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Practices and Procedures for the Alignment of Marine Main Propulsion Shafting Systems

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PRACTICES AND PROCEDURES
FOR THE
ALIGNMENT OF MARINE
MAIN PROPULSION SHAFTING SYSTEMS

Prepared by

PANEL M-16
PROPULSION SHAFT SYSTEMS

of the

SHIPS' MACHINERY COMMITTEE

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UNIT CONVERSIONS

The original document contained only English units. Metric units are now provided as well. Where the values in English units are general the converted values are rounded and approximate, but where precise numbers were given precision was maintained.

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DISCLAIMER

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SECTION 1.0 INTRODUCTION

1.1 INTRODUCTION AND DEFINITION OF PROPULSION SHAFTING SYSTEM ALIGNMENT

Common sense suggests that the bearings supporting marine propulsion shafting be aligned so they hold the shaft in a straight line, since offsetting one bearing with respect to another would bend the shaft. Moreover, industrial machinery is typically aligned so that the shafts of the driving and driven machines are collinear [1]. However, this approach is not always appropriate for marine main propulsion shafting because: a) its bearings move as conditions change, and b) the shafting is never really straight because it deflects under its own weight and the weight of the propeller and other attached components. Therefore, the vertical and athwartships position of the bearings must be aligned so that the limits established to ensure proper system operation are not exceeded for all normal ship operating conditions. It follows that shaft alignment could also be called bearing alignment, since the alignment is a direct result of the position of bearings that support the shafts.

1.2 HISTORICAL PERSPECTIVE

In the early 1960s, it was discovered that serious misalignment of propulsion shafting caused gear skew on large, steam turbine bull gears, which in turn initiated pitting on the second reduction gear teeth. At that time shafting systems usually had too many bearings, and a straight-line alignment was considered to be a good alignment. However, the straight-line alignment tended to place a large amount of the attached line shaft weight on the aft bull gear bearing in the cold condition. Then, as the gear heated up, even more shafting weight shifted onto this bearing, while the forward gear bearing became more unloaded. This unequal loading of the gear bearings caused gear skew. Computerized shafting analysis was used to "misalign" the bull gear in the cold condition relative to the propulsion shafting, so that in the hot condition the vertical loads on the two bull gear bearings would be equal within an acceptable limit. The SNAME paper "Coordinated Alignment of Line Shaft, Propulsion Gear and Turbines", by Andersen and Zrodowski [2] firmly established the relationship between shaft misalignment and gear problems and offered practical solutions.

As a result of information accumulated since then, today's shipbuilders produce vessels with practically no built-in propulsion shafting alignment problems. Nonetheless, alignment problems still occur and propulsion shaft alignment remains a challenging task for both shipyards and designers. Some of these alignment problems are caused by: a) worn or failed bearings, b) extensive hull damage which may have disturbed the bearing settings, and c) improperly bored stern tubes.

1.3 PURPOSE OF THE ALIGNMENT GUIDE

Ship operators surveyed by SNAME, as well as the US Coast Guard and the US Navy, expressed a need for a technical reference guide on the alignment of main propulsion shafting [3]. This document is intended to provide these guidelines for personnel involved in the design, installation, and operation of marine propulsion shaftlines. This guide provides a general overview of shaft alignment for the novice, and more detailed information for the person with some knowledge on one facet of shaft alignment that needs more information on another alignment topic.

SECTION 2.0 DESCRIPTION OF MARINE SHAFTING SYSTEMS

"A shaft is a rotating or stationary member, usually of circular cross section, having mounted upon it such elements as gears, pulleys, flywheels, cranks, sprockets, and other power-transmission elements" [4]. While this definition was taken from a mechanical engineering design text it is nonetheless an accurate description of marine propulsion shafts. This same text also provides a good description of what a shaft must endure throughout its intended life span: "Shafts may be subjected to bending, tension, compression, or torsional loads, acting singly or in combination with one another. When these loads are combined, one may expect to find both static and fatigue strength to be important design considerations, since a single shaft may be subjected to static stresses, completely reversed stresses, and repeated stresses, all acting at the same time" [4]. From these excerpts, it is apparent that marine propulsion shafting has similar design considerations to other power transmission shafting.

A marine main propulsion shafting system must accomplish the following objectives:

- transmit the power from the main engines or reduction gear to the propulsor.
- transmit the thrust developed by the propulsor to the ship's hull.
- support the propeller or propulsor rotor.

In the present context, a marine propulsion shaft connects a driving device, such as a reduction gear, engine, or electric motor to a propeller or other propulsor. For example, a reduction gear power take-off shaft that drives a generator or pump is not considered a propulsion shaft, but the shaft connecting the final gear to the propeller is a propulsion shaft.

Marine propulsion shafts can be found in many sizes and configurations, serving installations from the smallest boat to the largest container ship, over a range from less than 20 hp (15 kW) to more than 87,000 hp (65,000 kW). The shafts are driven by many different types of prime movers, including diesel engines, gas turbines, and steam turbines. In many instances a speed reduction is required which may not only dictate different alignment criteria, but also may result in complex shafting arrangements such as V- and Z-drives.

Figures 1a and 1b illustrate typical shafting systems. Propulsion shafting is comprised of inboard and waterborne shafting. The inboard shafting is located inside the hull forward the shaft seal and consists of the line shafting and thrust shaft. The corresponding bearings are called line shaft or steady bearings. The waterborne shafting is located aft of the forward end of the shaft seal and consists of the propeller shaft or tail shaft section that supports the propeller and in some cases the stern-tube shaft, which passes through the stern tube.

The basic shaft system can be designed after the fundamentals of the vessel propulsion system have been determined, i.e. prime mover, the maximum propulsor speed, and shaft power. The classification societies have formulas for calculating the minimum required shaft diameter for the different shaft sections. After the shaft diameters have been estimated, the bearings can be located along the shaftline. The bearing loads, shafting flexibility, and the shafting system lateral natural frequencies should be considered when determining the number and location of the bearings.

Reference 5 includes a formula for estimating the number of line shaft bearings. This reference also indicates that as a general rule the ratio of bearing center distance to shaft diameter may range from 12 to 22, although the final ratio must be based on the requirements for vibration, shaft strength, and flexibility requirements to maintain alignment during service conditions.

A typical shaft system may contain several different types of bearings, each with different load and lubrication requirements. There are two principle types of bearings used with marine shaft systems: anti-friction (ball or roller type) bearings and hydrodynamic (journal or stave type) bearings. Within each type there are numerous variations involving different bearing materials. Each different configuration will have distinct criteria with regard to the methods of lubrication, maximum and minimum allowable load, cooling requirements, and wear characteristics.

SECTION 3.0 THE IMPORTANCE OF PROPER PROPULSION SHAFTING SYSTEM ALIGNMENT

The shafting system can be considered as a beam supported by multiple, unevenly spaced bearings. The bearing positions relative to the shaftline, or "offset," must be set so that the loads on the bearings and the shaft stresses are within acceptable limits. Performing regular alignment checks to maintain proper shafting system alignment is essential for reliable ship operation.

a. **Proper shaft alignment is necessary for correct internal alignment of main reduction gear slow speed elements and direct-connected diesel engine bearings.** One of the primary goals of shafting system alignment is to ensure that the main reduction slow speed gear (bull gear) or direct-connected main engine is properly aligned and that the bearing loads are within the acceptable limits set by the gear or engine manufacturer. In fact, in geared installations, much of the emphasis on achieving an acceptable shaft alignment is based on the need to prevent damage to the main reduction gears. In addition to the usual maximum and minimum bull gear bearing load limits, reduction gear vendors also prescribe a maximum load differential between the forward and aft bull bear bearings in the static condition. This bull gear bearing static load differential relates to bull gear-to-pinion tooth stresses in the dynamic (running) condition. An excessive differential can produce unacceptable gear tooth stresses, which can cause destructive gear tooth wear, pitting, and failure. Overstressed gear teeth can fail due to fatigue after many years of operation. However, the propulsion shaft-to-reduction gear alignment is not as critical for a system with a flexible coupling between the reduction gear or direct-connected engine and shafting system that isolates the gear or engine from the affects of shafting system alignment. Nonetheless, the bearings and propulsion train components forward of the flexible coupling should still be included in the alignment analysis.

b. **Proper shaft alignment prevents overloading of the shafting system bearings**. All bearings have a maximum allowable pressure load stipulated by the bearing manufacturer. Exceeding these limits can lead to premature bearing wear or failure.

c. **Proper shaft alignment prevents excessive shaft vibration from a lightly or unloaded bearing.** When a shaft is misaligned, a support bearing in the system can become lightly loaded. Then the ship can experience "pounding" of the bearing, as the load on the bearing varies from a loaded to unloaded condition during each shaft revolution. When the bearing actually unloads, the bearing span of the system dramatically increases and this increased span - from the bearings forward and aft of the unloaded bearing - can cause excessive lateral (whirling) vibration in the shafting system.

d. **Proper shaft alignment prevents shaft failure from excessive bending stresses.** Shafting systems are designed to achieve the allowable shaft bending stress levels defined by the classification societies or other good engineering standards. The shafting alignment has a direct effect on these shaft stresses and a misaligned shafting system can have overstresses that lead to eventual shaft fatigue failure. Misaligned shafting systems can produce very significant bending loads [5].

SECTION 4.0 CRITERIA FOR THE ACCEPTABLE ALIGNMENT OF A MARINE PROPULSION SHAFTING SYSTEM

The benefits of proper shafting system alignment, as discussed in Section 3, logically lead to criteria that will be used to achieve the proper alignment on a new construction or refurbished shafting system. Obviously, the criteria have engineering, economic, and scheduling ramifications. The shafting system should be designed and aligned to meet the following criteria for all normal ship operating conditions, including the changes caused by bearing weardown, thermal growth, ship loading, and the other factors which affect alignment as discussed in Section 5.

a. **The maximum allowable bull gear bearing load differential shall not be exceeded.** The shafting system alignment must be designed so it meets the criteria for the bull gear bearing differential load. This differential applies to the athwartship, as well as the vertical loads. The reduction gear manufacturer will usually provide the allowable differential. However, if this value cannot be obtained, then a general rule is that the static load differential should be less than 25% of the combined total load on the two bull gear bearings in all operating conditions to ensure that the gear tooth meshing contact is not misaligned by a skewed attitude of the gearwheel in its bearings. Sometimes it may be cost effective to perform a skew analysis to evaluate the acceptability of the measured gear loads if the gear manufacturer's load differential appears overly restrictive.

   To meet this load-differential criterion, the shafting system designer must provide for a sufficiently flexible system at the shaft/gear interface using an appropriate bearing spacing. A flexibility factor (or allowable setting error) is commonly used to determine whether the shafting system has reasonable flexibility. The flexibility factor is determined from the values for maximum allowable bull gear differential and the influence coefficients of the aft and forward bull gear bearings on themselves. A minimum absolute value of 0.010 inch (0.25 mm) for the flexibility factor is considered acceptable. Sections 5.1.a and 6.3 contain more information on flexibility factors and bearing influence coefficients, respectively.

   Some gear manufacturers use the gear/line shaft gap and sag to specify the alignment limits (See Section 7.4 for information on the gap and sag method). This is an older method of aligning the gears to the propulsion shafting, although it is still used during new construction for making an approximate alignment until the shaft sections are connected. At least one gear manufacturer specifies that the gear be aligned with zero gap and sag to the free-hanging shaft in order to prevent any moment or shear from being imposed on the gear shaft. However, when there is a long span from the coupling to the first line shaft bearing, this alignment can overload the line shaft bearing and/or produce high bending stresses in the line shaft. Some gear manufacturers and almost all engine manufacturers specify limits on the moment and shear applied to their couplings (which can be measured by the strain gage method discussed in Section 7.3)

b. **No support bearing in the system shall be loaded above its maximum allowable pressure.** Generally, the support bearings are sized to accommodate the shaft vertical design loads for all anticipated alignments and ship conditions. The maximum allowable bearing loads are usually defined in terms of pressure and are specified by the bearing vendors or based on historical results. The maximum allowable load for a particular bearing is usually determined by using the projected area of the bearing (the product of the effective bearing length and the shaft outer diameter), although not all classification societies compute the pressure load on the bearing in this manner.

(1) Line shaft bearings will run well with pressure limits of 75 psi (c. 500 kPa) for disc lubricated bearings, and 50 psi (c. 350 kPa) for ring lubricated bearings. It is very important that the bearings have a uniform load distribution in both the vertical and horizontal planes since very high-localized pressures can occur on a bearing that is tipped down or skewed athwartships relative to the shaft. The load distribution on the line shaft bearings can be checked by applying a strip of bluing axially along the journal and rotating the shaft through the bearing. All the bluing should be wiped off; if not, the area with the bluing is not in contact with the bearing. The end of the bearing should be shimmed until the bluing is evenly removed when the shaft is rotated. The bearing's athwartships load distribution can also be estimated by removing the bearing cap, and measuring the side skew at the four corners with a feeler gage. A hot-running line shaft bearing usually indicates an overloaded bearing if the lubrication system is operating satisfactorily [6].

(2) In general, an anti-friction bearing large enough to fit the shaft diameter is capable of supporting a much higher load than can be applied from a reasonable alignment. The B_{10}* life for this type of bearing typically exceeds 100,000 hours. Therefore, almost any positive, downward load will be acceptable. Although anti-friction bearings are not nearly as sensitive to angular alignment as sleeve bearings, some care needs to be taken during installation so that even self-aligning bearings start out with the proper angular alignment.

