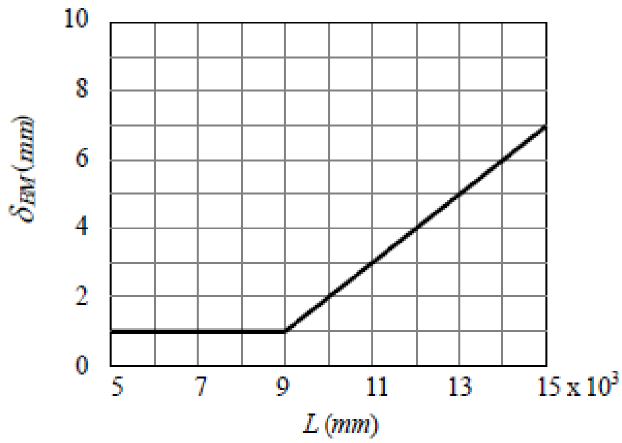

Figure 65: Allowable lower limit as a function of the distance from the support point of the aft most engine bearing to the aft most bulkhead of the engine room, ClassNK [24]

The values of δ_{B2}, δ_{B3} can be calculated for either rigid or elastic support assumptions, as explained below:

$$\delta_{Bi} = \frac{-R_i}{S_i}$$, where:

- i: The number of the bearing of interest, as counted from the aft of the engine,
- R_i: Reaction force at engine bearing Number i (in kN),
- S_i: Influence number of engine bearing Number i, when the hull deflection at the aft most bulkhead of the engine room becomes equal to -1mm (in kN/mm), which is calculated using the following formula:
  $S_i = \sum_{n=1}^{a-1} C_{b+i-1,n} * (1.5 * X_n - 0.5) + \sum_{n=1}^{b-1} C_{b+i-1,n} * x_n{}^{1.5}$, where $x_n = X_n/L$, and:
  - n: Support point number of the shafting (counted from the aft of the shafting),
  - α: number of the nearest support point forward of the aft most bulkhead of the engine room (counted from the aft of the shafting),
  - b: Support point number of the aftmost engine bearing (counted from the aft of the shafting),
  - X_n: Distance from the support point b to the support point n (in mm),
  - L: Distance from the support point b to the aft most bulkhead of the engine room (in mm),
  - $C_{m,n}$: Influence number at the support point m when the relative displacement at the support point n becomes -1 mm (in kN/mm)

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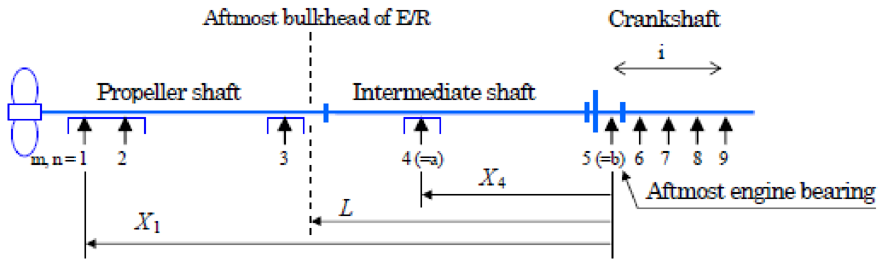

Figure 66: Numbers of main engine bearings and support points, ClassNK [24]

In cases where rigid support was assumed, δ_{B2}, δ_{B3} can be calculated by simultaneously solving the following equations:

Where:

- K: Stiffness of the bearing support (in kN/mm)
- S_i: Influence number for engine bearing i,
- $C_{i,j}$: Influence number for engine bearing i when the relative displacement at engine bearing j becomes equal to -1 mm (in kN/mm) (counted from the aft of the engine),
- δ_i (i=1, 2, 3, 4, 5): Elastic relative displacement at each engine bearing resulting from the relative displacements δ_{B2}, δ_{B3}.

LR (2023)

Lloyd's Register (2023) requires that in cases of bearings constructed by synthetic materials, evidence shall be given to Lloyd's Register which indicate that the diameter, ovality and straightness after installation of the bearing are verified to be safe for operation under all conditions of the vessel and its machinery, as well as they should also be verified to be within the acceptable tolerances regarding ovality and straightness (longitudinally and circumferentially). Moreover, the intermediate shaft bearing shall not exceed the reaction force limit which is set to 80% of the manufacturer's allowable bearing load (for plain journal bearings), in accordance to the projected area of the bearing. In general, the bearings' loads limits are following the manufacturer's limitations.

DNV (2021)

As DNV (2021) states, the bearing loads must comply with the limits set by the manufacturer. In cases where zero or significantly loads on bearings are present, they should not be further investigated only if they have no adverse effect regarding oil-whirling phenomena.

BV (2015)

BV (2015) requires that the Bearing Loads and the Load Distribution amongst the bearings shall not exceed the limits imposed by the Bearing Manufacturer.

BV (2015), regarding Bearings' Reactions, states that the calculations performed during the design stage of Shaft Alignment must ensure that no oil-whirling phenomena are going to be encountered. To ensure this, "a global whirling calculation of line shafting and ship structure which are connected through the oil film stiffness and damping" shall be carried out. BV (2015) also requires, that the calculations which concern the reaction forces of the bearings, are carried out using the Hertz Contact Theory for static conditions, and by integration of the oil film pressure, as it is given by the Reynold's equations for journal bearings, for running conditions.

When in static condition, according to BV (2015), the characteristics of the contact (such as stiffness, reaction, length of contact, maximum pressure and squeezing) shall be defined via conducting a Hertz Theory approach.

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Hertz theory approaches the issue by assuming a finite length cylindrical socket which contains a cylinder which has a load applied on it. In this approach, the socket represents the bearing at the supporting point and the cylinder represents the shaft. The contact pressure and the load are the parameters of interest which need to be calculated, and the shaft's displacement inside the bearing is considered as known. Moreover, BV (2015) states that all of the bearings, except the aft stern tube one, shall be modeled using only one supporting point.

Hertzian Contact Theory

At this point, a reference in the Hertzian theory shall be made. Hertz theory, as a classical theory of contact mechanics, derives from the analytical solution of elasticity theory equations under half space approximation, and is often used in order to correlate the properties of a system to the developed stresses which occur. The assumptions one shall make while applying the Hertz theory, are :

- The developed strains are considered small and within the elastic limit.
- The surfaces of the objects in contact are assumed to be continuous and non-conforming (which practically means that the contact area is significantly smaller than the dimensions of the involved bodies).
- Each body is considered as an elastic half-space.
- The surfaces are frictionless.

Hertz theory primarily focuses on non-adhesive problems (which means that no tension is allowed to occur in the contact area, i.e.: the bodies in contact can be separated without adhesion forces). Spherical contact is considered to be a special case of the Hertz theory of contact. Generally, there are two distinguishable cases of non-adhesive contact problems: A conforming contact, where the bodies are in touch at numerous points before any deformation takes place, and the non-conforming contact (e.g. the geometry of the surfaces of the two involved bodies are different enough), when the bodies involved are in touch only at one point or along a line before any load is applied.

If only vertical forces are examined in a Hertz Theory of contact problem (which in our case is true; we assume that the cylinder is subjected to a load and the displacement of the shaft inside the cylinder socket is considered as known), elastic deflections on the surface under applied pressure is given by the formula:

$$u_z(x,y) = \frac{2\pi}{E\prime} \iint \frac{p(x',y')}{\sqrt{(x-x')^2+(y-y')^2}} dx'dy',$$ where:

- $u_z(x,y)$: the elastic deflection,
- $\frac{1}{E\prime} = \frac{(1-v_1{}^2)}{E_1} + \frac{(1-v_2{}^2)}{E_2}$: the reduced elastic modulus, and v_1, E_1, v_2, E_2 are the Poisson's ratio and Young's Modulus of the two bodies,
- $p(x,y)$: is the contact pressure

Due to the fact tha Hertz Theory of Contact assumes parabolic pressure distribution, which serves in high accuracy the purpose of investigating cylindrical, elliptical or spherical body contact, the following equation may be extracted:

$$p(x,y) = \sqrt{p_0 * (1 - \frac{r^2}{a^2})}, \text{ where:}$$

- *r:* the distance to the arbitrary point on the surface,
- *a:* Hertz Contact Radius; unknown parameter,
- p_0: Maximum Hertz Pressure; unknown parameter.

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By substituting the concluded formula for the contact pressure p(x, y) in the equation for the elastic deflections, the following occurs:

$$u_z = \frac{\pi * p_0}{4 * E' * a} * (2 * a^2 - r^2), r \leq a$$

Furthermore, for a rigid sphere penetrating an elastic half space, the elastic deformation of the flat surface is given by the formula:

$$u_z = \delta - \frac{r^2}{2 * R}, r \leq a.$$

By equating the above equation with the first equation obtained for the elastic deformation, all of the unknown parameters are specified as follows:

$$a = \frac{\pi * p_0 * R}{2 * E'},$$

$$\delta = \frac{\pi * \alpha * p_0}{e * E'},$$

$$p_0 = \frac{2}{\pi} * E' * \frac{\delta}{\sqrt{R}},$$

$$F = \frac{4}{3} * E' * \sqrt{R} * \sqrt{\delta}$$

Where F: the applied load.

1.3.2. Aft Stern Tube Bearing Particulars

As previously discussed, the aft stern tube bearing requires extra attention and investigation during the design stage of the Shaft Alignment procedure, due to its larger length compared to the rest of the bearings, as well as because of the large cantilever load which is exerted by the propeller, and which directly affects the aft stern tube bearing. The Deflection curve can be utilizing in order to determine the actual misalignment angle of the shaft inside the bearing, measured from the reference line which is already established via means which will be later presented. Practically, the misalignment angle equals zero, only when the shaft is in contact with the bearing at both of its edges.

At this point, it is of vital importance to mention the following: The stern block of the vessel, is usually characterized by increased stiffness. Consequently, when both the aft and the forward stern tube are installed, the stern tube misalignment angle is not severely impacted by the hull deflections. However, in cases where the forward stern tube bearing is not installed, extensive investigation shall be conducted, and the results must be concluded in the decision-making process regarding the misalignment angle and the whole shaft alignment design.

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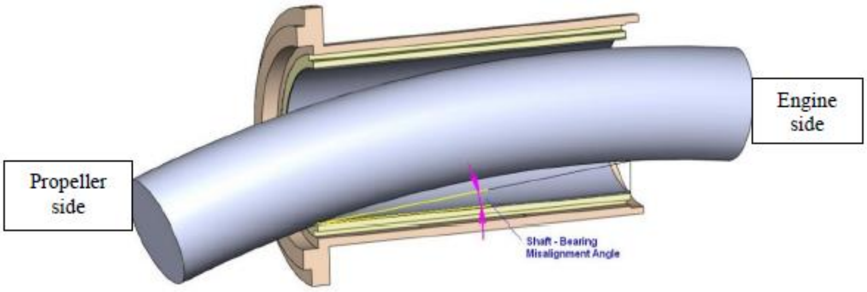

Figure 67: Aft stern tube bearing misalignment, ABS [12]

1.3.3. Definitions of Slope Boring and Bearing Inclination Methods

In cases where the limit which is set for the maximum misalignment angle between the shaft and the aft stern tube bearing is exceeded (0.3*10^{-3}rad), it is mandatory for either slope boring or bearing inclination techniques to take place. At this point, both techniques will be explained:

- Slope Boring: According to ABS,, *"Slope boring is a process where the bearing shell is machined so as to ensure that the center line of the bearing's inner bore is misaligned to the desired angle (defined by shaft alignment analysis). To allow provision for slope boring, the inner bearing diameter is initially pre-machined to the smaller diameter. The special boring machine is then attached to the stern block and aligned so as to match the required misalignment angle. Machining is then conducted by boring through the bearing in several passes, if required. Multiple passes may be necessary when larger amounts of bearing material are to be taken away because of a danger of bearing material overheating, as well as to ensure required machining tolerances".* The main disadvantage of this method is that is a slow and sensitive procedure which requires special equipment, as well as the fact that for significantly long bearings, the precision is reduced.

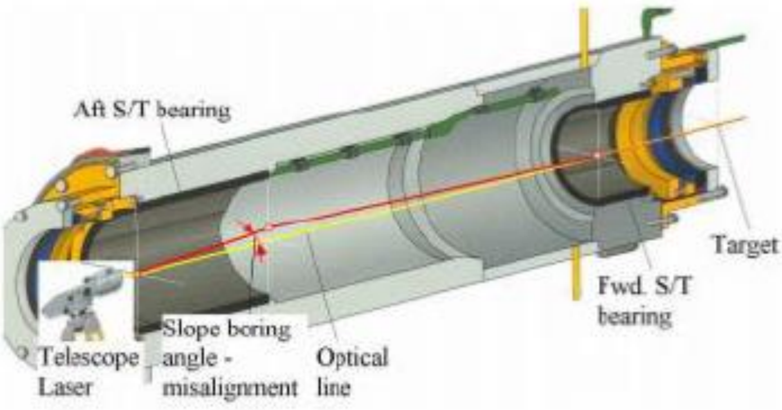

Figure 68: A slope boring arrangement, ABS [12]

- Bearing inclination: According to this method, the bearing is machined to its desired final diameter, prior its installation on the vessel. Another difference this method opposes in comparison to Slope Boring, is that the bearing's casing is fixed to the stern block of the vessel using epoxy resin, instead of shrink fitting. The main disadvantages of this method are that it is possible only for one inclination to take place, which has a major impact on the possibility of load distribution modifications.

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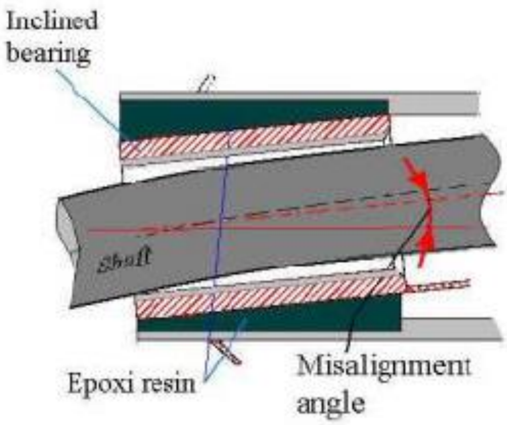

Figure 68: A bearing inclination arrangement, ABS [12]

The main reasons why the techniques described above are mandatory in cases where the maximum limit of the misalignment angle is exceeded, have as follows:

- Prevention of excessive edge loading of the tail shaft,
- Modification of the load distribution along the bearing's length,
- Treatment of the excessive propeller loads which result in bending deformation of the tail shaft,
- The axes of the bearing and the shaft are already "misaligned" because of the bearings' vertical offsets as well as due to the shaft's deformation,
- Need for accommodating a large variety of loads during the vessels operational life cycle.

1.3.4. Relative slope Limits

ABS (2019)

ABS (2019) requires that Slope Boring or Bearing Inclination methods are applied in cases where the limits regarding the angular misalignment are exceeded. The limit regarding the relative slope between the shaft and the aft stern tube bearing is set to $3*10^{-4}$ rad. The pre-mentioned techniques are widely known practices in the industry. ABS provides a software, which can conduct aft stern tube bearing analysis and evaluate the slope boring conditions and application.

Lloyd's Register (2023)

Lloyd's Register (2023) also sets the limit regarding the maximum allowable relative slope between the shaft and the aft stern tube bearing (in static conditions) to $3*10^{-4}$ rad.

BV (2015)

BV (2015) suggests that the mean relative slope in the aftmost bearing (θ_s) , between the shaft and stern tube inner axes, is to be less than the ratio of radial clearance divided by the bearing's effective length: $\theta_s < \frac{C_{radial}}{L_{eff}}$, where: C_{radial}; the radial clearance and L_{eff}; The effective length of the bearing.

DNV (2021)

DNV (2021) requires that in hot running or hot static conditions, the maximum value od the relative misalignment angle between the aft stern tube and the shaft does not exceed $3*10^{-4}$ rad (0.3mm/m) and 50% of minimum diametrical bearing clearance divided by the bearing's length, whichever is less (which is practically the same limit BV proposes).

$$\boldsymbol{\theta_{slope,max} = \min\{0.3 * 10^{-3}, 0.5 * \frac{C_{diam,min}}{L_{bearing}}\}}$$

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The aforementioned is only applicable for no-slope or single-slope designs.

1.3.5. Shear Forces and Bending Moments

ABS (2019)

As far as Shear Forces and Bending Moments of the propulsion machinery are of interest, ABS (2019) states that they shall always strictly remain in the limits defined by the machinery's manufacturer. Some Diesel Engine Makers define Moment and Force limits regarding the main engine's aft flange, with the purpose of preventing misalignments of the Main Engine which could prove to be devastating. An example of a maker's diagram regarding an aft flange of a main engine and the allowable shear forces and bending moments, is presented below:

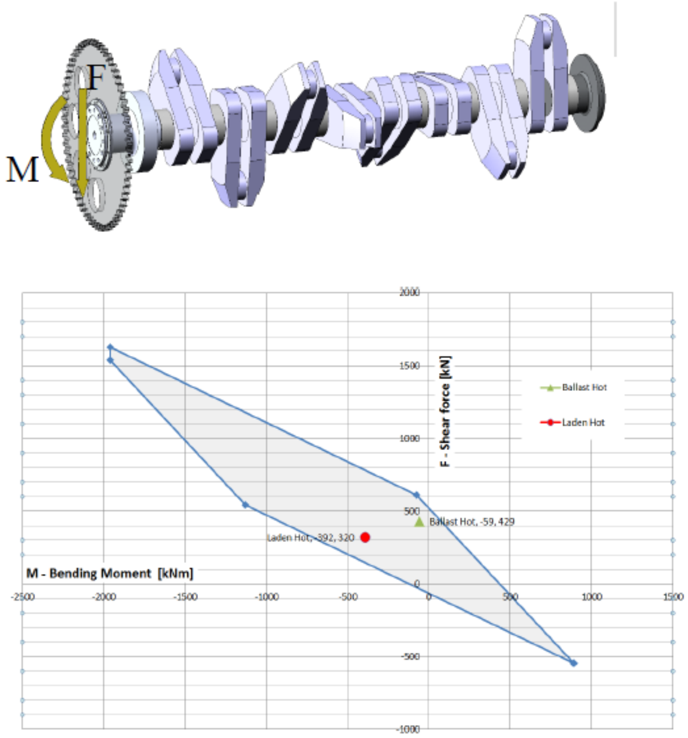

Figure 69: An example of Diesel Engine Output Flange Allowable Shear Force and Bending Moment Chart, ABS [12]

DNV (2021)

DNV (2021) requires that the Shear Forces and Bending Moments are also within the allowable limits set by the manufacturer of the specific installation.

BV (2015)

BV (2015) requires for the shaft bending stress and moments to be in compliance with the manufacturer's requirements.

ClassNK (2006)

ClassNK (2006) states that, the bending moments calculated at any bearing should at all costs be less than the calculated aft stern tube bearing Bending Moment.

LR (2023)

LR (2023) requires that the Shear Forces and Bending Moments shall be in compliance with the manufacturer standards.

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1.3.6. Aft Stern Tube Bearing Acceptable Pressure

ABS (2019)

Regarding the aft stern tube bearing's loads, it is in direct relation with the acceptable compressive stress levels of the material from which the bearing is constructed. The compressive pressure which ABS Rules refer to, correlate to the "compressive pressure estimated from the bearing load over the projected area of the bearing". Current ABS rules set the limits for stern tube bearings to:

Table 4: Limits regarding aft stern tube bearing pressure, ABS (2019)

For metallic bearings:	**0.8 [N/mm^2]**
For oil-lubricated synthetic bearings:	0.6 [N/mm^2]

This case is one more example of a designing issue which can be estimated and approached with high precision, by utilizing the ABS Software for conducting an aft stern tube bearing analysis and slope boring evaluation.

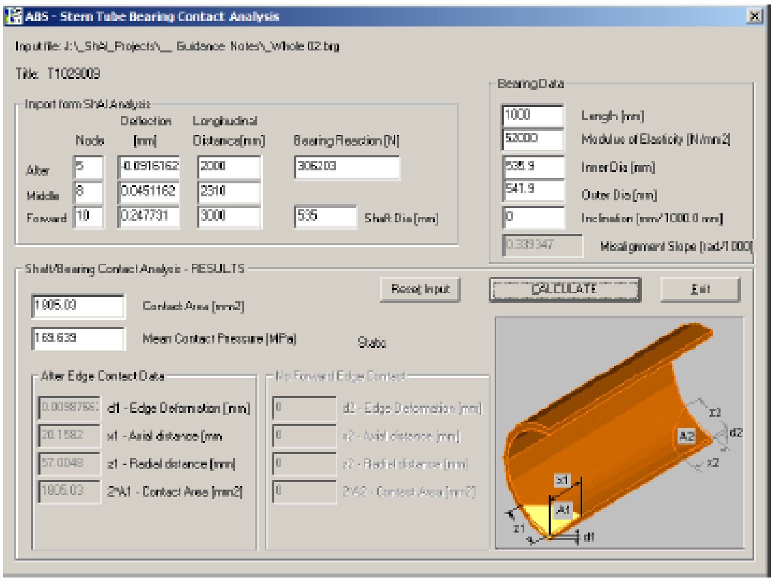

Figure 70: Example of ABS aft stern tube evaluation software, ABS [12]

BV (2015)

BV (2015) sets the limits for the value of the maximum local pressure on the stern bush to:

$$P_B \leq 110\ bars,$$

While the limit for the specific pressure on stern bushed is:

$$P_S < 0.8\ MPa, in\ cases\ where\ white\ metal\ is\ used\ as\ antifriction\ material, while$$

$$P_S < 0.6\ MPa, in\ case\ of\ other\ antifriction\ material\ types, where\ Specific\ Pressure\ is\ calculated\ from\ the\ formula: P_S = \frac{R_V}{L_{eff} * D_O}$$, where:

- R_V: Total vertical reaction of the considered bearing, in N
- L_{eff}: Effective length of the considered bearing, in mm
- D_O: Outer diameter of shaft in way of the considered bearing, in mm.

BV also suggests, that in the case of aft stern tube bearings, the load distribution in static condition should be: 2/3 of the reaction on the shaft part and 1/3 of the reaction on the forward part, as well as to model the Aft

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Stern Tube bearing using at least five supporting points, "in order to have detailed results at each section of the bearing".

ClassNK (2006)

According to ClassNK (2006), in cold static (cold engine) and light draught conditions, the nominal bearing pressure at the aft stern tube bearing should not exceed 0.8 MPa. Moreover, the maximum relative inclination between the tail shaft and the aft stern tube bearing is set to $3*10^{-4}$ rad. The maximum allowable pressure on the aft stern tube bearing is set to 40 MPa.

LR (2023)

As LR (2023) states, the limits regarding the aft stern tube bearing, its allowable minimum length and the bearing's pressure, regulations have as follows:

Table 5: Limits regarding aft stern tube bearing pressure and length, LR (2023)

For water lubricated bearings which are lined with lignum vitae, rubber composition or staves of synthetic material:	$L_{bearing}$>=4*D, where D; rule diameter of the shaft in way of the bearing
For water lubricated bearings lined with two or more circumferentially spaced sectors of synthetic material, in which it can be shown that the sectors operate under hydrodynamic principles:	$L_{bearing}$>=2*D, where D; rule diameter of the shaft in way of the bearing, and $P_{nominal}$<0.55 MPa (regarding the nominal pressure of the bearing)
For oil lubricated bearings of synthetic material (where the flow or lubricant is to be such that overheating, under normal conditions, cannot occur):	$P_{nominal}$: considered upon application, must be supported by the results of an approved test program. However, $P_{nominal}$ shall not exceed the limits set by the type of the synthetic material. $L_{bearing}$>= 2*D, where D; rule diameter of the shaft in way of the bearing.
For oil lubricated, white metal lined bearings, which are equipped with an approved type of sealing:	$L_{bearing}$ is to be approximately equal to 2*D, where D; rule diameter of the shaft in way of the bearing, and not less than 1.5*D. $P_{nominal}$<0.8 MPa
For bearings of cast iron and bronze, oil lubricated and equipped with an approved type of sealing:	$L_{bearing}$>=4*D, where D; rule diameter of the shaft in way of the bearing.
Grease lubricated bearings:	$L_{bearing}$>=4*D, where D; rule diameter of the shaft in way of the bearing. Other lengths may be considered upon application, subject to the provision of suitable supporting in-service or testing evidence at relevant shaft pressures and velocities.

Moreover, Lloyd's Register requires that the loading of the forward stern tube bearing in static conditions, is such that the unloading during static or running conditions (including severe weather conditions and maneuvering conditions) is prevented.

1.3.7. Additional Stern Tube Clearance Requirements, (ABS)

Regarding the stern tube clearances, the acceptable range depends on the effect it has on the bearing misalignment in each case. However, under all circumstances, the measured value cannot differ from the calculated one more than +/-0.1 mrad (ABS, 2019). The aft stern tube bearing clearance, highly depends on whether the installation has a forward stern tube or not. That is, because the existence of both stern tube bearings results in significantly different load distributions and contact areas. Moreover, when a forward stern tube bearing is installed, it could be considered as a "connecting link" between the propeller shaft and the intermediate shaft, while in installations which only have an aft stern tube bearing, the propeller shaft is

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directly engaged with the intermediate one, which serves as an additional complication factor in the design process regarding the shear forces, the load distributions, the vibrations etc.

More specifically, according to ABS Rules (2019) on Shaft Alignment.

In cases where both aft and forward stern tubes are installed, the calculations shall be conducted considering the following:

- Calculations referring to the propeller shaft only, supported on the aft and the forward stern tube bearing,
- Unrestrained shaft at both ends,
- Only gravity forces acting upon the propeller shaft; propeller weight is not considered,
- Aft and forward stern tube bearings are modelled using multiple support points.

While in cases where no forward stern tube bearing is installed, the calculations shall have as follows:

- Propeller shaft and intermediate shaft are connected and supported only on the aft stern tube bearing and the intermediate bearing,
- Propeller shaft is unrestrained on the aft end,
- Intermediate shaft is unrestrained on the forward flange,
- Aft stern tube bearing is designed with a double slope and modelled using multiple supporting points,
- Intermediate shaft bearing is modelled using multiple supporting points.
- The stern tube lubrication arrangement is designed with the fresh oil inlet located aft of the aft stern tube bearing,
- The calculation contains details needed for clearance verification and its comparison with the bore sighting measurements,
- A sag and gap calculation procedure on the propeller-shaft connection with the intermediate shaft is not required. Should this be included in calculation, the review engineer should include a review comment, and the surveyor should verify that the sag and gap procedure does not foresee any information required to conduct shaft alignment corrections.

In cases where the approach concludes in excessive load on the intermediate shaft's bearing or in excessive shaft stresses, the clearance calculation may be conducted by adding a support point on the forward end of the intermediate shaft.

1.3.8. Length of the Aft Stern Tube Bearing, (ABS)

As it was previously discussed, the aft stern tube bearing requires special attention in the design stage, as it is more complex and sensitive than the rest of the bearings, due to its increased length, the dynamic and static loads of the propeller with which the propeller shaft is directly engaged, and due to its special design, which usually includes the slope boring technique.

These days, the design trends lean more and more towards a double-slope design, which is thought to have numerous benefits regarding the life span, the behavior and the performance of the shaft system, as well as it contributes significantly in the avoidance of bearing damages. A single or double slope design could also be optimized using Fluid Structural Interaction (FSI) Analysis method for dynamic turning conditions.

The general benefits of designing an aft stern tube bearing with a double slope, are:

1. Smooth transition from static to running condition, faster reaction of an oil wedge creation and thus minimization of the time of the bearing's exposure to metal-to-metal contact.

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2. Maintenance of sufficient oil film thickness during all operating and maneuvering conditions of the vessel, including cases where the downward dynamic forces of the propeller reach their maximum values.

The whole point of optimizing the design of an aft stern tube bearing, is to minimize the contact pressure. It is important to note, that when investigating static contact pressures which practically engage only the gravitational forces of the equipment. Thus, the value of the static contact pressure may be higher than the pressure exerted on a dynamic condition, when the oil film is developed and contributes to relieving the bearing and carrying a part of the load.

The ideal condition would be maintaining minimum contact pressure under all circumstances and conditions, which could be translated into maximum static contact area. Furthermore, the most preferable case engages entirely symmetrical distribution of the contact area on the edges of the aft stern tube bearing. It is obvious, that the most preferable condition when taking the prementioned under consideration, is the one where the angular misalignment between the shaft and the bearing is equal to zero. However, this is not always feasible, so the design engineer should turn for solution to a low angular misalignment design.

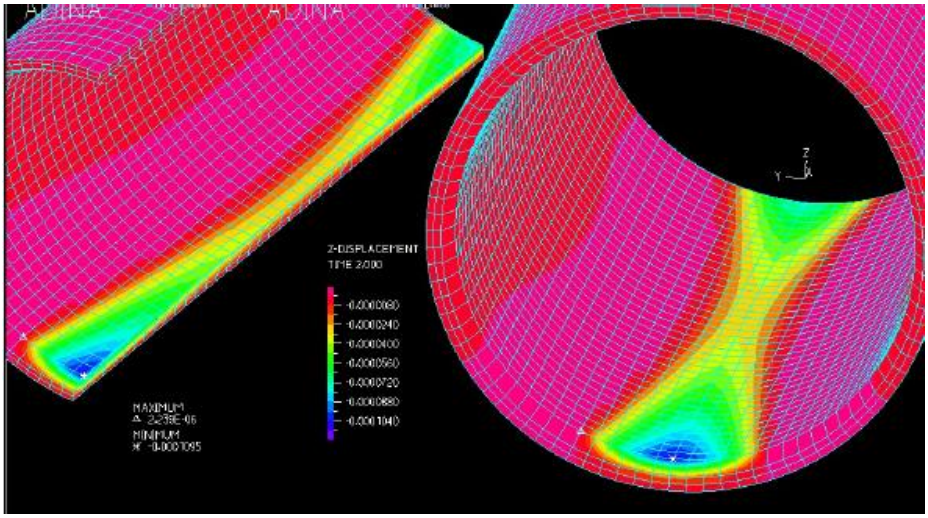


Figure 71: Desired static contact area between the shaft and the bearing, fully symmetrical, no misalignment present, ABS [12]

ABS Rules (2019) proposes the following regarding the ratio of the bearing's length to the shaft's diameter:

$L_{bearing} \geq 1.5 * D_{shaft}$, for oil lubricated bearings, while

$L_{bearing} \geq 4 * D_{shaft}$, for water-lubricated bearings.

According to ABS Rules, the design of the aft stern tube bearing is directly impacted by the approach taken during the design stage of the shaft alignment process. The approach correlates to the relative position of the intermediate bearing and the Main Engine's bearings, in comparison to the aft stern tube bearing. The approach could be: Zero Offset Alignment condition (refers to a straight-line situation), Positive Offset Alignment condition (most of the system's bearings are positioned above the aft stern tube bearing) or Negative Offset Alignment condition (most of the bearings are located below the aft stern tube bearing).

Below, the results of the investigation ABS conducted on a Diesel Engine driven VLCC are presented. The most preferable condition in this case study, in terms of relative slope angle and minimum contact pressure, is represented by the "Negative Offset Alignment" condition.

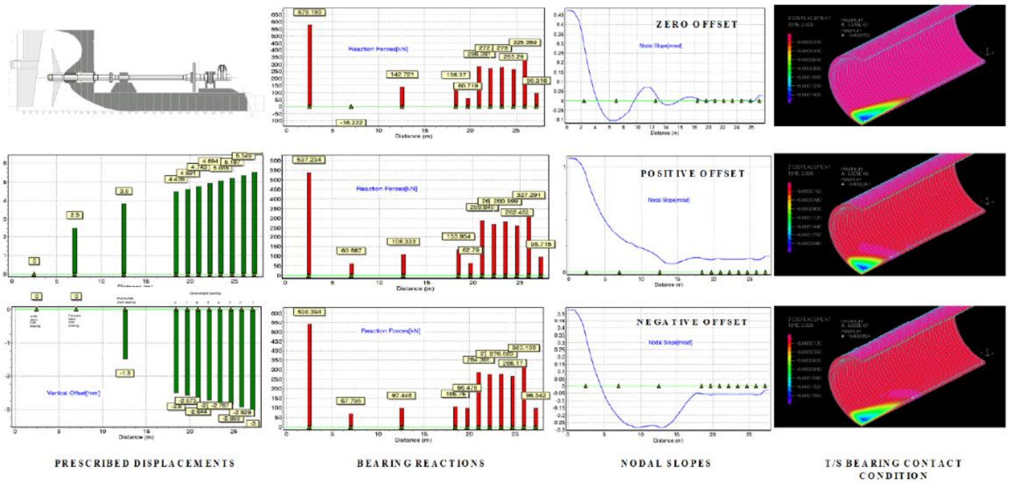


Figure 72: Bearing Reactions, Nodal Slopes and Contact Area of aft stern tube bearing, in relation to the Alignment Design, ABS [12]

1.3.9. Single Slope Design of the Aft Stern Tube Bearing, (ABS)

The basic principles based on which ABS suggests the Aft Stern Tube Bearing with Single Slope to be designed, are presented below:

- The slope boring design should be conducted taking into consideration only the static loads.
- In cases where it is desired for the dynamic loads to contribute to the design, this shall only happen with the requirement of a full-scale dynamic analysis being carried out using the Fluid-Structure Interaction software (FSI).
- The dynamic loads exerted from the propeller which are to take part in the calculations, should be the ones which fall into the worst-case scenarios regarding the service condition that the vessel is being subjected to, which is with the vessel turning hard-over at full speed.

The main scope of the optimization process of the aft stern tube design featuring a single slope installation, is to provide an adequate hydrodynamic lift of the shaft and to facilitate the procedure of the oil wedge forming. The most common means to achieve this, is to minimize as much as possible the misalignment angle of the design for static conditions or low revolutions. It also is a common practice to apply a slope design only at the bottom part of the bearing, and in the most cases, the static approach of the single slop design process is satisfactory for dynamic conditions as well.

1.3.10. Single-point Contact and Multi-point Contact, (ABS)

Another subject of the design stage of an aft stern tube bearing and its slope boring characteristics, is the "dilemma" between the Single Point Contact Approach vs. Multi Point Contact Approach.

Traditionally, during the design phase of the shaft alignment procedure, the shaft was modeled using the singe point contact approach, where the precise location of the contact points practically represented the actual location of the bearing. However, after 2015, the tendency in major Classification Societies was to highlight the importance of a two-point or multi-point contact approach.

The single-point approach is sufficient enough in cases where no slope boring is required, however in situations which do not fall into the aforementioned case, this approach proves to be inadequate. Moreover, even the two-point contact or even the multi-point contact approach, as an assumption, differs significantly from the reality, as the contact between the bearing and the shaft is actually established over an area of the bearing that the shaft penetrates into.

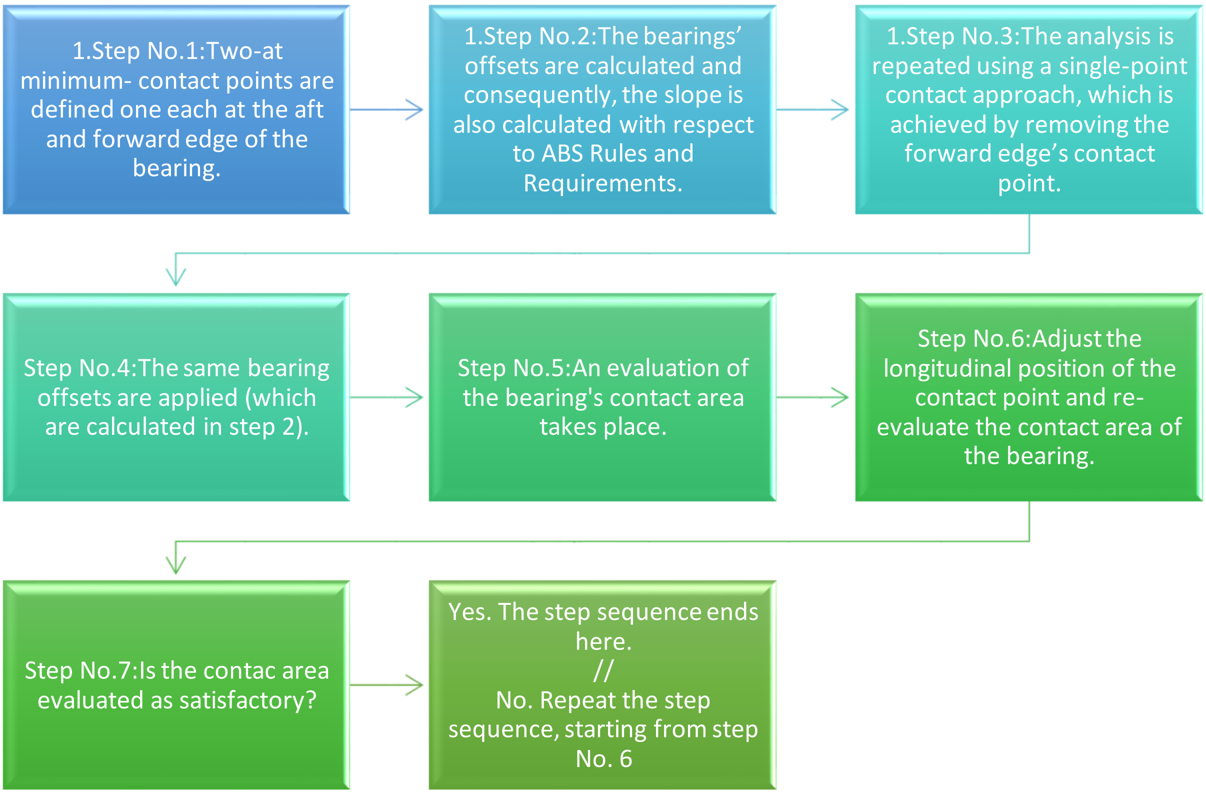
The importance of the decision between a single point contact or a multiple contact approach has to do with the fact that the methodology used has an effect on the reaction's intensity, as well as on the slope bearing design.

ABS (2019) suggests that under all circumstances, at first, a single point contact investigation is conducted in order to obtain the bending curvature of the shaft, as well as the zero-misalignment bearing slope. Once this information is determined, a two-point approach investigation is conducted, using the bearing slope calculated from the single point contact approach, which aims in defining the offsets of selected points along the bearing's length. More specifically, these points have to be the aft and forward edge of the aft stern tube bearing, and the maximum pressure point has to be selected between D/3 and L/4 of the aft bearing edge (where; D: shaft diameter and L: bearing length).

The two basic methods regarding single point or two-point contact ABS suggests, are:

1. Initially assuming a combined approach, entangling both single- and two-point contact, or
2. Conducting from the start, a pure single point contact approach.

It is important to note, that even when the two point contact approach is utilized, it is only used as a transient case until the single point contact is deemed to be satisfactory and adequate. More precisely, the basic steps ABS suggests regarding the two-point contact approach methodology have as follows:



ABS suggests engineers who utilize ABS Software for evaluating the contact pressure on the aft stern tube bearing, that the results of the procedure described above could "be considered as satisfactory when the co-ordinate x of the selected single point of contact is equal or slightly smaller than the calculated distance d, as shown in the bearing model above".

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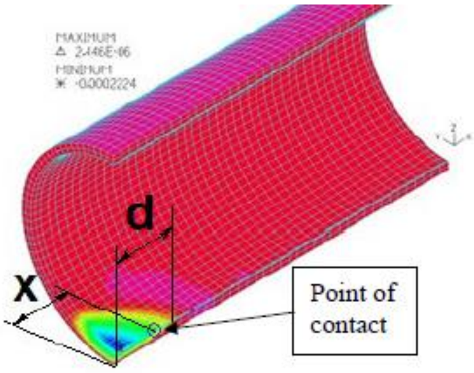

Figure 73: A Screenshot from ABS Software for evaluating contact pressure on the aft stern tube bearing, single contact point model, ABS [12]

Additionally, ABS (2022) in Enhanced Shaft Alignment Guide, requires that the slope angle between the shaft and the aft stern tube bearing should be calculated with the aft stern tube bush being in hot condition (static). More precisely, the following should be decided:

1. Slope angle between the shaft and aft stern tube bearing (misalignment angle) before and after the application of slope boring angle or inclination angle in the hot static condition.
2. Estimated Contact Area between the shaft and the bearing bush.
3. Estimated Mean Actual Contact Pressure between the shaft and the bearing bush

For the purposes of the contact evaluation, the software ABS provided can be used in order for the desired parameters to be calculated.

1.3.11. Double-Slope Design of the Aft Stern Tube Bearing

ABS (2019)

It is highly advised by ABS, to design the Double-Slope in order to reassure that the heaviest reaction load is applied precisely on the transition point (knuckle point) between the two different slope angles, and that the loads in the aft and forward edge of the bearing are approximately equal to zero. In this case, thorough investigation and analysis should be carried out, aiming to determine the exact location of the knuckle point and the exact values of the two different slope angles. It is advised, when applicable, to optimize the design by taking into account the dynamic forces exerted by the propeller during running condition.