(3) Waterborne-shaft bearings may be either oil- or water-lubricated. Oil-lubricated bearings usually have a length-to-diameter (L/D) ratio of 2. Bearings using synthetic material are usually longer, with an L/D between 2 and 4. The American Bureau of Shipping allows oil-lubricated, white-metal lined outboard bearings to be shorter (but not less than L/D = 1.5) as long as the bearing nominal pressure is not more than 116 psi (800 kPa). Likewise, oil-lubricated synthetic bearings may be less than 2D in length (limit is L/D = 1.5) if the bearing nominal pressure is not more than 87 psi (600 kPa).

The following pressures are the maximum allowable limits for each bearing type:

Oil-lubricated synthetic liner:	87 psi (600 kPa)
Oil-lubricated babbitt liner:	116 psi (800 kPa)
Water-lubricated synthetic liner:	70 psi (c. 500 kPa)
Water-lubricated rubber or wood liner:	40 psi (c. 300 kPa)

(4) The propeller bearing is located just forward of the propeller and its primary function is to support the propeller and shaft weight, and counteract the propeller induced hydrodynamic forces. The slope (or cant) of the propeller bearing is critical and the bearing must be aligned with the tail shaft so that the load is distributed over most of the available bearing area. Otherwise, the overhung weight of the propeller will deflect the tail shaft and the total propeller weight will be concentrated on a small area at the aft end of the bearing. A change in the bearing offsets rarely causes a significant change in the load on the propeller bearing and many more propeller bearings have failed from a poor load distribution than from an excessive overall

* B_{10} is the life in hours that a group of apparently identical bearings will exceed or complete.

load. This is particularly true for oil-lubricated metal-lined bearings, which have less tolerance for slope misalignment because of their stiffness.

One way to align the propulsor bearing considering its slope in relation to the theoretical shaft slope through the bearing is to run the shafting alignment computer program with the propulsor bearing located at its fore-and-aft center. The propulsor bearing is then set to a vertical slope corresponding to 60%-100% of the calculated slope. A rule-of-thumb for the slope limit on oil-lubricated bearings is 0.003 radians, and a rule-of-thumb for water-lubricated strut or stern-tube bearings is that the bearing may be out of parallel to the shaft by no more than the design minimum bearing clearance. A particularly poor alignment is one where the shaft bears downward at one end of the propulsor bearing, and upward at the other end of the bearing, passing through the bearing clearances.

Propulsor bearings with a misaligned slope also need a higher shaft rotational speed to develop an adequate hydrodynamic lubrication film, and therefore are more apt to generate stick-slip noise at slow shaft speeds. The shaft speed required to prevent stick-slip noise increases as the slope misalignment worsens.

(5) In general, the load on a forward stern tube bearing varies considerably with the vertical or athwartship offset. It is just as important to achieve a good load distribution across a forward stern tube bearing as it is for a propulsor bearing. This bearing should also be sloped if the output from the alignment software shows a difference in slope of more than 0.003 radians between the shaft and a non-canted forward stern tube bearing. Sometimes sloping the forward stern tube bearing can be avoided by lowering the propulsor bearing, which achieves the same net result.

c. **No support bearing in the system shall be loaded less than its minimum allowable load.** A minimum allowable load is required for each bearing to stabilize the shaft in the bearing and prevent shaft vibratory problems. Determining the minimum load is a somewhat arbitrary decision, although zero or negative (upward) loads on the bearings are not acceptable. The minimum load can be chosen so that the bearing will not become unloaded because of the normal, expected shaft eccentricities (runout). One guideline is that the minimum load shall be at least 50% of the bearing's initial design load.

d. **The maximum allowable stresses in the shafting shall not be exceeded.** Obtaining acceptable shaft bending stresses is a high-priority alignment criterion since the shaft can fail from excessive bending stresses. Since the alignment directly affects the bending moments and bending stresses in the shaftline, the alignment affects the overall shaft stress and shaft diameter calculations. However, the classification societies, such as the American Bureau of Shipping (ABS), generally calculate the minimum shaft diameters based on a torsional stress limit without regard to the bending moments in the shaft (or other loads, such as thrust, which produces relatively low stress). Therefore, the classification society equations use other means to compensate for the loads not included in their formulas. For example, the ABS accounts for the higher bending moments experienced by tail shafts and stern tube shafts by: a) using an appreciably larger "shaft design factor" than for line shafting, and b) limiting the material strength used in the tail shaft calculations.

When the bending stresses are high, it is important to evaluate the fatigue risk. For instance, the US Navy's shaft stress formulas [7] require a more detailed analysis than those used by the classification societies, and include the bending moments determined from the alignment analysis and the resulting fatigue stresses. The US Navy design rules specify the minimum factors of safety at various points along the shaftline, and limit the maximum bending stress in steel waterborne shafting to 6000 psi (41,400 kPa). The factor of safety calculation accounts for stresses caused by: a) steady state torque, b) steady state thrust, c) alternating (cyclic) bending moments determined from the computerized shaft alignment, and d) alternating torques. The minimum allowable factor of safety depends on whether the vessel is a surface ship, icebreaker, or submarine, and whether the shaft section is waterborne or line shafting.

e. **The design alignment criteria for directly connected engines, special couplings, or other equipment, shall not be exceeded.** The manufacturers of large main engines directly connected to propulsion shafting usually specify shaft alignment limits to prevent excessive crankshaft deflections. A limit on the bending moment and shear at the engine coupling is a relatively common alignment requirement. The strain gage method is an ideal method (although not the only way) to measure the coupling moment and shear. Existing shafting installations that put excessive moment or shear loads on the engine coupling can usually be corrected by a vertical or athwartships adjustment of the line shaft bearings. The engine manufacturer may also specify limits on the coupling gap and sag (See Section 7.4 for information on this method). This is a valid requirement if strain gages are not used, although it is necessary to uncouple the shaft and measure the gap and sag. If strain gages are used, it is worthwhile to ask the engine manufacturer to provide the equivalent moment and shear limits. Otherwise, the alignment software will produce the equivalent moment and shear corresponding to the specified gap and sag. If the shafting and bull gear contain special couplings, such as flexible or spline couplings, then the coupling alignment limits should be obtained from the coupling vendor.

SECTION 5.0 FACTORS THAT AFFECT THE ALIGNMENT OF MARINE PROPULSION SHAFTING SYSTEMS

The actual alignment of a shafting system changes as its bearings move relative to one another because of changes in the condition of the ship and its environment. The goal is to obtain a satisfactory alignment so that the bearing reactions, shaft stresses, and clearances between the shaft and adjacent components and structures remain within acceptable limits for all normal operating conditions. The alignment analysis and the alignment measurements must quantify and account for the various factors that affect the alignment. Any effect that might move one or more bearings relative to the others must be considered. After the individual effects are quantified, they can be combined using superposition. The practical aspects of shipyard construction must also be considered when specifying the alignment. If the alignment criteria are too stringent, then excessive costs will be incurred during construction. If some criteria are too impractical, then they should be re-evaluated.

5.1 FACTORS THAT AFFECT THE ANALYSIS OF THE SHAFT ALIGNMENT

a. **Flexibility of the shafting system.** The alignment of shaft systems in land-based installations relies on the foundations of the equipment to be fixed relative to each other. However, ships are inherently flexible and so there is an almost continuous movement of the shafting system's supports. The required flexibility of the shafting system depends on the flexibility of the ship along the shaft length, and as a general rule, the shaft system should be more flexible than the ship. The flexibility of a marine shaft system has a distinct effect on its performance and a shaft whose flexibility is well suited to its installation will have little problem, regardless of the vessel's load or other conditions.

   Some ships are much more sensitive to shaft alignment than others, and the flexibility of the shafting system – or conversely its stiffness – is one of the most important factors. A short, rigid shaft system is very sensitive to the small changes in alignment that occur normally, such as thermal growth or vessel loading. A stiff shaft is generally less forgiving of excursions outside its design envelope, and may be damaged by a permanent or transient foundation change caused by sea conditions, grounding, or loading. For example, a large diameter shafting system of short overall length tends to be more sensitive to changes since it is less flexible. In contrast, a soft shaft with high lateral flexibility relative to the hull structure is less sensitive to thermal growth, ship loads, and sea conditions. Accordingly, it is preferable to install as few bearings as possible in order to produce a lower bending stiffness. A shaft system that is designed to be flexible and insensitive to small changes will generally have fewer problems caused by shaft misalignment.

   A large number of vessels with direct-coupled propulsion systems (propeller-shafting-diesel engine) have very rigid shafting, and are very sensitive to changes in the bearing offsets that may result from hull deflection, bearing wear, or thermal growth. Fortunately, this shafting is generally installed in a very rigid bottom structure, so there is less hull deflection, which somewhat reduces this effect. Nonetheless it is strongly recommended that the shaft alignment analysis for certain categories of vessels (in particular those with rigid direct-coupled propulsion systems) consider the influence of the surrounding structure on the bearing offset.

The issue of shaft stiffness will become increasingly important as ship powers continue to increase. Until the 1980s the general limit for power transmitted on one shaft was 60,000 hp (45,000 kW). Formally naval vessels had the highest-powered shafts, which usually operated with relatively high shaft speeds. Since the shaft torque decreases with increasing shaft speed (rpm) and the shaft diameter is determined in part by torque (and not power), naval shafts do not have inordinately large diameters. Naval shafts are also relatively long, incorporating several line shaft, stern-tube, and strut bearings. Therefore, naval shafts are fairly soft since they have a relatively small diameter and long length.

The engine powers in recent commercial ships have increased to over 90,000 hp (67,000 kW) at less than 95 rpm. Furthermore, modern long-stroke diesel engines can fully exploit the efficiency of slower turning propellers. Consequently, large diameter, stiff shafts are required to transmit the higher torque from heavy propellers. Also, the trend to place the main engines further aft on commercial vessels reduces the shaft lengths, and the shafting becomes stiffer. In some cases, the minimum shaft diameter is larger than normally required by the design rules to prevent torsional vibrations and fatigue failure. The propulsion shafting is therefore stiffer, and large shear force and bending moment changes that can jeopardize the machinery alignment are induced by variations in the hull deflections. This does not imply that only high-powered, low-speed shafts are stiff, because the stiffness of a shafting system depends on the shaft diameters and lengths, as well as on the bearing location and support stiffness.

In addition to the cases already cited, the shaft alignment needs to be evaluated more carefully for the following alignment- sensitive installations:

- propulsion shafting with a reduction gear where the bull gear is driven by two or more ahead pinions
- propulsion shafting with a power take-off or with booster power arrangements
- propulsion shafting for which the tail shaft bearings are to be slope bored.

As discussed earlier, the flexibility factor can be used to evaluate the flexibility of the shafting in way of the reduction gear, and therefore, the sensitivity of the gear to improper alignment [3]. The flexibility factor represents the total of the permissible error in estimating the thermal rise of the slow speed gear bearings relative to the line shaft bearings and the permissible error in setting the gear to the line shafting without exceeding the maximum allowable difference in the static slow speed gear bearing reactions. An absolute minimum value of 0.010 inch (0.25 mm) for the flexibility factor is considered acceptable [5]. See reference 8 for more information on the flexibility factor. The flexibility factor (FF) is mathematically defined as:

$$FF = \Delta R / |I_{11} - I_{22}|, \text{ where} \qquad (5\text{-}1)$$

ΔR = maximum allowable difference between the two reduction gear bearing static reactions, lbf (N)
I_{11} = influence coefficient of forward reduction gear bearing on itself, lbf/mil (N/m)
I_{22} = influence coefficient of aft reduction gear bearing on itself, lbf/mil (N/m).

The influence coefficient matrix can be used as a rough guide for determining whether a shaft is stiff or soft, but there are no established criteria. However, gear bearing influence coefficients over 1000 lbf/mil (175 N/m) or line shaft bearing influence coefficients over 500 lbf/mil (88 N/m) may indicate a rigid system.

b. **Thermal growth.** Thermal expansion of equipment causes relative movement of the bearings, which changes the shafting system alignment. The thermal growth occurs primarily at the bull gear, thrust bearings, and in direct-connected diesel engines, where hot lubricating oil sumps located below the machinery lift it when going from the cold to the hot operating condition. Thermal changes in the line shaft and tail shaft bearings can be ignored since they are generally small compared to the engine or gear thermal growth.

Propulsion shafting is generally aligned when cold (and it may be very cold in northern shipyards). The cold alignment must be set so that the desired bearing loads are achieved when the ship is in the hot, running condition. Since the thermal growth of the bull gear changes the bearing loads, the thermal growth must be calculated and integrated into the alignment analysis. This leads to the establishment of design cold and hot alignments.

The bull gear has two bearings relatively close together, so the influence coefficients are high, but not equal, and sensitive to errors in the assumed thermal growth. The gear manufacturer may estimate the thermal growth of the gear bearings. However, in some practitioner's experience, the gear manufacturer's load tolerance is often unnecessarily tight, and rarely do the two bull gear bearings undergo the same thermal growth. This is particularly true on large, double reduction, steam turbine gears. The first reduction gear often runs 10°F (6°C) or so hotter than the second reduction gear because of the higher pitch-line velocities, which will cause the bull gear bearing located underneath the first reduction gear to expand more that the bearing on the other side of the bull gear. The gear bearing thermal growth can be calculated using the equation below, with different operating temperatures for the two bearings as appropriate.

The following general rules are useful:

(1) The gear manufacturer's estimate of the thermal growth can usually be accepted. When this is not available, the thermal growth, Hv, for steel can be calculated as:

$$Hv = k \cdot Dv \cdot \Delta T, \text{ where} \quad (5\text{-}2)$$

k = coefficient for thermal expansion of steel, $6.6 \cdot 10^{-6}$/°F ($11.7 \cdot 10^{-6}$/°C)

Dv = vertical distance from shaft centerline to the bearing foundation, inch (mm)

ΔT = difference between the hot operating and cold temperatures, °F (°C)

When the hot operating temperature is unknown, then usually a value of 120°F (50°C) may be assumed [9].

(2) To satisfy the reduction gear load differential criteria it is necessary to make the best thermal growth corrections possible, but then check the hot alignment. This can be done on a sea trial using the strain gage method described in Section 7.3. Ideally, telemetry could be used to measure the shaft strains underway, but it is possible to get a fairly good representation of the hot loads without telemetry by shutting down the propulsion system to measure the strains after running several hours so that the gear casing and foundations are at their operating temperatures.

(3) Although the thermal growth of the line shaft can be ignored, there may be an exception to this where the line shaft bearing foundation is located near equipment that has some measurable thermal impact.

It is generally recommended that the criteria in Section 4 for the maximum bearing load, minimum bearing load, and bull gear bearing load differential be met for both the hot and cold aligned conditions, although there is some debate over the need to meet these criteria for the cold condition. Some practitioners believe that the cold condition can be disregarded since it is not an actual operating condition. Others believe that since the ship actually initiates its operations in the cold condition, that this condition be considered a short-term operating condition. A reasonable compromise is to design for acceptable alignment in the cold condition as much as is practical.

c. **Weardown of waterborne bearings.** Outboard bearings are often water-lubricated and have staves or liners that experience significant weardown over long periods of time, which can have a significant influence on the alignment. Extreme bearing weardown can cause the tail shaft and line shaft to fail. Inboard bearings are often oil-lubricated, metal-lined bearings that wear very little over their lifetimes. The weardown of inboard or outboard oil-lubricated bearings can usually be ignored, although occasionally an oil-lubricated bearing will wipe and cause a small change in alignment.

The design shafting alignment must account for the change in loads from bearing weardown; an acceptable alignment must be maintained throughout the allowable weardown range. Generally, weardown tends to decrease the waterborne bearing loads while increasing the inboard bearing loads. A bearing weardown analysis is usually performed as part of the computerized shaft alignment analysis. The weardown analysis examines the effects on bearing loads at wear intervals typically 0.005 to 0.010 inches (c. 0.13 to 0.25 mm) throughout the allowable weardown range. It is usually assumed that the water-lubricated bearings wear at a rate proportional to their relative pressure loading, although many other factors can accelerate the weardown. For example, if the pressure load on the main propulsor bearing is twice that of the aft stern tube bearing, then the propulsor bearing will wear at twice the rate of the stern tube bearing.

Since bearings can wear unevenly in a shaftline with multiple bearings, several weardown scenarios are possible. The most conservative approach is that each bearing may reach its wear limit while the others are nearly new. This approach is probably not realistic, except if one bearing is replaced while the others remain worn. Another approach, as discussed above, is to perform a collective weardown analysis, which assumes that all the bearing staves or liners are new or replaced before the weardown begins. Starting with this assumption, the unit load on each bearing subject to weardown is calculated. One of the bearings is then assumed to wear a fixed incremental amount. The incremental weardown on the other bearings is then calculated assuming that they wear in proportion to their unit load. The unit load on each bearing is then recalculated from the incremental wear on each bearing. This iterative calculation is repeated until one or more bearings reaches its weardown limit or the bearing load goes outside of its acceptable range.

It is relatively easy to predict the alignment changes that will occur over the weardown range when there is only one wear-prone waterborne bearing. The load and bending stresses will increase at the next bearing forward as bearing wear progresses. This next bearing will usually be a line shaft bearing, which can be lowered to keep its load or bending stress within limits. However, if there is only one line shaft bearing (or none) between the tail shaft and the gear or engine, then the resulting misalignment can have serious consequences on that equipment. In that case, periodic alignment checks should be made to prevent adverse effects on the bearings, shafting, gear, or engine.

Alignment changes in a shaft system with more than one wear-prone waterborne bearing are much more problematical because it is more difficult to predict the wear progression in each wear-prone bearing. Initially, the aft bearing is prone to wear faster than the forward bearing because of heavier unit pressures, possibly poor load distribution, and the ingestion of sand or silt in the aft bearing. As the aft bearing wears, the load increases at the forward bearing, which may or may not cause it to wear. There will be a significant change in the tail shaft to line shaft alignment that is not easy to plan for without experience in this area. Therefore, as in the single wear-prone bearing case, periodic alignment checks are needed to monitor the weardown.

d. **Drydock versus waterborne alignment.** The drydock alignment condition can be an important consideration during the alignment design process since it may be necessary to align the shafting in drydock. The shaft alignment varies between the drydock and waterborne conditions, mostly because of buoyancy effects. The alignment will also change for different drydock/keel blocking arrangements. The shafting should be aligned in drydock so it achieves the proper alignment when the ship is afloat. In this way, the shipbuilder avoids the more cumbersome alignment adjustments in the waterborne condition.

The shafting alignment software calculates the change in bearing loads caused by the buoyancy of the waterborne shafting and propulsor. However, the vessel's hull deflection along the length of the shaft is also important because it determines the relative location of each bearing. Advances in computer technology and structural analysis give the designer a much better understanding of how changes in the hull deflection affect the shaft alignment. The hull deflection along the shaftline should be evaluated for all normal waterborne ballasted and loaded conditions and for all building ways and drydocked conditions where alignment work might be performed. The hull deflection of a ship in drydock or under construction is affected by the number, position, and type of blocks in the docking or construction support plan.

Since the hull deflection calculation is subject to relatively large tolerances compared to the bearing offsets, it is preferable to obtain the necessary hull deflection data from similar ship designs. If these data are not available, then a naval architect's analysis of the hull deflections can be used to estimate the position of the bearing foundations. ABS requires that alignment calculations consider the effect of the deformation of the aft end of the hull, where known. A common case concerns ships with waterborne shafting, where hull deflection in way of the strut may be of the order of several inches (centimeters). However the strut bearing has a relatively low influence coefficient and the shafting system is not extremely sensitive to errors in the hull deflection calculation.

When work that will affect the shafting alignment will be performed in drydock, the alignment should be checked while the ship is waterborne before going into drydock and then again immediately after drydocking. Then the correlation between the waterborne and drydock bearing loads can be used to predict the waterborne bearing loads when the ship comes out of drydock. It is important to perform a final waterborne alignment check and adjustment after undocking and under the normal draft and load conditions. If this cannot be done, then it is necessary to make the alignment adjustments in drydock based on the correlation between the drydock and waterborne loads. When coming out of drydock, the ship's hull should be given some time to stabilize (up to two days, if practical) before performing the final alignment check.

e. **Ballast, cargo, and fuel loads.** Although the ship's ballast, cargo, and fuel loads affect the hull shape they are often neglected when considering factors that affect the shaft alignment. Their effect on shaft alignment varies with the type and size of the vessel and the shafting arrangement. Cargo loading can have a significant impact on the shaft alignment, especially for ships that have a wide range of loading options that affect the hull deflections and the relative bearing positions. Unfortunately, many ships so affected tend to have relatively stiff, short, and stubby shafting systems that are sensitive to misalignment. There have been cases where ship classes whose designs did not account for hull loading had to be assigned restricted loading patterns to maintain acceptable propulsion performance.

The analysis of ship loading effects on shaft alignment can be difficult, which may explain why this factor is often ignored. As a minimum, the alignment analysis for cargo ships should account for the "no-load" and "full-load" conditions. However, for some ships, the intermediate cargo loading condition may create a shaft alignment that falls outside of the no-load and full-load conditions, and so the intermediate ship loading should also be analyzed. If there is a large variation in results, then the shafting should be aligned to the mean draft condition to minimize the bearing load changes on both sides of the mean condition, which will hopefully produce an acceptable alignment.

Since the calculation of these ship-loading effects may be inaccurate, it is preferable to obtain the necessary data from similar ship designs. If the engineer is fortunate, there will be measurements of the ship load versus shaft bearing loads, or hull deflection data that can be related to the relative bearing foundation positions. If data from similar ship designs is not available, then a naval architect's analysis of the hull deflections for the various load conditions can be used to estimate the positions of the bearing foundations. Here are other guidelines:

(1) Short, stiff vessels can probably be safely aligned in the light-draft condition, which is apt to be the condition in a shipyard after undocking.

(2) The alignment of ships with aft engine rooms short shafting systems will vary more with aft draft than with cargo load. Unless experience says that the load variation is not significant, it is wise to measure the bearing loads at both extremes of normal operating draft aft.

(3) Shaft bearing loads on ships with amidships engine rooms can be expected to vary significantly with cargo load.

The storage of fuel affects the hull deflections and the shaft alignment in a manner similar to cargo loading and ballast. The shaft alignment measurements should be made under similar fuel load conditions when possible. The US Navy rule-of-thumb is to measure the alignment with about 80% of the fuel and/or cargo load.

f. **Vessel design.** Different types of vessels and propulsion trains have different influences and concerns. For example, vessels with water-lubricated propulsor and stern tube bearings experience weardown, while oil-lubricated bearings have practically no weardown and can accept higher load limits. Consequently a shaft to be installed in a water-lubricated bearing should be designed to be more flexible than one intended for an oil-lubricated bearing, since the former shaft will have a larger range of deflection as the bearing wears down.

Vessels that incorporate a gear in the drive train have a combination of design criteria that must be satisfied. Gear manufacturers place limits on the allowable bearing loads and allowable bull gear load differential. After a shaft is hard coupled to the final gear, the gear shaft becomes part of the shaft system and its bearing loads are influenced by the entire shaft system. The shaft must be relatively soft so that the allowable loads and load differential are not exceeded when the gear bearing offsets change because of by thermal growth, oil film thickness, and torque reaction.

Direct-coupled diesel engines have their own design criteria. The engine structure is relatively stiff and the shaft section coupled to it must be relatively soft. Movement of the shaft will influence the engine bearing loads and the engine movement will influence the shaft bearing loads. Unlike gear elements, the engine crankshafts usually have many bearings. While slight misalignment of a gear system may cause tooth contact problems, slight misalignment of a direct-coupled diesel system may cause lightly loaded or unloaded bearings at idle or stopped conditions. These same unloaded or lightly loaded bearings, however, may be forced into their normal configuration once the engine load increases causing excessive crank web deflections and stresses. This phenomenon is becoming more prominent as diesel engine shaft systems become stiffer.

g. **Propulsor hydrodynamic effects and the bore slope at the aftermost bearing.** A significant consideration is whether to slope the aftermost bearing supporting the overhung propulsor. The hydrodynamic force on an operating propulsor is off-center relative to its centerline. These hydrodynamic forces are imposed on the shaft, which may significantly alter its relative position within the aft bearings and create horizontal reactions. This eccentricity creates a moment that varies in magnitude and direction, depending primarily on the stern configuration and vessel draft. Since this moment, in deep water when underway, acts opposite to the weight moment, the slope of the shaftline will decrease relative to the static conditions, and may even reverse. It is important to ensure that the shaft journal remains in contact with the aft end of the bearing, because if the effective support point moves forward when underway, then the longer overhang of the propulsor can cause dynamic pounding. Thus, if the bore is to be sloped, the slope should usually not be the full angle predicted by the alignment analysis, and a bias in favor of the aft end support is recommended.

h. **Ship speed and power**. The range of shaft speeds and powers affect the running position of the shaft within the bearings. By knowing the range of operating positions within each bearing, the designer can place each bearing in the proper location. The shaft speeds and powers also determine the range of thrust loads and the off-thrust loads, which can have a significant effect on the running position of the aftermost bearing journal. In some applications the off-center thrust can affect the load on the propulsor bearing. The propulsor designer should be able to provide information on the off-center thrust. The shaft speed and power affect the shaft response to various forms of vibration.

i. **Dynamic conditions.** The shafting system alignment changes during ship operations because of ship flexing and reduction-gear torque roll. In most cases, the shaft alignment is measured in the static condition and dynamic effects are ignored because it is much more difficult to analyze and measure them. However, the US Navy has evaluated the dynamic alignment when the reduction gear is supported on resilient mounts for structure borne noise attenuation. Analytic studies and dynamic alignment measurements performed on ships with resiliently mounted reduction gears indicate that dynamic torque roll can be significant and should be included in the alignment analysis. The reduction gear manufacturer should provide an analysis of dynamic torque roll, which can be used to offset the reduction gear in the cold static condition to compensate for this dynamic motion. This compensation will most likely be primarily in the horizontal direction.

j. **Deflection of the main thrust bearing and foundation.** In addition to experiencing axial deflections, the thrust bearings may rise or be compressed as the applied thrust load increases, depending upon the their foundation design. This effect should be small or negligible for a properly designed foundation.

k. **Effect of pressure and temperature at depth for submersible vessels.** The combined effect of increasing pressure on the hull and decreasing water temperature at depth is a significant factor affecting the shaft alignment of submersible vessels. Both the high water pressure and the cold water cause the pressure hull of a submersible vessel to contract. As the depth increases, the bearings mounted on top of foundations fixed to the lower portion of the pressure hull will rise relative to bearings supported either uniformly from above and below or by unaffected non-pressure hull structure. Therefore, the deflection of the affected inboard bearing foundations caused by pressure and temperature at the maximum design depth must be calculated.

5.2 FACTORS THAT AFFECT ALIGNMENT MEASUREMENTS

The following factors should be considered when making alignment measurements, although they are not normally included in the shaft alignment process.

a. **Hull temperature.** The temperature of the hull can affect its shape. The water temperature for a waterborne ship or the air temperature for a drydocked ship will influence the hull temperature. The significance of the hull temperature on shaft alignment depends in part on the type of ship; for some designs, it may be negligible.