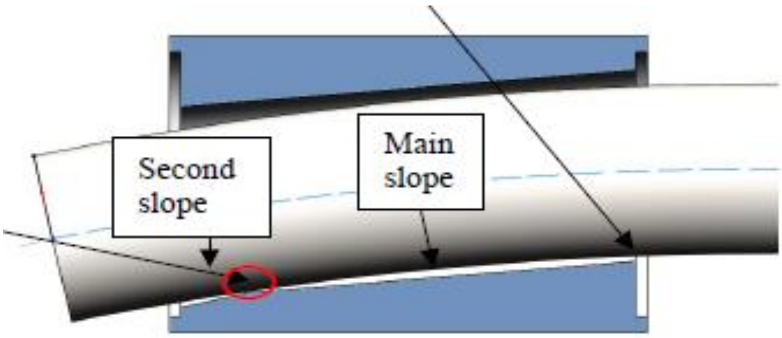

Figure 74: Typical Double Slope Design

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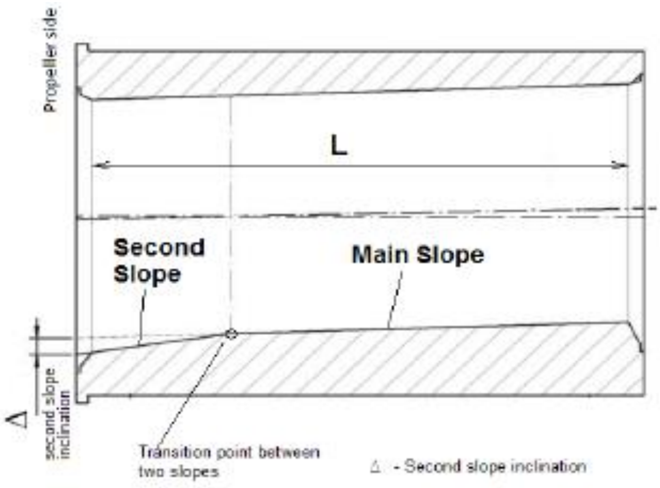

Figure 75: Typical Double Slope Design

Same way as it was approached in the Single-Slope Design, in this case, the transition point which shall support the heaviest load, is to be between D/3 and L/4 distance from the aft bearing edge.

At this point, it is of vital importance to highlight the fact that ABS Rules regarding Shaft Alignment clearly and explicitly demand that, in installations with no forward stern tube bearing, a double-slope design is applied. The exact location of the knuckle point is of great importance. This is because, the main scope of the design of a Double-Slope is to ensure that the shaft is forced to be in touch with the bearing only at the transition point, thus making the loads at the aft and forward edges of the bearing equal to zero and making the control over the acting load and the calculations remarkably easier.

The main benefits of a double-slope design have as presented below:

- As the condition transforms from static to dynamic, two oil wedges are developed: one aft and one forward of the knuckle point, which reduces significantly the metal-to-metal contact and improves the behavior of the system in terms of shaft lifting.
- The double-slope design is less sensitive to human and/or production errors during the installation, as well as less susceptible to the machining procedure.
- During the bedding-in process, plastic deformation is formed around the area which supports the higher loads, which in this case is the transition point. These deformations do not have a negative impact on the design, as long as they are absorbed and neutralized by the enlargement of the contact area, in order to minimize the contact pressure.

In cases where the shafting system is considered to be susceptible to problems related to the alignment (e.g. large twin screw and large single screw vessels or designs with no forward stern tube bearing), significant benefits are extracted by the conduction of a more detailed analysis using FSI. However, in order for the responsible engineer to carry out a Fluid Structure Interaction analysis, the precise dynamic loads have to be known, which are obtained by utilizing Computational Fluid Dynamics (CFD) Software. In these cases, the condition of interest is the transient hard turning at full speed condition of the vessel. According to ABS Rules (2019), FSI calculations shall be conducted for Full Ballast Condition and Laden Condition.

DNV (2021)

Moreover, DNV (2021) requires that installations with no forward stern tube bearing are designed with a multi-slope aft stern tube bearing. However, in cases where the shaft's diameter is less than 400mm (single screw)

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or 300 mm (twin screw), if the aft bearing lubrication criteria are fulfilled, a single-slope design could be accepted by the Classification Society.

The aft stern tube lubrication criteria which DNV introduces (2021) are based on the verification of the actual shaft's speed (*n*) being greater than the minimum speed required for hydrodynamic lubrication (n_0), for both low speed and full speed conditions.

Low speed criterion

In this case, the minimum speed of the shaft which ensures hydrodynamic lubrication conditions are calculated in hot static conditions, and the following must be verified:

$$n_{min} \geq n_{0,stat}$$

Full Speed Criterion

In this case, two different hot running conditions are participating in the calculations, which differ in terms of the vertical hydrodynamic bending moment acting on the propeller.

Hot running condition No. 1

In this case, the hydrodynamic bending moment acting on the propeller shall be calculated incorporating 30% of full torque downwards (for single stern tube arrangements), and 15% of full torque downwards (for other conventional designs).

Hot running condition No. 2

In this case, the hydrodynamic bending moment on the propeller shall be calculated incorporating 30% of full torque upwards.

It is important to note, that for unconventional hull forms and/or novel propulsion arrangements, the range of the propeller loads which participate in the calculations may need to be further extended, but this is an issue which needs to be specifically investigated for each installation which falls in the prementioned categories uniquely.

Regarding the Full speed criterion, the following must be fulfilled:

$$n_{full} \geq \max\{n_{0,dyn1}, n_{0,dyn2}\}$$

The value of the parameter n_0 is in all cases defined from the following formula:

$$n_0 = \frac{28 * 10^3 * c * h_0 * p_{eff}}{{c_{visc}}^{v} * D * L_{eff}}$$

Where:

- n_0: minimum rotational shaft speed ensuring hydrodynamic lubrication [rpm]
- $h_0 = \frac{D^{0.43}}{760}$, minimum required lubrication film thickness [mm]
- $p_{eff} = \frac{10^6 * W}{L_{eff} * D}$, effective bearing pressure [N/mm²]
- $L_{eff} = L * K_D * K_L * \left[\left(0.1 + 0.17 * \frac{W_{min}}{W_{max}}\right) - \left(0.32 - 0.02 * \frac{W_{min}}{W_{max}}\right) * \log(a)\right]$, $required\ that\ L_{eff} \leq L$, length of locally pressurized area [mm]
- $K_D = 0.53 * 10^{-6} * D^2 - 1.08 * 10^{-3} * D + 1.55$, dimensionless size factor
- $K_L = 0.33 * \left(\frac{L}{D}\right)^2 - 1.5 * \left(\frac{L}{D}\right) + 2.66, provided\ that \frac{L}{D} \leq 2$, dimensionless length to diameter ratio

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Moreover:

- n_{min}: actual shaft speed for continuous slow speed operation [rpm]
- n_{full}: actual max shaft speed for continuous operation, usually at MCR [rpm],
- C: Diametrical bearing clearance [mm]
- L: Bearing length, or in case of multi slope bearing, segment's length
- C_{visc}: Viscosity parameter [mineral oil: 1, EALs: 0.75]
- v: Kinematic viscosity at 40° [cSt] of the lubricant. It is used as the minimum viscosity acceptable for the installation.
- D: Bearing journal diameter [mm]
- W: radial bearing load (W1+W2, as depicted in the figure below) [N]
- W_{max}: Maximum value of W1+W2 [N]
- W_{min}: Minimum value of W1+W2, [N]
- a: Calculated relative slope between the shaft and the bearing at W_{max}, either a_1 or a_2 [mm/m] (as depicted in the figure below)

"The hydrodynamic propeller loads are defined as vertical bending moments as percentage of full speed torque for conventional hull forms and propeller arrangements". (DNV, 2021)

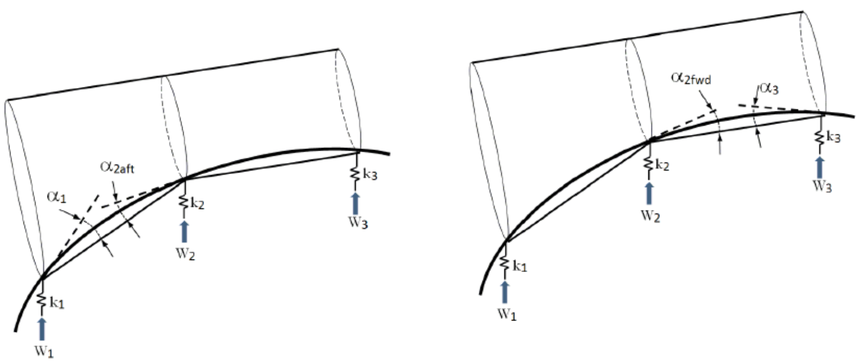

Figure 76: Depiction of reaction forces and misalignment angles in a double-slope design aft stern tube bearing, DNV [16]

At this point, in order to enrich the conversation regarding the stern tube lubrication, requirements proposed by DNV (2021), some additional details should be presented.

More specifically, it is important to note that the calculations regarding the verification of hydrodynamic lubrication conditions, are based on a quasi-empiric solution of the Reynold's equation for journal bearings and take under consideration special circumstances regarding the load distribution and the misalignment angle of the tail shaft inside the aft stern tube bearing, with a scope of ensuring hydrodynamic lubrication in the areas which suffer most in terms of pressure. Based on the calculations, a minimum operational shaft speed limit as well as a minimum viscosity of the lubricant limit shall be set. The dynamic condition calculations with a hot engine ensure that the lubrication system is capable of adapting and successfully operating in a wide range of loads.

Moreover, it should be highlighted, that in cases of double slope design bearings, the center load will be eventually distributed at both of the bearing parts, as there is oil film developed at both segments. The load will consequently be distributed proportionally on the two parts of the bearing, and from this the following are concluded:

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$$W_{2,aft} = \frac{W_1 * W_2}{W_1 + W_3}, W_{2,fwd} = \frac{W_2 * W_3}{W_1 + W_3}$$

Additionally, in addition to the Shaft Align Class Regulations by DNV (2021) two Enhanced Shaft Align notations are available, which include two “levels” of requirements, which are not mandatory: Shaft Align (1) and Shaft Align (2), both revised in 2023. More specifically, Shaft Align (1), if applied, imposes a requirement of the aft stern tube bearing being of multi slope design, irrespective of the misalignment angle between the shaft and the aft stern tube bearing. Moreover, DNV (2023) requires that the aft stern tube bearing is constructed using a white metal alloy material with tin as the major constituent. Similarly with Shaft Alignment basic regulations, DNV suggests that for installations with a shaft diameter of less than 400mm (single screw) / 300mm (twin screw), a single slope stern tube bearing design may be applicable, provided that it ensures compliance with the hydrodynamic lubrication criteria (for the same range of hydrodynamic propeller loads), as they were earlier described.

Moreover, in Shaft Align (1) DNV (2023) requires that fully immersed propeller is assumed under continuous operating conditions, and that in cases where the propeller is not fully immersed, means of warning shall be provided in the wheelhouse and in the central alarm panel. Furthermore, if inboard shaft bearings are installed on top of heated tanks, means of awareness in cases where the maximum temperature regarding thermal expansion are exceeded, shall be installed.

If one desires to meet the qualifications regarding Shaft Align (2) as well, in addition to Shaft Align (1) requirements, the following shall be met:

A CFD analysis taking into account hydrodynamically induced propeller forces and bending moments must be carried out, for the following conditions:

- Straight ahead running at MCR at design draught
- Transient turning conditions which at minimum include a ‘hard over’ turn with rudder angles of 35 degrees to both port and starboard. The turn shall originate form MCR straight ahead condition at design draught
- In cases where under normal operating conditions a partially immersed propeller is present, the conditions which is assumed to lead to “worst case scenario” propeller loads (in terms of local bearing pressure) must be thoroughly investigated
- Other critical conditions (such as ballast condition or crash stop maneuvers) which are decided by the designer, shipyard or Classification Society, should also be investigated.

Additionally, for straight ahead running conditions, hydrodynamic lubrication criteria calculations should be conducted by utilizing the results of the CFD investigation regarding the propeller loads. However, an additional 10% MCR Torque should be applied to the predicted upward bending moment. In terms of the calculations regarding the “Hot Running condition Number 1”, the propeller load which participates in the calculations should be similar to the bending moment of at least 30% of MCR torque.

Shaft Align (2) requirements incorporate a Finite Element Analysis regarding the contact area and the pressure distribution on the aft stern tube bearing. As minimum, the results should include:

1. The results which concern the straight-ahead condition with propeller bending moments derived from the CFD calculations, with additional 10% MCR torque added to the predicted upward bending moment, as well as the results which concern the “Hot running condition Number. 1” with a propeller bending moment of at least 30% of MCR torque.
2. The results concern Hot Static Condition, not incorporating hydrodynamic propeller loads.

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As far as Hull Deflections are of interest, DNV in Shaft Align (2) states that, is requested by the Classification Society, they should participate in the calculations regarding all relevant loading conditions. DNV (2021) states that the suggested method of estimating Hull Deflection is based on FEA Models of the vessel.

1.3.12. Designs with no Forward Stern Tube Bearing, (ABS)

Generally, it is beneficial in terms of avoiding possible unloading of bearings due to changes in the loading condition of the vessel, to have less bearings throughout the shaft's length. It is commonly endorsed in the industry for the acceptable distance between to subsequent bearings to be of a value equal to $400 * \sqrt{D}$ or more, with D being the shaft's diameter. The applied span between bearings, however, shall take into serious consideration the possibility of oil-whirling and generally vibration phenomena, which could turn out to be devastating for the shaft system's reliability and cause severe damages. From this point of view, an installation with no forward stern tube bearing comes with benefits, especially for relatively short shafts (such as the ones encountered in tankers and bulk carriers) which are seen in vessels which are characterized by a reduced distance between the Main Engine and the propeller. Installing a propulsion shaft system with no forward stern tube bearing can also make the installation, verification and the control of the clearances and the misalignment angle between the shaft and the bearing significantly easier. Furthermore, the system becomes more flexible, thus reducing the adverse impact of hull deflections on the shaft alignment design.

On the other hand, however, in systems with no forward stern tube bearing, if the calculations and the design stage is not thoroughly examined and carefully investigated, the results could be severe. Because of the fact that in these cases the tail shaft is directly engaged with the intermediate shaft, the exact longitudinal location and vertical offset of the intermediate shaft bearing plays a huge role in the successful design and installation of the aft stern tube bearing and its misalignment angle, and it should be carefully verified after the installation procedure in order to avoid possible loss of control over the misalignment angle. This type of installations are moreover characterized by the increased possibility of failure, damage or wear of the forward stern tube bearing seal, due to the shaft's bending which is imposed by the larger distance between the stern tube bearing and the intermediate shaft. This could lead to undesirable vibration phenomena, or more precisely, oil-whirling. Furthermore, according to ABS (2019), in these cases, the intermediate shaft bearing shall consist of a lower and an upper bearing shell. As far as the installation process is of interest, in cases where there is only an aft stern tube bearing, the intermediate shaft's bearing exact position should be defined and marked by sighting, and after the insertion of the propeller shaft into the stern tube, the line bearing shall be placed at the exact same location which was previously defined.

For the adequate modelling of a double-slope aft stern tube design, three points of support-at minimum-should be utilized: One in the transition point, and one each at each bearing's edges. Additionally, three support points (at least) are desired in the case of the intermediate shaft bearing, in order to obtain precise information regarding the misalignment angle and the reaction loads on the bearing.

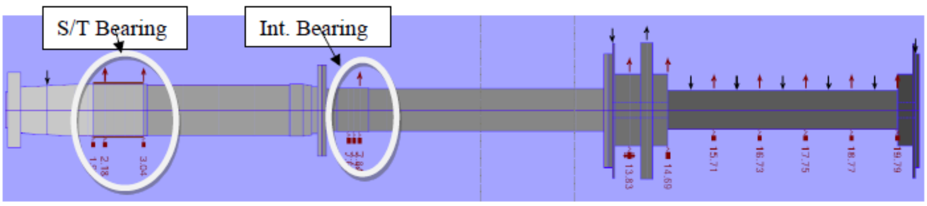

Figure 77: Example of modelling an installation with no forward stern tube bearing; three support points are assumed in both the aft stern tube bearing and the intermediate shaft bearing, ABS [12]

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1.3.13. Modelling of Shaft Line

BV (2015)

According to BV's guidance (2015) the shaft line shall be modelled based on the exact same geometry and characteristics as the original shaft line which is to be installed on the vessel.

The shafts should be modelled using conical or cylindrical beam elements. Moreover, BV (2015) requests that the propeller's mass participates in the shaft model, as an additional mass, in way of the propeller's center of gravity. The guidelines for the propeller's modelling in the Line Shafting investigation, have as follows:

1. Buoyancy effect in water is to be considered.
2. The exact immersion ration is to be considered, which depends on the loading condition of the ship.
3. The vertical and transverse hydrodynamic forces of the propeller are to be part of the calculations, which also depend on the exact loading condition of the vessel.

ClassNK (2006)

As ClassNK (2006) requires, while modeling the shaft line, static loads must be considered and the buoyancy working on the shafting must also be considered as a load. Moreover, the tensile force which occurs due to the cam shaft driven chain, as it is given by the manufacturer, must also be included in the calculations.

Number of bearings

ClassNK (2006) introduced a new parameter of interest regarding the modeling phase of the shaft. More precisely, on the “Guidelines on Shafting Alignment”, the effect of the number oof the main engine bearings included in the calculations is discussed. Traditionally, since one of the most highlighted issues of the design stage of the Shaft Alignment was the investigation of the Aft Stern Tube Bearing in terms of avoiding edge loading, no close attention was paid to the Main Engine's Bearings. In the diagram below, the Main Engine Bearings considered are presented, in correlation with the Main Engine's number of cylinders up until the year 2006.

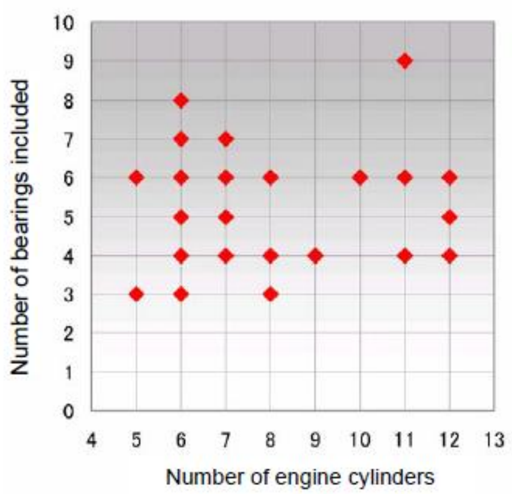

Figure 78: Number of main engine bearings taken into account in Shaft Alignment calculations, ClassNK [24]

Nevertheless, ClassNK through extensive research reaches to the conclusion that the severity of excluding Main Engine bearings from the calculations is highly remarkable, especially on the calculated values of the three aftmost main engine bearings, which are additionally considered to be the Main Engine bearings of the most interest.

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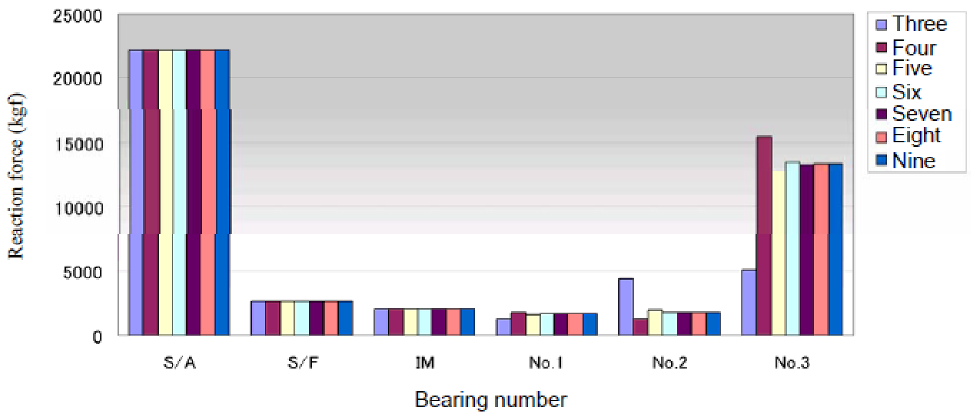

Figure 79 Effect of number of main engine bearings in the calculated bearing reactions, ClassNK [24]

Moreover, ClassNK (2006) suggests that the intermediate or tail shaft bearings where not severely affected by the number of Main Engine bearings taken into account.

As seen in the diagram above, which is an example regarding a seven-cylinder engine, the calculation process begins to stabilize only after five or more main engine bearings are taken under consideration in the calculations. However, this conclusion does not only affect seven-cylinder main engines, as the number of cylinders is completely irrelevant to the effect of the prementioned calculation approach. More precisely, what ClassNK (2006) suggests, is that if the reaction forces of e.g. 4 Main Engine bearings need to be calculated precisely, at least two more bearings shall be included in the modelling and calculations.

1.3.14. Crankshaft Modelling

As previously discussed, the small distance the main engine bearings have with each other, in combination with the mismatch between the flexibility characteristics between the bedplate, the engine's structure and the crankshaft, render the main engine's bearings of typical 2-Stroke crosshead diesel engines more sensitive to shaft alignment. Moreover, establishing an approach of developing a crankshaft model which consists of a simple beam with the same diameter the actual crankshaft has, may lead to remarkable deviations between the calculated measures and the real ones. A beam model with the exact same geometry as the actual crankshaft, has significant differences in terms of bending stiffness. When modelling and investigating the crankshaft's behavior in terms of shaft alignment, a suitable equivalent model of the crankshaft shall be used. In many cases, the Main Engine's manufacturer provides the equivalent crankshaft model. Nevertheless, if the equivalent model of the crankshaft is not available, alternative methods of generating it exist.

ABS (2019)

One of these methods is recreating the crankshaft's equivalent model by utilizing Finite Element Analysis of one-crank throw model of the crankshaft, and then generating several beam models. Following a trial-and-error sequence, the beam model which is characterized by the same bending deflections at the crank pin, shall correspond to the equivalent crankshaft model which should be utilized. (ABS, 2019)

Normally, the deviations which occur due to simplifications regarding the crankshaft's geometry are not of remarkable significance and do not adversely impact the shaft alignment's results. However, the load variation of the crankpins as a function of the crankshaft's angular position are of more significance and should be established by the responsible engineers during the procedure of the evaluation of the calculation results in comparison to the precise measurements.

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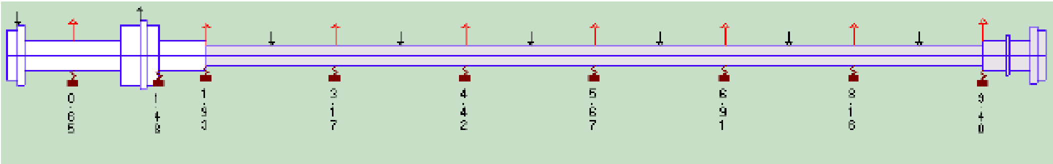

Figure 80: Crankshaft's equivalent model, ABS [12]

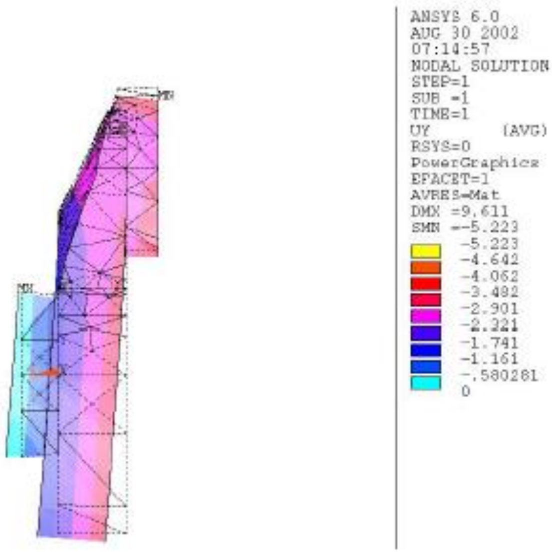

Figure 81: Finite Element Analysis of half of the crank, ABS [12]

At this point, some characteristics of the procedure of measurements on the main engine bearings should be discussed. When the measurements are taken, it is common for the crank-throws to be turned 90 degrees to port and starboard. Consequently, the piston does not remail in the TDC, as assumed during the calculation stage, which actually means that the assumption of non-variable loads of the crankpins is collapsed. As a result, the engineers responsible should be prepared to encounter a deviation between the calculated and measured values of the main engine's reactions, and they should take these deviations under consideration during the Shaft Alignment design stage in order to avoid encountering unloading of bearings during the vessel's operational life cycle. Another parameter which contributes to the pre-discussed deviations, is the stiffness variation of the crankshaft. According to ABS Rules (2019), the magnitude of these deviations is expected to be:

- Approximately up to 10% for the two aft most main engine main bearings, and
- Approximately up to 20% for the main bearings within the engine.

Extreme simplifications during the crankshaft's modelling phase may lead to significant damages, and consequently, time and expenses. A commonly seen example of a misleading, non-beneficial simplification is the case where a Partial Equivalent model of the crankshaft is utilized. In this case, the number of main engine bearings which are being investigated reduces dramatically, and the loads exerted from the crankshaft's components (such as pistons, piston rods, crossheads etc.) are completely overlooked. The consequences of such methods entail highly inaccurate information on the main engine's bearing loads, increased likelihood of

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design errors which may lead to unloaded bearings and inaccuracy in the sag and gap data. A common industry practice is to utilize at least four main engine bearings. (ABS, 2019)

BV (2015)

According to BV (2015), if the case is of a direct-drive installation of a low-speed Diesel/Gas engine, the crankshaft's model shall be created with an equivalent model, only if the exact stiffness matrix of the crankshaft is not given by the manufacturer (or in any case where it is unknown). When creating the equivalent model of the crankshaft, all masses, including propeller, wheels, gears, couplings etc. shall participate in the calculations, as well as the shaft's weight. If practicably possible, for the parts of the shafting machinery which function in water or oil environments, buoyancy could be taken into consideration. Furthermore, the following external forces are to participate in the equivalent model's investigation as well:

1. For geared installations: Tooth forces and moments in each direction
2. For direct coupled low speed engines: Chain forces, cylinder weights.

ClassNK (2006)

As ClassNK (2006) suggests, there are two possible ways of determining an equivalent model for the crankshaft: A Finite Element Analysis numerical calculation, or an Approximate Analytical Expression method.

FEA Numerical Calculations

At this method, ClassNK suggests that firstly a detailed FEA Model of the crankshaft is developed. The model had its left end restrained and an enforced vertical displacement is given at each supporting point (each supporting point actually represents a bearing). The reaction forces are then calculated. Simultaneously, a FEA Model of a circular bar is created, which is developed by using the exact same boundary conditions. By gradually decreasing the circular bar's diameter, a point is finally reached where the reactions at the supporting points of the Circular Bar ere equal to the ones calculated in the FEA Model of the crankshaft. The diameter of the circular bar at this stage, is the equivalent diameter of the crankshaft. An empirically established value of the crankshaft's equivalent diameter is a percentage of 60% of the actual diameter of the crankshaft. In the figure below, the calculated bearing reactions regarding the aforementioned procedure are presented.

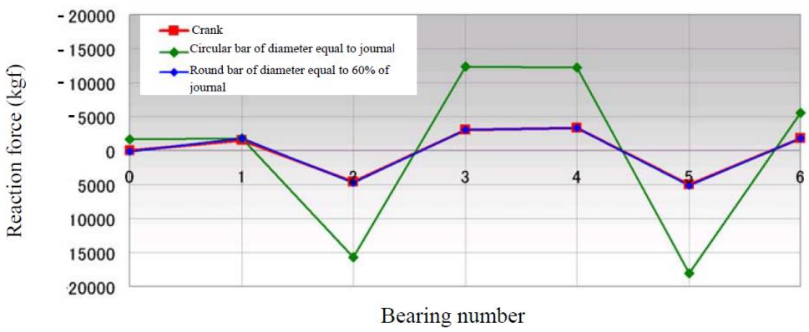

Figure 82: Calculated reaction forces on the support points in the cases of the pre-mentioned approaches, ClassNK [24]

Analytical Approximations

According to ClassNK (2006), assuming "that the diameter of the crankpin is equal to that of the crank journal", the equivalent diameter of the crankshaft can be calculated as follows:

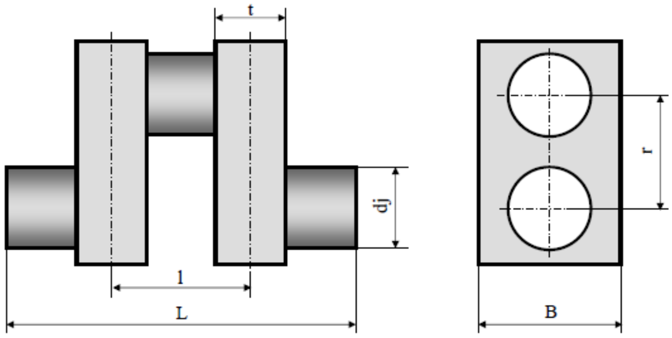

Figure 83: The dimensions needed in order to calculate the equivalent diameter of a crankshaft, ClassNK [24]

According to ClassNK (2006), assuming "that the diameter of the crankpin is equal to that of the crank journal", the equivalent diameter of the crankshaft can be calculated as follows:

$d_{eq} = [\frac{1}{2} * \left(\frac{1}{1+A_w} + \frac{1}{1+B_w+B_P}\right)]^{1/4} * d_j$, where:
$A_w = \frac{3*I_j}{2*I_w} * \frac{q_1*r}{L} * \frac{1+\left(\frac{l}{L}\right)^2}{1+0.65*\left(\frac{d_j}{L}\right)^2},$
$B_w = \frac{3*(1+v)*I_j}{I_{wp}} * \frac{q_2*r}{L} * \frac{1+\left(\frac{l}{L}\right)^2}{1+0.65*\left(\frac{d_j}{L}\right)^2},$
$B_p = \frac{3*(1+v)*I_j}{I_{jp}} * \frac{l*r^2}{L^3} * \frac{1+\left(\frac{l}{L}\right)^2}{1+0.65*\left(\frac{d_j}{L}\right)^2}.$ Also,
$I_j = \frac{\pi*{d_j}^4}{64},$
$I_{jp} = \frac{\pi*{d_j}^4}{32},$
$I_w = \frac{B*t^3}{12},$
$I_{wp} = \beta * B * t^3$, *and:*
$\beta = 0.0004057 * \left(\frac{B}{t}\right)^3 - 0.0082857 * \left(\frac{B}{t}\right)^2 + 0.0605809 * \left(\frac{B}{t}\right) + 0.1449623,$
$q_1 = 1 - \frac{0.1*\frac{{d_j}^4}{B*t^3}}{\left(\frac{r}{d_j}\right)*(1+0.1*\frac{{d_j}^4}{B*t^3})},$
$q_2 = 1 - \frac{0.45*\frac{{d_j}^4}{B*t^3}}{\left(\frac{r}{d_j}\right)*(1+0.45*\frac{{d_j}^4}{B*t^3})},$

v: The Poisson's ratio of the crankshaft's material.

However, ClassNK (2006) suggests that the manufacturer's equivalent model, if given, is utilized.

During the calculations which are based to the equivalent model of the crankshaft, ClassNK (2006) requires that the following are considered:

1. The piston weight shall be accounted for, either by increasing the specific gravity of the shaft or by adding the (self-) weight differential to loads representing it.
2. The propeller's weight shall be taken into account by estimating the buoyancy corresponding to the draft predicted in the condition in which the shaft alignment calculation is performed.
3. Loads representing the (self-) weight of the propeller shaft should also be modified by reducing the specific gravity to reflect the effect of buoyancy of lubricating or sea water, depending on the type of lubricating system used for stern tube bearings.

1.3.15. Wear

ABS Rules (2019) regarding Shaft Alignment, state that, while the bearings' condition in terms of wear does not have a significant impact on the alignment of the shaft, in order for a detailed design process to be conducted, it should be taken under consideration. The rate of the wear development depends on the contact area between the shaft and the bearing, and the conditions which govern the lubrication process during the vessel's operation, where the shaft executes rotational movement. Consequently, the angle of misalignment between the shaft and the bearing also has an impact on the wear down of the bearings.

The tail shaft (propeller shaft) of the shafting system, is considered to be extra-susceptible to wear down, due to its edge loading, caused by the propeller's overhang, which in turn leads to increased bending of the tail shaft.

Additionally, the monitoring of a bearing's condition in terms of wear can be tricky. A common industry practice is for the shaft seal manufacturer to supply a poker gauge or wear-down gauge in order for the measurements of stern tube bearing's wear down to be conducted. This tool is explicitly manufacturer uniquely for its installation, and severe issues may show up if it is lost.

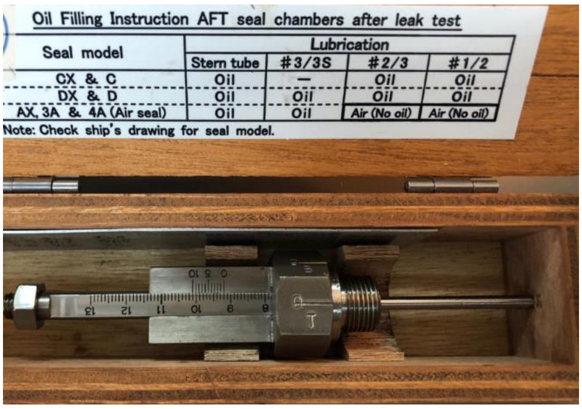

Figure 84: A wear-down gauge

1.3.16. Aft Stern Tube Clearances, (ABS)

Regarding the calculation of the clearances of the aft stern tube bearings, according to ABS (2019), two different situations can be distinguished:

1. Installations with both aft and forward stern tube bearings installed, and

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2. Installations with no forward stern tube bearing installed.

In the cases where both aft and forward stern tube bearings are present, the clearance of the aft stern tube is calculated by summarizing the shaft's deflection on the aft edge of the aft stern tube bearing (which is defined through the displacement vector table) and the absolute value of the bearing slope at the same location.

In order for the calculations to be conducted, two bearings are required. Consequently, in cases where no forward stern tube bearing is installed, it is replaced by the intermediate shaft bearing (as the intermediate shaft is directly coupled with the propeller shaft through bolts).

1.3.17. Requirements regarding Hydrodynamic Lubrication Criteria

It is generally observed, that Classification Societies which do not entail criteria regarding the number of contact points may entangle in their calculations for purposes of evaluating the contact pressure on the aft stern tube bearings, often use requirements which entangle lubrication criteria.

BV (2015)

BV (2015) requires that, for running conditions, and while taking under consideration the vessel's loading condition, the engine room temperature (ambient) and the shaft's speed (low, medium or high), an evaluation of the "risk of oil film break-up or excessive pressure on the antifriction material" is carried out. It is also encouraged to entangle the propeller's effort values in these calculations, if possible. The main scope of these calculations which correspond to running conditions, is the calculation of:

- The maximum local oil film pressure,
- The minimum oil film thickness,
- The relative position of shaft centers with respect to oil grooves,
- The distribution of reactions,
- The squeezing of the antifriction layer,
- The shaft's deflection and slope,
- The shaft's bending moment,
- The shaft's shear force, and
- The shaft's bearing stresses.

As BV (2015), the input data of this procedure shall have as follows:

1. Initial position of shaft in its bearings.
2. Offsets of support taking into account thermal expansion, pre-sag and structural deformation.
3. Effective length of the bearings.
4. Diameters of shell sleeves and antifriction material layer.
5. Young's Modulus and Poisson's ratio of shell and antifriction material layer.
6. Outer and inner diameters of shafts in way of supporting points.
7. Young's modulus and Poisson's ratio of shafts in way of supporting points.
8. Stiffness matrix of shaft line in way of supporting points.
9. Oil viscosity.
10. Rotational speed of shafts.
11. External forces and moments (as listed in the requirements concerning input data).
12. Flexibility matrix of steel work.

The acceptance criteria as BV (2015) demands are:

1. Maximum local oil film pressure: $P_O < 80\ bars$
2. Bearing loads: according to manufacturer's requirements.

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3. Specific pressure on stern bushes: $P_S < 0.8\ MPa$ in cases where the antifriction material is white metal, and $P_S < 0.6\ MPa$ if the antifriction material is of other type.
4. Shaft position in aft bush with respect to oil grooves: As recommended in the figure presented below. Zone 0 is considered to be forbidden and Zone 10 is considered to be optimal.
5. Minimum oil film thickness: $h_{min} > 30\mu m$.
6. Mean relative slope θ_s: The relative slope between the shaft and the stern bush inner axes is to be less than the ration of radial clearance divided by the bearing's effective length: $\theta_s < \frac{c_{radial}}{L_{eff}}$, where: c_{radial}; the radial clearance and L_{eff}; the effective length of the bearing.
7. Shaft Bending Stress and Moments: In compliance with manufacturer's requirements.

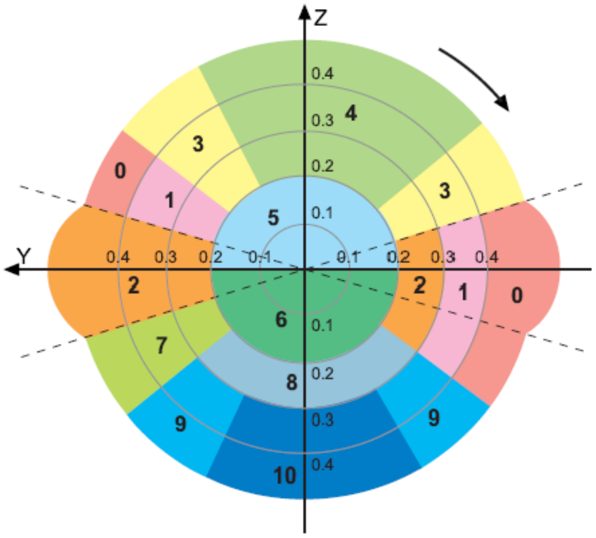

Figure 85: Shaft severity zones as a function of the shaft's location, BV [23]

The calculations regarding the Oil Film Characteristics shall be conducted by utilizing the Reynold's equation and a geometric equation which defines the oil film's height, with reference to the relative position between the deformed journal of the shaft and its machined profile. Based on these equations, the stiffness, the reactions, the oil pressure and the damping characteristics will be calculated. The idea behind the process, is to define through an iterative process the characteristics of the equilibrium position of the shaft inside the bearing (running conditions), and by utilizing the results, calculating the characteristics of the shaft's behavior inside the bearing (via Hertz Theory application or Oil Film Calculations). The mechanical parameters of all the bearings as well as the external forces (gravity, propeller, engine, gearing etc.) are introduced in the problem by the [K]: Global Stiffness Matrix and the parameter F_{ext}, thus reducing the problem in a way of supporting points. The general form of the Equilibrium Equation has as follows:

$$[K] * U + B_{sb} + F_{ext} = 0,$$ where:

- $[K]$: Global Stiffness Matrix, which is a combination of the following.
  $[K_s]$: Shaft Line stiffness matrix
  $[K_{sb}]$: Stiffness matrix of contact in static contact or running oil lubricated contact
  $[E_h]$: Hull flexibility matrix
- U: Vector of displacements, including the following components:
  U_b^0: Vector of initial bearing center position with reference to shaft center (without gravity), depending on the loading condition, temperature and alignment procedure.

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U_s: Vector of shaft center displacements in way of the supporting points relatively to the reference line

U_{sb}: Vector of shaft center relative displacements in way of the supports.

- B_{sb}: Vector considering the non-linearity of the contact conditions:
  $B_{sb} = F_{sb} - ([K_{sb}] * U_{sb})$, where F_{sb}: the contact force vector
- F_{ext}: External load vector, including gravity and other external efforts (propeller, engine, gearing) reduced in way of the supports.

By conducting iterative process (using an initial value for the displacement U_{sb}^0 close to the equilibrium solution in order to calculate the main contact characteristics: K_{sb}^0, F_{sb}^0 and B_{sb}^0), the factor U_{sb}^1 is defined.

The criterion which has to be met in order for the iterative process to be considered as successful, has as follows:

$$\left| {U_{sb}}^{i} - {U_{sb}}^{i+1} \right| < 0.001\ mm$$

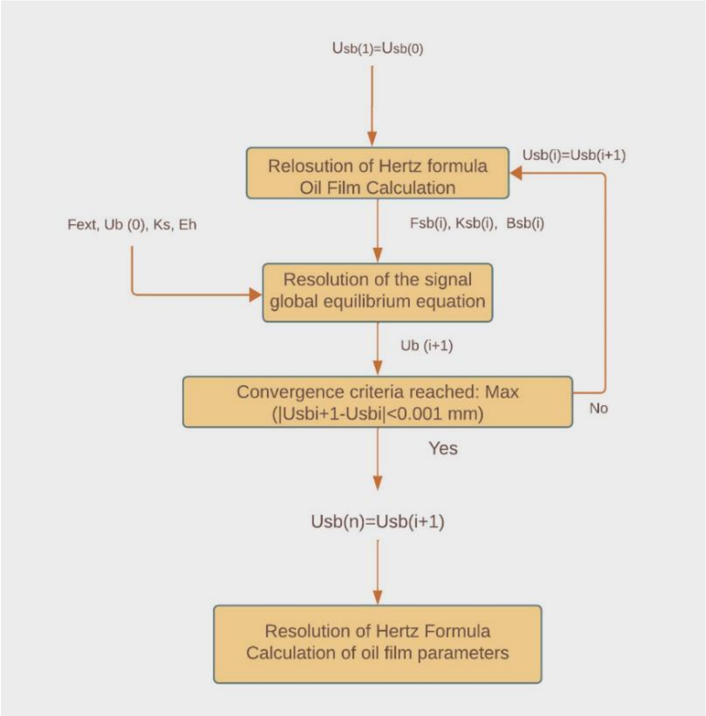

Figure 86: Flow chart of the iterative process, as BV (2025) suggests

1.3.18. Gear Driven Propulsion Installations

ABS (2019)

Regarding Gear Meshes, which means installations where the propeller is directly engaged with the gearbox, failures in the proper designing of Shaft Alignment may result in Gear Mesh misalignment or Gear Shaft bearing misalignment/overloading. Moreover, the conditions which are applied in order to serve Shaft Alignment purposes, affect the misalignment conditions between the gear and the pinion. According to ABS (2019), it is required for these installations to always maintain a uniform contact area on approximately 90% of the effective face width of the gear teeth. Additionally, the gear-pinion misalignment conditions shall be limited to the precise range defined by the gearbox's manufacturer.

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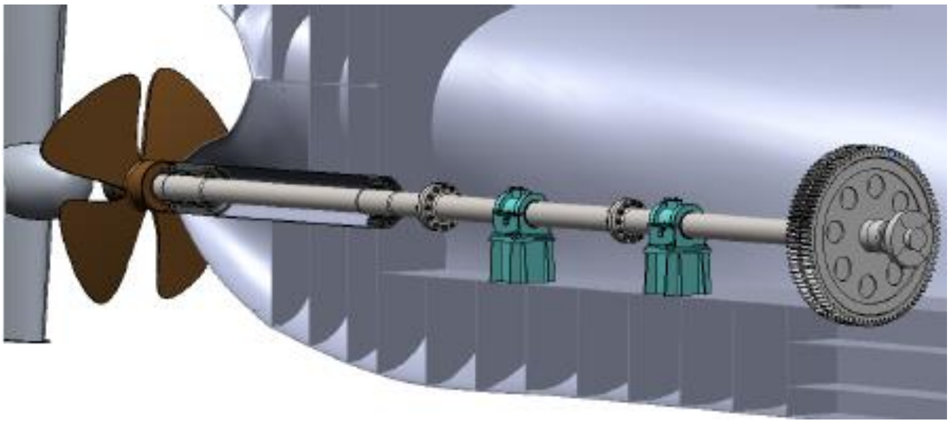

Figure 87: A typical Gear Driven propulsion system, ABS [12]

In order to conduct a more thorough investigation regarding Gear Driven Propulsion systems, the dynamic gear loads may be applied in combination with the static loads, in a pseudo-dynamic analysis procedure. More precisely, as ABS suggests (2019), "steady acting dynamic loads at gears are incorporated in the static shafting alignment model om top of the existing static forces". However, it is suggested that the results of a pseudo-dynamic investigation in Gear Driven Propulsion systems serves informative purposes, concerning the misalignment condition of the gears during running conditions.

ClassNK (2006)

ClassNK (2006) in cases of Shafting with Reduction Gear, states that the shafting from the propeller to the wheel gear is to participate in the calculations.

DNV (2021)

As far as installations which incorporate reduction gear(s) are of interest, DNV (2021) states that the limits regarding output shaft bearing loads and load distribution between the bearings, are set by the manufacturer of the reduction gear.

1.3.19. Possible Errors

During the design and calculations phase of the Shaft Alignment procedure, various errors may occur which can lead to difference between the calculated and measured values of numerous parameters, or even later lead to severe damages in the propulsion shafting mechanism. However, errors could happen during the on-board verification stage or in the installation phase. According to ClassNK (2006), some of the most common reasons which may lead to errors, cold be:

- Approximations and simplifications during the modelling of the shaft, which is later utilized for calculation reasons, such as the number of bearings taken into account (especially main engine bearings), modeling of the stern tube bearing, the diameter of the equivalent model of the crankshaft etc.,
- The accuracy of the Sag and Gap method (possible deviations between the calculated and measured values), and
- The accuracy of the measurements regarding Bearing Reactions.

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2.SHAFT ALIGNMENT PROCEDURE

Following the Design Stage of the Shaft Alignment process, which was previously discussed, the time comes for the application phase to take place. Every parameter of interest which was defined during the Design Stage must be verified to match the actual measured value after the installation of the shafting mechanism.

The installation process usually begins simultaneously with the installation and assemblance of the first stern blocks. However, several Classification Societies, including ABS (2019), suggest that the Shaft Alignment Procedure is initiated only after "*all of the heavy stern structure is in place, including the Main Engine*" (ABS, 2019). Furthermore, ABS (2019) requires that all of the welding work is completed when the aforementioned procedure begins. BV (2015) similarly requires that, "*only when the welding works of the neighboring parts of the aft structure of the vessel are completed, shall the procedure of the relative alignment of the aft stern bushes be conducted*". The Shaft Alignment Procedure is usually commenced under the responsibility of the Assigned Designer or Builder. According to DNV (2021):

"All large welding work in the vicinity of the shafting must be completed before sighting process and insertion of propeller shaft. All large and heavy structure elements shall be in place before final verification of shaft alignment".

2.1. Parameters to be addressed during the Shaft Alignment Procedure:

ABS (2019)

During the Shaft Alignment Procedure, the following parameters are addressed (ABS, 2019):

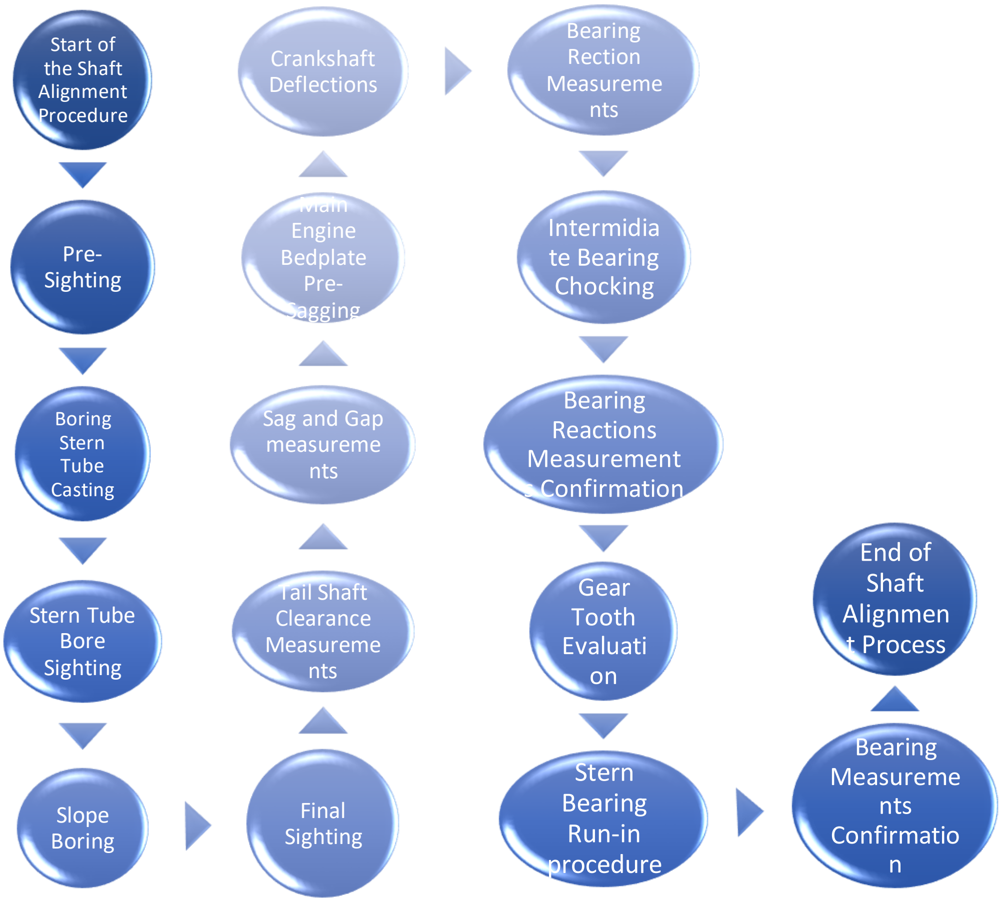
1. Pre-Sighting,
2. Boring of the aft stern tube casting,
3. Stern Tube bore sighting,
4. Slope Boring,
5. Final Sighting,
6. Tail Shaft clearance measurement,
7. Sag and Gap measurement
8. Main Engine bedplate pre-sagging,
9. Crankshaft Deflections,
10. Bearing Reaction Measurements,
11. Gear Tooth Evaluation,
12. Main Engine Chocking,
13. Intermediate bearing offset adjustment and chocking,
14. Stern rube bearing running in procedure,
15. Sea trial.

From the aforementioned, any issue which falls into the Measurement category, will be explained in the following chapter dedicated explicitly to Measurement Conduction.

As the Procedure begins while the vessel is in dry dock, the sag and gap measurements may take place under these conditions, but they should at all costs be in compliance with the calculated values regarding Dry Dock condition. Moreover, Hull Deflection measurements could also be conducted while the vessel is in dry dock, as long as the state of the vessel (which is not operating) are taken into account in the calculations (ABS, 2019).

Additionally, when the vessel is launched, the deflections of the crankshaft and the reaction forces of the bearings must be measured and be compared to the calculated values respectively, and they shall not exceed the allowable limits, as they were stated in the previous chapter. During the Sea Trial stage, according to ABS (2019), the reaction forces of the bearings must be measured and verified to be in compliance with the calculated values, for specific loading conditions, as they were described in the previous chapter. The Gear Tooth contact and the Crankshaft Deflections are investigated for the vessel in ballast condition and the Main Engine in hot static condition (ABS, 2019).

"Shaft alignment is considered acceptable when all the measured results are within the acceptance criteria and tolerances of the approved shaft alignment calculations." (ABS, 2019)



In the diagram shown above, the step-by-step procedure of the Shaft Alignment conduction phase is described, according to ABS (2019).

According to ABS (2022) the following should be submitted for review regarding the Shaft Alignment Procedure:

• Bore sighting before and after installation of stern tube bearings

• Stern tube bearing fitting pressure verification

• Tail shaft bearing clearance measurement

• Sag and gap measurements

• Bearing load measurements

Additionally, as long as the Shaft Alignment Procedure is of interest, ABS (2022) in Enhanced Shaft Alignment Guide, requires that the Shipyard keeps a detailed log with the recording of all the installation steps, which will be sent to the ABS Engineering Office for approval. Additionally, it is required that the following are applied with the vessel in Drydock or Very Light Ballast Condition:

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1. The shaft alignment procedure is not to commence before the vessel stern blocks are fully welded and all of the heavy stern structure is in place, including the main engine.
2. Only then should the reference line for positioning the shafts, bearings, main engine and gear box be established.
3. As far as possible, laser or optical checks are to be performed at night in order to avoid undesirable light or temperature disturbance.

BV (2015)

BV (2015) on the other hand, requires that the sighting, measurement and installation details of the Shaft Alignment procedure are provided to the Classification Societies for approval. Extra attention shall be given to the method, which is followed during the on-board measurements stage, according to BV. More specifically, as BV (2015) states: "The detailed Shaft Alignment procedure should be submitted for approval. It should include the corresponding calculations and description of each step that will be performed onboard: sightings, shaft installation fitting procedure and applied measurement tolerances." However, no specific requirements are established in the Guidelines on Elastic Shaft Alignment regarding the methods to be used or the specific conditions under which they should be performed, BV (2015) just provides some recommendations concerning the Shaft Alignment procedure. At this point, further explanation on the pre-mentioned objectives of the Shaft Alignment Procedure will be given.

2.2. Sighting Through/ Bore sighting

The precise calculation and implementation of the vertical offsets of the bearings is one of the most crucial parameters of the Shaft Alignment process. In order for these offsets to be practically defined, as was mentioned in previous chapters, a reference line must be established, based on which all of the offsets will be measured and applied.

ABS (2019)

The Sighting Through/Bore Sighting process is usually conducted by utilization of one of the following methods (ABS, 2019):

1. Piano Wire Method
2. Optical Telescope Method
3. Laser Method

Later in this Thesis, further details regarding the specific implementation guidelines for each method will be given.

It is worth mentioning, that ABS (2022) in Enhanced Shaft Alignment Guide, requires that engine's distance from the foundation is measured, so that the measured values match the ones assumed in the design phase, and consequently, result in the same offsets. If the engine is inclined, laser techniques may be used in order to verify the inclination with respect to the reference line. Final sighting readings are required from the Main Engine in order to finalize its positioning at the correct inclination angle and offset, as they were stated in the Shaft Alignment calculations. If bedplate pre-sagging is utilized, the bedplate's alignment should also be verified by piano wire or laser method in order to ensure compliance with manufacturer's recommendations.

BV (2015)

BV (2015) states that, after the sloping and the fitting of stern bushes is completed, *"The exact position of their centers should be precisely checked by optical or laser sightings".* The measured vertical and horizontal offsets of the stern bushes should correspond to the ones assumed during the calculation phase of the Shaft Alignment process. BV (2015) sets the limit regarding the measured and the calculated offsets at 0.05mm or less.

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DNV (2021)

According to DNV (2021), the Surveyor must be present in order for the stern tube bearing's geometry to be measured and reported after mounting the bearings but prior to insertion of the propeller shaft. The straightness, slope and ovality must be verified to be within the specified tolerances. DNV (2021) also requires that: *"Each stern tube bearing shall be checked at minimum three longitudinal positions covering the whole length of the bearing. Equal procedure shall be applied on each segment in case of multi slope bearing ".* Additionally, DNV (2021) suggests that for single stern tube designs laser aided or similar sighting methods should be used and that a minimum of two bearings should be included in the process.

2.3. Piano Wire Method

This particular method utilizes a wire (usually of 0.5mm or 0.7mm diameter) made out of steel. In cases where the Main Engine is installed at the time of the conduction of the Piano Wire Method, the one end of the steel wire is located on the aft part of the Main Engine, while in cases where the Main Engine is not yet installed, it is placed in a temporary support which represents the exact location of the Main Engine. The other end of the steel wire extends outside of the aft part of the end stern tube. The steel wire needs to be tensioned, and for this reason, a known weight is placed on the aft and forward ends of the wire. Moreover, the wire is centered using a Centering Spider at the aft and forward ends of the aft stern tube bush. Since the locations of the bearings at this stage are known, precise measurements are taken regarding the port, starboard, bottom and top distances between the piano wire and the aft stern tube casting in the way of the bearings' locations.

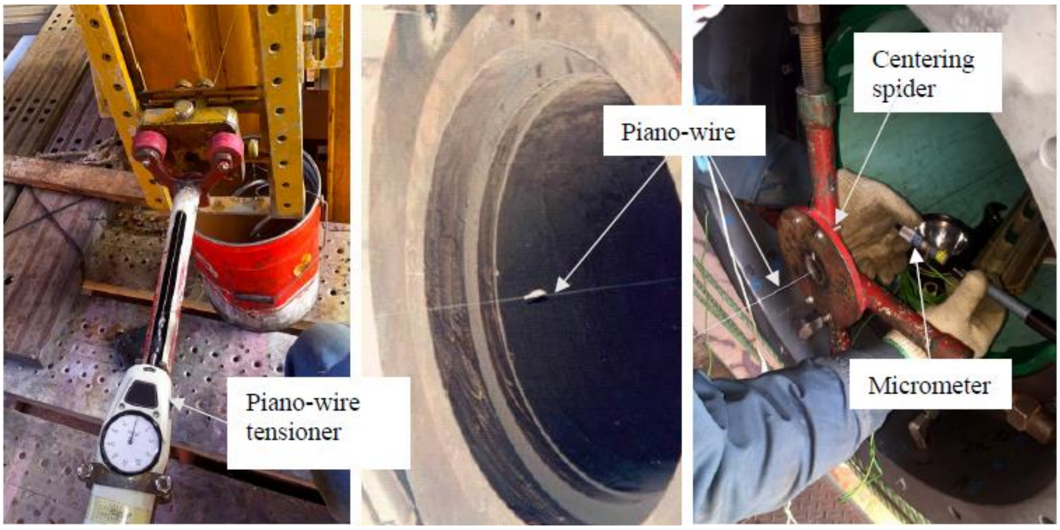

Figure 87: Piano Wire Method, ABS [12]

2.4. Optical Telescope Method

This method engages a precision telescope in order for the reference line to be established. Before the measurements are carried out, the instrument is placed on a tripod and leveled on a horizontal plane. In this method, a point which represents the reference point is set at one end, which should be completely undisturbed throughout the whole measurement process. Then, a set of points are set in locations which represent the center of the casting bore. The measurements taken concern the physical distance between the established reference line and the aforementioned set of points.

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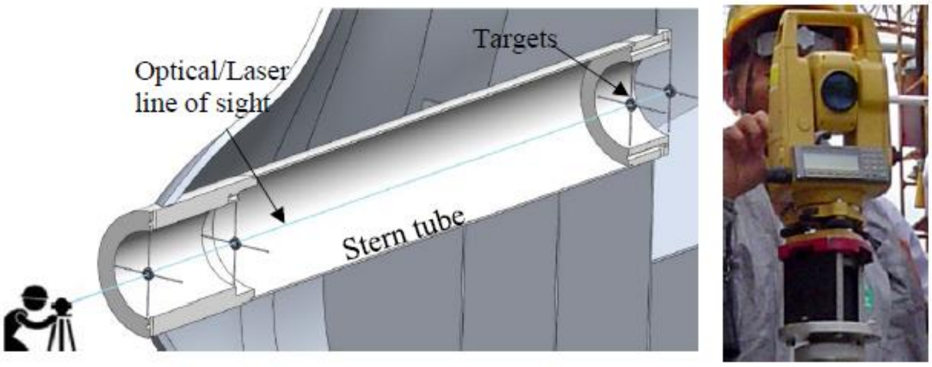

Figure 88: Optical Telescope Method: Instrument and Process depiction, ABS [12]

2.5. Laser Method

According to the Laser Method, a laser transmitter is placed in the precise center of a “reference diameter”. Consequently, the reference targets are defined, and a receiver target is set. While the process takes place, in order for all of the desired measurements to be conducted, the receiving target is moved in the positions of interest, always along the reference line. When the process is completed, the results, which are recorded digitally, can be presented in table or graph form.

Figure 89: Laser Method, ABS [12]

At this point, the main advantages and disadvantages of the three aforementioned methods are briefly presented:

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Table 6: Main Benefits and Drawbacks of Sighting Through Methods

Sighting Through Method:	*Benefits:*	*Disadvantages:*
Piano Wire Method	Low-Cost method. Easy to Applicate. Shipyards have several years of experience on this specific method.	Excessive need for precise measurements using a micrometer, thus a skilled worker is required. The need to take into account the natural or forced sagging of the wire makes the process more demanding. Measurement errors which occur due to the usage of Micrometers and other measurement instruments. Wire vibrations, the condition of the surfaces or other external factors may lead to errors in the measurement procedure.
Optical Telescope Method:	Easy-to-set up and conduct method. Easy to read results. High accuracy (of about ±0.01 mm).	The process is highly impacted by the precise locations at which the target point are set to, thus requiring high accuracy from the worker responsible. The telescope's leveling should be highly accurate.
Laser Method:	Measurements of high accuracy (of about ±0.005 mm). The results are extracted faster and easier than they do if other methods are utilized. Continuous real-time data acquisition through Computers. Automatic calibration of the sensor unit. Elimination of human errors thanks to the leading role of technology.	High-Cost method. Requires specialized staff, which usually cannot be found in a Shipyard's employees. The final results and measurements are significantly affected by the proper setting of the receiver and the transmitter units. The method is sensitive to temperature, noise and vibration alterations.

2.6. Pre-Sighting

During the Shaft Alignment procedure, it is of crucial significance to define the tolerances regarding the stern tube casting, as the stern tube is not an easy to access location, and consequently, all of the measurements, dimensions and procedures shall be precisely calculated to be in compliance with the design stage, so that undesirable errors and failures are avoided.

More Precisely, Pre-Sighting is the phase of the Shaft Alignment procedure stage, which takes place before the installment of the bearings, and aims in determining the machining tolerances of the stern tube casting.

It is important to highlight, that in cases where a Bearing Inclination method which engages epoxy resin choking is applied, the stern tube casting is machined in a larger diameter, as it has to be able to house the stern tube

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bearings as well as the surrounding epoxy. Consequently, the machining tolerances do not have such a significant impact, thus making the Pre-Sighting procedure optional (ABS, 2019).

The process of Pre-Sighting is followed by the machining procedure of the Stern Tube Casting.

2.7. Stern Tube Bore Sighting and Final Sighting

When the stages of Pre-Sighting and Machining of the Stern Tube Casting are completed, it is important for the bearing manufacturer to make some final adjustments to the bearing's outer diameter. For this to happen, precise measurements shall be taken, by using a Micrometer, in order to record and verify the final inner dimensions of the stern tube. It is a common industry practice for the Shipyard's personnel to conduct a verifying sighting procedure, when the vessel is in Dry Dock, and before the bearings are installed, in order for the aforementioned to be defined and for the tolerances (regarding the machining of the outer diameter of the stern tube bearing) to be determined.

ABS (2019)

According to ABS (2019), it is mandatory for "a surveyor to witness sighting of the stern tube before bearings are fitted".

Consequently, when all of the above are completed, and the bearings as well as the Main Engine and other significant components of the Propulsion Shaft are installed, a Final Sighting procedure is conducted. During this phase of the Shaft Alignment Procedure, some of the most important parameters of interest are verified to be in compliance with the calculated and desirable values. As ABS (2019) states, these parameters have as follows:

- The fitted position of the stern tube bearings is confirmed, with respect to the engine's position and the other shaft line bearings' positions,
- The slope boring misalignment angle is confirmed, with respect to the established reference line,
- The horizontal misalignment of the stern tube bearings and the main engine are confirmed,

At this point, some extra attention shall be drawn to the process of installing the intermediate shaft bearing. More specifically, in cases where both aft and forward stern tube bearings are present, the intermediate bearing is not fixed to its position during the Final Sighting procedure, as they are removed when the time comes for the propeller to be fitted, and consequently, they are re-installed. However, in cases where there is no forward stern tube bearings, the procedure differs. In such cases, the aft most intermediate bearing is utilized as a "control point" for the misalignment angle of the aft stern tube bearing. Its offset is verified during pre-sighting and final sighting and compared to the calculated value.

It is worth mentioning, that in Enhanced Shaft Alignment Guide, ABS (2022) states that the bore sighting procedure should be commenced prior to the fitting of the bearings. The prementioned is not required in cases where the aft stern tube bearing is installed by resin chocking. The bore sighting procedure of the stern tube bore should verify the following:

- The stern tube bore actual dimensions: in order to define dimensions and tolerance for the aft stern tube bush outside diameter's machining.
- The stern tube bore actual misalignment or offset, if any, vertical or horizontal; in order to define angular corrections for stern tube bearing outside diameter's machining.
- Whenever applicable, it is recommended that all subsequent corrections are done by machining the outside bush diameter, rather than correcting the stern tube bore.

Moreover, a bore sighting procedure after the installation of the stern tube bearings is required, so that the following are defined:

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- The aft bush slope, as installed. The measurement is to be taken with reference to the forward stern tube bush.
- The horizontal misalignment between the aft and the forward stern tube bearing.

Additionally, ABS (2022) in Enhanced Shaft Alignment Guide requires that sighting readings are taken from the aft end of the stern tube, in order to verify that the aft and forward stern tube are correctly aligned relative to each other.

DNV (2021)

DNV's Enhanced Shaft Align (1) Guidelines (2023) state that "Laser aided sighting of vertical and horizontal offsets of the stern tube housing in way of the bearings (Stern Tube bearings) shall be submitted for review by the attending surveyor. A minimum of 5 reference points shall be used covering the aft most bearing housing and 3 reference points for the forward bearing housing. Moreover, DNV (2023) at Enhanced Shaft Align (1) states that*: "The laser reference line shall be made concentric with the stern tube and independent of the bearings".* Last but not least, at the Guidelines regarding the Enhanced Shaft Align (1), DNV (2023) suggests that alternative measurement methods can be used during the sighting procedure, but only after equivalent accuracy is verified by the Classification Society.

2.8. Slope Boring

The most common industry practice is for the Slope Boring to be applied by the bearing's manufacturer. According to ABS (2019) the acceptable tolerances regarding the slope boring, are set to a maximum value of ±0.1 mrad. The most common methodology of applying a Slope Boring, is for the slope to appear only in the inner diameter of the bearing. This practice facilitates the replacement of a worn or damaged bearing, as well as significantly reduces the cost of the procedure. In cases where refurbishment or alterations are needed concerning the bearing's fitting, the slope of the aft stern tube bearing may be machined in the shipyard's facilities. However, the pre-mentioned is not considered to be highly desirable, as the process is highly impacted by the temperature, the environmental conditions, the accuracy and expertise of the workers and other parameters.

2.9. Bearing Inclination

In cases of Epoxy Resin chocking, the bearing's inclination is applied prior to the chocking procedure and the misalignment angle is verified during the Final Sighting process. When the application evolves a double slope design, the bearing inclination is usually conducted by the bearings' manufacturer. There are two proposed methods of Epoxy Resin Chocking: Either individually chocking the aft and forward stern tube bearing or chocking the whole stern tube altogether. The Bearing Inclination method is considered to be characterized by many benefits. As ABS (2019) describes them, they have as follows:

1. Pre-sighting is not required,
2. Stern tube bore does not require machining,
3. Stern tube bearing outer diameter machining is not required

On the other hand, the epoxy resin's presence reduces the heat conductivity between the structure of the stern and the bearing, which according to ABS (2019) is considered to be a remarkable disadvantage of the Bearing Inclination method. Moreover, the sealing mechanism which is going to be used in the stern tube outboard area, must ensure water tightness. However, if properly addressed, the pre-mentioned should not be of concern. According to ABS (2019) the Surveyor's presence is mandatory only in the final slope boring or bearing inclination verification.

2.10. Tail Shaft Bearing Clearances, (ABS)

As ABS (2019) in the Guidelines regarding Shaft Alignment states, the clearances of the bearings of the tail shaft should be measured prior to the fitting of the propeller. At this stage, the clearances of the bottom

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parts of the bearings is expected to be greater than zero. Once measured, the clearances should be compared to the calculated values for this specific condition of the tail shaft. When the propeller is fitted, the clearances must be remeasured for verification reasons, and at this instance, the clearances of the bottom parts of the bearings is expected to be equal to zero.

Additionally, in Enhanced Shaft Alignment Guide, ABS (2022) requires that the measurements of the clearances of the tail shaft, which are taken with the propeller uninstalled and the tail shaft fitted but unrestrained on the forward flange, lie within ±0.05 mm of the calculated values. After the installation of the propeller shaft in stern tube together with forward and aft seal assemblies, measurements shall be taken regarding the bearing clearances at the ends of the aft and forward stern tube bearings (at 4 positions, 90 degrees apart). The bottom clearances must be in compliance with the approved calculation report which was submitted to ABS. If this is not the case, the propeller shaft must be pulled out and corrective actions should be taken.

2.11. Sag and Gap Procedures

The main purpose of the sag and gap procedures is to bring the actual values of the sag and gap measurements as closer as possible to the values which were calculated in the design stage of the Shaft Alignment. The pre mentioned shall only be utilized as a counterbalancing method to the jack-up measurements regarding the fine tuning of the bearings in terms of Bearing Reactions and Offsets. However, great effort should be made in order to verify the compliance of the Sag and Gap values to the calculated ones. The adjustment of the Sag and Gap values is accomplished via the altering of the height of the lineshaft supports, as the adjustments takes place prior to the final assembly of the Shafting, in an open flange system. Hence, the procedure usually entangles various temporary supports as well. According to ABS (2019), the acceptable tolerances between the calculated and the measured Sag and Gap values is set to ± 0.1 mm. Moreover, ABS (2019) states that in cases where a possibility of the tail shaft not sitting on the bottom of the forward stern tube bearing exists, a jack down force is applied, which is precisely calculated based on the unique conditions of each situation.

At this point, it is important to note that in order for the calculated values to be compared in a valid way with the measured values, some specifications and details should be accounted for and stated. More precisely, according to ABS (2019), the sign convention and the exact location and measurement method which are used should be addressed.

Gap values are defined based on the difference between the top and bottom edges of a pair of uncoupled flanges. "Gap at each flange is calculated from the angular inclination of the shaft at flange location and the flange diameter. Total gap is obtained by linear summation of the gaps at both flanges". (ABS, 2019)

"Sag is calculated by taking the bending displacement at each flange location and subtracting that from the deflection of the mating flange" (ABS, 2019)

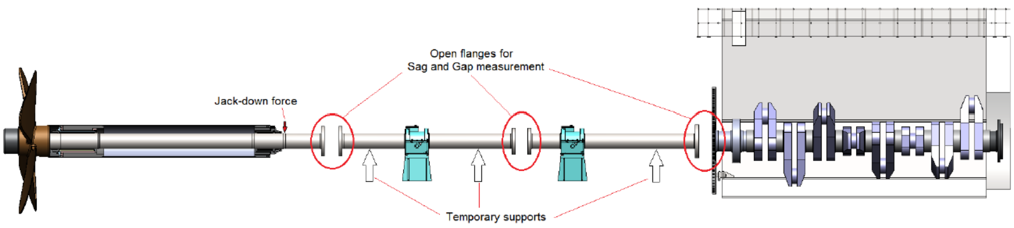

Figure 90: Visualization of the Sag and Gap procedures, ABS [12]

ABS (2019)

According to ABS (2019) the precise details of the shafting system during the phase of the calculation of the Sag and Gap values should be defined, regarding the Sag and Gap data, the locations of the temporary supports

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and the condition of the vessel. Moreover, it is mandatory that the Sag and Gap values should be measured in dry dock condition or after launching in a light afloat condition.

Note: According to ABS (2019), in cases of vessels with no forward stern tube bearing, the sag and gap values between the intermediate and tail shaft should not be verified.

However, ABS (2022) in Enhanced Shaft Alignment Guide, states that after the installation of the intermediate shaft is completed (and after the fine tuning of the system is also completed by altering the positions of the intermediate shaft and its temporary support with respect to the measurements taken from the main engine), sag and gap measurements should be taken at both flanged connections. The measured values should then be compared to the ones calculated in the report which was approved by ABS. The limits regarding the deviations of the sag and gap values are the same as previously discussed in this Thesis.

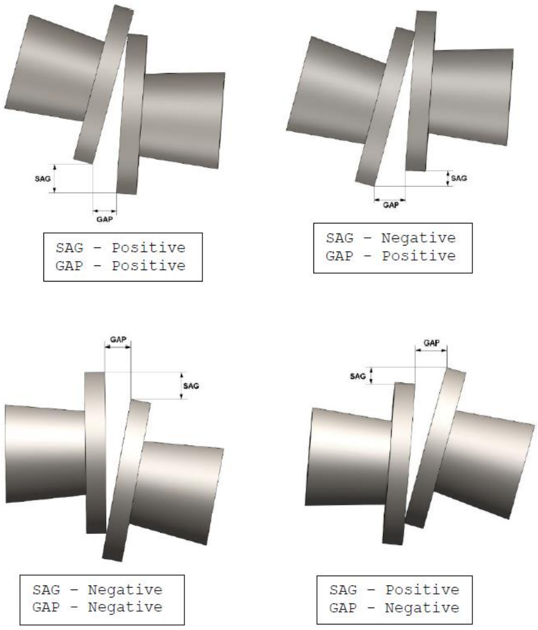

Figure 91: Four different positions of interaction between mating flanges, as defined by ABS [12]

BV (2015)

Additionally, BV (2015) states that in cases where the Sag and Gap method is utilized in order to fine-tune the shaft's bearings, the requirements have as follows:

1. Sag and Gap values should be calculated and submitted in the calculation report
2. Tolerance for Sag and Gap values attained on board should be typically within a range of 0.05 mm or less compared to the values calculated during the Design Stage.

2.12. Engine Plate Pre-Sagging

The Pre-Sagging of the Engine's Plate is a common industry practice which is usually utilized in order to minimize the effect of thermal deviations during the running-hot operating condition of the vessel. As previously explained in this Thesis, Thermal Deviations could be analyzed in two different deflections. One of them is characterized as a hogging deformation which is significantly more intense in the places of the engine's space where the temperatures are higher. The Engine's bedplate is a structural element which is impacted by the prementioned. The hogging of the bedplate is treated by taking advantage of the "Engine Plate Pre-Sagging" method. The Pre-Sagging is applied according to the guidelines and recommendations of the Engine's manufacturer.

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2.13. Crankshaft Deflections Measurements

According to ABS (2019), measurements concerning Crankshaft's deflections are conducted with the main purpose of verifying that the stress limits regarding the Crankshaft are within the limits as they are specified by the Engine's Manufacturer. The Deflections of the Crankshaft are expressing the magnitude of the adjacent web's opening during a crankshaft's full revolution. Most manufacturers prefer for the crankshaft's deflections to be as close to zero as possible, for the vessel in service condition, loaded and with a hot engine. Moreover, they provide the sign conventions regarding the measurements.

2.14. Ship Afloat Condition (BV)

According to BV (2015), with the vessel in afloat condition, the following shall be performed in order to assure that the hydrostatic pressure and the hull deformation it causes are properly taken into account:

1. Intermediate bearing positioning
2. Prime mover positioning
3. Shaft bolting
4. Load tests

Moreover, regarding the prime mover, BV (2015) states that during the pre-positioning of the Gearbox or the prime mover, the vertical and horizontal offsets should be measured and compared to the values of the Shaft Alignment calculations. If the measured values do not match the calculated/assumed ones, they should be adjusted "according to the values determined by elastic alignment study".

Additionally, with the vessel afloat, BV (2015) recommends that the bearings should be placed. In cases where the sag and gap method is utilized in order to adjust the shaft bearings, the upcoming should be followed:

- Gap/sag values should be calculated and submitted in the calculation report
- Tolerance for sag/gap attained onboard should be typically within a range of 0.05 mm or less compared to the calculated values

The recommendations BV (2015) makes regarding the Bearings' Reactions are presented in the following chapter.

2.15. Bearing Reactions

The reaction forces on the bearings are one of the most crucial parameters of the Shaft Alignment Design and Procedure and they indicate the successfulness of the Shaft Alignment. However, the bearings' reactions are significantly impacted by numerous variables such as the Hull Deflections, the Thermal Deviations, the flexibility of the bearings' foundations and the offsets of the bearings.

ABS (2019)

As a result, it is of high significance that the measurements of the bearings' reactions correspond to the values calculated from the Shaft Alignment model. According to ABS (2019), the prementioned measurements are conducted from the Shipyard personnel, with the vessel on light afloat condition. In order for this verification to take place, all of the accessible bearings are measured, using the jack-up method. Afterwards, the measurements are plotted in a jack-up diagram in order for the actual values of the bearings' reactions to be determined. At this point, it is important to note that according to ABS (2019), the measured values are to be confirmed once again during the sea trials, in hot static condition.

As far as the limits are set regarding Hull Deflections, ABS (2019) suggests that all of the bearing's reactions must never exceed the limits set by the manufacturer and by the documentation of the Shaft Alignment calculations.

Regarding the other of the system's bearings, ABS (2019) indicates the following:

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- “If Hull Deflections are considered in the design, measured bearing reactions must be within ±20% of the calculated values and must not exceed ±80% of the manufacturer’s maximum allowable limit.”
- “If Hull Deflections are not considered in the design, measured bearing reactions may not exceed 80% of the manufacturer’s maximum allowable limit.”

Consequently, it is mandatory, that prior the evaluation of the values extracted from the jack-up diagram, the Surveyor must discuss with the builder in order to determine whether Hull Deflections were taken into consideration in the design or not (ABS, 2019). Moreover, an estimation must take place, in order to predict the behavior of the bearings’ reactions when the vessel operates in fully ballasted or fully laden condition (ABS, 2019). Last but not least, if some bearing reactions are close to zero or close to the upper limit, the Surveyor and the builder should determine whether during increased draft and operating conditions problems will occur (ABS, 2019).

However, ABS (2022) in Enhanced Shaft Alignment Guide, states that: when the vessel is in Drydock (or very Light Ballast Condition), and after the fine tuning of the system by altering the intermediate bearing and its temporary support offsets is completed, then the bearings reactions must be measured at 4 rotational angles 90 degrees apart from each other (for each one of the bearings). If the measured values deviate more than 20% from the calculated ones, the intermediate bearing’s offset can again be utilized as a fine-tuning mechanism, until acceptable load distribution is achieved. Additionally, as ABS (2022) states: *“Alternatively, if any of the measurable bearings is lightly loaded per the calculations, the above percentage is superseded, and the measured reaction is to be within ±5 tons”.* The above-mentioned measurements are to be conducted with the engine on cold condition and the aft peak tank empty, as well as with the engine in hot condition and the aft peak tank full (as permitted by the vessel’s loading manual). After all of the above are managed, the chocking of the main engine may be conducted, under the guidance of the resin chock engineer and as per the approved engine chock calculations.

BV (2015)
As BV (2015) suggests, the final verification of the bearing reactions shall take place in afloat condition after the shaft is bolted. The load checks should be performed in the accessible bearings by utilizing the Jack-Up method. BV (2015) suggests that, is possible, load checks should be performed for:

- The forward bush
- Intermediate bearings
- Aft gearbox bearings (journal type only)
- Three aftmost main engine bearings

All of the calculation and the correction factors used must be submitted in the report (BV, 2015).

LR (2023)
According to Lloyd’s Register (2023), the builder is responsible for ensuring that the position and the construction of the bearings works in a way which enforces the minimization of the effects of hull deflections under any of the ship’s operating conditions, under the scope of optimizing the bearing load distribution. In cases of bearings constructed of synthetic materials, evidence shall be provided in order to verify that the tolerances stated by the manufacturer are being met, both prior and after the installation. Additionally, the forward stern tube bearing static load *“is to be sufficient to prevent unloading in all static and dynamic operating conditions, including the transient conditions experienced during maneuvering turns and during operation in heavy weather”* (LR, 2023).

DNV (2021)
According to DNV (2021), the measured sag and gap values and/or the jacking loads with force displacement diagrams and/or alternative verification data should be reported in presence of the surveyor.

ClassNK (2006)

ClassNK (2006) also suggests that the bearings' loads should be verified by utilizing the Jack-Up method.

2.16. Jack-Up measurements

The Jack-Up measurement method is one of the most widely used methods in the Maritime industry, along with the Strain Gauge measurement method. Contrary to the Strain Gauge method, the Jack-Up method is direct and preferable in cases where the object of measurement is accessible. The principle behind the Jack-Up method relies on the lifting and lowering of the shaft, and measuring the jacking load applied at the jacking location, close to the bearing of interest, which is indicated by the hydraulic pressure which is applied. It is important to note, that the surveyor or the engineer who will assess the jack-up diagram should be properly trained, as each different type of bearing has its own Jack-Up diagram in terms of appearance, which makes adequate training, technical knowledge and expertise mandatory, in order for the assessment to be accurate.