The effect of uneven heating of a ship's hull structure caused by solar radiation, whether the ship is in drydock or waterborne, is more important. Tests and experience have demonstrated that thermal growth caused by solar heating can deflect the hull enough to significantly affect the shaft alignment. Daylight heating tends to expand the topside of the hull relative to the bottom. These hull deflections affect the relative positions of the shaft bearings and the shafting alignment. Therefore, the shaft alignment should be measured several hours after sunset and before sunrise, or with the ship inside a covered facility.

The shafting system should be designed so that the solar radiation and hull temperature do not affect the shaft alignment when the vessel is waterborne. If necessary, the alignment should be checked under the two extreme temperature conditions: a) under a hot sun, and b) in the late evening or early morning. Changes to the design alignment may be required if the alignment is unsatisfactory in some climates.

b. **Structural work.** Structural work, such as major welding and equipment change-outs near the shaft bearing foundations, can affect the shaft alignment. Thus, all structural work near bearing foundations should be completed before the final alignment measurements.

c. **Transient ship operating conditions and sea conditions.** Although transient conditions are not always included in the shafting system alignment and stress analyses, the designers should consider how they affect the shaft alignment. Transient conditions are usually extreme situations that exist for short intervals but may still accumulate a significant number of load cycles. Transient ship operating conditions include loads from hydrodynamic control surfaces and high sea states. In general, rough seas do not affect the alignment because they cause random transient events. The magnitude of such transient effects can be quite large, depending upon the particular ship's operating profile.

d. **Journal alignment issues.** In the past, the shaft alignment was normally only concerned with maintaining the correct bearing loads within the range of static journal positions. While the weardown of water-lubricated stave bearings was included in this range, the hydrodynamic or off-center thrust influence on the journal position was seldom included. This latter influence causes the journals to shift within the confines of the bearing much more than previously suspected, and in some cases the journal positions are considerably different than accounted for in a static alignment. This issue is particularly important for ice-class ships where there are ice impact forces on the propeller.

   Traditionally, the terms shaft alignment and bearing alignment have been almost interchangeable because shaft alignment is almost always considered from the viewpoint of positioning the bearings. However, shaft alignment should not be understood solely as bearing alignment, rather shaft alignment should be viewed as journal alignment. Only in the static condition does the bearing position determine the shaft position. When both static and dynamic influences are included, the shafting must be aligned using the journals and not just the bearings.

SECTION 6.0 DEVELOPING THE DESIGN ALIGNMENT

6.1 THE DESIGN ALIGNMENT PROCESS

Because the position of the bearings is not constant, the design alignment must meet the design criteria discussed in Section 4 for all anticipated ship operating conditions. The design alignment must consider the condition of the ship during the initial alignment, and how the alignment will change over time because of various ship operating conditions and long-term effects such as bearing wear. Consider the following two examples, which address the effects of hull deflections and bearing weardown.

- Under a certain ship ballast condition, the hull deflection causes an outboard intermediate bearing to rise relative to the adjacent bearings. As the bearing rises, the load on it increases, and the bending moment in the shaft at the bearing also increases. At the same time, the loads on adjacent bearings decrease. Initially lowering the intermediate outboard bearing by one-half of the predicted motion can compensate for this effect, although the impact on the propulsor bearing must be considered before this is attempted.
- The weardown of a water-lubricated propeller bearing of 0.250 inches (6.5 mm) is allowed before its staves or liners must be replaced. In this case, initially raising the bearing by one-half of the allowable weardown will minimize the allowable variation from the design alignment.

Propulsion shaft alignment has advanced from considering the individual sections or components as separate pieces of equipment, which are simply aligned to each other on a straight-line, to considering the shafting system as a continuous beam. The continuous beam analysis allows designers to better understand the interaction of the complete shafting system, how the movement of one bearing support affects the others, and how minimum stresses are achieved. The purpose of the design alignment analyses is to determine the acceptable bearing reactions, shaft-bending moments, shaft deflections, and slopes for all known alignment conditions. The following general outline summarizes a process for developing an acceptable design alignment. Simpler approaches may be used for simpler systems. These items noted below are discussed in more detail in the subsequent paragraphs.

- Create a mathematical model of the shafting system and perform a static analysis.
- Identify all the factors affecting the shaft alignment and determine the effect each one has on the shaft offsets at the bearings.
- Identify the ship operating conditions for which the design criteria must be met.
- Assume an initial alignment of the bearings for a known baseline condition. For example, assume that all bearings are in a straight-line when the ship is waterborne at dockside.
- Calculate and tabulate the shaft offsets at the bearings for each ship operating condition.
- Calculate and tabulate the bearing reactions and shaft bending moments, deflections, slopes, and bending stresses for each ship operating condition for the assumed initial alignment. Compare the maximum and minimum bearing reactions, and the maximum shaft bending stresses for the assumed initial alignment with the design criteria.
- If the design criteria are not met, calculate a revised alignment for one or more bearings to meet the criteria. Even if the criteria are satisfied, it may be possible to calculate an optimal alignment that will minimize the life-cycle costs by reducing the bearing loads, shaft stresses, or number of bearings.
- After the design alignment is determined, calculate the builder's settings such that the shipbuilder can achieve a proper alignment of the bearings.

6.2 CREATING A MATHEMATICAL MODEL

The mathematical model should provide a reasonable representation of the shafting system's geometry, stiffness, weight, boundary conditions, and applied loads. The results of the static analysis will only be as good as the mathematical model. The following documents are needed to create the mathematical model for the shafting alignment calculation that will be analyzed with either an alignment computer program based on continuous beam theory or a general finite element analysis program:

(1) Shafting arrangement drawing
(2) Shafting detail drawing
(3) Propeller drawing, including the propeller centerline, weight, and center of gravity
(4) Reduction gear drawing (if applicable), with bull gear weight and dimensions
(5) Diesel engine manufacturers' project guide (if applicable)
(6) Bearing design data for the line shaft and waterborne bearings
(7) Shaft seal detail drawing

The mathematical model includes the shafting outside and inside diameters, the shaft sleeves, the concentrated weights (such as the propeller and bull gear), the bearings, and the modulus of elasticity and density of the shaft material. The shaft bore plugs, raised lands in way of the bearings and sleeves, and non-metallic shaft coverings are usually ignored. Tapered bores are modeled by assuming an abrupt change in the bore diameter at the midpoint of the taper.

The bearings are typically modeled as rigid knife-edge supports, although in some cases a bearing may be modeled as a distributed elastic support for a more accurate computation of the shaft bending moment at the bearing. The bearing support point is placed at the bearing centerline, except for the bearing just forward of the propulsor. The relatively long length of this bearing makes it more difficult to accurately define the net reaction point. The tail shaft has a significant slope at this bearing because of the propulsor weight and so the reaction point tends to be in the aft part of the bearing. For waterborne bearings with L/D ratios of about 4, the bearing reaction point is assumed to be one shaft diameter forward of the aft bearing face [5]. The US Navy specifies that the propulsor bearing support point be located forward from the aft end of the bearing the larger of one-fourth of the bearing length or one shaft outer diameter [7]. For oil-lubricated bearings, with an L/D ranging from 1 to 2, the bearing reaction point is assumed to be one-half shaft diameter forward of the aft bearing face [5].

The weights of propellers, collars, and other components attached to the shafting that have little effect on the overall system flexibility are modeled as concentrated loads. The effect of buoyancy on the propulsor and waterborne shafting is included in the analysis. The model should include a station at each change in shaft geometry, bearing support points, flange faces, concentrated weights, hydraulic jack locations (for subsequent bearing load measurements), shaft seals, and other locations as appropriate. Additional stations may be required between long lengths of shafting to determine the bending moments along the shaft.

Static analyses of the mathematical model are required to determine the bearing reactions and shaft bending moments, stresses, deflections, and slopes caused by the system's own weight, applied loads, and deflections. Analyses for both the buoyant and airborne (non-buoyant) conditions are usually performed. The US Navy requires the analysis of the straight-line, non-buoyant (in air) condition [7].

6.3 INFLUENCE COEFFICIENTS

Since the bearings' relative position effects the bearing reactions and shaft stresses, it is useful to determine the influence that a unit deflection of the shaft at each bearing has on the reaction at each bearing and on the bending moments, deflections, and slopes throughout the shaftline. These calculated influence coefficients (or influence numbers) are solely a function of the shafting system's geometry and stiffness, and the effect of gravity is not relevant to this analysis. After the influence coefficients have been determined, the bearing and shaft reactions can be calculated for any ship operating condition or proposed alignment. A similar approach can be used to determine the influence of a unit load applied to the shaft. For example, since the system is linear, the influence of a 1000 lbf (4.5 kN) load applied at the propeller can be used to determine the effect of a heavier propeller or a hydrodynamic load applied to the propeller during a turn.

The bearing reaction influence coefficient matrix is an N x N symmetric matrix, where N is the number of bearings and each element in the matrix is an influence coefficient of the form I_{ij}. Mathematically, the influence coefficient I_{ij} is defined as $I_{ij} = \Delta R_i/\Delta_j$, where ΔR_i is the change in the bearing reaction on the i^{th} bearing for a unit displacement Δ_j of the j^{th} bearing. Tables 1a and 1b are examples of bearing reaction influence coefficient matrices. For this example, raising line shaft bearing no. 1 by 1 mil (0.025 mm) increases the load on itself by 30.89 lbf (137 N), decreases the load on the aft bull gear bearing by 112.35 lbf (500 N), increases the load on the forward bull gear bearing by 92.58 lbf (412 N), and so forth. Since the influence coefficient matrix is symmetric, the affect of a given movement of a bearing A upon a bearing B is the same as the affect of the same movement of bearing B on bearing A. For example, in Tables 1a and 1b, raising line shaft bearing no. 1 by 1 mil (0.025 mm) decreases the load on line shaft bearing no. 2 by 18.50 lbf (82 N), and raising line shaft bearing no. 2 by 1 mil (0.025 mm) decreases the load on line shaft bearing no. 2 by 18.50 lbf (82 N).

Table 1a. <u>Example of an Influence Coefficient Matrix</u> (English Units)

	Bearing Reaction Influence Coefficients, lbf per mil (lbf/mil) of bearing displacement						
Bearing	Forward bull gear	Aft bull gear	Line shaft no. 1	Line shaft no. 2	Line shaft no. 3	Stern tube bearing	Propeller bearing
Forward bull gear	535.34	-612.53	92.58	-21.98	8.01	-1.64	0.22
Aft bull gear	-612.53	704.03	-112.35	29.78	-10.84	2.22	-0.30
Line shaft no. 1	92.58	-112.35	30.89	-18.50	8.97	-1.84	0.25
Line shaft no. 2	-21.98	29.78	-18.50	27.54	-23.88	8.15	-1.10
Line shaft no. 3	8.01	-10.84	8.97	-23.88	33.04	-20.78	5.50
Stern tube bearing	-1.64	2.22	-1.84	8.15	-20.78	21.59	-7.71
Propeller bearing	0.22	-0.30	0.25	-1.10	5.50	-7.71	3.15

Table 1b. <u>Example of an Influence Coefficient Matrix</u> (Metric Units)

	Bearing Reaction Influence Coefficients, Newtons per millimeter (N/mm) of bearing displacement (*multiply by* $1 \cdot 10^{-3}$)						
Bearing	Forward bull gear	Aft bull gear	Line shaft no. 1	Line shaft no. 2	Line shaft no. 3	Stern tube bearing	Propeller bearing
Forward bull gear	93.75	-107.27	16.21	-3.85	1.40	-0.29	0.039
Aft bull gear	-107.27	123.29	-19.68	5.22	-1.89	0.39	-0.053
Line shaft no. 1	16.21	-19.68	5.41	-3.24	1.57	-0.32	0.044
Line shaft no. 2	-3.85	5.22	-3.24	4.82	-4.18	1.43	-0.19
Line shaft no. 3	1.40	-1.89	1.57	-4.18	5.79	-3.64	0.96
Stern tube bearing	-0.29	0.39	-0.32	1.43	-3.64	3.78	-1.35
Propeller bearing	0.039	-0.053	0.044	-0.19	0.96	-1.35	0.55

Every bearing in a shafting system is affected when the offset of one bearing is altered, although the effect becomes smaller as the distance from the altered bearing increases. Observe that I_{ii} is always positive since lifting a bearing always increases the load on it. Also, note that the sum of the elements of each row or column of the influence coefficient matrix equals zero. In general, the signs of the adjacent terms in any row or column should alternate, e.g., from positive to negative to positive, although there are certain shaft arrangements for which this may not occur.

Influence coefficient matrices can also be determined for the bending moments. In this case, the coefficients represent the change in moment (e.g., N-m) for a 1 mil (0.025 mm) change in the bearing position. The bending moment influence coefficient matrices are not symmetrical.

As a minimum, static alignment analyses should be performed to determine:

- The influence coefficient matrix for a unit deflection of the shaft at each bearing
- The bearing reactions and shaft bending moments, deflections, and slopes for the following conditions:
    a. straight-line, dry-docked
    b. cold aligned, dry-docked
    c. straight-line, waterborne
    d. cold aligned, waterborne
    e. hot aligned, waterborne
    f. cold aligned with collective weardown, waterborne
    g. hot aligned with collective weardown, waterborne

Once the static analyses are complete and the results have been tabulated, the following checks are recommended to confirm the validity of the results. If the static analyses results fail any of these tests, then the mathematical model input file should be checked for errors.

- Perform an independent calculation of the total weight and longitudinal center of gravity of the shafting system, both in air and waterborne, using the geometry, material properties, and applied loads specified in the mathematical model. The calculated system weight should equal the sum of the bearing reactions determined by the static alignment analysis. The sum of the moments of the bearing reactions about any point should equal zero.
- Confirm that the bearing reaction influence coefficients are in an N x N symmetric matrix. The sum the elements of each row or column of the influence coefficient matrix should equal zero. For most systems, the signs of the adjacent terms in any row or column should alternate.
- Plot the bending moment and deflection diagrams. The resultant curves should be continuous and make sense for the system configuration.