In the following table, the advantages and drawbacks of the Jack-Up method are presented:

Table 7: Benefits and Drawbacks of Jack-Up method

Jack-Up Method:	
Advantages:	**Drawbacks:**
Directly calculates the bearing's reaction	Time Consuming: The entire preparation procedure must be conducted for each different object of measurement
Widely Used method in the Maritime industry, preferred in cases where the objects are accessible	Requires correction coefficients, in order to minimize the impact of the difference between the measured value (close to the bearing) and the actual value (center of the bearing)
	Different appearance of diagrams for different types of bearings: The responsible engineer must be cautious
Accuracy and hysteresis can be improved (using of load cells)	Reduced accuracy and wide hysteresis may occur due to several factors (e.g. misalignment of the hydraulic jack). A measurement may need to be repeated several times
Simple and ready-to-use equipment, easy installation	Can't be used for measurements in inaccessible bearings.
Low-cost Method	

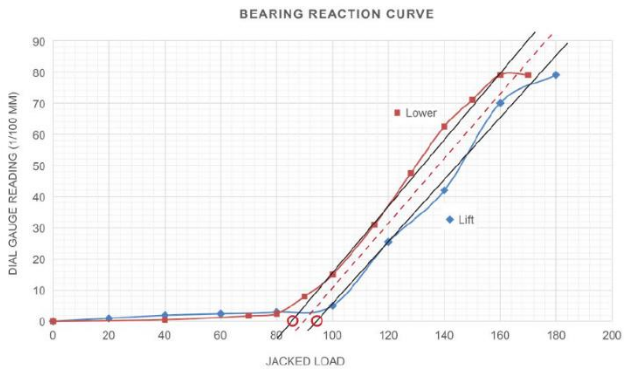

Figure 92: A typical Jack-Up measurement diagram, ABS [12]

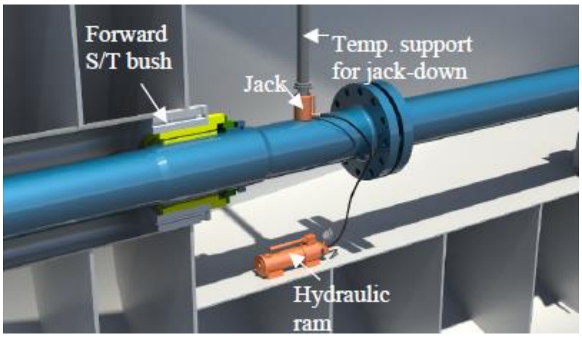

Figure 93: Jack down force application, ABS [12]

2.17. Strain Gauge Method

The Strain Gauge method is generally preferred in cases where the objects of measurement are not accessible. This method incorporates a reverse engineered logic, which is based on the measurement of the vertical and transverse loads of the bearing. The principle is based in the following procedure:

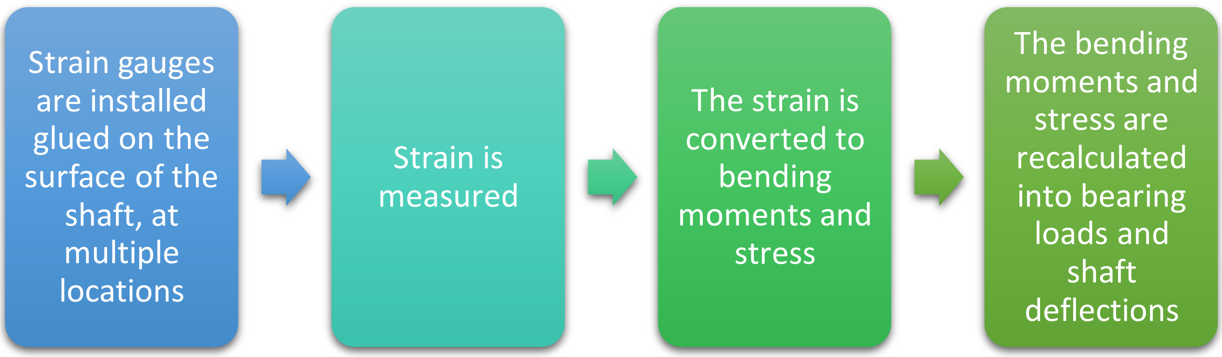

Figure 94: Strain Gauge measurement method: basic principles

It is of vital importance to highlight the fact that the Strain Gauge method requires increased installation time compared to the Jack-Up method, however, once installed, measurements are easy to repeat.

Moreover, this method requires personnel of high expertise and advanced knowledge, and the installation is more difficult.

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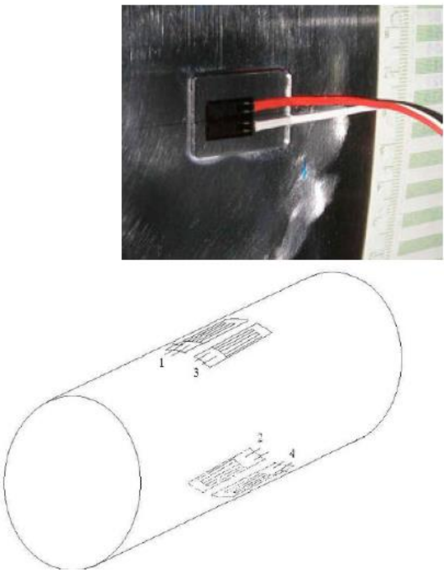

Figure 95: Strain Gauge installation, ABS [12]

According to ABS (2019), for installations with one intermediate bearing, at least three strain gauge locations are recommended. Moreover, ABS (2019) suggests that two pairs of strain gauges are glued in pre-selected locations 180 degrees apart. Last but not least, ABS (2019) requires, that: “Bearing loads obtained by the strain gauge method should be reconfirmed on accessible bearings with the jack-up method.”

In the following table, some of the advantages and disadvantages of the strain gauge method are presented:

Table 8:Advantages and disadvantages of the Strain Gauge method

Strain Gauge Method:	
Advantages:	**Drawbacks:**
Traditionally highly accurate in cases where the quality of the equipment and the installation is adequate, the reverse engineering model is highly developed, and the parameters used in the analytical model are accurate	Time Consuming: The installation procedure takes time and requires high expertise personnel
Suitable for installations where no accessibility can be granted, where the parameters of interest can be calculated through reverse calculation methods	The equipment as well as the entire installation is significantly sensitive, expensive and sophisticated. Requires experienced personnel.
Provides data about both horizontal and vertical loads of bearings	
Once installed, repeated measurements are easy	Accuracy depends on analytical model utilized during reverse calculations

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If strain gauges are installed with telemetry system, information, data and performance characteristics can be monitored during the vessel's operation	The method shall always be used in conjunction with the Jack-Up method (ABS, 2019)

2.18. Intermediate Bearing Offset

The intermediate bearing and its offset have a significant impact on several parameters of the Shaft Alignment process, as explained in the previous chapters of this Thesis. As a result, it is commonly utilized as a way of fine tuning the entire propulsion system in terms of Shaft Alignment. As previously discussed, small alterations in the offset of the Intermediate Shaft Bearing results in alterations in the Main Engine's bearings, the Crankshaft's deflection or the Forward Stern Tube's reaction. However, in systems with the characteristics of the traditional designs of Oil Tankers or Bulk Carriers (short and rigid shafting with one intermediate bearing) the problem of optimization of the bearings' reactions by adjusting the intermediate bearing's offsets becomes extremely complex, due to the sensitivity of the system.

At this point, the importance of the differences between a design with no Forward Stern Tube bearing and a "Traditional" design which incorporates both Stern tube bearings, must be highlighted.

ABS (2019)

More specifically, according to ABS (2019), in designs with both Aft and Forward Stern Tube bearings, the misalignment angle of the Aft Stern Tube bearing is less sensitive to alterations of the intermediate bearing's offset, which results in a more "reliable" and "steady" system. On the other hand, as was extendedly discussed in previous chapters, the system is more susceptible to unloading of the bearings due to hull deflections.

Contrary to the prementioned, ABS (2019) states that a system with no Forward Stern tube Bearing installed is more flexible and thus less susceptible to hull deflections. Moreover, in practical terms, the misalignment angle of the Aft Stern tube bearing can be modified before the propeller fitting procedure. As far as the disadvantages of this system are of interest, it should be highlighted that the misalignment angle of the Aft Stern tube bearing is directly affected by the alterations made in the intermediate bearing's offset, which may lead to unloading of the aft stern tube bearing, rendering this practice as both helpful and risky.

ABS (2019) suggests that after the offset adjustments are completed, gear tooth contact inspections should be conducted. These inspections must take place prior to the chocking of the gearbox, in order for the proper functioning of the gear and pinion to be verified. ABS (2019) requires that the maximum value of the relative misalignment between the gear and the pinion does not exceed 0.1 mrad. This limit is set due to the increased sensitivity of the gearbox bearings, as they are placed very close to each other. The gearbox misalignment is difficult to measure. However, one of the most common industry practices is the Gear Contact Evaluation with Dye method, during which the pinion teeth coated with a contrasting dye and a full revolution of the gears is made. An acceptable misalignment state is considered to be one in which the results of the Gear Contact Evaluation *show "visible markings across 90% or more of the face width of the gear teeth"* (ABS, 2019). According to ABS (2019), a surveyor is required to attend and record the gear tooth contact in cases of propulsion gear units larger than 1120 kW or 1500 HP.

Additionally, ABS (2022) in Enhanced Shaft Alignment Guide, states that in designs with no forward stern tube bearing, the intermediate bearing should serve as a reference point in means of sighting procedures. However, the responsible Surveyor must verify that the intermediate bearing's offset is not changed at all after the bore sighting is completed. The entire installation procedure for Enhanced Shaft Alignment according to ABS (2022) has as follows:

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The intermediate bearing is placed, and its vertical offset is defined based on the sighting process. Then, the Main Engine bearings' offsets are also set, based on sighting only. Consequently, the bearing bore sighting procedure is utilized as a way of verifying the stern tube's bearing slope and the vertical offsets of the intermediate bearing and of the Main Engine bearings. In order to avoid deviations regarding the misalignment of the aft stern tube bearing, the offset of the intermediate bearing should be adjusted to match the calculated value. As ABS (2022) states: *"This offset (vertical distance from the foundation) should be measured by a calibrated micrometer at several (minimum 4) points and recorded".* After that, in order for the propeller shaft to be fitted, the intermediate bearing is removed. Once the installation of the propeller shaft is conducted, the intermediate bearing must be placed in the exact same position as before, and its offset should not change after this point. Then, the propeller shaft and the intermediate shaft are assembled. If Sag and Gap measurements are taken and utilized, it should only be for reference and verification purposes and no alterations must be made based on them in order to tune the intermediate bearing's offset. Afterwards, the following steps must be taken according to ABS (2022):

1. *"Verify shaft bearing reactions at 4 (four) rotational angles at 90 (ninety) degrees apart. If any of the bearing reactions are more than plus or minus 20% off from the calculated value, do not adjust the intermediate shaft bearing offset. Instead use the engine mounting system to adjust the engine bearing offsets until an acceptable bearing load distribution is achieved, that is, before the Main Engine is choked. Alternatively, if any of the measurable bearings is lightly loaded per the calculations, the above percentage is superseded, and the measured reaction is to be within ±5 tons. This measurement is to be conducted with the engine cold and the aft peak tank empty and with the engine hot and with the aft peak tank full or as filled up as permitted by the vessel's loading manual".*
2. *"Only once the bearing reaction measurements are satisfactory should the engine be choked. In this case, no further adjustments may be possible in subsequent bearing reaction checks in light/full ballast/fully laden conditions unless the engine remains un-chocked. However, the bearing reaction measurements are still required to be recorded for these conditions".*

LR (2023)

According to LR (2023), *"Intermediate shaft bearings' loads are not to exceed 80 percent of the bearing manufacturer's allowable maximum load, for plain journal bearings, based on the bearing projected area*".

2.19. Run-In Procedure, (ABS)

The Run-In Procedure is usually conducted prior to the Sea Trials. As ABS (2019) requires, a surveyor must witness the Run-In Procedures in vessels which incorporate a design with no Forward Stern tube bearing or a design with a double sloped aft stern tube bearing. As far as the Shaft Alignment process is of concern, the Run-In Procedure helps preparing the shafting system for full operational conditions by slowly and gradually exposing the aft stern tube bearing to operational loads.

Regarding the Run-In Procedure, ABS (2019) requires that:

"*Before commencing the Run-In procedure, the vessel is brought to deeper waters in open sea, anchored and ballasted to the approved sea trial draft. If the vessel proceeds from the Shipyard for sea trials in a light ballast condition with a partially immersed propeller, then low engine RPM and minimum rudder angle movements are recommended."*

2.20. Sea Trial Stage

The Sea Trial Stage is the last stage of the Shaft Alignment procedure, as the vessel is tested for its seaworthiness. All of its systems, machinery and structural components are monitored for malfunctions or unwanted behaviors.

ABS (2019)

ABS (2019) requires that during the Sea Trial stage, confirmatory measurements shall be taken regarding the Bearings' Reactions in specific loading conditions. Moreover, *"The Surveyor should review the vessel's pre-sea trial transit draft condition and the corresponding amount of propeller immersion. In addition, the Surveyor also confirms the operational functionality of shaft bearing temperature, stern tube oil, and cooling sea water sensors."*

In cases where vessels with no forward stern tube bearing present problems regarding the bearings' reactions during the Sea Trial phase, modifications in the intermediate bearing's offset are not allowed according to ABS (2019). If drastic measures and corrective actions are needed, they should be conducted after the Sea Trial, when the vessel has returned to the Shipyard. Otherwise, in vessels with the traditional design which incorporates both stern tube bearings, adjustments on the intermediate bearing's offset can be made by the Shipyard's personnel.

Additionally, according to ABS (2019), for designs that have not taken under consideration the hull deflections, additional confirmatory measurements on the bearings' reactions shall be taken for at least one additional service condition, as long as the condition is enlisted on the vessel's loading manual. *"The measured bearing reactions may not exceed 80% of the manufacturer's maximum allowable limit for any draft condition."* (ABS, 2019)

During the Sea Trial stage, according to ABS (2019) the following should be closely monitored:

- Aft Stern Tube bearing temperature, during any test requiring higher rudder angles (zig-zag maneuvering, turning circle etc.)
- During Starboard maneuvers on systems with a clockwise rotating propeller or Port turns for counterclockwise propellers, the Surveyor should pay close attention to the stern tube and intermediate bearing temperatures.

ABS (2019) requires the following regarding the First Vessel in the Series in terms of Sea Trial measurements and verification of the Bearings' Reactions:

First Vessel in Series:
Confirmatory bearing reaction measurements at the drydock or lightship condition and in a selected service draft condition
If hull deflections are accounted for, the measured bearing reactions at the drydock or lightship condition must be within ±20% of the calculated values and may not exceed 80% of the limits set by the manufacturer.
When measured reactions at drydock or lightship condition are found outside of ±20% of the calculated values, adjustment of intermediate or main engine bearings is considered an option. Revised Shaft Alignment calculations should be submitted.
When the criterion for ±20% deviations in the values of the bearings' reactions cannot be met, the acceptance criteria regarding designs which do not take under consideration hull deflections shall be applied.

Figure 96: Requirements regarding Bearing Reaction measurements and verification for First Vessel in Series, ABS [12]

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ABS (2019) requires the following regarding Subsequent Vessel in the Series in terms of Sea Trial measurements and verification of Bearings' Reactions:

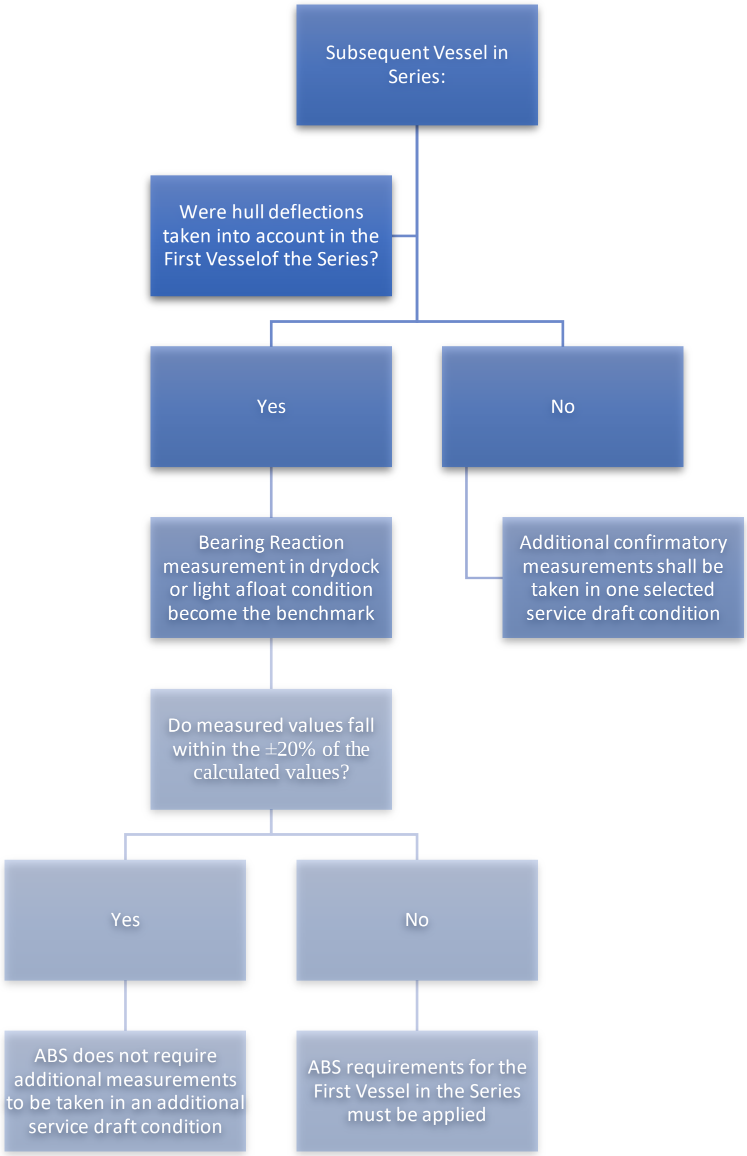

Figure 97: Requirements regarding bearing reactions measurements and verification for Subsequent Vessel in Series, ABS [12]

Additionally, ABS (2022) in Enhanced Shaft Alignment Guide, requires that during the Sea Trial Stage and with the vessel in Light Ballast and Full Ballast conditions, and the main engine chocked, the following must be done:

The crankshaft's deflection must be measured in order to verify compliance with the manufacturer's limits. Moreover, the shaft bearing reactions must be measured at 4 rotational angles at 90 degrees apart. "If any of the shaft bearing reactions are more than plus or minus 20% off from the calculated value, adjust the intermediate shaft bearing(s) offset until an acceptable load distribution is achieved. Alternatively, if any of the measurable bearings is lightly loaded per the calculations, the above percentage is superseded, and the measured reaction is to be within ±5 tons. This measurement is to be conducted with the engine cold and the aft peak tank empty and with the engine hot and with the aft peak tank full or as filled up as permitted by the vessel's loading manual" (ABS, 2015).

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When it comes to the Fully Laden condition during Sea Trials, ABS (2022) in Enhanced Shaft Alignment Guide requires the following for oil carriers only:

The cargo tanks must be filled up with water, in order to simulate as closely as possible, the fully laden condition. The aft peak must also be filled as much as possible and in accordance with the vessel's loading manual. Then, the 1st step of the prementioned procedure must be conducted: The crankshaft's deflection must be measured in order to verify compliance with the manufacturer's limits. Moreover, the shaft bearing reactions must be measured at 4 rotational angles at 90 degrees apart.

"If any of the shaft bearing reactions are more than plus or minus 20% off from the calculated value, adjust the intermediate shaft bearing(s) offset until an acceptable load distribution is achieved. Alternatively, if any of the measurable bearings is lightly loaded per the calculations, the above percentage is superseded, and the measured reaction is to be within ±5 tons. This measurement is to be conducted with the engine cold and the aft peak tank empty and with the engine hot and with the aft peak tank full or as filled up as permitted by the vessel's loading manual" (ABS, 2015).

Additionally, according to ABS (2022), during the Sea Trials stage, a Test Procedure must be conducted. During this procedure, a set of parameters must be monitored closely at about 5-minute intervals. The parameters of interest should be: The temperature of all of the shaft bearings (including the three aftmost main engine bearings) as well as the rudder angle in degrees and the shaft's rotational speed in RPM. The records must then be handed to ABS for record purposes and future reference. The data should be trend plotted. An example is given in the figure below.

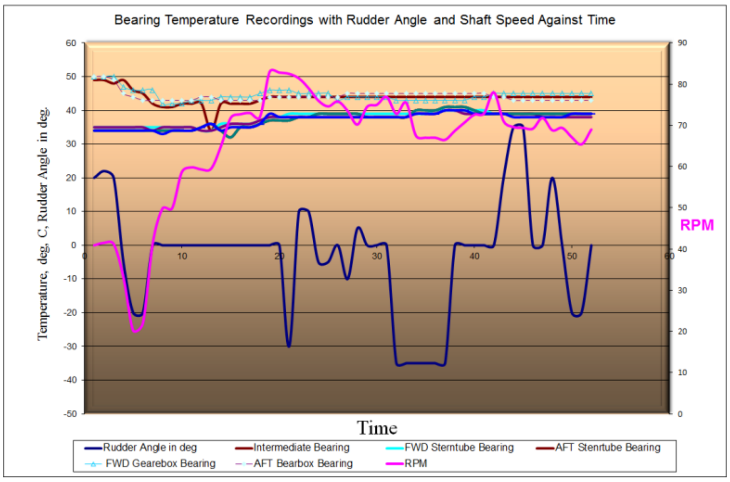

Figure 98: Bearing temperatures monitoring, ABS [13]

BV (2015)

Regarding the Sea Trial stage, BV (2015) suggests that the following courses should be tested:

- Straight
- Zig-zag
- Turning

"Test sequences should be sufficiently spaced in time in order to permit necessary dissipation of heat generated by the succession of severe loadings, thus avoiding overheating of stern bearings" (BV, 2015).

During the Sea Trial, the lubricating oil flows, bearing temperatures and vibration signs should be closely monitored (BV, 2015).

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DNV (2021)

According to DNV (2021), during the Sea Trial stage, the propeller stern tube bearing lubricant to be used should be the one that is expected to be used under normal operating conditions. During the Sea Trial, the lubricant's viscosity should be confirmed and reported. Additionally, DNV (2021) requires the following:

"Suitable signboards shall be pasted at operating locations in machinery spaces and on the stern tube system tanks reflecting the minimum approoved viscosity of the oil. The signboards shall diferentiate between mineral oil and EAL".

Additionally, to the aforementioned, DNV (2023) in Shaft Align (2) requires that before the Sea Trial stage begins, the lubricant's viscosity should be confirmed to correspond to the approved value. Moreover, a verification of the established sampling point regarding the lubricant shall take place.

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3.SHAFT ALIGNMENT MEASUREMENTS AND MONITORING

After the Design Stage and the Installation Procedure of the Shaft Alignment Process are completed, the precise ways of monitoring and confirming the desired values of the parameters of interest during the vessel's life cycle, as well as during the construction of the vessel, should be described. The aforementioned process is defined as the Measurements and Monitoring stage of the Shaft Alignment procedure, and as ABS (2019) states, its scope is the verification of the values of the following parameters:

1. Bearing Reactions
2. Bearing vertical offsets
3. Misalignment angles
4. Crankshaft's web deflections
5. Gear Misalignment

ABS (2019) divides the Measurement and Monitoring of the Shaft Alignment in two parts: The Alignment condition confirmation during and after the construction of the vessel, and the long-term shaft alignment condition monitoring. More precisely, ABS (2019) suggests, that by monitoring the performance of the aft stern tube bearing, the overall safety and seaworthiness of the vessel is promoted, and the bearing maintenance can be scheduled upon the specific characteristics and behavior of each vessel.

In the following chapter, the precise methods of measuring and monitoring shall be described and explained.

3.1. Bearing Reaction Measurements

Jack-Up Method

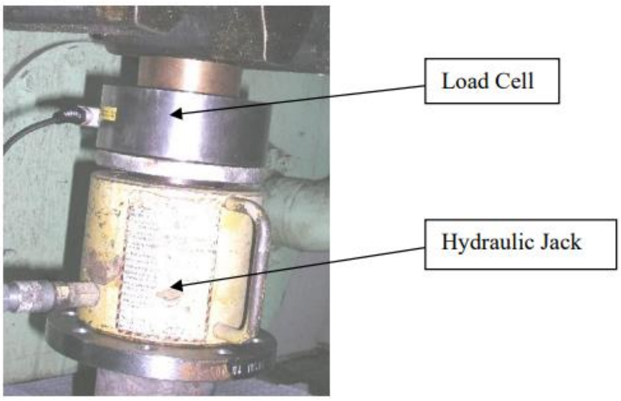

Figure 99: Hydraulic Jack WITH Load Cell, ABS [12]

According to ABS (2019) as well as ClassNK (2006), Jack-Up measurement method is suggested for measuring the Bearing Reactions. As previously discussed, the Jack-Up method is widely established and preferred in the Maritime industry, due to its simplicity, effectiveness and low cost (compared to other methods). Via utilizing the Jack-Up measurement method, one can determine with high accuracy the load of a bearing, as well as one can determine whether the bearing is unloaded. DNV (2021) also lists the Jack-Up measurement method as a valid method regarding the measurement of Bearing Loads. Moreover, DNV (2021) suggests that alternative methods can be used, without specifying them. During the measurements, the results shall be reported with the presence of a surveyor (DNV, 2021).

The equipment needed for the Jack-Up method consists of the Hydraulic Jack, A Dial Gauge with a magnetic stand and, if wanted, a load cell in order to try to eliminate the hysteresis effect. At this point, it is important to note that by the term "Hysteresis", the difference between the loads calculated in the lifting and the lowering process is described. More specifically, during the lifting process of the shaft, due to the friction,

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which is present between the cylinder and the rod, the Jacking loads tend to be remarkably higher than the ones in the lowering process (considering that the external load is the same in both occasions). Consequently, according to both ABS (2022) and ClassNK (2006), in order to eliminate the effect of the prementioned friction and the subsequent hysteresis has on the measurements, it is proposed that an average value should be used in the calculation and measurement process of the Jack-up load measurement.

As far as the procedure is of concern, ABS (2019) suggests that the Hydraulic Jack is placed as close to the bearing as possible and seated upon a stiff base (such as a stiffener which belongs to the double-bottom of the vessel). Moreover, ABS (2019) states that it is important that the Dial Gauge is not based on a flexible structure, as this could have severe impact on the results of the Jack-Up method. Additionally, ClassNK (2006) suggests that:

"In the Jack-Up test, the dial gauge used to measure the jack displacement should be properly secured so that it is affected neither by neither the rise of the shaft nor the deformation of the floor plate".

As ABS (2019) explains, during the Jack-Up measurement procedure, a lifting curve and a lowering curve are produced in order for the load of the bearing to be defined. Moreover, the gradient of the average line between the lifting and lowering curve is defined, as it can serve purposed of verification of the measurement procedure. One of the most important parts of the method is the production and the interpretation of the Jack-Up diagram. The Jack-Up diagram consists of two sections: The Lowering and the Lifting Curve. The Lifting Curve's characteristics have as follows: In the beginning of the lifting of the shaft, when the bearing still supports some of the shaft's weight, the curve seems to slowly increase. When the weight of the shaft is completely supported by the Hydraulic Jack (Breakaway Point), the gradient increases and seems to continue in a constant way (Steeper Curve). Once the Breakaway point is reached, more measurements are taken, whose number is defined in a way which ensures accurate measurement of the curve's gradient.

A typical Lifting Curve of a Jack-Up diagram is presented below:

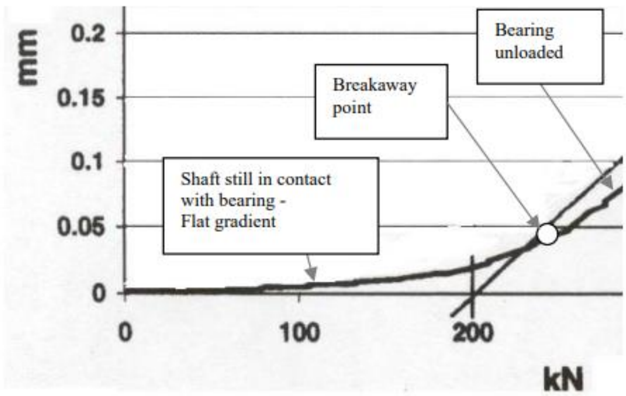

Figure 100: Typical Lifting Curve of a Jack-Up diagram, ABS [12]

Once the lifting of the shaft is completed, the lowering of the shaft takes place. As ABS (2019) suggests, the procedure is identical to the Shaft's Lifting procedure, with a key-difference being noticed on the Lowering Diagram's depiction. More precisely, the Lowering Diagram consists of "two parts": The part where the shaft is lowered but its weight is completely support by the Hydraulic Jack, and the part which begins with the Breakaway point and marks the distribution of the weight between the Hydraulic Jack and the bearing. During the 1st part, the curve is characterized by an almost constant gradient. During the 2nd part, the gradient of the lowering curve changes significantly.

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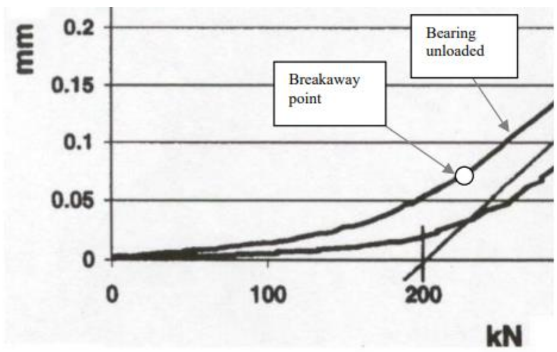

Figure 101: Depiction of the Lowering Curve of a Jack-Up diagram, ABS [12]

Due to the fact that the Hydraulic Jack is placed really close to the bearing, but not in the center of it, a correctional coefficient must be applied in order for the actual value of the Bearing Load to be obtained. As ABS (2019) suggests, the actual Bearing Load is calculated by the equation:

$$\boldsymbol{F_b = C_f * F_j}$$

According to ABS (2019), the correctional coefficient is derived from the design stage of the Shaft Alignment.

According to ClassNK (2006), the coefficient of correction is calculated by the following formula:

$$C = -\frac{I_{BB}}{I_{BJ}}$$

Where:

- I_{BB}: The reaction influence number of the bearing when the jack is regarded as a supporting point
- I_{BJ}: The reaction influence number of the bearing to the jack supporting point

At this point, it is important to remind that ClassNK (2006) by the term “Reaction Influence Number” refers to the widely known Influence Coefficients, as they were thoroughly explained in previous chapters of this Thesis.

BV (2015) requires that the Bearing Load tests are carried out by utilizing the Jack-Up measurement method, after the shaft is bolted and the final load checking are completed. If possible, BV (2015) demands that Bearing Load tests are performed regarding the following bearings:

1. Forward bush
2. Intermediate bearing(s)
3. Aft gearbox bearings (journal type only)
4. Three aftmost main engine bearings

Moreover, the calculation method which is followed by the responsible engineers in order for the correction factor to be determined, as well as the whole measuring procedure in details, should be submitted in the final report of the Shaft Alignment.

Additionally, it is important to highlight the importance of the Gradient Line of the Jack-Up Diagram, as it can be used in order to confirm the accuracy of the measurements. More specifically, as ABS (2019) states: “For the purposes of Jack-Up measurements, the average line between the lifting and lowering curve is used to calculate the gradient”, which is defined as follows:

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$$Gradient = \frac{Force\ Change}{Displacement\ Change}\ \frac{kN}{mm}$$

As one can easily notice, the "definition" of the Curve's Gradient regarding the purposes of Jack-Up measurements describes the influence coefficients, as they were explained in previous chapters of this Thesis.

In cases where a bearing is unloaded, which are undesirable, the Jack-Up diagram will not have a visible Breakaway point, as the weight of the shaft will be immediately supported by the Hydraulic Jack. The lack of the Breakaway point will be present in both the lifting and lowering curves of the Jack-Up diagram. A depiction of a Jack-Up diagram of an unloaded bearing is presented below:

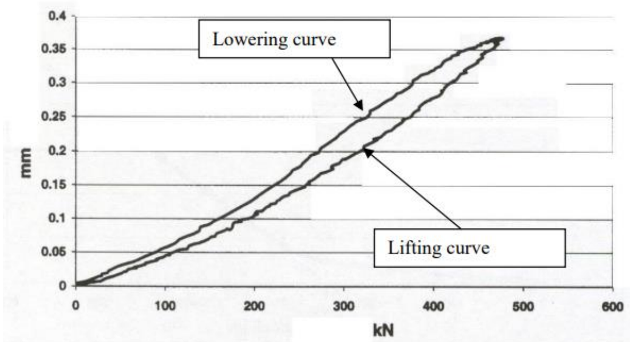

Figure 102: An example of a Jack-Up diagram of an unloaded bearing, ABS [12]

According to Lloyd's Register (2023), the loads of the system's bearings should be verified to be in compliance with the calculated values. More specifically, as Lloyd's Register (2023) states: *"In general the measurements will be carried out by the jack-up measurement technique using calibrated equipment".* Additionally, if the vessel of interest is the first in series, Lloyd's Register (2023) requires that a program of static shaft alignment measurements is carried out in order to verify compliance of the actual system with the one designed during the preliminary stages of Shaft Alignment. The program should focus on matters of Hull Deflections and Temperature Deviations. In order for the verification of the system's conditions regarding the Shaft Alignment, the bearing loads should be measured in several operating conditions, which also depend on the ship type. According to Lloyd's Register (2023), where practicable, the operating conditions should include: Combinations of light ballast cold, full ballast cold, full ballast hot and full draught hot with aft peak tank empty and full.

On the other hand, for vessels which are subsequent in series, if the satisfactory operation of the first vessel of the series is established, a minimum set of parameters of considerations should be submitted and examined. The only limitation lies in the fact that, at least one hot representative condition should be included.

Last but not least, Lloyd's Register requires that in cases where during the design stage of the shaft alignment increased sensitivity (connected with the service condition of the vessel) of the system is established, Shaft Alignment should be verified during the Sea Trial stage via utilizing a Strain Gauge technic approved by the Classification Society.

Strain Gauge Method

The Strain Gauge method is the most popular alternative to the Jack-Up method in the Maritime industry, it is suggested by ABS (2019) and ClassNK (2006) and it presents several benefits as well as drawbacks. The main reason for which it is preferred to the Jack-Up measurement method, is the fact that it can be utilized in cases where the bearings are not accessible. As a measurement method, the Strain Gaige method is based on a reverse engineering logic, which correlates the strain, the bending moments and stress of a beam (shaft) to the loads applied to it. As previously stated, several gauges are glued on the shaft's surface. An excitation

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voltage is applied, and the response voltage is measured and documented. The bending moment of the shaft is calculated by the formula (ABS, 2019):

$$M = E * \varepsilon * W_p$$

Where:

- *M: The bending moment*
- *E: Young's modulus*
- *ε: Strain*
- W_p: *Section Modulus*

The strain may be calculated by utilizing the formulas (ABS, 2019):

$$\varepsilon = \frac{V_0}{V_{ek}} * \frac{1}{k}$$

$$\varepsilon = \frac{\Delta R}{R} * \frac{1}{k}$$

Where:

- V_0: *The measured voltage*
- V_{ek}: *The excitation voltage*
- ΔR: *Change in bridge resistance*
- R: *Initial bridge resistance*
- k: *Bridge Factor (a common value is equal to 2)*

When the initial voltage is applied on the bridge, depending on the flexing of the shaft, the output voltage changes proportionally to the shaft's strain. According to ABS (2019), since Voltage and Resistance are correlated as follows:

$$\frac{\Delta R}{R} = \frac{V_{out}}{V_{in}}$$

The bending moment of the shaft is calculated as presented below:

$$M = E * \varepsilon * W_p = E * W_p * \frac{V_{out}}{V_{in}} * \frac{1}{k} = E * W_p * \frac{\Delta R}{R} * \frac{1}{k}$$

3.2. Bearing Vertical Offset Measurement

ABS (2019)

According to ABS (2019), the vertical offsets of the bearings are set "either during the bore sighting or during the sag and gap procedure". When the bearings offset adjustments are completed, ABS (2019) suggests that the intermediate bearing and the main engine bearing are choked. However, the effect of extra steel work on the vessel, hull deflections or thermal deviations on the bearing offsets should not be underestimated. After the chocking, the following methods are suggested by ABS (2019) in order to measure the vertical offsets of the bearings:

- Optical methods
- Laser methods
- Hydraulic jacks

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- Strain gauges
- Crankshaft deflections
- Combined methods, hydraulic jacks, strain gauges and crankshaft deflections

More precisely, regarding each one of the aforementioned methods:

- *Optical Methods*: Feasible in cases where visual contact is granted. Relatively inaccurate method which gives information contained in the part which is visible from the engineer.
- *Hydraulic Jack-Up:* As it was previously described, this method is usually utilized in order to calculate the bearing loads. By using reverse engineering calculation methods, the vertical offsets of the bearings may also be extracted following this method. In cases where the bearings are not accessible, the Jack-Up method may be used combined with another method. However, it is not believed to be a highly accurate method for the purposes of measuring the vertical offsets of the bearings.
- *Strain Gauge:* This method is suitable in cases where the bearings are not accessible. For the purposes of measuring the vertical offsets of the bearings, reverse calculations must take place, and their accuracy may be significantly improved if Strain Gauge method is combined with another method (e.g. Jack-Up method, crankshaft deflection measurements)
- *Crankshaft deflections:* If the 3D model of the crankshaft is available by the manufacturer, measurements regarding the crankshaft deflections may be utilized in a reverse calculation base in order to extract the vertical offsets of the bearings
- *Combined Measurement Methods*: The combination of methods, according to ABS (2019), is highly preferred in cases where the vertical offsets of the bearings are to be recalculated from other system parameters.

ABS (2019) warns for some extra attention when it comes to the Reverse Shafting Alignment calculation of Bearing Offsets. As ABS (2019) states:

"The reverse shafting alignment is an analysis where bearing offsets are recalculated from measured bearing reactions, bending moments and crankshaft deflections. It is an inverse approach to the regular shaft alignment calculation".

More specifically, theoretically, the system of equations which correlates the vertical offsets of the bearings to the Moments and Forces applied on the bearings, has as follows:

$$\{R\}_{Nx1} = [K]_{Nx1} * \{r\}_{Nx1}$$

Where:

- $\{R\}_{Nx1}$: Nodal Load Vector (moments and forces)
- $[K]_{Nx1}$: System stiffness matrix
- $\{r\}_{Nx1}$: Nodal displacement vector

Typically, the process of calculating the system's offsets when knowing all of the system's moments and forces is trite, as the number of unknown variables equals to the number of known variables. However, in reality, not all of the systems moments and forces are considered to be known, as many of the bearings are not accessible for measurements. As a result, there are multiple solutions to the pre described system of equations, unless the unknown moments and forces are predicted or assumed.

DNV (2023)

Additionally, DNV (2023) in Shaft Align (2) regarding the vertical and horizontal offsets of the stern tube housing, states that:

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“Laser aided sighting of vertical and horizontal offsets of the stern tube housing in way of the bearings shall be submitted for review by the attending surveyor. A minimum of 5 reference points shall be used covering the aft most bearing housing and 3 reference points for the forward bearing housing. The laser reference line shall be made concentric with the stern tube and independent of the bearings”.

Moreover, DNV (2023) allows alternative measurement methods to be used, as long as their equivalent accuracy is verified by the Classification Society.

3.3. Bearing Misalignment Measurements, (ABS)

The term “Bearing Misalignment”, as previously discusses, refers to the difference between the centerlines of the shaft and the bearings. When the intermediate bearing(s) is/are of interest, the bearing misalignment is easily measured by utilizing feeler gauges. However, ABS (2019) suggests that several measurements are taken along a full rotation of the shaft, in order to verify the compliance of the bearing misalignment and exclude cases of shaft run-out. The procedure, according to ABS (2019), is described as follows:

“Misalignment is defined by subtracting the bearing clearance measurements fore and aft (vertical misalignment), and port and starboard side (horizontal misalignment) of the bearing”.

The prementioned apply also to the Main Engine bearings. However, when it comes to the aft stern tube bearings, matters become a little bit more complicated. More specifically, the bearing misalignment calculations may only be conducted after the propeller is fitted. The measurement procedure regarding the Misalignment angle of the Aft Stern Tube bearing will be thoroughly discussed in an upcoming chapter of this Thesis. Additionally, ABS (2019) suggests that the Intermediate and Main Engine bearings are choked to their final positions only after verification of the clearances and the bearing reactions have taken place.

3.4. Crankshaft Deflection Measurements

Generally, Crankshaft Deflection measurements are taken in several stages of the vessel’s construction and operation. It is important to highlight, that the tolerances and the limits regarding the Crankshaft’s deflections are set by the Engine’s manufacturer. Moreover, the engine’s manufacturer supplies the vessel with an engine tool set with a deflection gauge. The measurements are conducted by utilizing a dial indicator which is placed on a predefined location between the webs. Then, the crankshaft is rotated, and measurements are taken in several locations. It Is common, that the measurements are taken in 5 (five) different angles of the crankshaft.

3.5. Sag and Gap measurements

The Sag and Gap method, as was mentioned and discussed in the previous chapters of this Thesis, has as its main scope the configuration of the shafting, the bearings’ offsets, and consequently, the bearings’ loads, to the calculated values. In order for that to happen, the actual Sag and Gap measured values should correspond to the ones calculated during the Design Stage of the Shaft Alignment procedure. However, the Sag and Gap method should only be used as a counterbalancing method to jack-up measurements regarding the fine tuning of the bearings in terms of Bearing Reactions and Offsets. Moreover, the instruments and the equipment used for Sag and Gap measurements, Dial Gauges, Feeler Gauges and Depth Micrometers, lack in ways of accuracy.

ABS (2019)

More specifically, according to ABS (2019) the tolerances regarding the prementioned equipment has as follows:

- *Dial Gauges*: Accuracy up to ±0.001 mm
- *Feeler Gauges:* Accuracy up to ±0.05 mm, but errors often exceed 0.1 mm

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- *Depth Micrometers*: For *analog* micrometers, accuracy is set up to ±0.01 mm and for *digital* micrometers accuracy is set up to ±0.001 mm

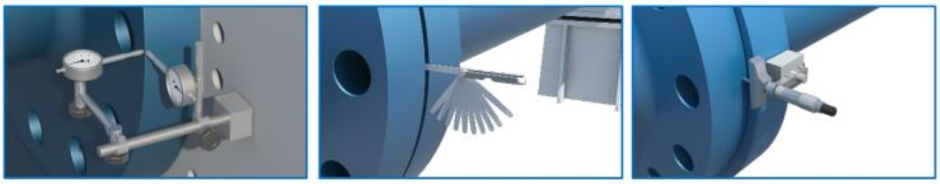

Figure 103: Sag and Gap measurement equipment, ABS [12]

DNV (2021)

DNV (2021) requires that all Sag and Gap measurement data are reported with the presence of a surveyor.