6.4 ASSUMED INITIAL ALIGNMENT

The bearings supporting the shafting move relative to one another as the condition of the ship and its environment changes. Section 5 discusses the common factors that affect shaft alignment. All factors that may significantly affect the shaft alignment should be identified, although the list will differ for different ship designs. The effect of each factor on the shaft offsets at the bearings should be quantified and tabulated.

The calculation of the alignment settings is somewhat a trial and error process. Therefore, an initial guess at the design alignment is needed and any reasonable assumption of bearing offsets for a particular ship condition will generally be satisfactory. Regardless of the assumption, the bearing offsets must be assumed for a specific ship condition and relative to a defined reference axis. A convenient first assumption might be that all bearings are in a straight-line either when the ship is waterborne at dockside or on the building ways.

The relative positions of the shaft bearings for a particular ship operating condition can be determined by evaluating all the factors that affect the alignment and then superimposing their effects on the initial alignment. The changes in shaft offsets at the bearings caused by various factors can be summed and added to the initial bearing offsets to determine the net shaft offsets for the specified condition. This is best illustrated by an example. Consider a ship operating in calm seas, with its machinery at normal operating temperature, its hull ballasted to a specific trim, with one outboard bearing worn to very near its limit, and the others worn in proportion to their unit load. The bearings supporting the shafting system were installed on the building ways to a specified design alignment and tolerance with respect to one another. The shaft alignment was verified to be within the specified tolerance. For this steady state condition, the shaft offset at each bearing is the sum of its design alignment offset on the building ways plus the following effects:

- the relative change in the hull deflection in way of the bearings that occurred in going from the building ways to the specified ballast condition
- the thermal growth of the bearing foundation that occurred going from the cold to normal operating temperature
- the weardown on the bearings, assuming that they were initially new
- the as-built deviation in the bearing offset from the design alignment allowed by the specified installation tolerance.

Note that the offsets discussed here may be relative to any straight-line reference axis, such as an optical line of sight. The axis used to define the offsets should be one that is convenient to the designer or shipbuilder. It can be the construction baseline for the ship. It can be any shaftline that can be referenced to the ship's structure at two points, such as machined lands in way of two different bearings. Rigid-body translations and rotations will have no effect on bearing and shaft reactions.

Note that there is a distinction between bearing offsets and shaft offsets at bearings. Hull deflections and thermal growth change the bearing offsets, which in turn affect the shaft offsets. Bearing weardown affects the shaft offsets, but has no effect on the offsets of the bearings themselves. The shaft offsets at the bearings, and not just the bearing offsets, affect the shaft alignment.

6.5 CALCULATING BEARING AND SHAFT REACTIONS

The bearing and shaft reactions for any ship operating condition can be calculated from the influence equations. These equations are formulated from the results of the static analyses (i.e. the deadweight bearing reactions, shaft bending moments, and the bearing influence coefficients) and the shaft offsets at the bearings determined for the different ship operating conditions. The influence equations can be used to calculate the bearing reactions or the shaft moments at any station for any set of bearing offsets. For example, for a system with N bearings, the reaction on bearing 3 (R_3) is given by an equation of the form

$$R_3 = R_{I,3} + \delta_1 \cdot I_{\delta 1,R3} + \delta_2 \cdot I_{\delta 2,R3} + \delta_3 \cdot I_{\delta 3,R3} + \ldots + \delta_N \cdot I_{\delta N,R3} \qquad (6\text{-}1)$$

or more concisely

$$R_3 = R_{I,3} + \Sigma\delta_j \cdot I_{\delta j,R3} \quad (j = 1 \ldots N). \qquad (6\text{-}2)$$

Likewise the moment in the shaft at station 24 is calculated using the equation

$$M_{24} = M_{I,24} + \Sigma\delta_j \cdot I_{\delta j,M24} \quad (j = 1 \ldots N) \qquad (6\text{-}3)$$

Using a general notation, the reaction at any bearing or can be determined using an influence equation of the form

$$R_i = R_{I,i} + \Sigma\delta_j \cdot I_{\delta j,Ri} \quad (j = 1 \ldots N) \qquad (6\text{-}4)$$

where i is the bearing or shaft station at which the reaction R_i is calculated, j is the bearing number (j = 1 ... N), and

$R_{I,i}$ = the straight-line or initial reaction at station i (from the static analysis results)
δ_j = the shaft offset at bearing j
$I_{\delta j,Ri}$ = the influence of a unit offset of the shaft at bearing j on the reaction at station i (from the static alignment analysis).

The number of influence equations can become quite large for a system with several bearings and shaft stations. However, the matrix formulation of the influence equations is readily adaptable to a spreadsheet format, which makes practicable an iterative approach for determining an “optimum” bearing alignment.

6.6 DETERMINING THE DESIGN ALIGNMENT

After the analysis is completed, identify the maximum and minimum bearing reactions and the maximum bending moments for the assumed alignment, and determine if any of the design criteria are violated. An acceptable alignment satisfies all the criteria, although this is not necessarily an optimal alignment. There is no one best alignment; theoretically there are an infinite number of alignment configurations that would produce the same bearing load and component stress values. Calculating the final design alignment settings can require trial and error, particularly if the system has several bearings. The ability to achieve a satisfactory alignment with a variety of configurations is the primary reason that shafting systems work on ships.

The influence coefficients are useful for determining how much a bearing must be offset to correct a problem. For example, consider a case where the load on a stern tube bearing exceeds its allowable limit by 5000 lbf (22 kN) under the worst-case ship operating condition. Repositioning the bearing lower with respect to the other bearings would reduce its load. Assume that the static analysis shows that the influence coefficient of the stern tube bearing on itself is 50 lbf/mil ($9 \cdot 10^{-3}$ N/mm). Therefore, setting this bearing 0.100 inches (2.5 mm) lower will reduce its load by 5000 lbf (22 kN), thus eliminating the overload condition on this bearing. The same result could have been achieved by raising one or both adjacent bearings. Unfortunately, the solution is rarely so simple since repositioning one or more bearings may solve one problem and create another. Realigning a bearing changes the load on that bearing and every other bearing in the system, (as seen from the influence coefficient matrix), and it also changes the bending moments, stresses, deflections, and slopes at every station along the shaft.

6.7 CALCULATING THE BUILDER'S SETTINGS

Once an appropriate design alignment is determined, it must be expressed in terms that the shipbuilder or maintenance facility can use so that the ship will have an acceptable alignment once it is waterborne. The engineer must consider how the ship will be supported when the bearings are installed and aligned, and how this differs from the baseline condition assumed in the alignment analysis. This includes an understanding of the expected design bearing clearances and the possible compression of the waterborne bearing staves. The engineer must also consider the method that will be used to align the bearings; e.g., optical alignment, gap and sag, etc.

6.8 INFORMATION THAT SHOULD BE DETERMINED FROM AN ALIGNMENT ANALYSIS

An alignment analysis report should be prepared for immediate use and for future alignment checks. An alignment drawing should be prepared with all the data necessary to generate the alignment. As a minimum, the alignment analysis report should include the following:

- A simplified sketch of the shafting system, showing the location of the bearings and the other data necessary to compute the influence coefficients.
- The influence coefficients.
- The bearing reactions for the conditions listed in Section 6.3.
- The bending moment and shear diagrams (at a minimum, the maximum values and their location). See Figure 2 for examples.
- The weardown limits for the waterborne bearings (if applicable) and the effect of bearing weardown on the bearing reactions, with all bearings at the maximum allowable wear and preferably the individual bearings at maximum allowable wear.
- The effect of thermal growth on the bearing reactions.
- The jack correction factors.
- The required offset and slope (cant) of bearings in the aligned condition and the instructions for boring the slope of the bearings, if required.
- The acceptable tolerances for bearing position.
- The flexibility factor, which should not be less than 0.010 inch (0.25 mm).
- The gaps and sags at the inboard shaft flange couplings.

An alignment procedure is typically prepared for a new construction project. This procedure should include data entry tables for each step in the process, beginning with the offsets at the ends and midpoint of the bore actually obtained for the aft bearings, the subsequent sags and gaps achieved for the uncoupled shafting, and finally, the reactions measured at each accessible bearing after all the couplings have been connected and the propeller installed, and with the vessel both in drydock and afloat.

6.9 VIBRATION CONSIDERATIONS

The propeller is a source of potentially serious vibratory excitation because of the nonuniform wake field in which it operates. Since the shafting system is inherently flexible it is extremely vulnerable to these vibratory excitations. Unless the overall design is already proven in service, it is necessary to perform axial, lateral, and torsional vibration analyses of the shafting system to ensure that the vibratory shaft stresses are below the allowable limits. This is best achieved by designing the shafting system so its natural frequencies do not coincide with the excitation frequencies. As with all dynamic systems, one type of vibration can influence the other types. For this reason, various types of coupled-vibration studies can also be performed, with a coupled axial-torsional analysis commonly performed for motor ships.

The excitation frequencies of concern include the shaft rotational frequency, the propeller blade frequency, and the engine firing frequency. While any of these frequencies can theoretically be changed, they are usually fixed by this point in the design, and the only practical option is to change the natural frequency of the shafting. This can be done by adjusting the shaft diameters, the bearing location and loading, and possibly the system damping. The best method for changing the shafting system natural frequencies depends on the individual ship and the type of vibration. For example, adding a torsional damper may be considered before any other change when there is a problem with torsional vibration on motor ships. If the system cannot be adjusted to avoid a resonance within the normal operating range, then a barred speed range may be mandated by the regulatory body.

SECTION 7.0 METHODS FOR MEASURING THE ALIGNMENT OF MARINE PROPULSION SHAFTING SYSTEMS

7.1 INTRODUCTION

There are several proven methods for assessing the alignment of main propulsion shafting. The most common procedures include the hydraulic jack, strain gage, gap and sag, and optical methods. The hydraulic jack method is a common technique used to measure the reactions of the line shaft bearings. The strain gage method is a more analytical technique developed to measure the inboard or outboard shaft bearing reactions using strain gages mounted at predetermined locations along the shaft. The gap and sag method is still used to determine the initial alignment settings, but this method generally does not provide the accuracy necessary for most systems. Optical alignment (or boresighting) is primarily used to analyze and adjust the position of the waterborne and/or inboard bearings in drydock, and to determine the initial position of the propulsion machinery. It is typically used prior to the initial installation of the propulsion shafting since it can only be used when some part of the shafting is missing. The following paragraphs discuss these methods in more detail.

7.2 HYDRAULIC JACK METHOD

The hydraulic jack method is the most widely used procedure for measuring the alignment of main propulsion shafting. This method can measure the reactions of the line shaft and bull gear bearings, detect bent shafts, and calculate the shaft runout. The hydraulic jack method is more straightforward and requires less training than other methods. It has sufficient accuracy, although a series of measurements must be performed on most inboard bearings to obtain a comprehensive view of the system alignment. Although the hydraulic jack method has shortcomings, it is valuable for measuring bearing reactions and shaft runout, and is often used to validate the results of the strain gage method.

The hydraulic jack method requires a hydraulic jack (or ram), a dial indicator with graduations of 0.001 inches (0.025 mm) or less, and a load cell with an electronic output display accurate to within 100 lbf (450 N). A calibrated jack can be used in place of a load cell, but this introduces additional errors of 10% to 20% because the jacks are sensitive to structural resistance and friction within the cylinder. Since a hydraulic jack measures pressure and not the force on the piston, the friction within the cylinder gives an inaccurate measurement of the piston load. A load cell installed between the shaft and hydraulic jack is more accurate than a calibrated hydraulic jack since the load cell measures the force applied to the shaft. Load cells use very accurate strain gages (0.25% of the full load range), have very low hysteresis, and their calibration can be easily checked with a universal test machine.

As shown in Figure 3, the hydraulic jack and load cell are positioned under the shaft, either forward or aft of the bearing, but as close as possible to the bearing. The jack must be supported by a jacking pad or a very stiff foundation member. Each line shaft bearing must have its own jack location. A dial indicator for measuring the shaft's vertical movement is installed on the top of the shaft directly above the jack. The dial indicator must be anchored to a structure that is not affected by the jacking process. Anchoring it to the bearing housing or foundation may produce errors. Prior to starting the measurements, the test operator should rotate the shaft several times on the turning gear. The operator should also lift the shaft at least once to: a) reduce hysteresis in the shaft, b) ensure that the shaft can be raised above the bottom of the bearing shell, c) ensure that no other bearings rise at the same time, and d) ensure there is sufficient clearance between the shaft and the upper bearing shell. The position of the shaft in the bearing clearances should also be checked to ensure that the shaft is in the bottom of the bearing and is centered athwartship.

For a vertical reaction measurement, the test operator lifts the shaft in approximately 0.001 inch (0.025 mm) increments, and plots the load and shaft movement. The operator must be attentive to prevent the shaft from lifting off (detaching) from adjacent bearings. The operator also records the load and displacement as the shaft is lowered in approximately 0.001 inch (0.025 mm) increments. The loading and unloading curves usually do not follow exactly the same line because of internal friction in the shafting and hydraulic cylinder, which causes nonlinear behavior and hysteresis in the system. The bearing reaction is determined by plotting the shaft displacement (mil, mm) versus the shaft load (lbf, N). Figure 4 shows typical plots of the jacking data in English units. In these figures, the shaft displacement is on the x-axis and the load is on the y-axis, although this can be reversed.