However, DNV (2023) in Shaft Align (2) specifically requires that the Sag ang Gap measurements are conducted via utilization of laser method, dial gauge method or any other method which ensures equivalent reliability and accuracy.

3.6. Stern Tube Bearing Clearance Measurements, (ABS)

Generally, as ABS (2019) suggests, prior floating of the vessel, the bearing clearances should be measured at four different positions along the shaft's circumference: Top, bottom, port and starboard. As ABS (2019) states:

"The total horizontal clearance is calculated by adding port and starboard clearance measurements. The vertical clearance is calculated by adding top and bottom clearance measurements".

As far as the aft stern tube bearing and its clearance is of interest, matters tend to be a little bit more complicated. More precisely, as it happens with every matter which is correlated to the aft stern tube bearing of a vessel, different rules and protocols apply based on whether a vessel is equipped with both aft and forward stern tube bearings or not. The interesting part of the clearances of the aft stern tube bearings, lays on the fact, that by measuring the bottom clearance of the aft stern bush, the bearing misalignment can be verified. More specifically, ABS (2019) suggests that the clearance as measured with the tail shaft unrestrained at both ends is "translated" into bearing misalignment and thus the slope angle is confirmed, only if the deviation falls within the limits set in the design stage of the Shaft Alignment. ABS (2019) requires that the Shipbuilder also verifies the bottom clearance, and consequently the misalignment angle, by utilizing feeler gauges. If a double slope design is present, the transition point's clearance shall also be measured.

Now, more specifics will be given regarding the two design possibilities: Design with both stern tube bearings and Designs which incorporate only the Aft stern tube bearing.

Designs With Both Stern Tube Bearings Installed

ABS (2019) requires that in such cases, the clearance must be measured with the propeller shaft installed, unrestrained at both ends and supported only by the stern tube bearings. The propeller should NOT be fitted. ABS (2019) clearly states that no other type of support should be used in installations with either stern tube bearings. ABS (2019) requires that the calculated value of the bearing clearance shall not deviate from the value calculated in the Design Stage by more than 0.1 mm.

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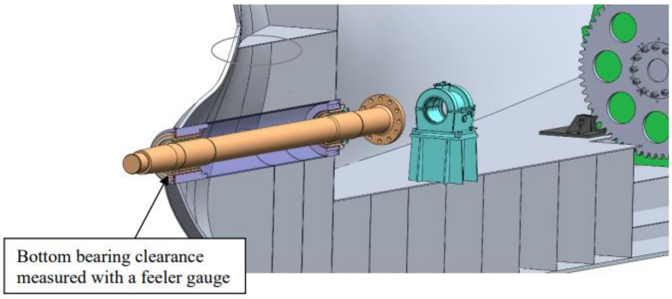

Figure 104: Clearance measurements in installations with both stern tube bearings, ABS [12]

Designs with no Forward Stern Tube Bearing installed

For installations with no Forward Stern Tube bearing, the procedure ABS (2019) suggests differs. More specifically, the procedure ABS (2019) proposes has as follows:

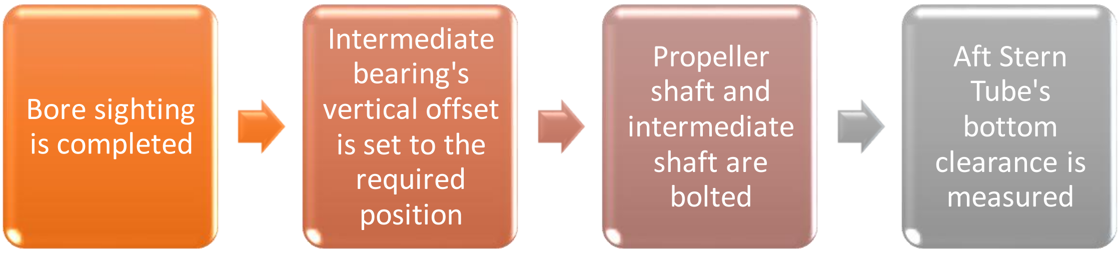

Figure 105: Stern Tube Bearing clearance measurement procedure for installations with no forward stern tube bearing as ABS suggests [Source; ABS, 2019]

Additionally, ABS (2019) proposes that the aft stern tube bearing is utilized as a means of controlling the bearings misalignment by adjusting the vertical offset of the intermediate bearing. In this occasion, the intermediate shaft bearing acts as the forward stern tube bearing would. However, in order for the prementioned to be applicable and allowed, all of the aforementioned requirements should be met.

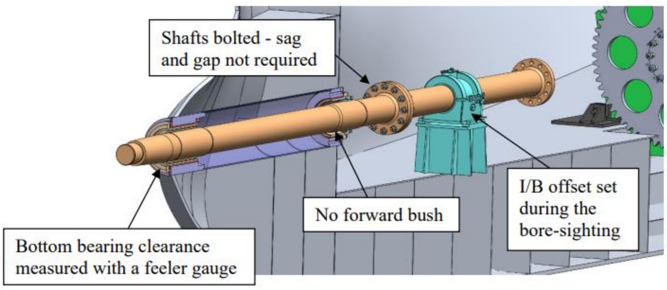

Figure 106: Aft stern tube bottom clearance measurements in installations with no forward stern tube bearing, ABS [12]

ABS (2019) suggests the adjustment of the bearing's inclination and bottom clearance of the stern tube bearing in installations with no forward stern tube bearing, as it ensures that the bottom bearing clearance is equal to

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the one calculated in the design stage, the Sag and Gap procedure can be omitted, and the procedure is marked by remarkable conveniency and accuracy. However, as ABS (2019) highlights, the following matters should be carefully handled in this occasion:

- *"Once the vessel is afloat, hull deflections impacting the intermediate bearing offset may still influence the stern tube bearing misalignment angle. It is advised to be particularly aware of this when the clearance measurement is conducted in an early block stage and with the main engine not yet in place".*

- *"The alignment calculation should be consulted to prevent potential overloading of the intermediate bearing and excessive bending of the forward end of the intermediate shaft".*

3.7. Eccentricity Measurements, (ABS)

Eccentricity of the shaft may result in severe effects regarding vibrational problems (e.g. whirling phenomenon) or horizontal loads on the bearings which may even result in bearing failures. Consequently, it is of high significance to ensure that excessive run-out of the shaft is not present. In order to ensure that shafting run-out is measured with a dial indicator. Moreover, ABS (2019) suggests that the shafting run-out is obtained via examining the strain and bearing reactions and their changes. In some cases, the shaft may need to be removed in order for the eccentricity measurements to be carried out.

Eccentricity measurements by utilizing dial indicators are conducted by placing dial indicators both on the vertical plane and horizontal plane of the shaft. The shaft then begins rotating, and the indicators are monitored in order for the responsible engineer to document deviations. Under ideal conditions, if the eccentricity equals to zero, the indicators will not display any deviation from the initially set value. However, practically, there always is a small deviation due to errors such as hysteresis or friction matters.

3.8. Stress Measurements, (ABS)

Generally, stress is present both on the shaft as well as on the bearings of the shafting mechanism of a vessel. However, the stress which is applied on the shaft is not severely affected by the shaft alignment condition, so it is not considered to be a matter of high significance regarding the specific topic of this paragraph. The stress in the bearings, on the other hand, is directly affected by the shaft alignment condition and the bearing-to-shaft misalignment and contact area. However, practically, it is almost impossible to carry out precise measurements regarding bearing stresses. The only possible solution for the responsible engineer, according to ABS (2019), is the evaluation of the contact area via optical examination of the bearing. After the specifying of the contact area between the bearing and the shaft, by utilizing proper software, a simulation may be carried out which will be capable of evaluating the dynamic performance and behavior of the bearing.

3.9. Monitoring of the Aft Stern Tube Bearing

The real-time monitoring of the Aft Stern Tube bearing may prove to be a significant factor which contributes in minimizing the risk of environmental contamination, personnel's safety and vessel's seaworthiness. The monitoring has as its immediate scope the evaluation of the bearing's performance in order to avoid bearing failure.

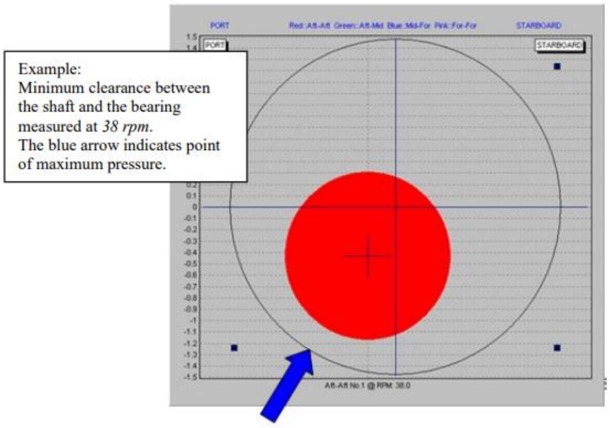

Figure 107: Real screenshot of a monitoring program for the aft stern tube bearing, ABS [12]

ABS (2019)

ABS (2019) suggests that proximity sensors are installed along a bearing's length, in order for the space between the bearing's inner surface and the shaft's outer surface to be determined and for the exact location of the shaft inside the bearing to be defined in any moment. Moreover, heat alarms as well as RPM, speed and rudder angle monitoring can be fitted along with appropriate alarms in order for a fully developed profile of the shafting system's behavior to be available at any time for the vessel's personnel and for possible malfunctions and damages to be avoided early enough.

DNV (2023)

Additionally, DNV (2023) in the Enhanced Shaft Align (2) regulations states that, the temperature of the aft stern tube bearings shall always be monitored via sensors and alarms should also be provided. More specifically, the alarms to be installed according to DNV's Enhanced Shaft Align (2), (2023), have as follows:

- Alarm regarding low value of percentage of propeller's immersion
- Alarm regarding high value or high rate of rise of Aft Stern Tube Bearing's temperature
- Alarm regarding high value of Heated Tank Space temperature below first inboard shaft bearing (if installed)

Moreover, at Shaft Align (1), DNV (2023) states that the aft stern tube bearing's temperature should be monitored, and alarms should be installed. Additionally, the lubrication system's design should verify compliance with sampling procedures regarding the lubrication oil of the aft stern tube bearing under all running conditions.

Almost the same apply in cases where compliance with DNV's Shaft Align (1), (2023), is desired. More specifically, according to DNV (2023) in order for a vessel to be in compliance with Enhanced Shaft Align (1), in cases where inboard bearings are placed on top of heated tanks, a high temperature alarm must be placed in the tank's space. According to DNV (2023) *"The alarm shall be set at the maximum temperature allowed for thermal expansion in the shaft alignment calculations".* Additionally, in Shaft Align (1), DNV (2023) states that means of monitoring should be present in cases of inadequate propeller immersion in the wheelhouse and the central alarm panel. Suitable signboards should also be placed at the operating locations.

3.10. Monitoring of the Intermediate Bearing, (ABS)

According to ABS (2019), the real-time monitoring of the intermediate bearing may also prove to be of high significance, and it can be accomplished by utilizing strain gauge equipment. Strain gauges may report the bearing load and the misalignment angle inside the bearing in real-time and under both static and dynamic conditions. ABS (2019) suggests that two (2) sensors are installed: A Data Acquisition sensor and a Display

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Unit. This way, the results extracted from the sensors can be directly displayed, even in the Engine Control Room. This way, any malfunction of problem regarding the intermediate bearing's operation can easily be detected and severe damage and accidents can be avoided. Moreover, as ABS (2019) states, the Data Acquisition software has the capability of alarming regarding Unloaded or Overloaded bearings, as well as bearings with excessive misalignment angle.

3.11. Recording of Bearing Reactions

ABS (2022) in Enhanced Shaft Alignment Guide, states that the Shipyard must keep a detailed log of all of the activities performed regarding the shaft alignment installation procedure. Moreover, a log of all the measurements taken as per "Enhanced Shaft Alignment Guide" must be kept. As far as the bearing reactions are of interest, a record which shows the measured values of the reactions of all of the accessible bearings, as well as of the three aftmost main engine bearings (or gearbox bearings), for all of the hull conditions for which measurements were taken must be kept. Additionally, the measured values of the reactions of the accessible shaft bearings compared to the calculated values during the design stage (for the same condition of the vessel) must also be documented and handed to ABS. The records can be in tabular or bar chart format. An example is given in the picture below.

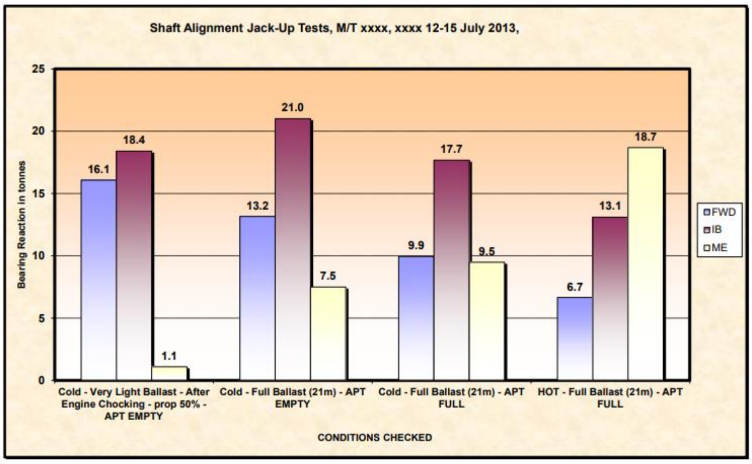

Figure 108: Bearing Recation values during Sea Trials for several vessel conditions (Forward Stern Tube Bearing, Intermediate Bearing, Aftmost Main Engine Bearing), ABS [13]

PART B

Evaluating LLM Performance on Technical Questions: Comparative Study

I. Introduction on LLM Assessment

The second part of this Diploma Thesis is focused on the capabilities of using Large Language Models (LLMs) for comprehensive reasons, including technical topics related to the profession of Naval Architects and Marine Engineers, specifically Shaft Alignment and the regulations that govern it. The primary goal of the study was to evaluate the usefulness and reliability of LLMs used to answer technical questions as well as inquiries on the Classification Societies' Shaft Alignment Regulations, for comprehensive and/or educational purposes.

Artificial intelligence is now being used in a variety of fields, engineering being just one of them. AI has several applications in engineering, including design optimization, robotics, automation, simulation, modeling, and natural language processing. It's worth noting that Natural Language Processing (NLP) and LLMs are closely related. More specifically, LLMs are regarded as both a subset and an advanced application of NLP. On the one hand, LLMs are created and trained using foundational NLP techniques and principles such as tokenization,

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semantic analysis, part-of-speech tagging, and syntactic parsing. For clarification reasons, the aforementioned terms related to Artificial Intelligence will be briefly explained:

- ***Tokenization:*** A core process in NLP. The term describes the process of breaking down a text into smaller units, named tokens. The tokens can be groups of words, individual words, numbers, letters, or even special characters and punctuation marks. The specific form of the tokens is based on the precise task the model has to execute, as well as on the language used. The tokens are then used in order for the text to be more easily analyzed by the computer.
- ***Part-Of-Speech (POS) Tagging:*** A process in NLP during which each word of a sentence is correlated and assigned to a specific grammatical role and part of speech (e.g. noun, verb, adjective, adverb, pronoun etc.). The POS Tagging procedure facilitates the understanding of how each word functions in a sentence and consequently is a core function in terms of understanding and generating human-like text, which is the function of LLMs. It is worth mentioning, that POS Tagging considers the contextual meaning of each word. For example, the word "close" can be both used as a verb or an adjective. POS Tagging enables the capability of differentiating the two words based on their syntactical role in the sentence.
- ***Syntactic Parsing:*** A process in NLP, during which the grammatical structure of a sentence and the relationship the words have with each other are analyzed. This procedure makes the construction of meaningful sentences from LLMs feasible. During the process of Syntactic Parsing, the text is broken down into units such as Words (each word is already categorized through the POS Tagging process as a specific part of speech) and Phrases. In reality, once the words are tagged to their form of speech (noun, verb etc.) through the POS tagging process, the process of Syntactic Parsing establishes the relationships between the several words in order to determine the hierarchical structure of a sentence. As a matter of fact, POS Tagging actually acts as the foundation of the Syntactic Parsing process, with the main scope of constructing a comprehensive view of a sentence's structure. POS Tagging and Syntactic Parsing combined, make the processing and comprehension of text in NLP applications feasible.
- ***Semantic Analysis:*** A process utilized in NLP in order to define the meaning of words. It goes beyond the process of categorizing the words based on their grammatical and syntactical role in a sentence. Semantic Analysis focuses on the meaning of words and on how they can be combined in order to express certain thoughts. During the Semantic Analysis procedure, synonyms, antonyms and contextual usage of words are also navigated. More precisely, as we all know, several words may have different meanings depending on the context in which they are used, and Semantic Analysis aims in clarifying their meaning. Moreover, Semantic Analysis entangles various tasks/techniques in order to achieve the aforementioned goals. Some of them are the following:
    - I. **Word Sense Disambiguation**: Determining the meaning of words, based on their context.
    - II. **Named Entity Recognition (NER)**: Classifying certain words such as Names, Locations, People etc.
    - III. **Semantic Role Labeling (SRL)**: Identifying the specific roles and relationships words have with each other (e.g. Agent and Patient roles)
    - IV. **Sentiment Analysis**: Determining the "footprint" of a text (e.g. positive, negative, neutral)
    - V. **Semantic Similarity**: Evaluating how similar and/or related two pieces of text are (in meaning).

On the other hand, LLMs are thought to be an advanced application within NLP. Due to their remarkable capability of generating, understanding and processing human-like text, which is accomplished through the utilization of deep learning architectures, LLMs are widely utilized in NLP. Furthermore, for the pre-mentioned reasons, LLMs are thought to be an integral part of NLP.

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During their studies, Naval Architects and Marine Engineers are asked several times to comprehend, analyze, compare and eventually apply disciplines which are based on large amounts of technically based documents, such as Classification Societies' Regulations. There is no doubt, that the utilization of LLMs with respect to facilitating the aforementioned process of handling and comprehending large amounts of documents would enhance the entire educational process in numerous ways. Moreover, such applications may also be used in a way which simulates the human interaction between a student and a professor and/or an expert on a specific technical matter, so that academic questions can be answered. Such applications would open the door for innovative academic procedures aiming to upgrade reinforce the whole process of comprehension in educational matters.

II. Establishment of QnA Dataset for Performance Evaluation of LLMs

For the purposes of this study, the course of action was decided to begin with the process of creating a set of questions which were answered wrongly, inadequately or were not answered at all by ChatGPT. The version of ChatGPT which was utilized in this study was ChatGPT-3.5. ChatGPT can be utilized for several tasks related to academic and educational tasks. Consequently, the process of defining the set of questions served the scope of establishing areas or patterns of behavior ChatGPT manifested, which indicated that ChatGPT was incapable of serving as a useful tool when it came to the precise issues of Naval Architecture, Marine Engineering and Shaft Alignment along with the Regulations which govern it.

The procedure of establishing this set of questions was based on testing and evaluating the answers in matters regarding Shaft Alignment and Regulations. More specifically, a large number of questions were addressed to the AI model, and for the purposes of the study, a set of approximately 100 Questions was defined as the "Questions of Interest". All of the questions which ChatGPT answered in a wrong way or inadequately are included in the final set of questions. Furthermore, a number of questions which ChatGPT answered correctly are also included in the final set, for the sake of assessment and comprehension reasons.

Additionally, it should be mentioned, that a number of "main categories" made up the final set of questions. More specifically, the main scope was for the set of questions to entangle as many qualities of questions as possible, in order for the assessment of the performance of the models to be as representative as it can. The main categories of questions in terms of nature were:

1. **1st Category:** Strictly theoretical questions (related to Physics knowledge, Mathematical equations and application of Math principles)
2. **2nd Category:** Questions which required fact retrieval (these questions were mostly targeted to Classification Society Regulations and limits regarding specific parameters)
3. **3rd Category:** Questions which entangled more comprehensive qualities and focused on assessing the performance of the models in terms of text generation, understanding of meanings etc. (e.g. questions related to differences between the approaches of several Classification Societies)
4. **4th Category:** Questions aimed to assess the performance of the models in more complex context. This category was made up of questions which included "tricky" terminology, complex configurations and meanings and multiparametric problems with an "optimization" nature. (e.g. questions regarding Double-Slope and Single-Slope Designs or Multi-Point and Single-Point contact or Shaft Alignment Design Stage/Shaft Alignment Procedure Stage/ Stages of Shaft Alignment Procedure Stage)

Many questions which belong to the final set simultaneously belong to more than one of the above categories.

II.I. A representative sample of each category questions based on the nature of the Questions

A representative sample of each Category's questions will now be presented:

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II.I.I. 1st Category: Strictly Theoretical Questions related to Physics knowledge and Mathematical equations and principles

Q: Explain the Hertzian Contact Theory.

Q: Explain the term “oil-whirling”.

Q: Explain the term “wiping”.

II.I.II. 2nd Category: Questions which required Fact Retrieval (related to Regulations and limits set by Classification Societies)

Q: Explain the term “aft stern tube misalignment angle”

Q: Explain the meaning of the term “Double Slope Design”

Q: Elaborate on Classification Societies’ Regulations regarding Double Slope Designs

Q: Elaborate on Classification Societies’ Regulation regarding vessel Designs with no Forward Stern Tube Bearing.

Q: Elaborate on Classification Societies’ Regulations regarding the maximum value of the relative slope angle between the tail shaft and the aft stern tube bearing.

II.I.III. 3rd Category: Questions which entangled more comprehensive qualities, with the scope of evaluating the model’s capabilities in terms of text-generation and understanding

Q: What is the impact Classification Societies have on the Shaft Alignment of a vessel?

Q: What changes in terms of Shaft Alignment depending on the Classification Society a Shipowner chooses for a vessel?

Q: Elaborate on key-differences between the Regulations each Classification Society provides regarding Shaft Alignment.

Q: How does the stiffness of the bearings affect Shaft Alignment?

Q: Name parameters which severely affect the Shaft Alignment of a vessel, and how the affect it.

II.I.IV. 4th Category: More complex questions which entangled difficult and “tricky” terminology and complicated configurations and meanings, thus requiring deeper understanding of the subject

Q: Is the flexibility of the shaft related to the flexibility of the vessel’s hull in Shaft Alignment?

Q: Explain the importance of “points of contact” when evaluating a bearing in Shaft Alignment.

Q: Name the Steps of the Shaft Alignment Procedure phase according to ABS.

Q: From which stages does the Shaft Alignment Process consist of?

Q: Describe the relationship and the importance the terms low running pressure and static contact area of a bearing have.

The questions which make up the final set, with respect to the response ChatGPT gave, were then classified based on their respective answers to the following main categories:

- Questions which ChatGPT answered Correctly
- Questions which ChatGPT answered Erroneously
- Questions which ChatGPT answered Inadequately/Too Generally
- Questions which ChatGPT answered Differently for Varying Question Prompt

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At this point, it is worth giving further information on some of the categories which were mentioned above:

- Inadequate/Too General answers: On the one hand, several technical questions were answered inadequately by the AI model (ChatGPT), due to lack of information. These are the cases to which the term “Inadequate Response” refers to. On the other hand, by the term “Too General Answer” in this study, the cases where repetitive Evasive Responses were given by the AI Model are described. It was observed that in numerous occasions, the Evasive Response the model provided were repeated, regardless of the question addressed. This phenomenon may be observed due to the fact that the AI Model generates responses based on probabilities and each next word or phrase is based on that. If a set of questions (inputs) triggers a specific statistical response/behavior, it is likely that they will have identical answers as well. Additionally, when a conversation is conducted with the AI Model, the previous answers and questions are taken into account. Consequently, if the AI Model due to lack of information or Understanding of the question classifies a set of questions as similar, there is a high possibility that the same response (output), even an Evasive one, will be given as well.
  The reason due to which Evasive Responses are given by the AI Model, in our case, are most probably related to the following:
  1. Lack of Information regarding the question which was addressed to the AI Model
  2. Lack of comprehending of the question which was addressed to the AI Model
  3. Training Data Limitations regarding the specific topic. If the model is not well-trained on the subject of the question, it may provide an Evasive Response
  4. Complexity of the question: The possibility of the deep understanding or the complex reasoning required in order for the question to be answered adequately being beyond the model’s capabilities is always present
- Questions which were answered differently for varying question prompt: It was observed, that some questions were given different answers depending on the way they were expressed (specific question prompt). By alternating the query, but without giving any extra information on the subject, the model gave different responses. Moreover, in several occasions, when a question was answered wrongly, the next question which was addressed to the model, included in its answer a “hint” for the previous wrongly answered question. After that, when the question which was answered wrongly was addressed to the model, the response given was correct or remarkably different form the first response.

II.II. A representative sample of each category’s Questions, based on ChatGPT’s responses

II.II.I. Questions with Correct Responses:

-Q: Which are the types of bearings found in a ship’s propulsion system based on their location?

-A: In a ship's propulsion system, various types of bearings are used to support and facilitate the rotation of shafts and other rotating components. These bearings are strategically located throughout the system to provide support, reduce friction, and ensure smooth operation. Here are the types of bearings commonly found in a ship's propulsion system based on their location:

Stern Tube Bearings:

Located at the aft (rear) end of the ship, stern tube bearings support the propeller shaft as it extends from the hull into the water. These bearings are typically hydrodynamic or hydrostatic bearings designed to withstand the axial and radial loads exerted by the propeller shaft.

Intermediate Shaft Bearings:

Intermediate shaft bearings support the intermediate shafts that transmit power from the main engine to the propulsion system components, such as gearboxes or reduction gears. These bearings are located within the

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ship's hull and are designed to withstand high radial and axial loads […]These are some of the common types of bearings found in a ship's propulsion system, each serving a specific purpose and located at different points along the shafting and propulsion machinery. Proper selection, installation, and maintenance of these bearings are essential for ensuring the safe and efficient operation of the vessel.

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-Q: What does the term "Hull Deflections" mean?

-A: "Hull deflections" refer to the structural deformations or distortions experienced by a vessel's hull under various loading conditions. These deformations can occur due to external forces acting on the hull, such as wave action, wind, hydrodynamic forces, and operational loads, as well as internal factors like cargo weight, machinery vibrations, and changes in draft.

Hull deflections can manifest in different forms, including bending, twisting, hogging (upward curvature amidships), sagging (downward curvature amidships), and torsion (twisting along the longitudinal axis). These deformations can affect the overall structural integrity, stability, and performance of the vessel.

Understanding and managing hull deflections are essential for ensuring the safety, seaworthiness, and longevity of a vessel. Structural design considerations, including hull material selection, reinforcement, and stiffening arrangements, aim to minimize the effects of hull deflections and ensure that the vessel can withstand anticipated loads and environmental conditions throughout its service life.

Hull deflections are also considered in the context of shaft alignment, as they can influence the relative positions of shafting components and affect alignment integrity. Proper alignment practices account for hull deflections to ensure that shafts remain aligned within acceptable tolerances under various operating conditions.

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II.II.II. Questions with Erroneous responses:

-Q: How can someone compare the loading state of two bearings with different dimensions?

-A: Comparing the loading states of two bearings with different dimensions requires considering various factors to ensure a meaningful comparison. Here's how you can approach this comparison:

Load Capacity: Determine the load capacity of each bearing based on its design specifications, including its load rating, material properties, and dimensions. Compare the load capacities of the two bearings to assess their ability to withstand applied loads.

Load Distribution: Evaluate how the applied loads are distributed across the bearings. Consider factors such as the position of the bearings relative to the load application point, the stiffness of the bearing supports, and the alignment of the shaft. Bearings experiencing concentrated or uneven loading may be subjected to higher stresses than those with more evenly distributed loads.

Bearing Geometry: Take into account the geometric differences between the bearings, such as their diameter, length, and contact area with the shaft. Bearings with larger contact areas or higher diameters may distribute loads more effectively and exhibit lower stresses than smaller bearings under similar loading conditions.

Operating Conditions: Consider the operating conditions of the bearings, including rotational speed, temperature, and lubrication. Bearings operating at higher speeds or temperatures may experience different loading states than those operating under lower-speed or ambient conditions.

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Finite Element Analysis (FEA): Perform finite element analysis to simulate the loading states of the bearings under various scenarios. FEA allows for detailed modeling of bearing geometry, material properties, and loading conditions, enabling a more comprehensive comparison of their performance.

Factor of Safety: Calculate the factor of safety for each bearing to assess its margin of safety against failure under the applied loads. Bearings with higher factors of safety are generally considered to be more robust and less susceptible to damage or premature wear.

Real-world Testing: Conduct real-world testing or field measurements to validate the predicted loading states of the bearings. Monitoring factors such as bearing temperatures, vibration levels, and wear patterns can provide insights into their actual performance under operating conditions.

By considering these factors, you can compare the loading states of two bearings with different dimensions and make informed decisions regarding their selection, installation, and maintenance in various applications.

Author Note: The desirable response should obviously mention the Sommerfeld Number

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-Q: What does the term "Slope Boring" mean in Marine Engineering?

-A: In marine engineering, the term "slope boring" may not have a specific application or meaning as it does in civil engineering or construction. However, there are drilling techniques and methods used in marine construction and offshore operations that involve drilling at angles or inclinations. These methods are typically adapted to the unique challenges and requirements of marine environments. Here are a few drilling techniques used in marine engineering that may be relevant:

Directional Drilling: Directional drilling is a method used to control the direction and inclination of the borehole in offshore drilling operations. It allows drilling engineers to reach targets located beneath the seabed or to avoid obstacles such as existing infrastructure or geological formations. Directional drilling is commonly used in offshore oil and gas exploration and production.

Subsea Drilling: Subsea drilling involves drilling wells from a floating or fixed platform located on the ocean surface to access hydrocarbon reservoirs beneath the seabed. The drilling equipment is typically installed on the platform and lowered to the seabed, where drilling operations take place. Subsea drilling may involve drilling at angles or inclinations to reach reservoir targets.[…] While the term "slope boring" may not be commonly used in marine engineering, drilling techniques adapted for marine environments often involve similar principles of drilling at angles or inclinations to achieve specific objectives. These techniques play a vital role in offshore exploration, production, construction, and infrastructure development in marine environments.

Author Note: Improper and out-of-context definition of slope boring

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-Q: Elaborate on double slope designs in Shaft Alignment.

-A: In shaft alignment, a "double slope" design refers to a method of compensating for thermal expansion and contraction in a shafting system. This design involves incorporating two slopes or bends into the shafting arrangement, typically near the bearing supports, to accommodate changes in shaft length due to temperature variations. Here's an overview of double slope designs in shaft alignment:

Purpose: The primary purpose of a double slope design is to provide flexibility in the shafting system to accommodate thermal expansion and contraction without causing excessive stresses or misalignment.

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Temperature changes, such as those resulting from changes in operating conditions or ambient temperature, can cause the shaft to expand or contract longitudinally. Without proper compensation, this thermal growth or shrinkage can lead to misalignment, increased bearing loads, and premature failure of components.

[…]Overall, double slope designs are a valuable tool in shaft alignment, particularly in applications where temperature variations are significant or frequent. By incorporating two slopes or bends into the shafting arrangement, engineers can effectively mitigate the effects of thermal expansion and contraction, ensuring optimal performance and reliability of the shafting system over its operational life.

Author Note: Double Sloping is referred to as a way of compensating for thermal deformation

II.II.III. Questions with General or Evasive Responses

-Q: What is the “Sighting Through” procedure in Shaft Alignment?

-A: The "sighting through" procedure in shaft alignment is a visual alignment method used to assess the parallelism or alignment between two rotating shafts. It involves visually sighting through a straight-edge or line of sight from one shaft to another to determine if they are aligned properly. This procedure is often used in conjunction with other alignment techniques to achieve precise alignment between the shafts. Here's how the sighting through procedure works:

Preparation: Before performing the sighting through procedure, ensure that the shafts are stationary and securely supported. Remove any guards or obstructions that may block the line of sight between the shafts.

Selection of Reference Points: Choose reference points on each shaft that can serve as alignment indicators. These reference points should be visible and easily identifiable from the opposite shaft. Common reference points include coupling hubs, shaft shoulders, keyways, or other machined features.

Positioning of Straight-edge or Line of Sight: Place a straight-edge or establish a line of sight between the selected reference points on the two shafts. The straight-edge can be a ruler, straightedge bar, or a laser alignment tool. Ensure that the straight-edge is positioned parallel to the shaft axis and perpendicular to the direction of movement.[…]The sighting through procedure provides a quick and effective visual method for assessing shaft alignment. However, it is essential to combine this procedure with other alignment techniques, such as dial indicator measurements or laser alignment, for more precise alignment assessment and adjustment.

Author Note: It is not mentioned that the “Sighting Through” procedure establishes the reference line utilized for the Shaft Alignment Procedures. In fact, it is generally described as a “visual alignment method”. The response is in-context, but too general.

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-Q: What is the impact Classification Societies have on the Shaft Alignment of a vessel?

-A: Classification societies have a significant impact on the shaft alignment of a vessel in several key areas:

1. Standards and Regulations: Classification societies set stringent standards and guidelines for the alignment of shafts. These standards ensure that the alignment is within acceptable limits to prevent excessive wear, vibration, and potential damage to the propulsion system.
2. Approval of Alignment Procedures: Before shaft alignment procedures are carried out, they must be approved by the classification society. This involves submitting detailed plans and methodologies that comply with the society’s standards. The approval process ensures that the alignment work will be performed correctly and to a high standard.

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3. Surveys and Inspections: Classification societies conduct surveys and inspections during the construction of new vessels and periodically throughout the vessel's service life. These surveys include checks on the shaft alignment to ensure it meets the required standards. Any deviations or misalignments identified during inspections must be corrected to maintain the vessel's classification status.
4. Certification: Upon satisfactory completion of the shaft alignment and verification that it meets the society's standards, the vessel receives certification from the classification society. This certification is crucial for the vessel's operational legitimacy and for insurance purposes.
5. Maintenance Requirements: Classification societies often require regular maintenance and checks on the shaft alignment as part of their ongoing survey regime. This ensures that the alignment remains within the acceptable limits over the vessel's operational life, thus preventing long-term damage and ensuring safety.
6. Documentation and Record-Keeping: Classification societies require comprehensive documentation of the shaft alignment process, including initial alignment during construction and any subsequent adjustments. These records are crucial for verifying compliance during inspections and for future reference in case of alignment-related issues.
7. Safety and Reliability: By enforcing strict shaft alignment standards, classification societies enhance the overall safety and reliability of the vessel. Proper alignment minimizes the risk of mechanical failures, reduces maintenance costs, and extends the lifespan of the propulsion system components.
8. Impact on Vessel Value: Vessels that comply with classification society standards, including those for shaft alignment, typically have higher resale values and better insurance terms. This compliance is seen as a mark of quality and reliability.

In summary, classification societies play a crucial role in ensuring that shaft alignment is carried out to the highest standards, which in turn supports the safety, reliability, and operational efficiency of vessels.

Author Note: The response consists of general areas of interest regarding Classification Societies, while it should mention the influence Classification Societies have on the design, implementation and verification procedures of Shaft Alignment in terms of proposed methodologies and design features. CHAT-GPT recognizes some key words and produces an extensive text which mentions generic information regarding the topic of interest.

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-Q: What changes depending on the Classification Society of a ship regarding Shaft Alignment?

-A: The classification society a ship is registered with can influence several aspects of shaft alignment, including standards, procedures, and operational practices. Here are the key factors that may change:

Alignment Standards and Tolerances: Different classification societies may have varying standards and acceptable tolerances for shaft alignment. These standards dictate the permissible limits for misalignment, which can affect how precise the alignment must be.

Approval and Documentation Requirements: The documentation and approval process for shaft alignment plans and procedures can differ. Some societies may require more detailed documentation and stricter approval processes, including comprehensive reports on alignment methods and results.

Inspection and Survey Frequency: The frequency and thoroughness of inspections and surveys related to shaft alignment can vary. Some classification societies might require more frequent or detailed inspections, influencing how often alignment checks and adjustments need to be performed.

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Technological Recommendations: Classification societies may have different recommendations or requirements for the technology and tools used for shaft alignment. This can include the types of laser alignment tools or other specialized equipment that are approved for use.

Certification Criteria: The criteria for obtaining and maintaining certification for shaft alignment can differ. Each classification society has specific requirements for what constitutes satisfactory alignment, which must be met to achieve certification.

Training and Expertise Requirements: The qualifications and training required for personnel performing shaft alignment may vary. Some societies might have higher standards for the expertise and experience of technicians.[…]

[…]Compliance Audits: The approach to compliance audits and how deviations or non-conformities in shaft alignment are handled can differ. Some classification societies might have more rigorous follow-up procedures for addressing and rectifying alignment issues.

Overall, the choice of classification society can significantly influence the approach, rigor, and cost of maintaining proper shaft alignment on a vessel. Each society's unique standards and practices must be carefully considered when planning and executing shaft alignment procedures.

Author Note: The response consists of general areas of interest regarding the responsibilities of a Classification Society, while it should focus on the aspects of Shaft Alignment which are influenced by the Classification Society of choosing (e.g. proposed methodologies, equipment, limitations regarding design aspects). Once again, the model recognizes key-words related to the topic of interest and produces an extensive generic reply.

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-Q: From which stages does the Shaft Alignment Procedure of a vessel consist of?

-A: Shaft alignment procedures in vessels involve a series of steps to ensure that the propulsion system's shafts are correctly aligned to minimize vibration, reduce wear on bearings, and optimize the system's efficiency. While specific procedures may vary depending on the vessel type, propulsion system configuration, and alignment tools available, here is a general overview of the typical steps involved in shaft alignment:

Preparation:

Gather necessary tools and equipment for the alignment procedure, including alignment lasers, dial indicators, feeler gauges, shims, and measurement instruments.

Review the vessel's documentation, including alignment specifications provided by the manufacturer and any relevant regulatory requirements.

Ensure that the propulsion system is shut down and safely secured before beginning the alignment procedure.

Initial Assessment:

Inspect the shafting components, including shafts, couplings, bearings, and support structures, for signs of wear, damage, or misalignment.

Conduct a visual inspection of the alignment condition, looking for obvious signs of misalignment such as shaft deflection, coupling misalignment, or bearing wear.[…]

[…]Address any alignment issues or deviations promptly to prevent excessive wear, vibration, or damage to shafting components.

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By following these procedures, vessel operators can ensure that the shafting system is properly aligned, thereby enhancing the reliability, safety, and efficiency of the propulsion system. Regular alignment checks and maintenance are essential for preserving alignment integrity and minimizing the risk of premature component failure or operational disruptions.

Author Note: The desired response should mention clearly the three basic stages of Shaft Alignment: Design, Procedures and Measurements/Verification.

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II.II.IV. Questions with Different Responses:

-Q: Which aspects of Shaft Alignment are correlated to the classification society the Shipowner decides to comply with?

-A: The aspects of shaft alignment correlated to the classification society one chooses to comply with include the following:

1. Standards and Guidelines: Different classification societies might have slightly varying standards and guidelines for shaft alignment. These guidelines dictate acceptable limits for misalignment, tolerances, and the methodologies to be used during the alignment process. Choosing a particular classification society means adhering to their specific standards, which can impact the procedures and equipment used.
2. Approval Process: The process for getting alignment procedures and plans approved can vary between classification societies. Some may have more rigorous or detailed approval processes, requiring more comprehensive documentation and justification of the methods used.
3. Inspection and Survey Requirements: The frequency, scope, and detail of inspections and surveys related to shaft alignment can differ between classification societies. Some societies may have more frequent or detailed inspections, impacting how often alignment checks and adjustments need to be performed.
4. Certification Criteria: The criteria for certification, including the specific measurements and documentation required, can vary. The classification society's specific requirements for what constitutes satisfactory alignment must be met to obtain and maintain certification.
5. Documentation Standards: Different classification societies may require different levels of detail in the documentation of the shaft alignment process. This includes initial alignment reports, periodic maintenance records, and any adjustments made over the vessel's operational life.
6. Technical Support and Guidance: Classification societies often provide technical support and guidance for shaft alignment. The level and type of support available can vary, influencing how alignment tasks are approached and resolved.
7. Technological Requirements: The specific technologies and tools that are approved or recommended for shaft alignment may differ. Some societies might endorse certain advanced technologies or specific brands of alignment equipment, affecting the choices available to shipowners and operators.
8. Compliance Costs: The overall cost of compliance, including alignment procedures, inspections, documentation, and potential adjustments, can be influenced by the classification society's requirements. More stringent or frequent checks could result in higher costs.
9. Training and Expertise: The level of training and expertise required for personnel performing shaft alignment can be influenced by the classification society's standards. Some societies might have higher requirements for the qualifications and experience of technicians.
10. Impact on Operational Flexibility: The specific requirements and recommendations of the classification society can impact the operational flexibility of the vessel. For instance, stricter alignment tolerances might necessitate more frequent adjustments, affecting the vessel's operational schedule.