Each curve in Figure 4 can be characterized by two regions. The first region is a high stiffness region, where the shaft is supported by both the bearing and the hydraulic jack. The second region occurs as the shaft is raised and the load is transferred from the bearing to the hydraulic jack, which causes a sharp change in the slope of the curve as it starts to follow the line of lower stiffness. Any sharp change in the slope of the lifting load curve after the low stiffness region is reached indicates that an adjacent bearing(s) is being detached or that the shaft has contacted the upper half of the bearing shell.

The jacking load, which corresponds to the reaction at imaginary bearing located at the hydraulic jack, can be defined by a "mean load line" positioned between the lifting and lowering lines. The US Navy technical manual for bearing reaction measurement [9] only requires that lifting line data be used; however, it is considered good practice to use the average of lifting and lowering data. If there is a large difference between the lifting and lowering load lines, then this is an indication that something is amiss and it should be investigated.

The reaction at the hydraulic jack is determined by the intersection of the mean load line with the line of zero shaft displacement, i.e., the y-axis. The load at the bearing is then determined by multiplying the hydraulic jack reaction by the jack correction factor (C), defined as the ratio between the bearing reaction and the reaction at the jack location. The jack correction factor is determined from the computerized alignment analysis by including the jack in the analysis, and is calculated as I_{ii}/I_{jacki}, where I_{jacki} is the influence coefficient of the jack on the i^{th} bearing. The slope of the mean load line is approximately equal to the influence coefficient of the bearing on itself (I_{ii}) multiplied by C.

The bearing reactions so determined are usually within 10% of the actual value, however, the accuracy of the influence coefficients is appreciably less. Consequently, the calculated influence numbers should be used to adjust the bearing offsets and not those determined from these measurements [5]. This process is repeated, so that it is performed at four locations 90° apart around the shaft's circumference for each bearing as shown in Figure 4. The average of the load lines for the 0°and 180° positions and the 90° and 270° positions should be nearly congruent. The bearing reaction is the average of the four corrected measurements. When it has been shown that the shaft is not bent, (see the discussion below), then the reactions at that bearing are measured at only two angular positions, 0°and 180° or 90° and 270°.

Since the bearing reactions are usually measured with the cold machinery plant, it is necessary to convert the measured reactions to the hot condition by considering the rise of the bull gear bearings caused by the thermal growth of their foundations and housings and the influence coefficients of these bearings, as discussed in Section 6.5.

The hydraulic jack method can also detect bent shafts and determine the runout of a shaft by comparing the bearing reactions at shaft positions 90° apart. A bent shaft should be suspected if the bearing reaction changes significantly with the shaft circumferential position, although loose flange-coupling bolts can cause the same symptom. The shaft runout at a bearing can be calculated from the hydraulic jack measurements at that bearing according to the following relation.

$$\mathrm{TIR} = \boxed{\frac{\sqrt{\left(R_0 - R_{180}\right)^2 + \left(R_{90} - R_{270}\right)^2}}{I_{ii}}} \qquad (7\text{-}1)$$

where TIR = total indicated runout, mils (mm)
C = jack correction factor
R_0 = measured bearing reaction at 0° position, lbf (N)
R_{90} = measured bearing reaction at 90° position, lbf (N)
R_{180} = measured bearing reaction at 180° position, lbf (N)
R_{270} = measured bearing reaction at 270° position, lbf (N)
I_{ii} = jack influence coefficient of bearing at the bearing, lbf/mil (N/mm)

The hydraulic jack method has been primarily used to measure the vertical reactions of inboard bearings. However, there are special circumstances where it may be used to measure the horizontal reactions. Although not nearly as practical or accurate, this method may be useful when athwartship misalignment is purposely established in the bearing system. This condition occurs if significant torque roll and/or thermal growth are expected during the running state. Using this method in the horizontal plane is very rare and the measurements can be quite cumbersome since special fixtures and procedures are typically required.

The hydraulic jack method has the following disadvantages.

(1) Many jacking measurements are required to obtain reasonable system alignment data and account for possible bends in the shaft. The entire process is time consuming and labor intensive when there are several bearings in the shaft system, and even more so when there are physical interferences around the jack locations.

(2) The bearing caps or casings on bull gear and motor bearings must often be disassembled for access to the main shaft. Also, a temporary stub shaft may need to be installed to provide a lift point for the forward reduction gear bearing.

(3) It is difficult to measure the bearing reactions on bull gears since their influence coefficients are large and it is easy to overlook the correct slope and unload the second bull gear bearing. (Therefore, the strain gage method is often used to measure the bull gear bearing reactions.)

(4) It is difficult to measure the horizontal bearing loads for evaluating athwartship alignment.

(5) Because of the physics of this technique, the hydraulic jack method does not accurately measure the bearing reactions on resiliently mounted reduction gears.

(6) The hydraulic jack method is not the optimal technique to use with the longer and more flexible rubber stave bearing; although some outboard bearings may be measured using this method.

(7) The following potential problems can cause inaccurate measurements and errors.
- The jack and load cell are not located directly under the shaft center and the dial indicator is not located directly over the shaft.
- The dial indicator support moves because of motion of the surrounding structure.
- There is hysteresis in the shafting system.
- There are shifting load centers in adjacent bearings.

7.3 STRAIN GAGE METHOD

The strain gage method is the only available technique for measuring the alignment of the entire connected propulsion shaftline with the vessel afloat. It provides a more complete assessment of the shaft's alignment condition than the hydraulic jack method and is generally more accurate. The strain gage method is primarily employed when the hydraulic jack method cannot be used to measure the bearing loads. The strain gage method has the following benefits:

- The reactions can be measured on bearings that are not accessible for jacking.
- Repetitive measurements are quick and easy after the strain gages are installed.
- The measurements can be made during ship operations.
- The strain gage method is adaptable for ships with resiliently mounted reduction gears, such as US Navy surface combatants, since there is no vertical shaft movement required during measurement.
- The reactions of bearings in both the vertical and horizontal directions can be calculated simultaneously. The athwartships shaft alignment can be assessed from the horizontal bearing reactions.
- The reactions at the bull gear and thrust journal bearings can usually be measured without disassembling these components.

The strain gage method provides shaft alignment information through the fundamental theory of flexural beam analysis. These principles can be used to derive mathematical equations for calculating the bending moments along the shaft from the measured strains, which can then be used to calculate the bearing reactions for the shafting system. There are two proven strain gage techniques for calculating bearing reactions, the free-body method [10, 11] and the moment-theorem method [12].

A typical shaft with N bearings is statically indeterminate by N-2 degrees, since there are only two independent static equilibrium equations useful for determining the vertical bearing reactions. The two static equilibrium equations are: (1) the sum of forces in the vertical direction is zero and (2) the sum of moments about any point is zero. Therefore, measuring the bending moments (or bearing reactions) at N-2 or more points provides the additional data needed to determine the bearing reactions along the shaft.

The free-body method judiciously divides the shaft into separate "artificial" free bodies at the strain gage locations, such as shown in Figure 5. A minimum of N-2 strain gage locations is required along the shaft. As shown in the figure, where the shaft section is accessible between bearings, a set of strain gages is located between the bearings, and where the shaft system is not accessible between a pair of bearings, then two sets of strain gages are located between the next pair of bearings. The moments determined from the strain gages can be used to calculate the bearing reactions (and the unknown shear forces) from the above two equations of static equilibrium as shown in the figure for several of the artificial free bodies. Observe that the shaft segment identified as the "free section" is statically determinate since it has no bearing, and therefore, this is the starting point for solving the equilibrium equations. The free-body method cannot be used when there are three bearings with inaccessible shafting, such as with a stern tube with a pair of bearings followed by a strut bearing. In this case, the moment-theorem method must be used.

The bending moments are calculated from the measured strains by $M = IE\varepsilon/c$, which is derived from $\sigma = E\varepsilon$ and $\sigma = Mc/I$ for the single axis stress condition on the shaft surface, where;

- M = bending moment in the shaft caused by the loads on the shaft and bearing offsets (if any)
- I = area moment of inertia of shaft
- E = modulus of elasticity
- ε = measured strain (peak)
- c = shaft outer radius
- σ = bending stress

The strain gages at each axial position are located in the same plane normal to the shaft centerline. The gages are installed 180° apart around the shaft circumference for measuring the axial strains, or 90° apart for determining if the shaft is straight. The gages should be located in accordance with the following guidelines.

- The strain gages should be located where the bending moment is very sensitive to changes in the bearing load, so that a small change in the bearing load will cause a significant change in the measured strain. Large absolute values of the moment influence coefficient obtained from the shafting bending moment diagram (usually determined from the computerized alignment analysis) identify suitable locations.
- The strain gages should be located at least one shaft diameter from shaft discontinuities, such as flanges, tapers, diameter or bore transitions, and concentrated weights to ensure that local stress concentrations do not affect the data.
- The strain gages should not be installed too close to a bearing since the exact location of the bearing reaction point is not known, which creates an error in the bending moment arm from the strain gage location to the bearing reaction point. Mounting a strain gage further from the bearing reduces the relative magnitude of this error.

The strain gages are typically installed by welding or bonding them to the shaft. Four gages can be wired into a full Wheatstone bridge or two gages can be wired into a half bridge at each longitudinal shaft location as shown in Figure 6. The two-gage configuration is usually adequate to minimize the temperature effects and the shaft longitudinal and torsional loads, although the four-gage configuration increases the sensitivity of the system to flexural strains. The bending tensile and compressive strains are recorded at 90° increments by rotating the shaft on a slow speed jacking gear. The shaft should be rotated at least one or two revolutions before any strain readings are recorded. The measured strains should stabilize such that the calculated bearing reactions do not differ significantly between several successive measurements. The strain gage signals may be transmitted via slip rings and brushes or by telemetry for measurements during ship operations.

Although the strain gage method has significant advantages, its complexity can be a drawback. For example, it is sometimes difficult for the novice to recognize improper strain gage readings, and on-site calibration and determination of the results are required to help eliminate this problem. It is recommended that the results for a sample of the bearings be determined before leaving the ship so that any errors can be determined and corrected. The strain gage results can also be compared with bearing reactions measured by the hydraulic jack method. The results should agree within 15% when the measurements have been taken under the same conditions. Errors in the strain gage method are caused by improper location of the strain gages, insufficient gage bonding, improper wiring techniques, damaged or defective strain gages, poor surface condition underneath the gage, or sign-convention errors. Any suspect gages should be replaced.

Another disadvantage of the strain gage method is that it cannot easily be used to measure the shaft runout, as can be done with the hydraulic jack method. An indication of the shaft runout can be determined from the strain gage measurements as follows, where a strain runout value of less than four microstrain (strain expressed as parts per million) is usually acceptable:

$$\text{Strain runout} = (\text{strain}_{90^\circ} + \text{strain}_{270^\circ}) - (\text{strain}_{0^\circ} + \text{strain}_{180^\circ}) \qquad (7\text{-}2)$$

The strain gage report should include the following: the strain gage locations and bridge configuration, strain readings, the calculated bearing reactions, the calculated strain-to-load conversion factors, the computer model input for the analysis of the strain gage results, the table of influence coefficients, the evaluation of results, and any realignment recommendations.

7.4 GAP AND SAG METHOD

The gap and sag method is a traditional technique primarily used to determine the preliminary alignment of inboard bearings and to establish the alignment of the machinery relative to the stern tube bearings during new construction prior to the final shaftline assembly. Figure 7 illustrates the gap and sag. The gap is the difference in opening between the top and bottom edges of the unconnected flange pair. The sag is the vertical distance between the centers of the unconnected flanges. The gap and sag are normally measured with two dial indicators making simultaneous readings on two mutually perpendicular surfaces of the coupling flanges as the shaft is rotated through 360°.

This method, once referred to as "drop and gap," is based on the theory that the alignment can be estimated by the relative position of the mating pairs of shaft flanges. The theoretical distance between the face and periphery of the unconnected coupling flanges determined from the alignment software can be compared with the actual measurements. Since the free end position of a shaft flange depends on the elevation and athwartship position of the adjacent bearings, this method can be used to establish and adjust the bearings' alignment. The alignment software can be used to determine a family of sag and gap combinations for a cold condition that will give acceptable bearing reactions in the hot condition.

The alignment must be checked for at least every 90° of shaft rotation through 360°. The tolerance on the flange measurements should be 0.002 – 0.004 inches (0.05 – 0.10 mm), depending on the size of the components and the length of the bearing spans, and the overall accuracy required for the particular shaftline.

This gap and sag method is adequate for establishing the initial shaftline position, especially for lengthy systems, but it has inherent limitations if it is not used in conjunction with a direct bearing reaction measurement method. As is the case for all shaft alignment methods, these measurements should be conducted at night when thermal distortion of the hull is a minimum. The advantages of the gap and sag method are that: (a) it uses simple measuring equipment, such as dial indicators and feeler gages, and (b) it is useful for evaluating both the horizontal and vertical alignment. The disadvantages of this method are that: (a) the load on the bearings and the deformation of the bearing supports may be different when the shafting system is assembled, (b) temporary supports may be required for the shafting, and (c) the bearing reactions are not measured directly, but are obtained from a plot of bearing reactions for various combinations of gap and sag.

The following requirements must be satisfied before starting the gap and sag measurements:

- all major hull work must be completed (i.e., the superstructure is in place, all major welding is completed)
- the prime movers and reduction gears are installed
- the propeller is at least 50% submerged for waterborne measurements
- the propeller shaft at the forward stern tube bearing is resting on the bearing's bottom shell.
- for direct-connected engines proper crankshaft and bedplate alignment has been confirmed.

The sag and gap method is highly sensitive to reading errors that increase with the number of aligned shafts. The alignment is checked at the flange surface, causing a reading error proportional to the ratio of the shaft length to flange diameter. Moreover, the method is sensitive to the manufacturer's practices, and the alignment error may increase if the alignment does not follow the basic alignment requirements listed above. However, there are situations where bearing reaction measurements cannot be made and in these cases the sag and gap method is used. These situations generally involve smaller craft where there may not be any line shaft bearings.

7.5 OPTICAL AND LASER METHODS

The optical method, also known as boresighting, is mostly used to check and correct the alignment of waterborne bearings before installing the main propulsion shafting. Although optical measurements do not provide direct bearing reactions, they are important for establishing the builder's bearing offsets and they dictate how the outboard bearing struts are bored prior to the bearing installation. The optical method can also be used to determine the preliminary location of inboard bearings when the shafting is not in place. The method is typically used to determine two separate alignment parameters: the location of the bearing supports relative to a datum (i.e., a line representing the straight-line alignment through all the bearing centers) and the localized alignment of the bearing commonly referred to as cant and skew. The optical method uses the line-of-slight relationship of the bearing bores to the shaft axis of rotation to establish an optical reference line and to determine the location of the shaft bearings relative to that line.

The location of each bearing can be verified using the optical method when the vessel is in drydock and the shafting is removed. The optical scope or laser may be set up outside the aft stern tube bearing looking forward, or mounted to a fixture on the driver output coupling looking aft. The offsets of all bearing centers are measured relative to an optical line-of-sight and compared to the output from the alignment computer model.

Optical methods can use a micro-alignment telescope (a borescope with a cross-hair lens) or a more elaborate computerized laser sensor and targets, depending on the shaft configuration, length, and alignment scheme. The laser method is based on the same technique as the optical scope method, except that the laser beam replaces the visual line-of-sight. The main advantage of laser alignment is that it can measure large distances with minimal loss of accuracy.

The scope or laser is attached to the flange of a reference shaft (bull gear, line shaft, direct-coupled engine drive flange, or propulsion motor) so that the line-of-sight or laser beam is perpendicular to the flange face. This should be checked with a target at the after end of the aftermost bearing. The flange with the scope or laser attached is then rotated through 360°. The line-of-sight or laser will trace a small circle on the target, and the mounting is corrected until the circle is reduced to a point. Once the correct mounting has been established, targets should be installed at the forward and aft ends of each bearing. Each bearing is set in place and adjusted until the light-of-sight or laser beam hits the target. As with all alignment measurements, the optical work must be performed at night when the ship's hull is least affected by temperature variations.

The optical line should be repeatable with good accuracy, and must be most accurate at the inboard end. It must be an extension of the shafting continuous beam to permit the mathematical analysis of the alignment data. The optical line must be a true projection of the center of rotation of the freely suspended after end of the existing portion of the propulsion system, i.e. reduction gear flange, direct-connected engine or propulsion motor drive flange, or line shaft. The deflection and slope must be calculated from the end of the shaft from which the optical line is projected. These theoretical data are used to compensate for the differences between the projected optical line and the calculated straight-line.

7.6 WIRE METHOD

The bearing alignment can be checked by anchoring a wire above the shaft at one end and tensioning it with a weight over a pulley at the other end of the bearing line to be examined. The location of the bearing in relation to the wire can then be measured. In most cases, the gear output flange or direct-connected engine drive flange is a good support for the wire.

The wire method is based on the same concepts as the optical method, although there are some drawbacks because the wire sags and vibrates, and must not be moved during the measurements. During installation of a bearing, a target can be installed in the bearing with the wire running through it. The hole in the target is sized to the allowable tolerance in the alignment procedure, which reduces the need to continually measure the bearing location relative to the wire [13]. The sag in the wire must be accounted for according to the following equation:

$$S = w \cdot x^2/(2 \cdot T) \qquad (7\text{-}3)$$

where: S = wire sag
w = weight of wire per unit length
x = one-half the distance between wire supports
T = wire tension, i.e., the weight of the suspended mass

The accuracy of the wire method can be within 0.004 inch (0.1 mm) over short distances, but the accuracy decreases over longer distances.

SECTION 8.0 ATHWARTSHIP ALIGNMENT OF PROPULSION SHAFTING SYSTEMS

8.1 INTRODUCTION

Ideally, propulsion shafting is aligned with the centers of all bearings in a straight-line in the horizontal plane so that horizontal loading in the hot operating condition is minimized. On some shafting and reduction gear designs, this may require a horizontal bearing offset in the cold, static condition. In some cases the torque roll of the engine requires that a system utilize a horizontal misalignment when cold in order to achieve a good running condition. Athwartships misalignment occurs when one or more bearings are positioned off of the determined ideal line, causing excessive bearing loads in the horizontal plane. The bearing reaction forces in the athwartship horizontal plane are typically smaller than in the vertical plane. Nevertheless, the horizontal reaction components of each bearing must be analyzed to achieve the overall alignment objective, and the position of each bearing within the horizontal plane must be considered to obtain a comprehensive alignment overview of the propulsion system.

In concept, the bearings are positioned horizontally so that the side clearances are equal. Because of these bearing clearances, the horizontal bearing loads are generally much less sensitive to athwartships misalignment than vertical misalignment, and a fair amount of athwartships misalignment may exist without causing excessive horizontal loads. The athwartship force component is added vectorially to the vertical force component to determine the resultant magnitude and direction of the bearing reaction. Athwartships bearing loads of up to 20% of the vertical loads are generally not a problem in line shaft or tail shaft bearings. For example, the total load on a bearing with a 30,000 lbf (133 kN) vertical load and a 6000 lbf (27 kN) athwartship load (20% of the vertical load), is 30,600 lbf (136 kN), which is an insignificant increase over the vertical load. However, horizontal misalignment can have a significant effect on the bull gear alignment, especially with respect to the allowable bull gear load differential, and on direct-connected engine alignment.

8.2 MEASUREMENT OF ATHWARTSHIPS ALIGNMENT

The following paragraphs summarize information pertinent to measuring the athwartship alignment. Section 7 provides more details on alignment measurement methods.

a. Strain Gage Method: The strain gage method is the most efficient and direct way of measuring the horizontal bearing loads.

b. Boresighting: Optical or laser alignment equipment can be used to determine the athwartships disposition of the gear, line shaft, or stern bearings. This method requires drydocking the vessel, and removing the shafting up to the gear or direct-connected engine to achieve a line of sight. Athwartships disposition of the gear shaft or engine relative to the stern tube bearings can be achieved by using a mirror target on the gear shaft or engine drive flange, mounting the alignment scope on the gear shaft or engine drive flange, or in the case of hollow gear shafts, by mounting boresight targets inside the gear shaft. An estimate of the bearing athwartships loads can be calculated from the boresight data and the bearing influence coefficients, but this is usually not very accurate because of the bearing clearances.

c. Clearance measurements: If the actual horizontal bearing reactions cannot be measured, then manual clearance measurements with feeler gages can be used to assess the transverse position of the bearings relative to the shaft. This method does not provide the athwartship loads on the bearings, and so it only gives an estimate of the athwartship alignment. Because of the characteristics of the bearings, it is difficult to make a direct correlation of the side clearances to transverse forces without direct measurements. Manual clearance measurements provide good data points for comparison with the reaction results obtained by direct methods such as the strain gage method.

   Usually this procedure requires that the line shaft bearing caps be removed, the gear bearings opened up, the end seal housings of journal bearings disassembled, and the fairings on waterborne bearings be removed. The side clearances reveal the localized alignment of the shaft through the bearing. More specifically, the angle of the shaft through the bearing, referred to as skew, is determined by comparing the clearances on opposite sides for each end of the bearing. If there are notable clearances on equivalent sides of each bearing end, then the transverse offset may need to be adjusted.

   For direct-connected engines proper crankshaft alignment should first be confirmed by taking crankshaft deflection readings, then, with the line shaft uncoupled from the engine flange, the crankshaft should be rotated with the turning gear to ensure that it is seated at the bottoms of the bearings. The line shaft is properly aligned athwartships when the flanges are centered with equal gaps on both sides.

d. Inspections: Inspecting the bearing wear contact surfaces may provide a good perspective of the overall horizontal alignment. Skewed contact patterns within a fluid film journal bearing may be traced to deficiencies in the bearing surface, but athwartship misalignment must also be considered. A skewed contact pattern within the waterborne stave bearings is more likely to be directly associated with transverse misalignment.

8.3 ATHWARTSHIPS ADJUSTMENT OF LINE SHAFT BEARINGS

The line shaft bearings are the first place to start when considering athwartships re-alignment, although the horizontal alignment adjustments on line shaft bearings are typically more involved than the vertical adjustments. The estimate of the horizontal adjustment is based on the same influence coefficient matrix used in the vertical plane, provided the shafting foundation is the same for the vertical and horizontal planes (i.e. there are no spring or resilient mounts in the vertical direction) since the influence coefficients are based on the shaft stiffness. However, the horizontal adjustments do not necessarily react as expected because of side clearances within the bearing that are not included in the influence coefficient matrix. After the alignment adjustments are complete, the bearing slope alignment should be checked and the bearing loads measured as described in Section 7.

In most cases, the horizontal alignment of line shaft bearings is acceptable if the bearing shell can be rolled in from either side of the bearing with the line shaft supported on a jack [13]. To correct athwartship misalignment it is usually necessary to release the bearing hold-down bolts and shift the bearing laterally. The bearing might then have to be re-chocked. The line shaft bearing housings are positioned to the foundation with fitted bolts. The transverse adjustments can be made by elongating the foundation holes and installing new fitted hardware, but this method is limited if the foundation footing is weakened in the process. As is the case with vertical alignment adjustments, the effects on adjacent bearings and seals must be considered.

8.4 ATHWARTSHIPS ADJUSTMENT OF STERN TUBE BEARINGS

If more adjustment is needed than possible using line shaft bearings, then athwartships adjustment of the stern tube bearings should be considered. This requires drydocking the vessel, and either boring the stern tube bores offset from their existing locations, or turning the outer diameter of new stern bearings eccentric relative to their bores. This method of athwartships adjustment is a practical solution if the stern bearings will also be adjusted vertically. Any vertical or athwartships changes in the stern tube bearing position are usually monitored using optical or laser alignment measurements. Although the US Navy does not permit the following method, a third alternative is to hold the stern tube bearing loosely in the stern tube bore using jack screws, align the stern tube bearing using optics, and then fix the stern tube bearing in place using poured epoxy. When using boresight data to determine the athwartships alignment, a fair curve alignment using a spline is helpful for determining the most expedient moves. In this way, it is even possible, with a sufficiently flexible shaft line, to leave the shafting with a slight bow in the horizontal plane, provided that a fair curve results. Whenever stern tube bearing adjustments are made, both ends of each bearing must be changed proportionately to achieve a good alignment of the bearing relative to the shaft.

8.5 ATHWARTSHIPS ALIGNMENT OF REDUCTION GEARS

The greatest concern for horizontal misalignment is the effect it has on the internal alignment of the bull gear shaft. The horizontal bearing misalignment must be corrected if it approaches or exceeds the gear manufacturer's limits or adversely affects the low speed tooth contact. Because of the increased risk of gear skew, large athwartships loads should not be allowed on gear bearings, particularly if they are opposing. These loads should be reduced or eliminated by athwartships alignment changes, or a gear skew analysis should be performed to determine if the athwartships misalignment will have a significant effect on gear life. For a main reduction gear bearing, the misalignment is related to the tooth load distribution at the gear to pinion mesh. The maximum increase in tooth load occurs at the extreme forward or aft end of each helix of the mesh.

Fluid film bearings are typically split along the horizontal plane and the shaft resultant reaction is designed to operate in the lower half of the bearing, so that minimal loading within the horizontal plane is desirable. Other cases exist where a journal bearing within a main reduction gear is split along the vertical plane. More athwartship loading is expected in this design, resulting from the combined forces of multiple mating gear elements. Other conditions exist where the reduction gear or drive motor bearings are purposely misaligned within the horizontal plane during the cold static condition to compensate for significant dynamic adjustments within the horizontal plane expected from the calculated torque roll and/or thermal growth. These dynamic effects should be verified with actual measurements, but this is rarely done because they can be cumbersome and expensive.

If there is an excessive athwartships load couple in the gear bearings, the first action should be to relieve this couple by an athwartships adjustment of the first one or two line shaft bearings. This may require applying, or increasing, the athwartships load on the line shaft bearings, but this is preferable to a gear load couple. As a last resort when there is a serious athwartships misalignment of the gear, the whole reduction gear housing is loosened from its foundation, and re-aligned in the horizontal plane.

SECTION 9.0 ADJUSTING THE SHAFT ALIGNMENT

9.1 NEW CONSTRUCTION VESSELS

Ideally, propulsion system alignment would occur after the vessel is complete and afloat. Unfortunately, physical and economic realities require that major alignment tasks be accomplished while the vessel is still on the building ways or blocks. For example, boring for the stern tube and strut bearings cannot be done with the vessel afloat.

Boring of the bearings should be done after all hot work in the lower part of the stern is complete and after all superstructure modules and other significant weights in the stern are placed in position. The boring reference line would be taken from the extended centerline of the rough-positioned reduction gear shaft or engine crankshaft. These bearing locations then become fixed references to which the other shaft, gear, and engine bearings must conform. Any attempt to finalize the alignment of these inboard components, prior to floating the vessel for the first time, involves uncertainty and risk.