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Choosing a classification society means committing to their specific standards and practices, which can significantly impact the approach to and management of shaft alignment on the vessel.

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-Q: What changes in terms of Shaft Alignment depending on the classification society pf choosing?

-A: Shaft alignment requirements can vary depending on the classification society overseeing the vessel's construction and operation. Classification societies such as the American Bureau of Shipping (ABS), DNV GL (now known simply as DNV), Lloyd's Register (LR), Bureau Veritas (BV), and others set standards and guidelines for marine engineering practices, including shaft alignment. While the fundamental principles of shaft alignment remain consistent, specific requirements and procedures can differ based on the classification society's rules. Here are some ways in which these requirements may vary:

1. **Alignment Tolerances:** Different classification societies may specify varying tolerances for shaft alignment. These tolerances can include limits on angular and parallel misalignment, bearing offsets, and acceptable levels of shaft runout. Tighter tolerances might be required for certain types of vessels or higher-speed applications.

2. **Measurement Methods:** The methods prescribed for measuring and verifying shaft alignment can differ. Some societies may specify the use of particular types of measuring equipment (e.g., laser alignment tools, dial indicators) or specific measurement techniques (e.g., strain gauge measurements, optical alignment).

3. **Documentation and Reporting:** Requirements for documentation and reporting of alignment procedures and results can vary. Some classification societies may have detailed forms and checklists that need to be completed, while others may require specific types of reports or certifications from third-party alignment specialists.

4. **Dynamic vs. Static Alignment:** Certain societies might place more emphasis on dynamic alignment (taking into account operational conditions such as thermal expansion and hull deflection) versus static alignment (performed when the vessel is in a static state). The requirements for how these factors are accounted for can differ.

5. **Frequency of Inspections:** The required frequency of shaft alignment checks and inspections can vary. Some societies may mandate regular checks at specific intervals, while others may only require inspections under certain conditions, such as after a grounding incident or major repairs.

6. **Hull Deflections and Environmental Factors:** Different societies may have distinct guidelines on accounting for hull deflections and environmental factors (e.g., temperature changes, loading conditions) in the alignment process. These factors are critical in ensuring the alignment remains within acceptable limits under various operating conditions.

7. **Approval of Alignment Procedures:** The process for obtaining approval of alignment procedures and results from the classification society can vary. Some societies may require prior approval of alignment plans and procedures, while others may conduct post-alignment inspections and verifications.

8. **Use of Software and Technology:** The acceptance and requirement for specific alignment software and advanced technological tools can differ among societies. Some may have preferences or requirements for using certain software programs or technologies for alignment calculations and documentation.

Here's a brief overview of how some prominent classification societies approach shaft alignment:

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American Bureau of Shipping (ABS)

- **Tolerance Specifications:** ABS provides detailed guidelines on acceptable shaft alignment tolerances for different vessel types and operating conditions.
- **Measurement Methods:** ABS specifies the use of precise measurement tools and techniques, such as laser alignment systems and dial indicators, to ensure accurate alignment.
- **Documentation:** Comprehensive documentation and reporting of alignment procedures and results are required, including detailed records of measurements and any corrective actions taken.

DNV (Det Norske Veritas)

- **Dynamic Alignment:** DNV places significant emphasis on dynamic alignment considerations, taking into account factors such as hull deflection, thermal expansion, and varying operational loads.
- **Measurement Techniques:** DNV may require the use of advanced measurement techniques and tools, and they provide specific guidelines on how to perform these measurements accurately.
- **Reporting:** DNV requires detailed reports on the alignment process, including the methodologies used and the alignment results, to be submitted for review and approval.

Lloyd's Register (LR)

- **Tolerance and Methods:** Lloyd's Register specifies clear tolerance limits and recommended methods for shaft alignment, often focusing on the precision and accuracy of the alignment process.
- **Frequency of Checks:** LR may have specific requirements for the frequency of alignment checks, particularly after certain events like dry-docking or structural modifications.
- **Environmental Factors:** LR provides guidelines on accounting for environmental and operational factors that may affect shaft alignment, ensuring the vessel operates safely under all conditions.

Bureau Veritas (BV)

- **Alignment Procedures:** Bureau Veritas provides detailed procedures and best practices for performing shaft alignment, including both static and dynamic considerations.
- **Approval and Inspection:** BV often requires prior approval of alignment plans and may conduct thorough inspections post-alignment to verify compliance with their standards.
- **Advanced Tools:** BV encourages the use of advanced alignment tools and technologies, ensuring precise and reliable alignment results.

Commonalities and Differences

- **Common Principles:** While the specifics can vary, all classification societies emphasize the importance of accurate alignment to prevent excessive wear, vibration, and potential failure of the propulsion system.
- **Variability in Detail:** The level of detail and the specific requirements for documentation, measurement techniques, and reporting can vary significantly.
- **Focus Areas:** Some societies may focus more on dynamic conditions and operational factors, while others might emphasize static alignment procedures and tighter tolerance limits.

Conclusion

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Understanding the specific requirements and guidelines of the chosen classification society is crucial for ensuring compliant and effective shaft alignment procedures. Operators should refer to the latest rules and guidelines published by their respective classification society to ensure they meet all necessary standards and maintain the vessel's operational integrity and safety.

Author Note: The "differences" between the Classification Societies which are mentioned in the model's response are untrue. However, the structure and the details provided in the response are different compared to the previous question.

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-Q: Elaborate the term "Aft Stern Tube Misalignment Angle"

-A: The term "Aft Stern Tube Misalignment Angle" refers to the angular deviation between the aft stern tube and the intended alignment axis. In marine propulsion systems, the stern tube is a hollow tube at the stern (rear) of the vessel through which the propeller shaft passes. Proper alignment of the stern tube is crucial for the efficient and safe operation of the propulsion system.

Here's what you need to know about the "Aft Stern Tube Misalignment Angle":

Definition: The misalignment angle is the angular difference between the actual alignment axis of the aft stern tube and the desired or intended alignment axis. It represents the deviation from the ideal alignment condition.

Measurement: The misalignment angle is typically measured using precision alignment tools such as laser alignment systems or dial indicators. These tools provide accurate measurements of the angular deviation between the stern tube's centerline and the reference axis.

Causes: Aft stern tube misalignment can occur due to various factors, including inaccuracies during installation, structural deflections, thermal expansion, hull deformations, or wear and tear over time. Improper alignment can lead to increased vibration, wear on bearings, shaft misalignment, and reduced efficiency of the propulsion system.

Impact: Excessive misalignment of the aft stern tube can have several adverse effects on the propulsion system and vessel operation. It can lead to increased friction, wear, and stress on the shafting components, resulting in premature failure of bearings, seals, and couplings. Additionally, misalignment can cause vibration, noise, and reduced fuel efficiency, affecting the overall performance and safety of the vessel.

Correction: Corrective measures may be required to address aft stern tube misalignment. This may involve adjusting the position of the stern tube, realigning the propeller shaft, or installing shims or other alignment aids to achieve proper alignment. Precise alignment ensures that the stern tube operates within acceptable tolerances, minimizing wear and maximizing the efficiency of the propulsion system.

In summary, the aft stern tube misalignment angle is a critical parameter in marine propulsion systems, and proper alignment is essential for ensuring the safe, efficient, and reliable operation of the vessel. Regular inspection, maintenance, and corrective actions are necessary to address any misalignment issues and maintain optimal performance.

Author Note: This specific question was once answered wrongly. When it was re-addressed to the model, the answer was different.

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-Q: Can a vessel have only one stern tube bearing?

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-A: Yes, a vessel can have just one stern tube bearing, although it's less common. In some ship designs, especially smaller vessels or those with specific operational requirements, a single stern tube bearing may be sufficient to support the propeller shaft. However, many vessels, particularly larger ones or those with higher power requirements, typically have multiple stern tube bearings to provide additional support and ensure proper alignment and load distribution along the propeller shaft. Having multiple bearings can also help reduce the risk of shaft misalignment and minimize vibration and wear on the shaft and bearing surfaces.

Author Note: This specific question was previously answered in a different way: The first answer was the following:

"In general, a vessel can have only an aft stern tube bearing if it is equipped with a single-propeller shaft propulsion system and does not have additional intermediate shafts or line shafts. However, the presence and configuration of bearings in a ship's propulsion system depend on various factors, including the vessel's design, propulsion arrangement, size, and intended use. Here are some scenarios where a vessel might have only an aft stern tube bearing: Single Propeller Shaft Configuration: Vessels with a single propeller shaft typically have one aft stern tube bearing to support the propeller shaft as it extends from the hull into the water. In this configuration, the aft stern tube bearing is the primary bearing supporting the propeller shaft. Direct Drive Propulsion: Some vessels employ a direct drive propulsion system where the main engine is directly connected to the propeller shaft without intermediate shafts or reduction gears. In such cases, only an aft stern tube bearing may be required to support the propeller shaft. Simpler Propulsion Systems: Smaller vessels or vessels with simpler propulsion systems may have fewer bearings to reduce complexity and maintenance requirements. In these cases, a single aft stern tube bearing may be sufficient to support the entire propeller shaft system. Specific Design Requirements: Certain vessel designs or applications may dictate the use of only an aft stern tube bearing. For example, vessels with limited space or weight constraints may opt for a simpler bearing arrangement to optimize performance and efficiency. While it's possible for a vessel to have only an aft stern tube bearing, it's essential to ensure that the bearing is properly designed, installed, and maintained to support the propeller shaft effectively. Proper alignment, lubrication, and monitoring are crucial to ensure the reliable and safe operation of the propulsion system. Additionally, larger or more complex vessels may require additional bearings, such as intermediate shaft bearings or line shaft bearings, depending on their propulsion configuration and operational requirements."

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III. Selection of Opensource LLMs for Comparison

For the purposes of this study, utilizing the GPT4All platform, the following Models where selected:

- GPT4ALL Falcon
- Llama 3 Instruct
- Nous Hermes 2 Mistral DPO

The selection of these specific models was based on the Performance Benchmark published on GPT4ALL's website. More precisely, each Model's performance is rated as shown in the table below:

Model:	BoolQ	PIQA	HellaSwag	WinoGrande	ARC-e	ARC-C	OBQA	Avg.
GPT4ALL Falcon	**77.6**	**79.8**	**74.9**	**70.1**	**67.9**	**43.4**	**42.6**	**65.2**
Llama 3 8B Instruct	**75.7**	**79.6**	**60.2**	**76.1**	**80.3**	**50.5**	**42.2**	**66.4**
Nous Hermes 2 Mistral DPO	**83.9**	**80.7**	**80.1**	**71.3**	**75.7**	**52.1**	**46.2**	**70**
Llama 7B	73.1	77.4	73	66.9	52.5	41.4	42.4	61

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Llama 13B	68.5	79.1	76.2	70.1	60	44.6	42.2	63
GPT4ALL-J 6B v1.0	73.4	74.8	63.4	64.7	54.9	36	40.2	58.2
GPT4ALL-J v1.1-breezy	74	75.1	63.2	63.6	55.4	34.9	38.4	57.8
GPT4ALL-J v1.2-jazzy	74.8	74.9	63.6	63.8	56.6	35.3	41	58.6
GPT4ALL-J v1.3-groovy	73.6	74.3	63.8	63.5	57.7	35	38.8	58.1
GPT4ALL-J Lora 6B	68.6	75.8	66.2	63.5	56.4	35.7	40.2	58.1
GPT4ALL LLaMa Lora 7B	73.1	77.6	72.1	67.8	51.1	40.4	40.2	60.3
GPT4ALL 13B snoozy	83.3	79.2	75	71.3	60.9	44.2	43.4	65.3
Nous Hermes	79.5	78.9	80	71.9	74.2	50.9	46.4	68.8
Nous-Puffin	81.5	80.7	80.4	72.5	77.6	50.7	45.6	69.9
Dolly 6B	68.8	77.3	67.6	63.9	62.9	38.7	41.2	60.1
Dolly 12B	56.7	75.4	71	62.2	64.6	38.5	40.4	58.4
Alpaca 7B	73.9	77.2	73.9	66.1	59.8	43.3	43.4	62.5
Alpaca Lora 7B	74.3	79.3	74	68.8	56.6	43.9	42.6	62.8
GPT-J 6.7B	65.4	76.2	66.2	64.1	62.2	36.6	38.2	58.4
Pythia 6.7B	63.5	76.3	64	61.1	61.3	35.2	37.2	56.9
Pythia 12B	67.7	76.6	67.3	63.8	63.9	34.8	38	58.9
Fastchat T5	81.5	64.6	46.3	61.8	49.3	33.3	39.4	53.7
Fastchat Vicuna 7B	76.6	77.2	70.7	67.3	53.5	41.2	40.8	61
Fastchat Vicunma 13B	81.5	76.8	73.3	66.7	57.4	42.7	43.6	63.1
StableVicuna RLHF	82.3	78.6	74.1	70.9	61	43.5	44.4	65
StableLM Tuned	62.5	71.2	53.6	54.8	52.4	31.1	33.4	51.3
StableLM Base	60.1	67.4	41.2	50.1	44.9	27	32	46.1
Koala 13B	76.5	77.9	72.6	68.8	54.3	41	42.8	62
Open Assistant Pythia 12B	67.9	78	68.1	65	64.2	40.4	43.2	61

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Mosaic MPT7B	74.8	79.3	76.3	68.6	70	42.2	42.6	64.8
Mosaic mpt-instruct	74.3	80.4	77.2	67.8	72.2	44.6	43	65.6
Mosaic mpt-chat	77.1	78.2	74.5	67.5	69.4	43.3	44.2	64.9
Wizard 7B	78.4	77.2	69.9	66.5	56.8	40.5	42.6	61.7
Wizard 7B Uncensored	77.7	74.2	68	65.2	53.5	38.7	41.6	59.8
Wizard 13B Uncensored	78.4	75.5	72.1	69.5	57.5	40.4	44	62.5
GPT4-x-Vicuna-13b	81.3	75	75.2	65	58.7	43.9	43.6	63.2
Falcon 7b	73.6	80.7	76.3	67.3	71	43.3	4.4	65.2
Falcon 7b instruct	70.9	78.6	69.8	66.7	67.9	42.7	41.2	62.5
Text-davinci-003	88.1	83.8	83.4	75.8	83.9	63.9	51	75.7

For clarification reasons, each benchmark dataset will be briefly explained:

Benchmark Datasets

BoolQ

BoolQ is a Benchmark Dataset used for training and evaluating natural language understanding models, like LLMs, which focuses on the Question Answering capabilities of the model in terms of YES/NO questions. Its purpose is to examine a model's capacity to determine the truth value of a statement (in response to a question). BoolQ represents an evaluation of how well a model can comprehend a passage and determine the correct binary answer.

PIQA

PIQA is a Benchmark Dataset which can be used in order to evaluate a LLM's capabilities in terms of understanding physical interaction issues. It focuses on testing models regarding their common sense and reasoning skills (how objects are used, how certain actions lead to specific outcomes). It represents the capability of the model regarding the understanding of practical tasks.

HellaSwag

HellaSwag is a Benchmark Dataset which can be utilized in order to assess the performance of a LLM in matters of common-sense reasoning as well as understanding and predicting human-like texts. It focuses on the understanding of physical interactions, social dynamics and general common-sense knowledge.

WinoGrande

WinoGrande is a Large-Scale Dataset with the scope of evaluating and advancing the commonsense reasoning capabilities of a model. More specifically, it focuses on evaluating models by testing their capabilities regarding common sense reasoning by resolving pronoun references in complex sentences. Consequently, WinoGrande assists in determining how well can a model perform when it comes to common sense reasoning.

ARC-e

ARC-e represents the "Easy" subset of AI2 Reasoning Challenge (ARC). Its center of attention is on assessing the problem-solving and reasoning capabilities of the model with a focus on science. Its main scope is to

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evaluate the performance of the model when it comes to elementary and middle school level science questions. ARC-e aims in assessing the model's capabilities in terms of reasoning in a structured format.

ARC-c

ARC-c represents the "Challenge" subset of AI2 Reasoning Challenge (ARC). This specific subset of ARC focuses on evaluating the performance of the model regarding advanced reasoning and problem-solving capabilities. In comparison to ARC-e, the dataset uses questions which require deeper understanding, inference and application of knowledge. ARC-c represents how well can a model perform beyond the simple fact retrieval tasks (understanding and applying of knowledge), when it comes to more complex questions.

OBQA

OBQA (OpenBookQA) is a Benchmark Dataset utilized in order to evaluate a model's performance regarding advanced reasoning and understanding when it comes to science questions. OBQA's scope is to test a model's capability in answering elementary level science questions, but the questions are designed so that more than a simple fact retrieval is asked from the model.

It is obvious, that ARC-e, ARC-c and OBQA are the Benchmark Datasets which present the more interest for the present study, as they focus on science-oriented capabilities, which are a key-factor when it comes to this study which focuses on science-based Regulations which encompass mathematical equations and Physics. However, all of the other Benchmark Datasets are of interest, as they represent the reasoning skills as well as the capability of the models in terms of forming structured answers with satisfactory reasoning by adequately understanding the Local Docs provided.

The models which were finally chosen were characterized by remarkable performance when it came to the ARC-e, ARC-c and OBQA Benchmark Datasets in comparison with the rest of the models which could be utilized.

At this point, it is important to mention that during the experimental phase with the LLMs, it was obvious that their performance regarding the present Study's purposes was significantly low when the Local Documents provided did not include the First Part of this Thesis. As a result, the Local Documents which the LLMs digested consisted of: The Frist Part of this Thesis and all of the Bibliography utilized for its composition.

IV. Assessment of the responses Large Language Models provided

In order for the evaluation of the performance LLMs note in matters concerning answering specific technical questions to be conducted, the same set of questions was addressed to the LLMs of choice and the responses were documented. For clarification reasons, the LLMs of choice are once again stated.

- GPT4ALL Falcon
- Llama 3 Instruct
- Nous Hermes 2 Mistral DPO

In this chapter, the performance of each model is presented.

IV.I. Llama 3 Instruct

As far as Comprehension related questions are of interest, Llama 3 was characterized by an overall satisfying performance regarding these type of questions **[1a].** However, in some cases, the response the model provided could have been more elaborated. Specifically, there were opportunities for the model to enhance its answers by more effectively integrating information from both the provided local documents and its pre-existing knowledge (as in information and patterns the model has learned from its initial training phase). On the other hand, Llama 3 responded with high competency to several questions which required combination and deep understanding of several distinguished topics provided through the Local Documents **[1b]**. It is worth mentioning, that when consequent similar questions with a comprehensive character were addressed to the model, significantly less repetitive same answers were given by Llama 3 **[1c]**, contrary to ChatGPT and the rest

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of the models utilized for the purposes of this Thesis (Nous Hermes 2 and GPT4ALL Falcon). This behavior correlates to more sufficient understanding of the questions which are addressed to the model. The reason for this behavior may be detected in the generation parameters of the model, such as higher temperature (<1) and/or Top-k sampling or Top-p sampling values. The performance of Llama 3 with respect to deep understanding of complex terminology and text generation was deemed as overall satisfying**[1d]**. Nevertheless, compared to Nous Hermes 2, some of the responses provided by Llama 3 were of remarkably lower quality. However, the model showed good signs which indicated confidently that via a fine-tuning process with an appropriate Dataset, the responses could be highly promising. This is because:

A) Llama 3 showed signs of highly adequate understanding of the "basics" of the regulations which were provided through the Local Documents and
B) Even in cases where the responses the model gave indicated insufficient understanding of the question or insufficient information, the responses were overall in context and deemed as satisfactory **[1e]** compared to the relevant responses of ChatGPT, Nous Hermes 2 Mistral DPO and GPT4ALL Falcon in same cases.

However, it should be mentioned that Llama 3 indicated signs of lack in terms of elaborating when it came to Regulation matters compared to Nous Hermes 2 Mistral DPO. However, the overall performance of the model with respect to questions related to Fact Retrieval and Regulation Limits was satisfactory. In fact, compared to Nous Hermes 2 Mistral DPO and GPT4ALL Falcon, even in questions which did not specifically require mentioning of the specific Regulations which govern the topic, Llama 3 several times provided the Regulatory Framework behind the subject of interest **[1f]**. This is a high-quality characteristic which cannot be overseen, as it can prove to be significantly valuable, especially if proper Fine-Tuning takes place regarding Shaft Alignment and Regulation matters. However, it should be mentioned that Llama 3 lacked in terms of further elaborating and explaining on the Requirements entangled in its responses, contrary to Nous Hermes 2 Mistral DPO.

When questions which belong to the "Complex" category were addressed to Llama 3, the overall performance was deemed satisfactory. Nonetheless, Nous Hermes 2 Mistral DPO showed signs of better performance when it came to this specific category of questions. Once again, it is worth mentioning that if proper Fine-Tuning with a suitable dataset is conducted, it is a confident estimation that the responses provided by Llama 3 will be of high quality and accuracy. More specifically, Llama 3 remarkably lacked compared to Nous Hermes 2 Mistral DPO when it came to questions which involved "tricky" terminology. Additionally, Llama 3 faced more difficulties in terms of completely understanding the specific context of more complex questions.

Conclusively, Llama 3 was characterized by a solid performance when it came to comprehensive questions, as it provided well-structured and well-reasoned responses. In comparison to Nous Hermes 2 Mistral DPO and GPT4ALL Falcon, Llama 3 outweighed in terms of responding to questions which entangled differences and contrasts between Classification Societies, as it several times even provided the Regulatory Framework governing the specific issue of interest, even if it was not directly requested in the question which was addressed to the model. This characteristic showcases its precedence compared to the rest of the models in terms of detail and combination of facts and knowledge. Moreover, in some cases, Llama 3 even provided in the context of elaborating on a subject solutions and proposals which were on context and overall satisfactory **[1g]**. Moreover, the performance of the model in questions which were strictly Science Related was also adequate. However, Nous Hermes 2 Mistral DPO in several cases demonstrated more elaborative responses. In conclusion, it is safe to say, that in "normal difficulty" questions the performance of the model was more than adequate and that through Fine-Tuning by utilizing an appropriate and tailored Dataset, it has the potential of becoming a reliable and useful tool in the hands of student and/or professionals with a scope of classifying and retrieving information from large amounts of documents specified in a precise technical subject, like Shaft Alignment.

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For clarification reasons, references regarding the conclusions extracted from the procedure of evaluating Llama 3 are now given:

[1a]: **Q1:** Can a vessel have only aft stern tube bearing?

A: Aft Stern Tube Bearing Only:

In some cases, it is possible for a vessel to have only an aft stern tube bearing, with no forward stern tube bearing. This design is often referred to as a "single-slope" stern tube design.
However, this configuration requires careful consideration and analysis by the engineer to ensure that the loads on the aft stern tube bearing are properly managed. The absence of a forward stern tube bearing means that the entire weight of the propeller and propulsion system is borne solely by the aft stern tube bearing.
Design Considerations:
To make an all-aft stern tube bearing design viable, the following factors must be carefully considered:
1. Increased load: The aft stern tube bearing will need to handle a significantly increased load due to the absence of a forward counterpart.
2. Double slope design: A double-slope design for the aft stern tube bearing is often necessary to ensure proper lubrication and heat dissipation.
3. Specialized lubricant: A specialized lubricant may be required to effectively manage the increased loads and temperatures in this configuration.
Practical Considerations:
While an all-aft stern tube bearing design is theoretically possible, it's essential to weigh the potential benefits against the added complexity and potential drawbacks:
1. Simplified maintenance: With only one stern tube bearing, maintenance might be simplified.
2. Cost savings: Fewer bearings may result in cost savings.
However, these advantages must be balanced against the increased risk of failure due to the higher loads on a single bearing.
Conclusion:
In summary, while it is possible for a vessel to have only an aft stern tube bearing, careful consideration and analysis are necessary to ensure that this design can effectively manage the loads and temperatures involved.
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[1a]: Q2: What does ABS suggest regarding single and multi point contact methodologies?
A: ABS Suggestion on Single-Point Contact (SPC) and Multi-Point Contact (MPC)
According to the American Bureau of Shipping (ABS), when performing shaft alignment analysis, you should consider both single-point contact (SPC) and multi-point contact (MPC) approaches. Here's what ABS suggests:
1. Initial assumption: Start by assuming a combined approach that includes both SPC and MPC.
2. Single-Point Contact (SPC): Use the SPC method as an initial estimate to quickly calculate the contact pressure between the shaft and bearing.
3. Multi-Point Contact (MPC): If necessary, refine your analysis using the MPC method, which accounts for distributed contact pressures along the length of the bearing.
ABS suggests this approach because:
1. SPC is simpler: SPC provides a quick estimate of the contact pressure, making it easier to perform.
2. MPC is more accurate: MPC takes into account complex interactions between the shaft and bearing, providing a more detailed understanding of the system's behavior.
3. Combined approach: By considering both methods, you can gain insight into the effects of different design parameters on the contact pressure.
By following ABS' suggestion, you'll be able to:
1. Quickly estimate the initial contact pressure using SPC
2. Refine your analysis with MPC if necessary
3. Gain a deeper understanding of the shaft alignment system's behavior

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Remember that this is just an initial assumption, and further refinements may be needed based on specific design requirements or experimental validation.
References:
* American Bureau of Shipping (ABS). (n.d.). Shaft Alignment Analysis.
* ABS Guide for Building and Classing Ships. (2020).
Please let me know if you have any follow-up questions!
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[1a]: Q3: Is the pre-sighting procedure optional when epoxy resin is applied?
A: When applying epoxy resin to a surface, presighting (also known as sighting) can be considered an optional step, but it's highly recommended for achieving optimal results.
Why Presight Before Applying Epoxy Resin:
1. Accurate Application: Sighting ensures the epoxy resin is applied precisely along the desired path or pattern, which is crucial when working with complex shapes or tight spaces.
2. Reduced Waste: By accurately sighting before applying the epoxy, you can minimize waste and avoid unnecessary excess material.
3. Improved Adhesion: A well-sighted application helps ensure a strong bond between the resin and the surface being coated.

When Presighting Might Not Be Necessary:
1. Simple Coatings: For straightforward applications where the coating is not critical to the overall performance or appearance of the part, presight might be unnecessary.
2. Small Areas: When working with small areas or simple shapes, the risk of error is lower, and presight may not add significant value.
In Summary:
While presighting before applying epoxy resin is an optional step, it's highly recommended for achieving optimal results in most cases. The benefits of accurate application, reduced waste, and improved adhesion make the extra effort worthwhile. However, if you're working with simple coatings or small areas, you might be able to get away without presight.
Do you have any specific epoxy resin applications or concerns? I'm here to help!
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[1b]: Q: Is the flexibility of the shaft connected to the flexibility of the vessel's hull? Does this relationship affect Shaft Alignment?
A: Yes, the flexibility of the shaft is indeed connected to the flexibility of the hull.
In a ship's propulsion system, both the shaft and the hull are subject to various forms of deformation (flexibility). The hull deformations can be caused by factors like:
1. Water pressure
2. Wind loads
3. Thermal expansions/contractions
These deformations can affect the alignment of the shaft, which is typically designed to operate within a specific tolerance range.
The connection between the flexibility of the shaft and hull lies in their coupling through various interfaces (e.g., bearings, couplings). When the hull deforms due to external loads or internal stresses, it can cause corresponding deformations in the shaft. This effect is often referred to as hull-induced shaft deflection.
The impact of this coupling on shaft alignment design is significant:
1. Shaft misalignment: The deformation of the hull and/or shaft can lead to changes in their relative positions, causing misalignment between the propeller and the main engine.
2. Increased vibration: As the shaft becomes misaligned, it may vibrate more severely due to the altered dynamics of the system.
To mitigate these effects, designers often employ various strategies:

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1. Stiffer hull design: Building a stiffer hull can reduce its deformation under load, which in turn minimizes the impact on shaft alignment.
2. Shaft flexibility compensation: Designing the shaft with increased flexibility or using specialized bearings/couplings to accommodate the deformations can help maintain proper alignment.
So, yes! The flexibility of the shaft is indeed connected to that of the hull, and their coupling has a significant impact on shaft alignment design in ship propulsion systems.
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[1c]: Q1: What impact do Classification Societies have on the Shaft Alignment of a vessel?
A: Classification Societies play a crucial role in ensuring the safe and efficient operation of vessels.
When it comes to shaft alignment, Classification Societies like ABS (American Bureau of Shipping), DNV GL (Det Norske Veritas - Germanischer Lloyd), LR (Lloyd's Register), RINA (Rina S.p.A.), and others have established guidelines and regulations that vessel owners/operators must adhere to. These societies set standards for the design, construction, and operation of vessels, including shaft alignment.
Here are some key ways Classification Societies impact shaft alignment:
1. Regulatory framework: Each society has its own rules and regulations governing shaft alignment, which vessel designers, builders, and operators must comply with.
2. Guidelines and recommendations: Classification Societies provide guidelines and recommendations for achieving optimal shaft alignment, taking into account factors like hull deflections, propeller pitch, and bearing selection.
3. Inspection and certification: During the construction or repair of a vessel, Classification Societies inspect the shafting system to ensure compliance with their regulations. Upon successful inspection, they issue certifications that validate the vessel's seaworthiness.
4. Periodic surveys: After initial certification, vessels must undergo periodic surveys (typically every 5-10 years) to verify continued compliance with regulatory requirements, including shaft alignment.
Some specific guidelines and recommendations from Classification Societies include:
* ABS: The American Bureau of Shipping recommends that hull deflections are considered when determining the optimal shaft alignment.
* DNV GL: Det Norske Veritas - Germanischer Lloyd suggests that hull deflections should be taken into account during the design phase to ensure proper shaft alignment.
By following these guidelines and regulations, vessel owners/operators can minimize risks associated with poor shaft alignment, such as increased wear on bearings, reduced propulsion efficiency, or even catastrophic failures.
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[1c]: Q2: Which aspects of Shaft Alignment are correlated to the Classification Society a Shipowner chooses to comply with?
A: Different Classification Societies have varying approaches to shaft alignment. Here's an overview of some key differences:
1. Hull Deflections: ABS (American Bureau of Shipping) emphasizes the importance of considering hull deflections when determining optimal shaft alignment. In contrast, DNV GL (Det Norske Veritas - Germanischer Lloyd) focuses more on the propeller pitch and bearing selection.
2. Shaft Alignment Tolerances: LR (Lloyd's Register) sets stricter tolerances for shaft alignment than RINA (Rina S.p.A.). For example, LR requires a maximum deviation of 0.5 mm/m from the ideal alignment, while RINA allows up to 1.0 mm/m.
3. Bearing Selection and Installation: Classification Societies have different guidelines for bearing selection and installation. For instance, ABS recommends using bearings with a higher load capacity than DNV GL, which emphasizes the importance of proper bearing installation and maintenance.
4. Shaft Alignment Procedure: The procedure for achieving optimal shaft alignment varies between societies. LR requires a more detailed analysis of hull deflections, while RINA focuses on propeller pitch and bearing selection.
Here's a brief summary:

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* ABS: Emphasizes consideration of hull deflections in the Shaft Alignment process.
* DNV GL: Focuses on Propeller Pitch and Bearing Selection for optimal shaft alignment.
* LR (Lloyd's Register): Sets stricter tolerances for shaft alignment (<0.5 mm/m) and emphasizes detailed analysis of hull deflections.
* RINA (Rina S.p.A.): Allows slightly higher deviations in shaft alignment (>1.0 mm/m) but stresses the importance of proper bearing selection and installation.
When choosing a Classification Society, vessel owners/operators should consider these differences to ensure compliance with regulatory requirements and optimize their shaft alignment procedures for efficient operation and reduced maintenance costs.
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[1d]: Q1: Do you know anything about the consequences if hull deformations are not taken into consideration during the design stage of Shaft Alignment?
Response: Yes, I do! If hull deformations are not considered during the design stage of shaft alignment, it can lead to a range of negative consequences. Here are some potential issues:
1. Misalignment: Failure to account for hull deformations can result in misalignment between the propulsion system components (e.g., bearings, couplings), leading to vibration, noise, and reduced efficiency.
2. Increased Vibration: Hull deformations can cause increased vibration levels due to changes in shaft alignment, which may lead to premature wear on equipment or even catastrophic failures.
3. Reduced Efficiency: Misalignment caused by hull deformations can reduce the overall efficiency of the propulsion system, resulting in decreased performance and potentially higher fuel consumption.
4. Increased Maintenance Costs: Neglecting hull deformation effects during design can lead to increased maintenance costs due to premature wear on equipment or frequent repairs.
5. Reduced Reliability: Hull deformations can compromise the reliability of the propulsion system by introducing unwanted stress concentrations, which may cause failures under various operating conditions.
6. Safety Risks: In extreme cases, misalignment caused by hull deformations can pose safety risks to personnel and vessels, particularly in situations where reduced performance or vibration leads to loss of control or stability.
To mitigate these consequences:
1. Conduct thorough analysis: Perform detailed calculations and simulations that account for hull deformation effects during the design stage.
2. Collaborate with experts: Work closely with experienced designers, engineers, and naval architects who have knowledge of hull deformations and their impact on propulsion systems.
3. Implement robust designs: Design propulsion systems to accommodate potential changes in hull shape or deflection, ensuring optimal performance under various operating conditions.
By considering the consequences of neglecting hull deformation effects during design, you can develop a more reliable, efficient, and safe propulsion system that effectively accommodates these deformations.
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[1d]: Q2: Which characteristics of Shaft Alignment have an effect on the oil film's characteristics?
A: The characteristics of Shaft Alignment that can affect the oil film's characteristics are:
1. Shaft position and orientation: The alignment of the shaft with respect to the bearing, including any deviations from perfect parallelism or perpendicularity.
2. Bearing offset values: Any differences in the distance between the shaft centerline and the bearing centers (i.e., bearing offsets) can influence oil film thickness and stability.
3. Shaft runout and vibration: The degree of deviation from a perfectly straight line, as well as any vibrations or oscillations that may occur during operation, can impact oil film quality and behavior.
4. Bearing clearance and preload: The amount of space between the bearing inner ring and the shaft (clearance) or the force applied to the bearings before they start rotating (preload), which affects the initial contact pressure and subsequent oil film development.
These Shaft Alignment characteristics influence the oil film's:
1. Thickness: Changes in shaft position, orientation, runout, and vibration can alter the oil film thickness.

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2. Stability: Bearing offset values, clearance, and preload affect the stability of the oil film, influencing its ability to maintain a consistent thickness and prevent breakdown or starvation.
3. Pressure distribution: The alignment characteristics influence the pressure distribution within the oil film, which affects lubrication performance.

By considering these Shaft Alignment factors, you can better understand how they impact the oil film's behavior and optimize your design for improved reliability, reduced wear, and increased overall system efficiency.

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[1d]: Q3: Does the specific location of the intermediate bearing play an important role in shaft alignment?
A: Yes, the specific location of the intermediate bearing can have a significant impact on shaft alignment.
The longitudinal placement of the intermediate bearing can affect:
1. Reaction forces: The span between the bearings has a strong effect on the reaction forces that apply to them. A well-placed intermediate bearing can help distribute these forces more evenly.
2. Misalignment angles: The exact location and vertical offset of the intermediate shaft bearing play a crucial role in determining the misalignment angle of the aft stern tube bearing.
Optimizing the location of the intermediate bearing is essential for achieving proper shaft alignment, as it:
1. Influences bearing reactions: A well-placed intermediate bearing can help reduce reaction forces on the bearings and minimize potential damage.
2. Affects shaft movement: The correct placement of the intermediate bearing helps to control shaft movement and reduces the risk of misalignment.
To ensure proper shaft alignment, consider the following guidelines:
1. Place the intermediate bearing at a location that minimizes the span between it and the aft stern tube bearing.
2. Ensure the vertical offset is within acceptable limits (typically ±10-20% of the bearing diameter).
3. Verify that the calculated misalignment angle meets the design requirements.
By carefully considering the specific location of the intermediate bearing, you can optimize shaft alignment and ensure a more reliable and efficient vessel operation.

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[1d]: Q4: How can the clearances of the aft stern tube be calculated?
A: Calculations of Aft Stern Tube Clearances!
The calculations for determining the clearance of an aft stern tube (ASTB) involve several steps:
Step 1: Determine the Shaft Deflection
* Calculate the shaft deflection at the aft edge of the ASTB using a displacement vector table or similar method.
* This value represents the distance between the shaft's original position and its current position due to thermal expansion, vibration, or other factors.
Step 2: Calculate the Bearing Slope
* Determine the absolute value of the bearing slope at the same location as Step 1 (aft edge of ASTB).
* The bearing slope is typically measured in degrees or radians and represents the angle between the shaft's axis and the plane of the bearing.
Step 3: Calculate Clearance
* Use the calculated shaft deflection and bearing slope to determine the clearance.
* Clearance = Shaft Deflection + Bearing Slope
The resulting value represents the total clearance at the aft edge of the ASTB. This calculation is typically performed using specialized software or spreadsheets, as shown in Figure 75 (Example of ABS aft stern tube evaluation software).
Additional Considerations:
1. Tolerances: Allow for tolerances and uncertainties in the measurements to ensure a realistic estimate.
2. Operating Conditions: Account for changes in operating conditions that may affect shaft deflection or bearing slope, such as temperature fluctuations or vessel speed variations.

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References:
1. ABS (2019). Rules for Building and Classing Steel Vessels - Part 2A: Hull Structures.
2. BV (2020). Rules for Building and Classing Steel Vessels - Part 2A: Hull Structures.
3. DNV GL (2020). Recommended Practice for the Design, Construction, and Operation of Marine Propulsion Systems.
Please note that this is a general overview of common industry practices, and actual calculations may vary depending on specific vessel design, operating conditions, and regulatory requirements.

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[1d]: Q5: Explain the importance of the intermediate bearing in systems with no forward stern tube bearing.
A: The Importance of Intermediate Bearings in Systems with No Forward Stern Tube Bearing
When designing propulsion shafting systems, the presence or absence of intermediate bearings can significantly impact vessel performance, maintenance, and overall design considerations. In particular, when there is no forward stern tube bearing (AT), an intermediate bearing plays a crucial role in ensuring proper system operation.
Why Intermediate Bearings are Important:
1. Shaft alignment: With no forward AT, the shaft may be more prone to misalignment due to hull deflections or other external factors. An intermediate bearing helps maintain shaft alignment by providing additional support and reducing stress concentrations.
2. Vibration reduction: The absence of a forward AT can lead to increased vibration transmission from the propeller to the vessel's structure. Intermediate bearings help absorb these vibrations, reducing their impact on the overall system.
3. Load sharing: When there is no forward AT, intermediate bearings share the load with the aft stern tube bearing (AT), ensuring that both bearings are not subjected to excessive loads and stress concentrations.
Design Considerations:
1. Location: The location of the intermediate bearing is critical in systems with no forward AT. It should be positioned to effectively share loads and reduce stress concentrations.
2. Type: The type of intermediate bearing used can impact performance, maintenance, and cost considerations. Common types include plain bearings, journal bearings, or thrust bearings.
3. Clearance control: Maintaining proper clearance between the shaft and the intermediate bearing is essential for optimal system operation.
Conclusion:
In systems with no forward stern tube bearing (AT), an intermediate bearing plays a vital role in ensuring proper shaft alignment, reducing vibration transmission, sharing loads, and improving reliability. By carefully considering design factors such as location, type, and clearance control, engineers can optimize the performance of these critical components.
What are your thoughts on this topic? Share your experiences or questions in the comments below!

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[1e]: Q: What is a common industry method utilized for wear measurements on aft stern tube bearing?
A: Wear Measurements on Aft Stern Tube Bearing!
In the marine industry, a commonly used method for measuring wear on aft stern tube bearings (ASTB) is:
Visual Inspection and Measurement
1. Recorded measurements: Record the initial measurement of the ASTB's axial clearance at regular intervals (e.g., every 1000 hours or as recommended by the manufacturer).
2. Wear rate calculation: Calculate the wear rate based on the measured changes in axial clearance over time.
3. Remaining operational life estimation: Estimate the remaining operational life of the ASTB based on its current condition and expected operating conditions.
Common Wear Measurement Methods:
1. Axial Clearance Measurements: Measure the distance between the bearing's inner ring and the shaft using a precision caliper or micrometer.

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2. Shaft Displacement Measurements: Measure the displacement of the shaft within the bearing housing using an LVDT (Linear Variable Differential Transformer) sensor or similar technology.
ASTB Wear Classification:
1. Normal wear: Axial clearance increases by 0.5 mm to 1.5 mm over a period of 1000 hours.
2. Moderate wear: Axial clearance increases by 1.6 mm to 3.9 mm over a period of 1000 hours.
3. Severe wear: Axial clearance increases by more than 4 mm over a period of 1000 hours.
Industry Recommendations:
1. ASTB inspection and measurement interval: Every 1000 hours or as recommended by the manufacturer.
2. Wear rate calculation: Based on measured changes in axial clearance over time, considering factors like operating conditions, vessel speed, and load.
3. Remaining operational life estimation: Estimated based on current condition, expected operating conditions, and wear rate calculations.
References:
1. ABS (2019). Rules for Building and Classing Steel Vessels - Part 2A: Hull Structures.
2. BV (2020). Rules for Building and Classing Steel Vessels - Part 2A: Hull Structures.
3. DNV GL (2020). Recommended Practice for the Design, Construction, and Operation of Marine Propulsion Systems.
Please note that this is a general overview of common industry practices, and actual procedures may vary depending on specific vessel design, operating conditions, and regulatory requirements.
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[1f]: Q: Can a reaction force of a bearing be negative in Shaft Alignment?
A: No, according to most classification societies (e.g., ABS, DNV GL, Lloyd's Register), the reaction force of a bearing in shaft alignment is not allowed to be negative.
In general, bearings are designed to operate with positive reaction forces, which indicate that the bearing is supporting the load and maintaining proper alignment. A negative reaction force would suggest that the bearing is actually resisting or opposing the movement of the shaft, rather than supporting it.
Most classification societies have specific guidelines for acceptable bearing reactions in shaft alignment designs. For example:
* ABS: "The calculated bearing reactions shall be positive under all service drafts" (ABS Guide for Building and Classifying Ships).
* DNV GL: "Bearing reaction forces should always be positive during normal operation" (DNV GL Rules for Classification).
Negative bearing reactions can indicate potential issues with the shaft alignment, such as:
1. Unloading: The bearing may become unloaded or partially unloaded due to misalignment or other factors.
2. Overload: The bearing may be subjected to excessive loads or stresses, which can lead to premature failure.
To ensure safe and reliable operation of a ship's propulsion system, it is essential to maintain positive bearing reactions during shaft alignment design and testing.
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[1g]: Q1 How do Thermal Deviations affect Shaft Alignment?
A: A great question!
What are Thermal Deviations?
In a vessel, thermal deviations refer to changes in temperature that can cause the engine structure and its components (like bearings) to expand or contract. This expansion/contraction affects the position of the bearings relative to each other.
How do Thermal Deviations Affect Shaft Alignment?
Thermal deviations can significantly impact shaft alignment by causing:
1. Bearing offset changes: As temperatures change, bearing positions shift due to thermal expansion and contraction. These changes affect the overall shaft alignment.
2. Shaft deflection changes: The changing temperature affects the stiffness of the engine structure, leading to altered shaft deflections (or movements).

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3. Coupling misalignment: Thermal deviations can cause coupling misalignments between adjacent bearings.
Consequences for Shaft Alignment
If thermal deviations are not considered in the design and calculation process:
1. Increased vibration levels: Misaligned shafts can lead to increased vibrations, which may affect overall vessel performance.
2. Reduced bearing life: Inadequate alignment can cause premature wear on bearings due to excessive stress and fatigue.
3. Inaccurate load sharing: Thermal deviations can result in uneven load distribution among the bearings, leading to reduced reliability.
Mitigating Measures
To account for thermal deviations:
1. Temperature monitoring: Install temperature sensors to monitor changes throughout the vessel.
2. Thermal expansion calculations: Use finite element analysis (FEA) or other methods to calculate thermal expansions and contractions of engine components.
3. Shaft alignment adjustments: Perform periodic shaft alignment checks and adjustments based on measured temperature variations.
By considering thermal deviations, you can ensure accurate shaft alignment, reduce vibrations, extend bearing life, and maintain optimal vessel performance!
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[1g]: Q2: How do Hull Deformations affect Shaft Alignment?
A: Hull deformations can significantly impact shaft alignment in several ways:
1. Changes in propeller pitch angle: Hull deformations can alter the propeller's pitch angle, which affects the mainshaft's position and potentially leads to misalignment.
2. Shaft bending or twisting: As the hull deforms, it can cause the shaft to bend or twist, leading to changes in its alignment with respect to the propulsion system components (e.g., bearings, couplings).
3. Bearing vertical offset changes: Hull deformations can alter the vertical offsets of the bearings, which affects the mainshaft's position and potentially leads to misalignment.
4. Coupling misalignment: Changes in hull shape or deflection can cause the coupling between the shaft and propeller to become misaligned, leading to vibration issues.
5. Shaft flexibility effects: Hull deformations can affect the shaft's natural frequency and damping characteristics, which can influence its behavior under various operating conditions.
These changes can have a cumulative effect on shaft alignment, making it more challenging to achieve accurate alignment during design or installation. In some cases, hull deformations may even lead to:
1. Increased vibration: Misalignment caused by hull deformations can result in increased vibration levels, which can affect the overall performance and reliability of the propulsion system.
2. Reduced efficiency: Hull deformations can reduce the efficiency of the propulsion system by altering the propeller's pitch angle or causing misalignment with other components.
To mitigate these effects, designers and builders must carefully consider hull deformations during the design process and ensure that:
1. Accurate calculations are performed: To predict the impact of hull deformations on shaft alignment.
2. Robust designs are implemented: To accommodate potential changes in hull shape or deflection.
3. Proper installation procedures are followed: To minimize misalignment caused by hull deformations during installation.
By understanding how hull deformations affect shaft alignment, designers and builders can develop more robust and efficient propulsion systems that operate effectively under various conditions.
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IV.II. Nous Hermes 2 Mistral DPO:

The overall performance of the model in handling questions of Comprehensive nature was commendable. More specifically, the responses were numerous times elaborated beyond the strict subject matter of the questions and the model enhanced the given responses with extra in-context information which exceeded the

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data provided by the Local Documents (e.g. In some cases the model's responses even provided valuable insights regarding health and safety risks of the ship's personnel and the environment) **[2a]**. However, in contrary to Llama 3, Nous Hermes 2 Mistral DPO frequently provided repetitive answers to similar consequent questions, which indicates a lack in terms of deep understanding of the question's scope **[2b]**. Furthermore, some specific pieces of text were generally often entangled in the model's responses in the exact same form, a behavior which was also encountered when experimenting with ChatGPT. In some other cases, while the model answered correctly regarding the specifics of the question with respect to definitions and terminology, the general "nature" of the phenomenon of interest was misinterpreted **[2c]**.

When answering more complex questions, the model's performance was more than adequate, especially compared to the responses Llama 3 provided which lacked significantly in terms of structure and quality **[2d]**. Conclusively, Nous Hermes 2 Mistral DPO performed more adequately in complex questions which required combined knowledge both from the Local Documents provided and from the pre-existing knowledge, demonstrating a deep understanding of the specifics of each question **[2e]**.

Additionally, a similarity is noticed between the answers provided by Nous Hermes 2 Mistral DPO and Llama 3. More specifically, both models provided accurate and in-context responses, even when the questions were not fully comprehended **[2f]**. On the other hand, a remarkable difference between the models' responses can be detected in the fact that Nous Hermes 2 Mistral DPO in some cases did not provide the Regulatory Framework of the specific subject of interest, even if it was entailed in the question which was addressed to it **[2g]**. Consequently, it is safe to say that Nous Hermes 2 Mistral DPO did not focus its responses on the Regulations and the limitations set by Classification Societies as much as Llama 3 did. More precisely, it should be highlighted, that when the questions addressed to the model incorporated differences between Classification Societies, the responses did not entail information regarding the specific Regulations in which the aforementioned differences can be detected. Furthermore, in comparison to Llama 3, Nous Hermes 2 Mistral DPO film some cases indicated slightly less adequate understanding of the information which was provided through the Local Documents **[2h]**. It was also established, that in questions of Comprehensive nature, Nous Hermes 2 Mistral DPO and Llama 3 incorporated the same key-aspects in their responses, with the only difference being that Nous Hermes 2 Mistral DPO explained, enhanced and elaborated better its answers **[2i]**.

Furthermore, Nous Hermes 2 Mistral DPO responses are evaluated as more than adequate in complex questions, as it provided well-structured and accurate responses. It should be highlighted, that the elaboration of the model in terms of user-friendly and human-like text generation as well as its performance regarding basic comprehension matters is slightly better that Llama 3. Of course, as expected, in some cases the responses given by Llama 3 on specific questions were accurate and overall correct, while the respective answers provided by Nous Hermes 2 Mistral DPO were not **[2j]**.

For clarification reasons, references regarding the conclusions extracted from the procedure of evaluating Nous Hermes 2 Mistral DPO are now given:

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[2a]: Q: Do you know anything about the consequences if hull deformations are not taken into consideration during the design stage of shaft alignment?
A: If hull deformations are not considered during the design stage of shaft alignment, several negative consequences may arise:

1. Inaccurate alignment calculations: Shaft alignment designs that do not account for hull deflections will likely result in misaligned components over time as the vessel experiences changes in loading conditions or other

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factors affecting its structure. This can lead to increased wear on machinery and potential failure of critical components, reducing overall system efficiency and increasing maintenance costs.
2. Reduced performance: Misalignment due to hull deformations can negatively impact a vessel's propulsion system performance by causing increased friction between shafting components or creating additional stresses within the system that reduce component life expectancy. This may result in decreased fuel efficiency, slower speeds, and reduced overall operational capabilities for the vessel.
3. Safety risks: Inaccurate shaft alignment can also pose safety risks to crew members and passengers aboard a vessel. Misaligned machinery components may cause vibrations or excessive noise levels that could lead to fatigue-related injuries among crew members over time. Additionally, improperly aligned systems might increase the likelihood of component failure during operation, potentially resulting in accidents or environmental hazards if oil or other pollutants are released into the surrounding waters.
4. Increased maintenance costs: As misalignment issues arise due to hull deformations, regular maintenance and repair efforts will become more complex and costly as technicians attempt to address these challenges without proper consideration of the vessel's unique structural characteristics. This may require additional equipment or specialized techniques that can further increase maintenance expenses over time.
5. Reduced service life: Misaligned shafting components are at an increased risk of premature failure due to excessive wear, stresses induced by hull deformations, and other factors related to improper alignment. This could lead to a reduced overall service life for the vessel's propulsion system, necessitating costly replacements or upgrades earlier than anticipated during the design phase.
6. Environmental impact: In addition to safety risks associated with misaligned shafting components, hull deformations that are not accounted for in the alignment design stage can also contribute to increased fuel consumption and emissions due to reduced system efficiency. This may result in higher greenhouse gas emissions or other pollutants being released into the environment over time.
7. Reputation damage: Vessels experiencing performance issues related to misaligned shafting components due to hull deformations can negatively impact a company's reputation within the industry, potentially leading to lost business opportunities and decreased market share as competitors offer more reliable, efficient vessels with properly aligned propulsion systems.
To minimize these negative consequences, it is crucial to consider potential hull deflection effects on shaft alignment during the design stage of a vessel and incorporate appropriate strategies for addressing them throughout its service life. This will help ensure optimal performance from the propulsion system while minimizing maintenance costs, safety risks, and environmental impacts associated with misaligned components over time.
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[2b]: Q1: What aspects of Shaft Alignment are correlated to the classification society the Shipowner chooses to comply with?
A: When selecting a classification society to follow for shaft alignment procedures, there may be differences in various aspects related to design, construction, and operation. These differences can impact how engineers approach shaft alignment during vessel construction and throughout its life cycle:

1. Standards and Guidelines: Each classification society has its own set of guidelines that govern shaft alignment procedures. The standards for tolerances, materials, methods, and equipment used in the process may differ between societies. Engineers must be familiar with their chosen society's specific requirements to ensure compliance.
2. Certification and Surveys: The process of obtaining certification from one classification society versus another might differ in terms of required surveys or inspections related to shaft alignment. Additionally, specific equipment or systems that can be certified by each society could also vary.
3. Acceptance Criteria: Different societies may have different acceptance criteria for shaft alignments during construction and throughout a vessel's life. These criteria might relate to factors such as allowable misalignment levels, acceptable rates of deflection over time, or the types of equipment that can be used in alignment processes.

4. Training and Qualifications: The training and qualification requirements for engineers performing shaft alignments may differ depending on the classification society they choose to follow. This could impact the availability of skilled personnel and the overall cost of compliance with specific society guidelines.
5. Technical Support: Each classification society provides varying levels of technical support and resources related to shaft alignment procedures. Some societies might offer more comprehensive guidance, training materials, or access to specialized tools and equipment than others.
6. Reputation and Marketability: Different classification societies have different reputations within the maritime industry, which can impact a vessel's marketability. Choosing a society with a strong reputation for safety and quality could potentially lead to higher resale values or more favorable insurance rates.
7. International Trade Compliance: Some countries may require vessels to be classified by specific societies in order to enter their ports or engage in trade activities. Depending on the vessel's intended trading routes, choosing a classification society that is widely recognized and accepted globally could be crucial for international trade compliance.
8. Continuous Improvement: The rate at which each classification society updates its guidelines and incorporates new technologies or best practices can vary. Choosing a society that actively invests in research and development may lead to more advanced shaft alignment procedures over time, potentially improving vessel safety and efficiency.
9. Cost Factors: Finally, the cost of compliance with different societies' shaft alignment requirements should be considered. This includes not only initial construction costs but also ongoing maintenance, inspection, and certification fees throughout a vessel's life cycle.

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[2b]: Q2: Is there any practical difference regarding the Shaft Alignment Process depending on the Classification Society of the vessel?
A: When selecting a classification society to follow for shaft alignment procedures, there may be differences in various aspects related to design, construction, and operation. These differences can impact how engineers approach shaft alignment during vessel construction and throughout its life cycle:

1. Standards and Guidelines: Each classification society has its own set of guidelines that govern shaft alignment procedures. The standards for tolerances, materials, methods, and equipment used in the process may differ between societies. Engineers must be familiar with their chosen society's specific requirements to ensure compliance.
2. Certification and Surveys: The process of obtaining certification from one classification society versus another might differ in terms of required surveys or inspections related to shaft alignment. Additionally, specific equipment or systems that can be certified by each society could also vary.
3. Acceptance Criteria: Different societies may have different acceptance criteria for shaft alignments during construction and throughout a vessel's life. These criteria might relate to factors such as allowable misalignment levels, acceptable rates of deflection over time, or the types of equipment that can be used in alignment processes.
4. Training and Qualifications: The training and qualification requirements for engineers performing shaft alignments may differ depending on the classification society they choose to follow. This could impact the availability of skilled personnel and the overall cost of compliance with specific society guidelines.
5. Technical Support: Each classification society provides varying levels of technical support and resources related to shaft alignment procedures. Some societies might offer more comprehensive guidance, training materials, or access to specialized tools and equipment than others.
6. Reputation and Marketability: Different classification societies have different reputations within the maritime industry, which can impact a vessel's marketability. Choosing a society with a strong reputation for safety and quality could potentially lead to higher resale values or more favorable insurance rates.
7. International Trade Compliance: Some countries may require vessels to be classified by specific societies in order to enter their ports or engage in trade activities. Depending on the vessel's intended trading routes, choosing a classification society that is widely recognized and accepted globally could be crucial for international trade compliance.

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8. Continuous Improvement: The rate at which each classification society updates its guidelines and incorporates new technologies or best practices can vary. Choosing a society that actively invests in research and development may lead to more advanced shaft alignment procedures over time, potentially improving vessel safety and efficiency.
9. Cost Factors: Finally, the cost of compliance with different societies' shaft alignment requirements should be considered. This includes not only initial construction costs but also ongoing maintenance, inspection, and certification fees throughout a vessel's life cycle.

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[2c]: Q1:Explain the term "misalignment angle" in the context of the Shaft Alignment process of a vessel.
A: In the context of a shaft alignment process of a vessel, the "misalignment angle" refers to the angular difference between two adjacent shafts or between a shaft and its bearing housing. This misalignment can occur due to various factors such as manufacturing tolerances, thermal expansion, settlement of foundation supports, or mechanical distortion over time.

The misalignment angle is typically expressed in degrees or millimeters per meter (mm/m) and represents the deviation from an ideal straight line or perfect alignment condition. Achieving proper shaft alignment is crucial for ensuring smooth operation, reducing wear on bearings and couplings, minimizing vibration levels, and maintaining overall structural integrity of the vessel's propulsion system.

During a shaft alignment procedure, technicians measure the actual misalignment angles between the shafts or between the shaft and its bearing housing using specialized tools such as laser aligners, optical devices, or electronic measuring systems. These measurements are then used to calculate and adjust the necessary corrections to achieve optimal alignment conditions according to the manufacturer's recommendations and industry best practices.

Author Note: The misrepresentation lies in the fact that the “misalignment angle” is explained as if it belongs to the general category of “undesirable phenomena” in Shaft Alignment, while it is generally used as a term to describe the misalignment between the aft stern tube bearing and the tail shaft.

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[2c]: Q2: Do you know anything about the influence coefficients in shaft alignment?
A: Influence coefficients are essential parameters in shaft alignment calculations for vessels. They help account for hull deflections and other factors that can affect the accuracy and performance of propulsion systems. The concept is based on understanding how different parts of a ship's structure interact with each other under various loading conditions, such as those caused by waves or cargo movement.

Influence coefficients are typically represented in an influence coefficient matrix (also known as "influence matrices" or "flexibility matrices"). This matrix contains numerical values that represent the degree to which one part of a ship's structure influences another part when subjected to external forces or loads. The higher the value, the greater the influence between two points on the hull.

These coefficients are used in calculations to determine the shaft alignment under different loading conditions and account for potential deformations in the hull due to factors such as wave action, cargo movement, or other operational stresses. By considering these influences, designers can develop more accurate and reliable propulsion systems that maintain proper alignment despite potential hull deflections over time.

The use of influence coefficients in shaft alignment calculations is a key aspect of ensuring the safety, efficiency, and overall performance of modern vessels. Classification Societies like ABS provide guidelines for

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incorporating these factors into design processes to minimize risks associated with hull deformations while maintaining compliance with regulatory requirements.

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[2c]: Q3:Calculate how many influence coefficients exist in a vessel with 10 bearings?
A: In a vessel with 10 bearings, there would be a total of 100 influence coefficients. This is because the number of influence coefficients equals the square of the number of bearings (N^2). In this case, N = 10, so N^2 = 10 * 10 = 100. Each bearing has an influence coefficient for every single bearing in the system, resulting in a symmetrical and square matrix with 100 coefficients.

Author Note: It can easily be noticed, that even if the general context of the response is on-point, the specifics are wrong. Moreover, when the model was asked to calculate the number of influence coefficients in a system with 10 bearings, the answer was correct.

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[2d]: Q1: Thermal deviations: how do they affect Shaft Alignment.
A: Thermal deviations can significantly impact shaft alignment in vessels due to the expansion and contraction of materials under varying temperature conditions. This is particularly relevant for components such as bedplates, which support the propulsion system's bearings and transmit forces between them. When temperatures change, these materials may expand or contract, leading to alterations in the geometry and dimensions of the supporting structures.

As a result, thermal deviations can cause changes in bearing offsets, affecting the overall alignment of the shafting system. This misalignment can lead to increased stresses on bearings, couplings, gears, and other components within the propulsion system, potentially causing premature wear or failure. Additionally, poor alignment due to thermal deviations may result in reduced efficiency, vibration issues, and even structural damage over time.

To account for these effects during shaft alignment calculations, it is essential to consider factors such as material properties (thermal expansion coefficients), temperature gradients within the vessel, and potential variations in bearing offsets due to thermal deviations while in service. This can be achieved through methods like finite element analysis (FEA) or by incorporating empirical data from similar vessels with known operating conditions. By taking these factors into account during shaft alignment planning, it is possible to minimize the impact of thermal deviations on the overall performance and reliability of the propulsion system.

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[2d] Q2: Elaborate on the following subject: Contact points in bearing evaluation procedure of Shaft Alignment
A: The bearing evaluation procedure within a shaft alignment design should consider contact points to ensure proper fit, function, and performance within the propulsion system. Contact points are crucial because they determine reaction intensity and misalignment slope between the shaft and the bearing. Here's how contact points relate to the bearing evaluation procedure in shaft alignment:

1. Shaft-to-bearing contact analysis: During the design stage, analyze the contact points between the shaft and bearings using specialized software or design tools. This step helps determine the optimal location for each contact point based on factors such as load distribution, shear stresses, bending moments, and other critical parameters that affect overall performance and component life.
2. Single-point vs. multi-point contact: In conventional shaft alignment analysis, a single point of contact is often used to represent the position of the bearing reaction. This approach can be satisfactory when the distance between the selected single point of contact (x) is equal or slightly smaller than the calculated distance (d). However, more complex scenarios may require considering multi-point contacts for greater accuracy in evaluating bearings and their reactions within the propulsion system.

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3. Misalignment slope determination: The location of the contact points directly influences the misalignment slope between the shaft and bearing. A proper evaluation of these contact points ensures that the misalignment angle remains minimal, reducing wear on bearings and couplings and maintaining structural integrity throughout the vessel's service life.
4. Installation considerations: When installing bearings on the shaft and within the coupling system, account for the location of the contact points to ensure proper alignment with respect to the shaft. This step minimizes misalignment angles, reduces wear, and maintains structural integrity throughout the vessel's service life.
5. Monitoring and maintenance: Regularly monitor changes in bearing offsets over time due to wear or other factors that may affect performance. Implement preventative maintenance procedures as needed to address any issues before they become significant problems. This step ensures ongoing optimization of the propulsion system's overall performance while minimizing wear on bearings and couplings, reducing vibration levels, and maintaining structural integrity throughout the vessel's service life.
6. Retrofitting and upgrades: If necessary, consider retrofitting or upgrading bearings during the vessel's service life based on changes in operating conditions, new technology advancements, or other factors that may impact performance. This step ensures ongoing optimization of the propulsion system's overall performance while minimizing wear on bearings and couplings, reducing vibration levels, and maintaining structural integrity throughout the vessel's service life.

By considering contact points in the bearing evaluation procedure of shaft alignment, you can ensure a robust and efficient propulsion system that meets classification society requirements while minimizing wear on bearings and couplings, reducing vibration levels, and maintaining structural integrity throughout the vessel's service life.

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[2d]: Q3: Elaborate on the following subject: The evaluation of a bearing as a procedure during the Shaft Alignment Design stage
A: Evaluating bearings is an essential part of the shaft alignment design process to ensure optimal performance and prevent potential issues with wear on bearings and couplings, vibration levels, and overall structural integrity. The following steps outline how to evaluate bearings as a procedure during the shaft alignment design stage:

1. Selection and specification: Choose appropriate bearing types based on factors such as operating conditions (e.g., speed, load, temperature), material compatibility with other components (shaft, seawater, lubricants), and manufacturer recommendations. Specify the required dimensions, tolerances, and surface finishes to ensure proper fit and function within the propulsion system.
2. Contact area determination: Utilize specialized software or design tools to determine the contact area between the bearing and shaft. This step is essential for calculating reactions, shear stresses, bending moments, and other critical parameters that affect overall performance and component life.
3. Simulation and analysis: Perform a simulation using appropriate software to evaluate the dynamic performance and behavior of the bearing under various operating conditions (e.g., speed, load, temperature). This step helps identify potential issues related to wear, vibration, or structural integrity before installation and allows for adjustments in design or material selection if necessary.
4. Installation: Follow manufacturer guidelines and industry best practices when installing bearings on the shaft and within the coupling system. Proper alignment of the bearing with respect to the shaft is crucial for minimizing misalignment angles, reducing wear, and maintaining structural integrity throughout the vessel's service life.
5. Monitoring and maintenance: Establish a regular monitoring program to track changes in bearing offsets over time due to wear or other factors that may affect performance. Implement preventative maintenance procedures as needed to address any issues before they become significant problems.
6. Retrofitting and upgrades: If necessary, consider retrofitting or upgrading bearings during the vessel's service life based on changes in operating conditions, new technology advancements, or other factors that may impact performance. This step ensures ongoing optimization of the propulsion system's overall

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performance while minimizing wear on bearings and couplings, reducing vibration levels, and maintaining structural integrity throughout the vessel's service life.

By following these guidelines for evaluating bearings as a procedure during the shaft alignment design stage, you can ensure a robust and efficient propulsion system that meets classification society requirements while minimizing wear on bearings and couplings, reducing vibration levels, and maintaining structural integrity throughout the vessel's service life.

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[2e]: Q: Can a vessel have only an aft stern tube bearing?
A: A vessel can indeed have only an aft stern tube bearing, but it is essential to carefully consider the design and its implications. A single-bearing system has been used in some modern ship designs, particularly for vessels with azimuth thrusters or controllable pitch propellers (CPP). However, this configuration requires additional attention to detail during the design phase to ensure proper functioning and longevity of the bearing.

When a vessel has only an aft stern tube bearing, it is crucial to implement specific requirements to compensate for the lack of forward support:

1. Double Slope Design: The stern tube bearing should be designed with a double slope, which helps distribute the load more evenly across the bearing and reduces stress concentrations at any single point. This design ensures that the bearing can handle the significant loads resulting from the weight of the propeller without excessive wear or damage.
2. Lubricant Inlet Location: The lubricant inlet for the stern tube should be located aft of the stern tube bearing to ensure an adequate supply of lubrication to the bearing and prevent overheating due to friction between the shaft and bearing. This location also helps maintain proper alignment with the shaft throughout the vessel's service life.
3. Intermediate Bearing Chocking: After installation, the intermediate bearing should be chocked or secured in place to minimize movement during operation. This step is particularly important for a single-bearing system as it ensures that the load distribution remains stable and even across the stern tube bearing throughout the vessel's service life.
4. Special Lubrication Systems: Depending on the specific design of the vessel, special lubrication systems may be required to ensure proper cooling and lubrication of the aft stern tube bearing. These systems can include water-lubricated bearings or other advanced lubrication technologies designed for high-performance applications.
5. Detailed Analysis: Before implementing such a design, it is crucial to conduct thorough analyses and simulations to ensure that the vessel's performance will not be compromised by having only an aft stern tube bearing. This analysis should consider factors like load distribution, vibration levels, and structural integrity throughout the service life of the vessel.

In summary, while it is possible for a vessel to have only an aft stern tube bearing, this design requires careful consideration and implementation of specific requirements to ensure proper functioning, longevity, and overall performance. These include double slope bearing designs, appropriate lubricant inlet locations, intermediate bearing chocking, special lubrication systems, and detailed analysis during the design phase.

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[2f]: Q: What parameters are defined during the Design Stage of Shaft Alignment according to ABS?
A: According to the American Bureau of Shipping (ABS) guidelines for shaft alignment procedures, several key decisions and considerations are made during the design stage. These design aspects aim to ensure safety, efficiency, and compliance with regulatory requirements:

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1.Equivalent model of propulsion system: Develop a simplified yet accurate representation of the propulsion system, including its main components (shafting, bearings, couplings, gears, etc.) and their interactions under various loading conditions. This model should consider factors such as material properties, geometry, and operational constraints.
2.Static and dynamic loads: Determine the expected static and dynamic loads acting on the shaft alignment system during normal operation and in extreme or abnormal situations (e.g., maneuvering, heavy weather). These load values will be used to design the structure, bearings, and other components for adequate strength and durability.
3.Slope boring design: Establish the slope of the bore in the machinery spaces that houses the shafting system. This ensures proper alignment during installation and helps maintain optimal performance throughout the equipment's service life. The slope should be designed to accommodate any expected deflections or deformations due to operational loads, thermal expansion, or other factors.
4.Consequence of misalignment: Assess the potential consequences of shaft misalignment on the overall propulsion system and its components (e.g., bearings, gears, seals). This information will help guide decisions related to tolerances, alignment procedures, monitoring systems, and maintenance strategies.
5.Alignment tolerances: Establish acceptable limits for shaft alignment during installation and throughout the equipment's service life. These tolerances should be based on factors such as component material properties, operational loads, and potential consequences of misalignment (as mentioned above).
6.Shaft alignment procedure: Develop a detailed plan for aligning the shafting in accordance with the designed parameters. This plan should include procedures for measuring and adjusting shaft positions, as well as any necessary monitoring or maintenance activities to ensure ongoing performance and safety.
7.Monitoring and maintenance strategies: Define methods for regularly checking and maintaining the alignment system throughout its service life. This may involve installing sensors or other measurement devices, establishing inspection schedules, and developing corrective actions in response to identified misalignment issues.
By addressing these design aspects during the shaft alignment procedure's development stage, you can ensure a safer, more efficient, and compliant propulsion system that meets regulatory requirements and minimizes potential risks associated with misalignment.
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[2g]: Q1: Can you list the requirements regarding the misalignment angle of the aft stern tube bearing?
A: It is crucial to follow general industry guidelines and best practices to ensure proper bearing installation and overall system performance. Here are some key considerations:

1. Load Capacity: Ensure that the chosen bearing can withstand the expected loads without exceeding its load capacity limits. This includes considering factors such as thrust forces, bending moments, and torque experienced by the stern tube during operation.
2. Material Selection: Choose materials for bearings and their supporting structures that are appropriate for the specific application and environmental conditions encountered during operation. Consider factors like temperature, humidity, corrosive environments, and potential exposure to contaminants like water or chemicals when selecting materials.
3. Safety Measures: Implement proper safety measures when adjusting bearing inclinations, including ensuring that all necessary tools and equipment are properly secured and that personnel involved in the process are adequately trained and protected from injury.
4. Documentation & Record Keeping: Maintain detailed documentation regarding all aspects of bearing installation, including dimensions, material specifications, load capacities, and safety measures taken during the operation. This information will be critical for future maintenance or repair work on the system in question.
5. Inspection & Testing: Verify that the completed installation meets industry standards and specifications before putting it into service. This may include verifying proper bearing alignment, checking for signs of excessive wear or damage, and assessing overall structural integrity throughout the system.

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6. Consult with OEMs: Collaborate with the original equipment manufacturer (OEM) of the stern tube system to discuss best practices, tolerances, and guidelines specific to their products. This collaboration will help ensure that you are following appropriate installation techniques for your particular application.
7. Monitor & Adjust as Needed: Continuously monitor the alignment of the stern tube bearings during installation or maintenance to ensure they remain within acceptable limits. If necessary, adjustments should be made in accordance with manufacturer guidelines and industry best practices.
8. Training & Education: Ensure that all personnel involved in the process are adequately trained on proper bearing installation techniques, safety measures, and alignment procedures. This will help minimize errors or misalignment issues during the project's execution.
9. Consider Hertzian Contact Theory Principles: Apply principles from Hertzian Contact Theory when designing and installing bearings to minimize stress concentrations, prevent unexpected failures, and maximize overall system performance throughout the vessel's operational life.
10. Alignment Procedures: Develop a detailed alignment procedure that addresses all aspects of shaft misalignment control during installation or maintenance processes. This should include steps for bore sighting, measuring clearances at the aft end of the tube, and verifying proper bearing inclinations throughout the system.

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[2g]: Q2: Can you list the requirements regarding oil film pressure in bearings?
A: The requirements for oil film pressure in bearings are essential for ensuring efficient operation, minimizing wear on components, and maintaining the overall performance of a vessel's propulsion system. The following factors should be considered when evaluating oil film pressure:

1. Adequate thickness: Maintaining an adequate oil film thickness is crucial to minimize contact between moving parts (shaft and bearing) and reduce friction-related wear. Generally, the recommended minimum oil film thickness ranges from 0.25 to 0.35 times the shaft diameter (D). This ensures that there is sufficient lubrication for proper operation while minimizing stress on components.
2. Uniform distribution: Oil should be evenly distributed across the bearing's contact area to maintain consistent oil film thickness and pressure throughout its length and width. Uneven distribution can lead to localized high-pressure zones, which may cause premature wear or failure of the bearing or other system components.
3. Optimal viscosity: Selecting the appropriate lubricant with the right viscosity is essential for maintaining an effective oil film pressure in bearings. The ideal viscosity will provide sufficient resistance to shear forces while allowing for efficient flow and distribution within the bearing's contact area. Consult your vessel's manufacturer or a lubrication specialist to determine the optimal viscosity grade based on operating conditions, temperature ranges, and other factors specific to your application.
4. Proper alignment: Ensuring proper shaft and housing alignment is critical for maintaining uniform oil film pressure across the bearing's contact area. Misalignment can lead to localized high-pressure zones, causing premature wear or failure of system components. Regularly inspect and adjust alignments as needed according to your vessel's maintenance schedule.
5. Sufficient flow rate: Ensuring an adequate supply of lubricant is available for continuous circulation through the bearing is essential for maintaining a consistent oil film pressure. This can be achieved by selecting the appropriate pumping system, filtering and cooling equipment, and monitoring fluid levels in reservoirs or sumps.
6. Cleanliness: Contaminants such as water, dirt, or foreign particles can compromise the effectiveness of an oil film pressure within bearings. Implement a regular maintenance program that includes cleaning and flushing systems to remove contaminants and maintain optimal performance.
7. Temperature management: High temperatures can cause thermal expansion of lubricating oils, leading to reduced viscosity and inadequate oil film pressures. Monitoring and controlling bearing temperatures through proper cooling systems or insulation measures is essential for maintaining consistent oil film pressure throughout the vessel's operational life.

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By meeting these requirements regarding oil film pressure in bearings, you can ensure efficient operation, minimize wear on components, and maintain overall performance of your vessel's propulsion system while minimizing risks associated with environmental contamination, personnel safety, and vessel seaworthiness.

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[2h]: Q1: Is the pre-sighting procedure optional when epoxy resin is applied?
A: When using an epoxy resin to create a watertight seal between the stern tube casting and the main engine, it's essential to ensure proper alignment before applying the resin. Pre-sighting is not optional; rather, it is a crucial step in the process to achieve accurate alignment and maintain water tightness.

During pre-sighting, you will use one of the methods mentioned earlier (piano wire, optical telescope, or laser instrument) to establish the reference centerline through the stern tube casting. This ensures that when the epoxy resin is applied, it forms a solid and accurate connection between the main engine and the stern tube casting, minimizing potential issues related to bearing wear, component failure, or environmental contamination during vessel operation.

In summary, pre-sighting is not optional but an essential step in ensuring proper alignment and water tightness when using epoxy resin for connecting the main engine with the stern tube casting.

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[2h]: Q2: How can the clearances of the aft stern tube be calculated?
A: Calculating aft stern tube clearances involves determining the distance between the shaft and the inner surface of the stern tube bearing. This measurement is crucial for ensuring proper system performance, minimizing stress on main engine components, and maintaining vessel seaworthiness throughout its operational life. To calculate aft stern tube clearances, follow these steps:

1. Determine design specifications: Consult your vessel's original blueprints or consult with the manufacturer to obtain the specified shaft diameter (D) and bearing inner diameter (B). This information will be necessary for calculating the clearance between the two components.
2. Measure actual dimensions: Using a caliper, micrometer, or other measuring tool, measure the distance between the centerline of the stern tube shaft and the inner surface of the stern tube bearing at multiple points along its length and width. Be sure to note down these values for future reference.
3. Calculate clearance: Subtract the shaft diameter (D) from the bearing inner diameter (B). This difference represents the designed clearance between the two components. To calculate actual clearances, subtract the measured distance between the centerline of the stern tube shaft and the inner surface of the stern tube bearing at each point along its length and width from the corresponding designed dimensions.
4. Compare clearances: Compare the calculated clearances with the design specifications to determine if there are any significant deviations that could impact system performance or create potential issues down the line. If clearance values differ significantly, further investigation may be required to identify root causes and implement corrective actions as needed.
5. Monitor clearances over time: Regularly measure and compare aft stern tube clearances with baseline data established in step 3 to track changes in dimensions over time. This information can help determine if the bearing is wearing at a normal or abnormal rate, which may indicate potential issues with the system that require further investigation and corrective action.

By following this common industry method for measuring clearances on an aft stern tube bearing, you can effectively monitor its health and take appropriate action to minimize risks associated with environmental contamination, personnel safety, and vessel seaworthiness.

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[2i]: Q: List some crucial parameters which significantly affect Shaft Alignment

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A: Several crucial parameters significantly impact shaft alignment. Proper consideration and management of these factors are essential in achieving accurate and efficient shaft alignment for optimal vessel performance:

1. Hull deflection: The flexibility of the hull can cause misalignment as it may not remain perfectly straight or symmetrical during operation. This deflection can lead to changes in the loads experienced by the propulsion system, affecting its alignment over time.
2. Load distribution: Uneven load distribution on the shafting system due to factors such as unequal thrust from azimuth thrusters or controllable pitch propellers (CPP) can cause misalignment and affect overall performance.
3. Shaft deflection at PBC or thrust block location: The deflection of the shaft at the propeller boss cap (PBC) or thrust block location can impact alignment, leading to potential issues with the propulsion system's efficiency and structural integrity.
4. Thrust block positioning: Proper placement and adjustment of thrust blocks are crucial for maintaining accurate shaft alignment according to vessel design specifications. Misaligned thrust blocks can lead to misalignment in the propulsion system and affect overall performance.
5. Shimming requirements: Vertical and horizontal shimming adjustments play a significant role in achieving proper shaft alignment. Incorrect or insufficient shimming can result in misalignment, affecting vessel performance and potentially causing damage to the propulsion system or structural integrity.
6. Propeller orientation: Ensuring that the propellers are within specified limits is essential for maintaining accurate shaft alignment and preventing potential damage to the propulsion system, structural integrity, or overall vessel performance.
7. Changes in operating conditions: Variations in loading conditions, such as changes in speed, direction, or cargo load, can impact shaft alignment over time. Regular monitoring and adjustments are necessary to maintain proper alignment under varying operational circumstances.
8. Maintenance procedures: Establishing ongoing maintenance procedures for the shafting system is crucial for maintaining accurate shaft alignment throughout a vessel's operational life. This includes regular inspections and prompt addressing of any misalignment issues that may arise during vessel operations.

By carefully managing these parameters, you can ensure proper shaft alignment in accordance with classification society requirements while considering your vessel design and other relevant factors.

Contrary to the response provided by Llama 3, for the exact same Question:

A: Based on various classification society guidelines and industry experts' opinions, here are some crucial parameters that can significantly impact shaft alignment:

1. Hull Deflection: Changes in hull shape or deflections due to loading conditions (e.g., cargo, ballast water) can alter the propeller shaft's position.
2. Shaft Length and Flexibility: Longer shafts with higher flexibility are more susceptible to misalignment caused by changes in vessel dynamics.
3. Propeller Pitch and Angle: Changes in propeller pitch or angle can affect the alignment of the mainshaft, leading to vibration issues.
4. Bearing Design and Condition: Worn-out or improperly installed bearings can lead to shaft misalignment due to uneven load distribution.
5. Shaft Material Properties: The material properties (e.g., strength, stiffness) of the shaft itself can affect its ability to withstand changes in loading conditions.
6. Vessel Speed and Load Conditions: Changes in vessel speed or operating loads can alter the propeller's position relative to the mainshaft.
7. Propulsion System Design: Inadequate design or installation of propulsion components (e.g., couplings, bearings) can lead to shaft misalignment issues.
These parameters are interconnected and can have a cumulative effect on shaft alignment. For example:

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* Hull deflection caused by loading conditions can affect the propeller pitch angle, which in turn affects mainshaft alignment.
* Worn-out or improperly installed bearings can exacerbate hull deflections and shaft misalignment.
Proper consideration of these factors is crucial for ensuring accurate shaft alignment during design, construction, and operation.

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[2j]: Q1: Is the pre-sighting procedure optional when epoxy resin is applied?
A: When using an epoxy resin to create a watertight seal between the stern tube casting and the main engine, it's essential to ensure proper alignment before applying the resin. Pre-sighting is not optional; rather, it is a crucial step in the process to achieve accurate alignment and maintain water tightness.

During pre-sighting, you will use one of the methods mentioned earlier (piano wire, optical telescope, or laser instrument) to establish the reference centerline through the stern tube casting. This ensures that when the epoxy resin is applied, it forms a solid and accurate connection between the main engine and the stern tube casting, minimizing potential issues related to bearing wear, component failure, or environmental contamination during vessel operation.

In summary, pre-sighting is not optional but an essential step in ensuring proper alignment and water tightness when using epoxy resin for connecting the main engine with the stern tube casting.

Contrary to the response provided by Llama 3:

A: When applying epoxy resin to a surface, presighting (also known as sighting) can be considered an optional step, but it's highly recommended for achieving optimal results.
Why Presight Before Applying Epoxy Resin:
1. Accurate Application: Sighting ensures the epoxy resin is applied precisely along the desired path or pattern, which is crucial when working with complex shapes or tight spaces.
2. Reduced Waste: By accurately sighting before applying the epoxy, you can minimize waste and avoid unnecessary excess material.
3. Improved Adhesion: A well-sighted application helps ensure a strong bond between the resin and the surface being coated.
When Presighting Might Not Be Necessary:
1. Simple Coatings: For straightforward applications where the coating is not critical to the overall performance or appearance of the part, presight might be unnecessary.
2. Small Areas: When working with small areas or simple shapes, the risk of error is lower, and presight may not add significant value.
In Summary:
While presighting before applying epoxy resin is an optional step, it's highly recommended for achieving optimal results in most cases. The benefits of accurate application, reduced waste, and improved adhesion make the extra effort worthwhile. However, if you're working with simple coatings or small areas, you might be able to get away without presight.
Do you have any specific epoxy resin applications or concerns? I'm here to help!

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[2j]: Q2: How can the clearances of the aft stern tube be calculated?
A: Calculating aft stern tube clearances involves determining the distance between the shaft and the inner surface of the stern tube bearing. This measurement is crucial for ensuring proper system performance, minimizing stress on main engine components, and maintaining vessel seaworthiness throughout its operational life. To calculate aft stern tube clearances, follow these steps:

1. Determine design specifications: Consult your vessel's original blueprints or consult with the manufacturer to obtain the specified shaft diameter (D) and bearing inner diameter (B). This information will be necessary for calculating the clearance between the two components.
2. Measure actual dimensions: Using a caliper, micrometer, or other measuring tool, measure the distance between the centerline of the stern tube shaft and the inner surface of the stern tube bearing at multiple points along its length and width. Be sure to note down these values for future reference.
3. Calculate clearance: Subtract the shaft diameter (D) from the bearing inner diameter (B). This difference represents the designed clearance between the two components. To calculate actual clearances, subtract the measured distance between the centerline of the stern tube shaft and the inner surface of the stern tube bearing at each point along its length and width from the corresponding designed dimensions.
4. Compare clearances: Compare the calculated clearances with the design specifications to determine if there are any significant deviations that could impact system performance or create potential issues down the line. If clearance values differ significantly, further investigation may be required to identify root causes and implement corrective actions as needed.
5. Monitor clearances over time: Regularly measure and compare aft stern tube clearances with baseline data established in step 3 to track changes in dimensions over time. This information can help determine if the bearing is wearing at a normal or abnormal rate, which may indicate potential issues with the system that require further investigation and corrective action.

By following this common industry method for measuring clearances on an aft stern tube bearing, you can effectively monitor its health and take appropriate action to minimize risks associated with environmental contamination, personnel safety, and vessel seaworthiness.

Contrary to the response provided by Llama 3:

A: Calculating aft stern tube clearances involves determining the distance between the shaft and the inner surface of the stern tube bearing. This measurement is crucial for ensuring proper system performance, minimizing stress on main engine components, and maintaining vessel seaworthiness throughout its operational life. To calculate aft stern tube clearances, follow these steps:

1. Determine design specifications: Consult your vessel's original blueprints or consult with the manufacturer to obtain the specified shaft diameter (D) and bearing inner diameter (B). This information will be necessary for calculating the clearance between the two components.
2. Measure actual dimensions: Using a caliper, micrometer, or other measuring tool, measure the distance between the centerline of the stern tube shaft and the inner surface of the stern tube bearing at multiple points along its length and width. Be sure to note down these values for future reference.
3. Calculate clearance: Subtract the shaft diameter (D) from the bearing inner diameter (B). This difference represents the designed clearance between the two components. To calculate actual clearances, subtract the measured distance between the centerline of the stern tube shaft and the inner surface of the stern tube bearing at each point along its length and width from the corresponding designed dimensions.
4. Compare clearances: Compare the calculated clearances with the design specifications to determine if there are any significant deviations that could impact system performance or create potential issues down the line. If clearance values differ significantly, further investigation may be required to identify root causes and implement corrective actions as needed.
5. Monitor clearances over time: Regularly measure and compare aft stern tube clearances with baseline data established in step 3 to track changes in dimensions over time. This information can help determine if the bearing is wearing at a normal or abnormal rate, which may indicate potential issues with the system that require further investigation and corrective action.

By following this common industry method for measuring clearances on an aft stern tube bearing, you can effectively monitor its health and take appropriate action to minimize risks associated with environmental contamination, personnel safety, and vessel seaworthiness.

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IV.III. GPT4ALL Falcon:

GPT4ALL Falcon tended to focus its responses on more general information and did not elaborate as much as the rest of the models on the specifics or the Classification Societies' Requirements regarding the subject of interest. However, the model in many cases demonstrated good performance in understanding the substance of the question **[3a]**. Furthermore, the responses CPT4ALL Falcon provided were usually in context and accurate but focused on the outlines of the subjects and not further elaborated **[3b]**. Moreover, the model's responses frequently were repetitive **[3c]** and short. Additionally, contradictory behavior was detected regarding the model's responses, as in some cases the answers were different while the subject of interest was the same **[3d]**.

GPT4ALL Falcon performed adequately when answering to more Complex questions, as its responses were generally accurate, but short **[3e]**. It is safe to say that the model provided significantly shorter answers to all the questions addressed, however they were mainly in the context and error-free **[3f]**. Of course, on many occasions the responses were deemed as inadequate due to the lack of information and elaboration **[3g]**.

On the other hand, in several cases GPT4ALL Falcon provided different points of view than the rest of the models through its responses **[3h]**. Nevertheless, the answers which the model gave completely lacked references to Regulations **[3i]** and in many cases the model showed signs of incompetency regarding Fact Retrieval and Comprehension capabilities **[3j]**. Some of the responses provided by the model were also wrong **[3k]** or indicated confusion regarding the subject of interest **[3l].**

Conclusively, GPT4ALL Falcon's performance in handling Comprehensive questions is evaluated as adequate, though the responses provided tend to be brief and lack detailed information and data (such as specific Requirements). In many cases, the model manages to present the core aspects of the topics effectively and accurately. Additionally, the performance of GPT4ALL Falcon in Science-related questions was also satisfactory, as it provided relevant and error-free responses. Overall, the model surely demonstrates potential as a useful tool in the hands of a student and/or professional aiming to handle and gain insights into large amounts of documents related to a specific technical subject like Shaft Alignment, but it would significantly benefit from further development via Fine-Tuning in order for its comprehensiveness and responses to be improved.

For clarification reasons, references regarding the conclusions extracted from the procedure of evaluating GPT4ALL Falcon are now given:

[3a]: Q: List some crucial parameters which significantly affect Shaft Alignment
A: The critical parameters that affect severely shaft alignment include:

1. Loads: The loads applied to the propulsion system can have a significant impact on the alignment of the shafts. Overloading of bearings, for example, can lead to fatigue and wear, which in turn can cause misalignment of the shafts.
2. Hull deflections: The hull deflections can also affect the alignment of the shafts. As the hull deflects, it places additional loads on the propulsion system, including bending moments on the shaft.
3. Overloading of bearings: Overloading of bearings can lead to severe wear and damages both for the bearings and the shaft, which shortens the system's life.
4. Misalignment of the shafts: Misalignment of the shafts can occur due to various reasons such as improper installation, misalignment during maintenance, or even due to changes in the vessel's operating conditions.
5. Shaft deflection: Shaft deflection can also affect the alignment of the shafts. As the shaft deflects, it places additional loads on the propulsion system, including bending moments on the shaft.
6. Misalignment of the propeller: Misalignment of the propeller can occur due to various reasons such as improper installation, misalignment during maintenance, or even due to changes in the vessel's operating conditions.

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7. Shaft misalignment: Shaft misalignment can occur due to various reasons such as improper installation, misalignment during maintenance, or even due to changes in the vessel's operating conditions.
8. Misalignment of the bearings: Misalignment of the bearings can occur due to various reasons such as improper installation, misalignment during maintenance, or even due to changes in the vessel's operating conditions.

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[3b]: Q: Can you elaborate on the requirements of the regulations regarding vessels with only one stern tube bearing? Give extra information regarding the following Classification Societies: ABS, DNV, LR, BV, CLASSNK
A: The American Bureau of Shipping (ABS) has specific requirements for vessels with only one Stern Tube Bearing in their publication "Guidelines for Shaft Alignment" (2019). The vessel's design and propulsion system must be analyzed to ensure that the vessel can safely operate with only one Stern Tube Bearing. The specific regulations on shaft alignment for vessels with only one Stern Tube Bearing may vary depending on the classification society and the specific requirements of the vessel's design and propulsion system.

The DNV also has specific requirements for vessels with only one Stern Tube Bearing in their standard "DNVGL-ST-A203" (2017). The vessel's design and propulsion system must be analyzed to ensure that the vessel can safely operate with only one Stern Tube Bearing. The specific regulations on shaft alignment for vessels with only one Stern Tube Bearing may vary depending on the classification society and the specific requirements of the vessel's design and propulsion system.

The LR publication "Guidance Notes for Shaft Alignment" (2018) provides guidance on the design stage of shaft alignment, including the selection of appropriate materials and manufacturing methods to ensure optimal performance. The specific regulations on shaft alignment for vessels with only one Stern Tube Bearing may vary depending on the classification society and the specific requirements of the vessel's design and propulsion system.

The BV standard "BV-GL-ST-A203" (2019) also includes requirements for vessels with only one Stern Tube Bearing in their standard. The specific regulations on shaft alignment for vessels with only one Stern Tube Bearing may vary depending on the classification society and the specific requirements of the vessel's design and propulsion system.

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[3c]: 1st Q: How do hull deformations affect Shaft Alignment?
A: Hull deformations can have a significant impact on the alignment of the shafts in a ship's propulsion system. The deformations cause changes in the vertical offsets of the bearings, which can lead to misalignment between the shafts and the propeller. This misalignment can result in increased shaft vibrations, reduced efficiency, and even damage to the bearings and other components of the propulsion system. In addition, the deformations can also affect the alignment of the shafts with respect to the hull, which can further increase the risk of misalignment and damage.

According to D. Sverko (2003), the coefficients, and as a result the shaft becomes really vulnerable to changes in the vertical offsets of the bearings. On the other hand, the flexible construction of the hull equals to increased deformations of the hull. As the hull deflections take place, the alignment of the shafts is affected, which can lead to misalignment between the shafts and the propeller.

In summary, the impact of hull deformations on shaft alignment can be significant, as it can cause changes in the vertical offsets of the bearings, resulting in misalignment between the shafts and the propeller. The deformations can also affect the alignment of the shafts with respect to the hull, which can further increase the risk of misalignment and damage.

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[3c]: 2nd Q: Do you know anything about how hull deformations specifically affect the Design Stage of shaft alignment?
A: Yes, hull deformations can have a significant impact on the design stage of shaft alignment. The deformations cause changes in the vertical offsets of the bearings, which can lead to misalignment between the shafts and the propeller. This misalignment can result in increased shaft vibrations, reduced efficiency, and even damage to the bearings and other components of the propulsion system. In addition, the deformations can also affect the alignment of the shafts with respect to the hull, which can further increase the risk of misalignment and damage.

According to D. Sverko (2003), the coefficients, and as a result the shaft becomes really vulnerable to changes in the vertical offsets of the bearings. On the other hand, the flexible construction of the hull equals to increased deformations of the hull. As the hull deflections take place, the alignment of the shafts is affected, which can lead to misalignment between the shafts and the propeller.

In summary, the impact of hull deformations on shaft alignment can be significant, as it can cause changes in the vertical offsets of the bearings, resulting in misalignment between the shafts and the propeller. The deformations can also affect the alignment of the shafts with respect to the hull, which can further increase the risk of misalignment and damage.

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[3c]: 3rd Q: Do you know anything about the consequences if hull deformations are not taken into consideration during the Design Stage of Shaft Alignment?
A: Yes, the consequences of not taking hull deformations into consideration during the design stage of shaft alignment can be significant. The misalignment between the shafts and the propeller caused by the deformations can lead to reduced efficiency, increased shaft vibrations, and even damage to the bearings and other components of the propulsion system. In addition, the misalignment can also affect the alignment of the shafts with respect to the hull, which can further increase the risk of misalignment and damage.

According to D. Sverko (2003), the coefficients, and as a result the shaft becomes really vulnerable to changes in the vertical offsets of the bearings. On the other hand, the flexible construction of the hull equals to increased deformations of the hull. As the hull deflections take place, the alignment of the shafts is affected, which can lead to misalignment between the shafts and the propeller.

In summary, the consequences of not taking hull deformations into consideration during the design stage of shaft alignment can be significant, as it can lead to reduced efficiency, increased shaft vibrations, and even damage to the bearings and other components of the propulsion system. The misalignment can also affect the alignment of the shafts with respect to the hull, which can further increase the risk of misalignment and damage.

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[3d]: Q1: Can a vessel have only aft stern tube bearing?
A: No, a vessel cannot have only an aft stern tube bearing. The forward stern tube bearing is necessary for propulsion shafting interaction with the surrounding systems and machinery. Without the forward stern tube bearing, the vessel's propulsion system would not be able to function properly. It is important to ensure that the vessel's design and propulsion system can safely operate with both stern tube bearings in order to avoid potential shaft alignment issues.

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[3d]: Q2: Is a vessel allowed, according to Classification Societies, and regarding Shaft Alignment, to have only one stern tube bearing?
A: Yes, a vessel is allowed by classification societies to have just one stern tube bearing for shaft alignment purposes. However, it is important to ensure that the vessel's design and propulsion system can safely operate

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with only one stern tube bearing. The specific regulations on shaft alignment for vessels with only one stern tube bearing may vary depending on the classification society and the specific requirements of the vessel's design and propulsion system.

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[3e]: Q1: Elaborate on the following subject: Contact Points in the procedure of bearing evaluation in Shaft Alignment

A: The single point contact model used in conventional shaft alignment analysis is a convenient way to represent the contact between the shaft and the bearing. However, this approach may not be sufficient for ships with complex propulsion systems, such as those with multiple bearings or intermediate shafts. In these cases, it may be necessary to use a more accurate method of determining the misalignment slope between the shaft and the bearing, such as using a multi-point contact model.

The multi-point contact model involves calculating the misalignment slope between the shaft and each individual bearing in the propulsion system. This approach is more accurate than the single point contact model because it takes into account the specific alignment requirements of each bearing in the system. However, this approach can be time-consuming and may require a significant amount of data collection and analysis.

In summary, while the single point contact model used in conventional shaft alignment analysis is a convenient way to represent the contact between the shaft and the bearing, it may not be sufficient for ships with complex propulsion systems. The multi-point contact model can provide more accurate results, but requires additional data collection and analysis.

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[3e]: Q2: Explain the importance of the intermediate bearing in vessels with no forward stern tube bearing

A: The importance of intermediate bearings in systems with no forward stern tube bearing cannot be overstated. These bearings provide an additional layer of support and stability for the main engine bearings, which can help to reduce the risk of damage or wear on these components due to hull deflections.

It is important for engineers to carefully consider the specific requirements and limitations of each design option when selecting the most appropriate method for installing a stern tube bearing on a vessel.

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[3e]: Q3: What does ABS suggest regarding single or multi point contact analysis?

A: ABS suggests that either the combined approach or the single point contact only should be used in shaft alignment analysis. They also suggest that point of contact analysis is a good practice for initial design, while double-slope design of aft stern tube bearing calculations are preferred.

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[3f]: Q1: What do you know about the parallel rise of bearing offsets due to thermal deviations in Shaft Alignment?

A: The "parallel rise of bearing offsets due to thermal deviations" is a common issue that can occur during the Shaft Alignment process. This phenomenon occurs when the bearings' vertical offsets increase due to thermal deviations, causing the bearings to become unloaded and potentially leading to damage or failure of the shafting system. To mitigate this issue, it is important to incorporate flexible bearing designs into the shafting system and use real-time monitoring and adjustment capabilities during the Shaft Alignment process.

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[3f]: Q2: Do you know anything about the influence coefficients in shaft alignment?

A: Yes, I am familiar with the concept of influence coefficients in shaft alignment. In general, these coefficients are used to represent the flexibility or stiffness of a system, such as a shaft alignment, and can be calculated based on various factors, including the geometry and material properties of the components involved.

In the context of shaft alignment calculations, the influence coefficient matrix is often used to account for the effects of hull deformations on the alignment process. By incorporating these coefficients into the calculation,

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it may be possible to more accurately predict the alignment behavior of a vessel under different conditions, including those associated with hull deformations.

Overall, the use of influence coefficients in shaft alignment calculations can help to improve the accuracy and reliability of these calculations, particularly when dealing with complex systems such as vessels with hull deformations.

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[3f]: Q3: Can you elaborate on double slope designs in Shaft Alignment?
A: Yes, I am familiar with double slope designs in shaft alignment. A double slope design involves two slopes on the shaft alignment, one at the stern and one at the bow. This design is preferred over a single slope design because it provides a more accurate measurement of the shaft angle, which can improve alignment accuracy and prevent alignment problems. The use of double slope designs is also recommended by some manufacturers for their propulsion systems.

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[3f]: Q4: Explain the term: aft stern tube misalignment angle
A: The aft stern tube misalignment angle is a measure of how much the stern tube bearing is misaligned from its intended position. This measurement can be used to determine whether the alignment is within acceptable tolerances or if corrective action is needed.

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[3g]: Q: Can you elaborate on the basic guidelines regarding Shaft Alignment from DNV?
A: DNV provides basic guidelines for shaft alignment in their publication DNV-CG-0283, which includes information on calculation methods and best practices for aligning shafts in propulsion systems.

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[3h]: Q1: What changes depending on the Classification Society of a ship regarding Shaft Alignment?
A: According to the most current published regulations of several Classification Societies, it is now recommended to perform shaft alignment analysis for maximum allowable alignment tolerances, resulting in a more unified approach on shaft alignment procedures. This includes the use of double slope designs and the calculation of misalignment angles to improve alignment accuracy and prevent alignment problems.

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[3h]: Q2: What are the proposed hull deformation measurement methods according to ABS?
A: ABS has developed a software that utilizes data collected from vessels of different types and sizes to estimate hull deformations. The software is based on a number of measurements taken from the vessel, including measurements of the hull girder deflection, as well as other relevant parameters such as the ship's roll angle and the vessel's speed. These measurements are then used to calculate the estimated hull deformation values, which can be used in the shaft alignment design process.

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[3i]: Q1: What changes depending on the Classification Society of a ship regarding Shaft Alignment?
A: According to the most current published regulations of several Classification Societies, it is now recommended to perform shaft alignment analysis for maximum allowable alignment tolerances, resulting in a more unified approach on shaft alignment procedures. This includes the use of double slope designs and the calculation of misalignment angles to improve alignment accuracy and prevent alignment problems.

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[3i]: Q2: Do you know anything about the specific procedures each Classification Society recommends regarding Shaft Alignment?
A: Yes, each classification society has its own set of guidelines for conducting shaft alignment on a vessel. These guidelines may include specific steps or procedures that must be followed to ensure accurate and

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reliable measurements. It is important to follow these guidelines to ensure compliance with the relevant regulations and standards.

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[3i]: Q3: Elaborate on the criteria in order for the shaft alignment calculations to be acceptable according to ABS

A: Acceptability criteria regarding shaft alignment calculations according to ABS should include a detailed description of the methodology used for the calculation, including any assumptions or simplifications made. The results should also include any relevant data or measurements that were used in the calculation, as well as any issues or challenges encountered during the process. Additionally, the results should be compared to the basic alignment requirements and any deviations from these requirements should be explained.

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[3j]: Q: Are there any rules regarding the length of a bearing in Shaft Alignment?

A: There are no specific rules regarding the length of a bearing in shaft alignment. However, it is generally recommended to use bearings with a sufficient length to accommodate the misalignment angle and provide adequate load distribution over the bearing length. The length of the bearing should also be considered when designing the shaft system to ensure proper alignment and prevent excessive edge loading.

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[3k]: Q: is the reaction force of a bearing allowed by Classification Societies to be negative in Shaft Alignment?

A: Yes, the reaction force of a bearing in shaft alignment is allowed by classification societies to be negative. In fact, the shaft alignment design procedure takes into account the expected deflection of the hull due to the ship's speed and heading, as well as any other factors that may affect the alignment. The shaft alignment conditions are re-examined if the measured value of the Bearing Reaction deviates from the calculated value more than 20%, indicating that the shaft alignment conditions may not be acceptable.

However, it is important to note that the negative reaction force of a bearing in shaft alignment does not necessarily indicate an unacceptable condition. The classification society's requirements for shaft alignment may vary depending on the specific vessel and its intended use. It is recommended to consult with the classification society or a qualified marine engineer for guidance on acceptable shaft alignment conditions.

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[3l]: Q: What are the limits set by ABS regarding bearing reactions?

A: According to ABS Rules, the maximum allowable bearing reaction for metallic bearings is set at 0.8 [N/mm2], while for oil-lubricated synthetic intermediate bearings, the offset is set at 0.8 [N/mm2]. These limits are intended to ensure that the bearing reactions do not exceed the calculated values and do not cause excessive stress on the shaft or other components of the system.

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V. Evaluation of Performance Process

In order for the evaluation of the performance of each LLM to take place in a quantified way, some Quality Indicators were established. More specifically, the Quality Indicators have as follows:

- **Accuracy:** Indicates the correctness of facts and terminology usage.
- **Relevance:** Indicates whether the responses were in-context.
- **Comprehensiveness:** Indicates the depth of understanding (regarding the question which was addressed to the model as well as the subject of interest in general) and the details provided in each response in terms of elaboration of the response.
- **Specificity:** Indicates the utilization of specific examples, details (e.g. proposed solutions and mitigating measures, proposed measurement tools, safety and environmental risks etc.) regarding the subject of interest and Regulatory Citations.

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- **Consistency**: Indicates internal consistency (e.g. the model provides contradictory information regarding the same subject) and Consistency with Source Documents (assessment of whether the information provided by the model are compatible with the Local Documents provided and do not deviate from them)

Each one of the responses the models provided was separately assessed based on a Scoring System. More specifically, every answer was rated on a scale from 1 (one) to 5 (five) regarding each Quality Indicator, with 1 (one) representing the lowest performance and 5 (five) representing the highest performance. Afterwards, in order for the overall performance of the Models-Llama 3, Nous Hermes 2 Mistral DPO, GPT4ALL Falcon- to be quantified and evaluated, the scores of each model's questions were aggregated. The Mean Values of the performance of each model regarding each one of the Quality Indicators was calculated, as well as the Standard Deviation of each model with respect to each Quality Indicator. The results occurred by a manual score marking conducted by the Author and they are presented below, in the form of Diagrams as well as Tables.

Table 9: Llama 3; Score

LLAMA 3	Mean Score	Standard Deviation	OVERALL MEAN PERFORMANCE SCORE
Accuracy	4.080	0.8739	**3.991**
Relevance	4.437	0.9553	
Comprehensiveness	4.368	0.9840	
Specificity	3.529	0.6581	
Consistency	3.540	0.8815	

Table 10: Nous Hermes 2; Score

NOUS HERMES 2 MISTRAL DPO	Mean Score	Standard Deviation	OVERALL MEAN PERFORMANCE SCORE
Accuracy	3.874	0.9322	**3.798**
Relevance	4.414	0.9414	
Comprehensiveness	4.345	1.0038	
Specificity	3.218	0.7018	
Consistency	3.138	0.7904	

Table 11: GPT4ALL Falcon; Score

GPT4ALL FALCON	Mean Score	Standard Deviation	OVERALL MEAN PERFORMANCE SCORE
Accuracy	3.566	1.0998	**3.465**
Relevance	4.614	0.7417	
Comprehensiveness	4.554	0.8396	
Specificity	2.024	0.6938	
Consistency	2.566	0.9841	

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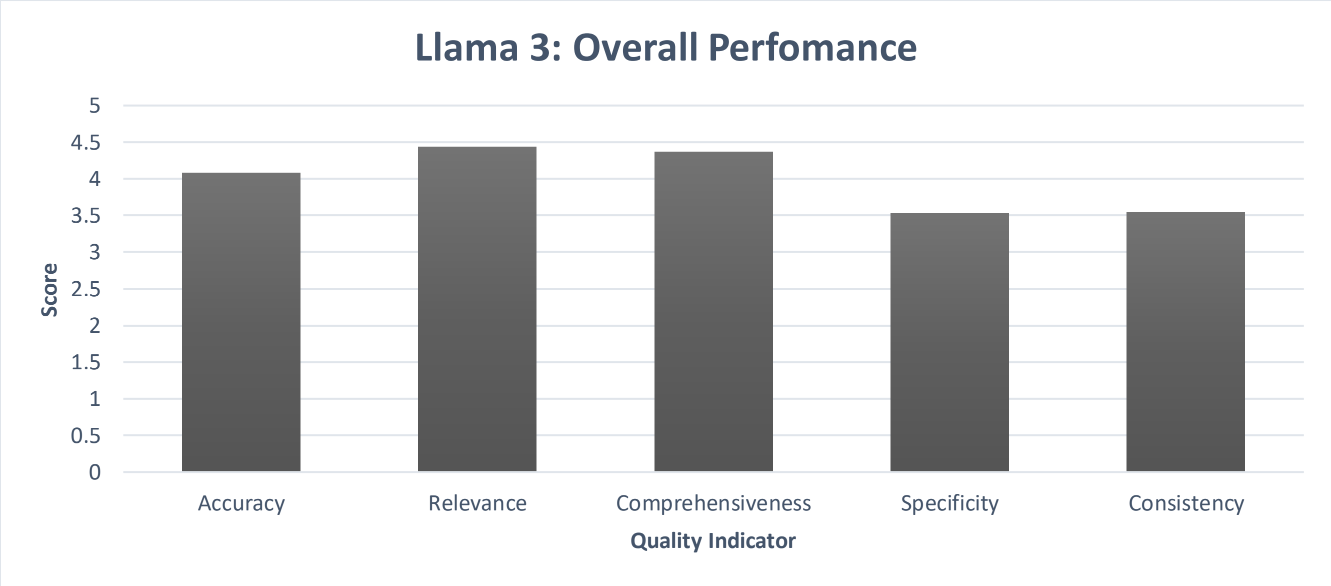

Figure 110: Llama 3, Overall Performance

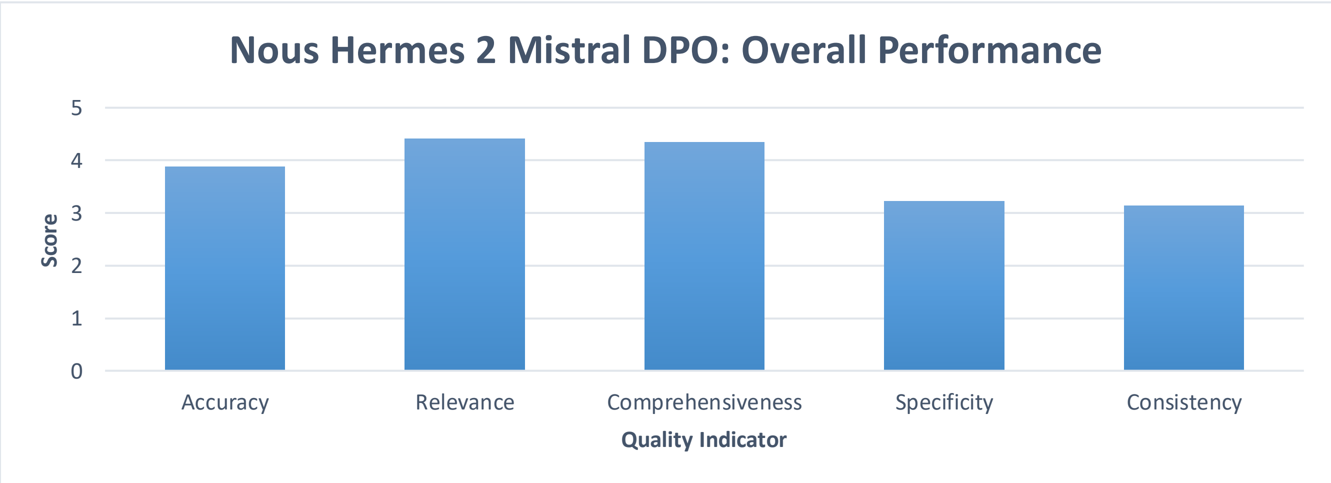

Figure 111: Nous Hermes 2 Mistral DPO, Overall Performance

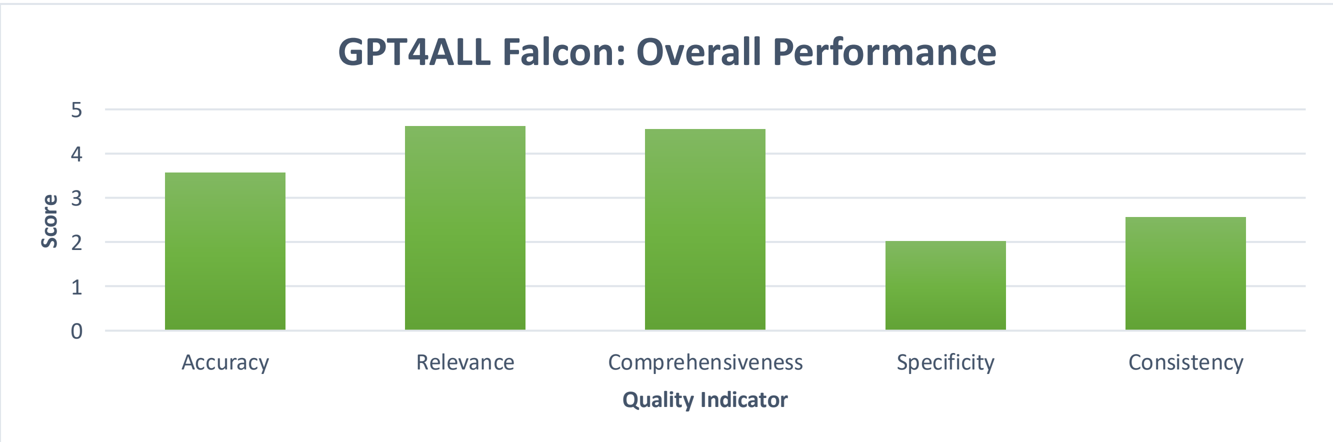

Figure 112: GPT4ALL Falcon, Overall Performance

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The results extracted from the quantification and evaluation process of the models' performances surely correspond to the observations stated in the previous chapter of this Thesis regarding each model's responses. More specifically, the following are established:

V.I. Evaluation of Llama 3

By studying the results of the assessment process based on Quality Indicators, it is obvious that Llama 3 demonstrated adequate performance across all the Quality Indicators. More specifically, the model responded satisfactorily in cases where comprehensive, fact-retrieval, specifying and complex questions were addressed. Additionally, it is important to highlight that the Standard Deviation of the model regarding each one of the Quality Indicators was generally low and compared to the Standard Deviations of the Nous Hermes 2 Mistral DPO, Llama 3 obviously exceeds in reliability and performance. In some cases, the Standard Deviations of GPT4ALL Falcon were slightly lower, but this is a subject which will be discussed later in this Thesis. The Mean Performance Scores along with the low Standard Deviations of the model, indicate that the Llama 3 provides responses which are tightly clustered around the mean values which are calculated for each Quality Indicator, something that indicates consistency in its answers. Consequently, it is safe to say that the High Mean Values of the Scores combined to the Low Standard Deviations indicate a reliable model with a behavior which is certainly preferred compared to the rest of the models.

Of course, the capabilities Llama 3 demonstrated can be further enhanced via Fine-Tuning with an appropriate Dataset which targets specific Shaft Alignment complexities related to Regulations and technical matters.

V.II. Evaluation of Nous Hermes 2 Mistral DPO

Nous Hermes 2 Mistral DPO is characterized by lower performances than Llama 3 regarding all of the Quality Indicators. However, the two models demonstrated closely comparable performances across the Quality Indicators of Interest, which was expected, as they were also characterized by similar trends in their responses. It is worth mentioning, that Nous Hermes 2 Mistral DPO exhibited a lower Standard Deviation than Llama 3 regarding the Quality Indicators of Relevance and Consistency, which demonstrates a less variable behavior on these matters. More specifically, it is concluded that Nous Hermes 2 Mistral DPO compared to Llama 3 was characterized by a more satisfying performance when it came to providing in-context and compliant with the Local Documents responses. The aforementioned observation demonstrates a minimization of the need for extensive validation of the model's responses.

In conclusion, even if Nous Hermes 2 Mistral DPO's performance was slightly inferior to Llama 3 regarding all of the Quality Indicators, the two models demonstrated similar trends and comparable overall performances. Along with the lower variability Nous Hermes 2 Mistral DPO demonstrated in Relevance and Consistency, it is safe to say that though a Fine-Tuning process its performance would be further improved to a point where the model could be a valuable, effective and reliable tool for Students and/or Professionals.

V.III. Evaluation of GPT4ALL Falcon

The results of the assessment process of GPT4ALL Falcon's performance are absolutely consistent with the observations made in the previous Chapter of this Thesis. More specifically, as it was thoroughly discussed previously, while the model was almost always in-context and demonstrated great capabilities in terms of comprehending the questions addressed and the subject of interest, its responses were always too short and not further elaborated. Moreover, it was established that the model lacked significantly in terms of Regulation Citing. All of the aforementioned are clearly demonstrated in the Mean Performance Scores of the model. The Mean Performance Scores of the model regarding Relevance and Comprehensiveness are remarkably higher than the rest of the models. However, regarding the Quality Indicators of Accuracy and Specificity, GPT4ALL Falcon lacks significantly compared to Llama 3 and Nous Hermes 2 Mistral DPO. Additionally, all of the prementioned observations are demonstrated through the Standard Deviations of the model, which are significantly lower regarding Relevance and Comprehensiveness. All of the aforementioned combined

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conclude that GPT4ALL Falcon demonstrates strong capabilities in providing responses which are contextually appropriate and correspond in the Local Documents provided. However, several areas of improvement are established in terms of Accuracy and Specificity. Even if the model indicated incompetency in many areas, its Relevance and Comprehensiveness performance underscores a potential for refinement through Fine-Tuning with an appropriate and targeted Dataset or via optimizing some of the model's parameters.

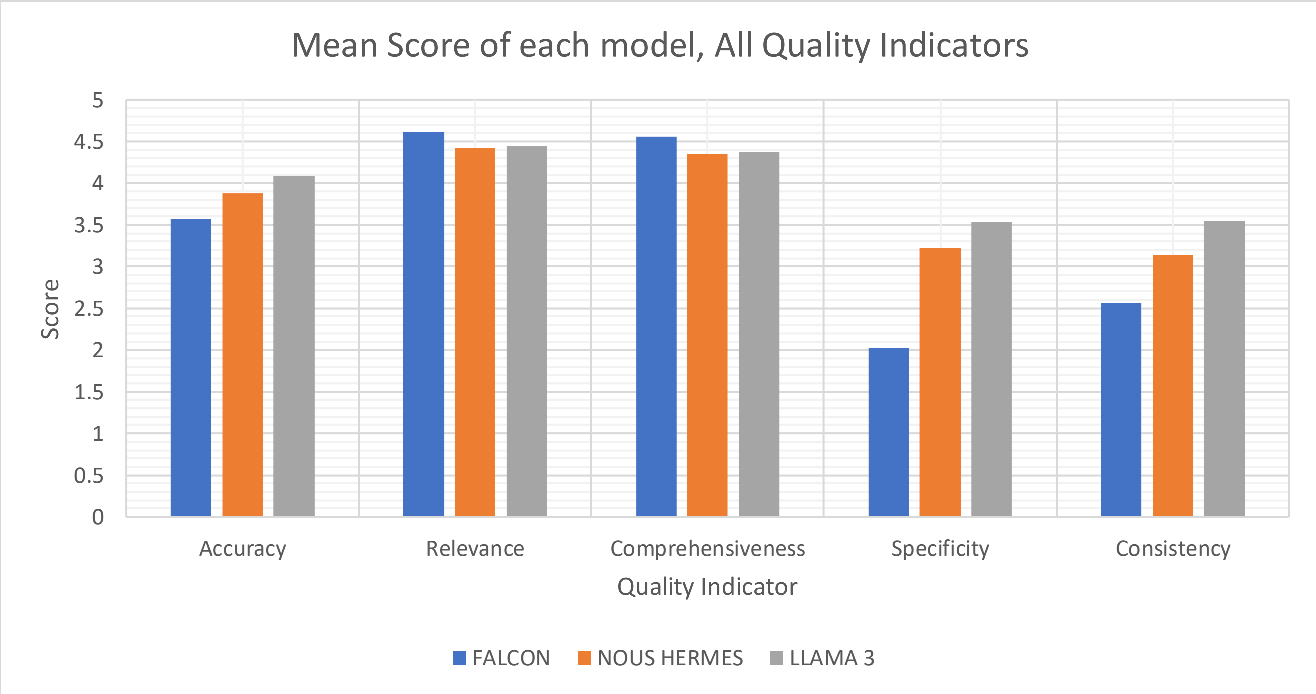

Figure 113: Mean Performance Score of each Model, All Quality Indicators

VI. Conclusion of the Assessment Process

Conclusively, it is safe to say that Llama 3 demonstrated superior performance among the evaluated models, as it was characterized by persistency and reliability across all of the Quality Indicators, combined with low deviation characteristics. The results of the assessment process indicate that Llama 3 is capable of providing in-context, detailed and accurate responses regarding Shaft Alignment and the Regulations that govern it. Optimization of the model through Fine-Tuning with a targeted Dataset will result in a reliable model capable of assisting Students and/or Professionals in handling and extracting information from large amounts of documents, such as Classification Societies' Regulations regarding Shaft Alignment.

Furthermore, it was established that the responses of the LLMs which utilized and indigested the Local Documents extracted from this Thesis were significantly more satisfactory than the answers Chat-GPT provided for the same questions. It is also worth highlighting for one more time, that the performance levels of the models were higher when the Local Documents included the First Part of this Thesis, something that leads to the conclusion that a swell-structured classification and organization of the Parameters of interest is of crucial importance for applications comparable to this Study's purpose. The responses of the models when the Local Documents did not include the First Part of this Thesis are not included in the present report, as they did not provide any valuable insights related to the scope of the Thesis.

In conclusion, the results extracted from this Study are satisfactory and the utilization of LLMs which are Fine-tuned with a tailored Dataset focusing on specific technical-related issues can definitely extend in various sectors of our profession, as Naval Architects and Marine Engineers are several times during their Studies and their Professional Life asked to comprehend and apply principles and guidelines which are entailed in large amounts of documents. The results of this study also indicated that the level of performance of the LLMs was

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significantly higher than ChatGPT's regarding Shaft Alignment and its Regulatory Framework, which demonstrates the need for more tailor-made AI Solutions when it comes to such specific and narrowed fields of knowledge.

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CONCLUSIONS AND FUTURE WORK

As it was established in the present Study, Shaft Alignment is a multiparameter and complex procedure, but at the same time crucial for the vessel's seaworthiness and reliability, as well as for the personnel's and the environment's safety. Classification Societies are responsible for monitoring, supporting and supervising every stage of a vessel's life, including Shaft Alignment. For this reason, Regulations which govern Shaft Alignment as well as Guides are published from each Classification Society. For the purposes of this study, the following major IACS Member Classification Societies were studied: ABS, DNV, BV, LR and ClassNK. The latest versions of each Class Rules and Guidelines was utilized for this Thesis.

The regulations relevant to Shaft Alignment are complex and thorough. However, the differences between each Class rules are not of significant magnitude. After all, they all have the same scope: The uninterrupted and safe operation of the vessel, as well as the safety of property, life at sea and environment. The differences were targeted in explanatory issues, as the essence of the Rules was similar. The limits set by each Class are generally either close to one another or similar, due to the fact that they usually arise from common industry practices, and they sometimes are experience-related.

However, the classification of the information, as well as the comprehension and the application of the principles of these large amounts of documents is indisputably hard, especially for Undergraduate Students, as they are characterized by a dense context with a lot of details. For this reason, AI Models enhanced in order to facilitate this process would be an extremely practical and useful tool. This is why, for the purposes of this Study, an evaluation of the performance of three opensource LLMs in terms of "comprehension" was conducted. These LLMs utilized also "Local Document" access as additional data source. The Local Documents consisted of "Part A" text of this Thesis as well as the Class Rules, Regulations and Guidelines utilized for its composition. Through experimenting, it was established that the model performance without providing them the comprehensive text of Part A from this Thesis was significantly low. The performance of the LLMs was then evaluated and quantified though a Scoring System based on the following Quality Indicators: Accuracy, Relevance, Comprehensiveness, Specificity and Consistency. The results indicated that two out of the three models which were utilized for the purposes of this Thesis, namely: Llama 3 and Nous Hermes 2 Mistral DPO, demonstrated strong capabilities in the area of comprehending large amounts of technical-related documents and providing accurate, detailed, explanatory answers with Regulatory Citations. The results extracted from this Thesis can be expanded into several technical-related matters and refer to Shaft Alignment issues as well as more generic technical queries. Consequently, it is concluded that providing a tailored Local Data pool for the purposes of specific technical subjects of interest, a very useful tool can be developed, aiming to assist not only students, but also professionals, in managing and comprehending large amounts of technical related documents. However, it is worth mentioning that during this Thesis, the importance of a careful and well-structured local document is highlighted.

In conclusion, this study provided very satisfactory results in terms of properly utilizing AI Models from a comprehensive perspective in the areas of education or business. The responses provided by the opensource LLMs, with access in targeted information, were more satisfactory than the relatively generic and often misleading answers provided by ChatGPT in the same questions. This indicates the need for tailor-made AI applications regarding specific technology-related issues. The results could be significantly improved, in the future, fine-tuning and retraining them with a tailored QnA Dataset, which would allow the model(s) to align with the latest specific requirements of Shaft Alignment related issues. The utilization of domain-specific fine-tuned AI models could facilitate and improve the "Lifelong Educational Experience" of Naval Architecture and Marine Engineering students in terms or comprehending complex principles, terminologies, requirements and configurations. This study aspires to lay the groundwork for further improvement and advancement in the area of personalized assistance in education, utilizing AI.

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