The concern that these realities cause the alignment engineer depends to a large extent on the type of ship and the shafting arrangement. For example, an aircraft carrier shafting system is very long and supported by many inboard line shaft bearings. These multiple line shaft bearings provide the engineer with "adjustment tools" that can be used to compensate for unintended relative movements of the strut and stern tube bearings relative to the rest of the system. This adjustment is accomplished by modifying the vertical and athwartship offsets of the line shaft bearings to achieve an alternative fair shaftline that produces acceptable reactions at all bearings. However, the bearing load changes predicted by the influence coefficient equations are not usually achieved in practice (i.e., the actual changes are less than the calculated changes) especially when large adjustments are necessary [9]. Often when a bearing is shimmed upward, the foundation will deflect downward, and so the shaft is only lifted a fraction of the desired amount. Nonetheless, the influence coefficient equations allow the initial adjustments to be made so that the desired overall alignment is nearly achieved and only slight additional adjustments are required.

Many commercial vessels have propulsion machinery well aft, with short shaftlines. The engine and gear are relatively close to the stern tube with few line shaft bearings. In this case, the engineer has fewer adjustment tools available to counter the effects of relative movements in way of the propulsion system supports after the stern tube and strut are bored. As a result, there is a much higher risk when attempting to enhance the construction schedule by finalizing the locations of the inboard propulsion system components. Adding line shaft bearings in an attempt to increase the adjustment tools is counter-productive, since placing bearings too close to the gear or engine increases the sensitivity of their bearing reactions to misalignment and thermal movements.

The following example illustrates the constant need for the alignment engineers on new-construction projects to anticipate downstream effects of alignment changes, to advise the construction department of areas of risk, and to make every effort to avoid major construction impacts.

> During the new-construction of a large crude carrier, a decision was made to install the seawater circulating piping to the main condenser before boring of the stern tube was completed. The piping was large in diameter, about 60 inches or 1.5 m, of copper nickel. The main condenser was attached to the low-pressure turbine, which

was aligned to the reduction gear. The reduction gear had been set on key chocks based on the intended extended stern tube bore center. After the boring was completed it was found that the reduction gear was no longer adequately aligned to the boring center. The misalignment was in the vertical plane and was such that, if the system were bolted up, the reactions at both the single line shaft bearing and the aft bull gear bearing would be too high. Adjusting the single line shaft bearing could not correct the problem. If this bearing's vertical offset were increased, then its own reaction would become even greater, and if its offset were decreased, the aft bull gear bearing would be more overloaded. In order to solve the problem the gear was tilted up at the forward end, which shifted the load from the aft to the forward gear bearing, and the gear was also translated vertically upward. However, as the gear case was tilted, the turbines had to be moved in order to remain aligned. Since the gear bearings are closely spaced compared to the distance between those bearings and the condenser connection for the large seawater pipes, the ratio of the movement in way of the pipes compared to the change in offset between the two bull gear bearings was on the order of four to one. Consequently, the seawater circulating piping had to be removed and reworked.

9.2 THE ALIGNMENT PLAN

The alignment plan, which stays with the vessel, should be updated before delivery of the ship to indicate the actual alignment. Data included in the as-built summary should include:

(1) The vertical and horizontal offsets measured at the centerline of the line shaft bearings, the low speed gear or motor bearings, and at the ends and midpoint of the waterborne bearings.
(2) The measured reactions at all accessible bearings, on-dock in the cold condition and afloat in the cold and hot condition, if possible.
(3) The sags and gaps for the unbolted couplings, both on-dock and afloat.
(4) The shaft bending moment distribution for the actual alignment.

With the amended alignment plan, the engineer responsible for alignment related work after the ship goes into service will have the information necessary to assess whether changes have occurred, and to restore the shafting to an acceptable alignment after the propulsion system has been disturbed or damaged. The offsets and bearing reactions can be measured directly and, if they are close to the design alignment values, it is not necessary to re-calculate and reassess the bending moments. However, if it is necessary to depart from the design alignment, or if the alignment plan cannot be located, then a new mathematical model of the system will have to be created and new calculations made. When new calculations are required for an existing shafting system, the criteria of most interest are the bearing reactions, the shaft bending moments, and the load distribution in long bearings.

9.3 WHEN TO MEASURE THE SHAFT ALIGNMENT FOR IN-SERVICE VESSELS

After the final alignment check for a new vessel shows compliance with the design alignment, then subsequent alignment checks for an in-service vessel are only necessary in the following cases:

(1) after work on the bearings or after the shafting has been removed.
(2) if structural damage may have changed the vertical or horizontal position of a bearing.
(3) when hot work near a bearing may have changed the bearing offsets because of thermal distortion.

(4) when misalignment is suspected from events such as repeated bearing failures, abnormal gear tooth contact patterns, unexplained shafting system vibrations, or a ship grounding.
(5) when the bearings, journals, or staves show uneven wear or evidence of overloading or unloading.

For case (1), the required alignment checks depend on which bearing was repaired or moved. If a stern tube or strut bearing is to be replaced, and the bearing is mounted in a symmetric shell (i.e., the bearing shell had not been turned eccentrically to adjust the alignment), then the bearing can be replaced with a new one whose concentricity has been checked. If the replaced bearing was eccentrically turned, then the original eccentricity must be used for setting up the new bearing in a lathe. After turning to the correct outer diameter, the bearing ends must be marked to indicate the required installation orientation.

If a line shaft bearing is being replaced with an exact duplicate, then the new shell can be rolled into the housing and no further bearing reaction checks are required, although the bearing clearances should be measured. However, if a line shaft bearing housing has to be moved or unbolted for any reason, then its reaction and horizontal shoulder clearances must be checked after it is bolted down. In addition, the adjacent line shaft bearings should be checked after the desired reaction is achieved at the disturbed bearing.

For case (2), if there is enough damage to question the alignment, then the hot work performed during the repairs will cause more changes to the alignment. In these cases, the shafting system should be completely removed and the bearing offsets should be checked after the repairs are completed.

Significant structural work in the stern of a vessel can affect the propulsion system alignment. In the case of major work involving structure in way of the engine and shaftline, the shafting should be removed and alignment should be monitored during the work. In less extreme cases, such as local piece-work steel renewals in tanks, changes in the bearing reactions and offsets of potentially affected bearings should be checked while the work is underway. If the bearings at risk are line shaft bearings, then periodically checking their reactions and shoulder clearances can suffice. For repairs affecting a stern tube, indirect measurements to detect changes in the bearing offsets are recommended.

Bearing re-alignment will also affect the alignment of the shaft seals relative to the shaft, depending on the location of the bearings relative to the seal assembly and the amount the bearings are adjusted. Radical changes in shaft position may also require re-positioning the seal housings as well. The seal vendor should be consulted for additional information.

Note that it is not necessary to change the existing alignment to match the design alignment. Since a shafting system has many potential satisfactory alignments, an alignment is acceptable when it does not impose adverse operating conditions on any component, while allowing the ship to operate without problems until the next scheduled alignment check or availability. When the measured reactions are different from the design values, then the system must be re-evaluated to determine what changes are required to achieve a satisfactory alignment [9].

FIGURES

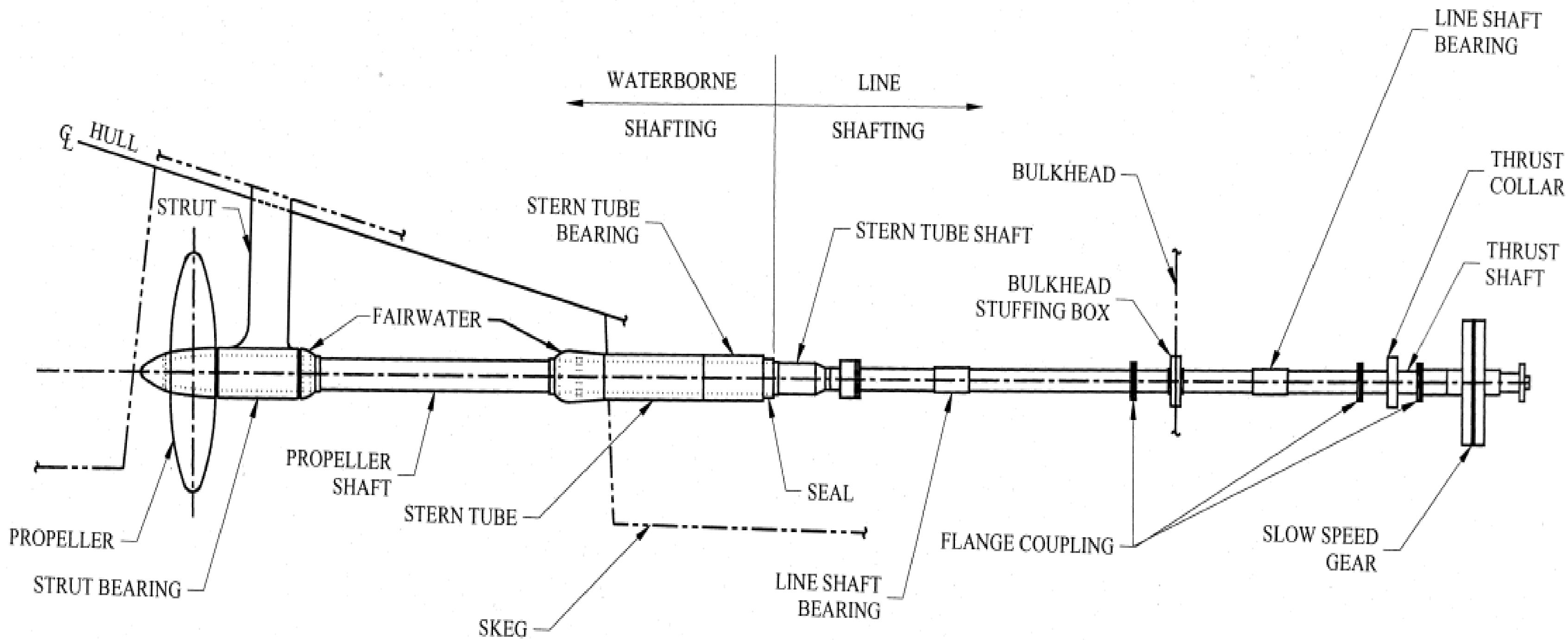


Figure 1a. Typical Shafting System.

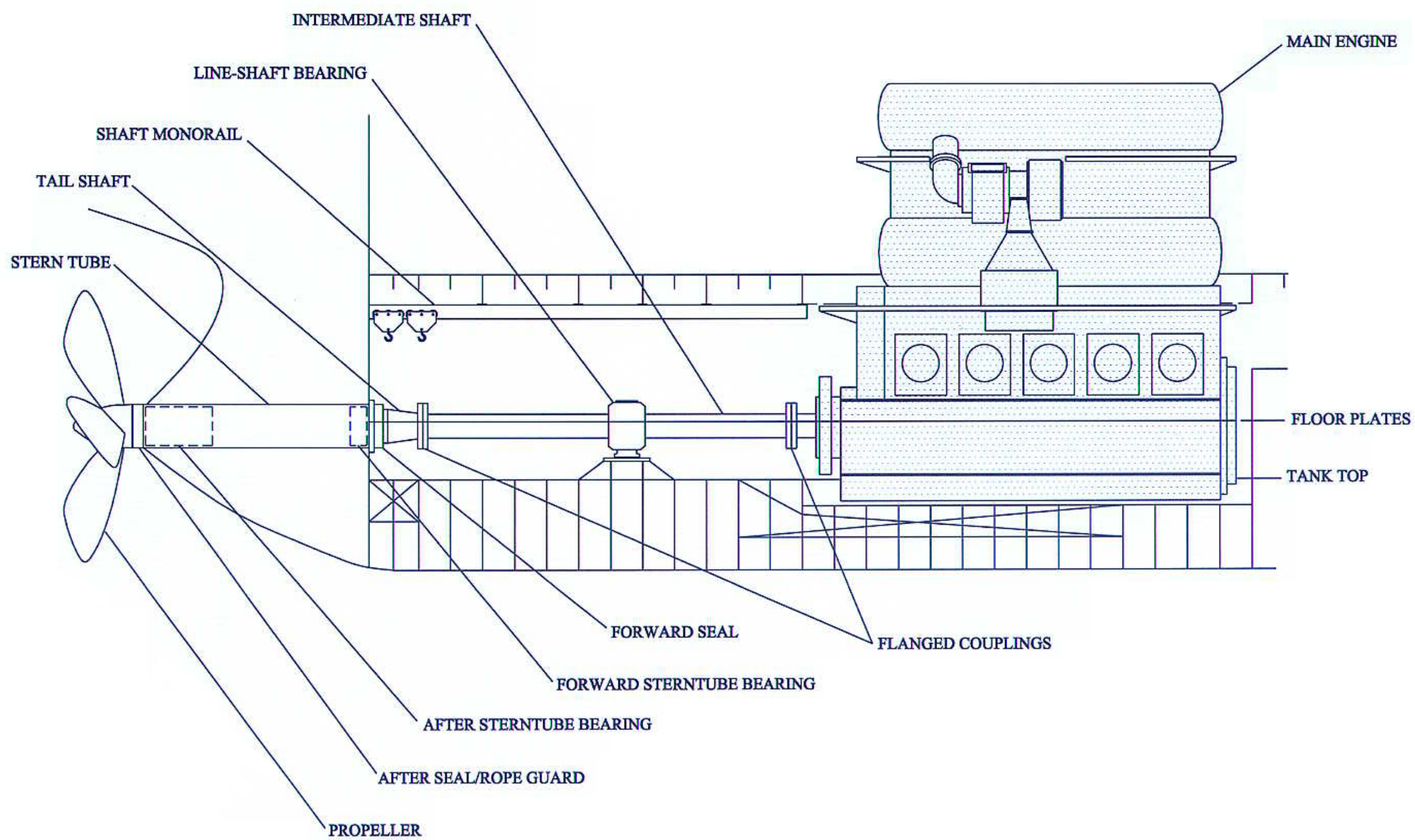


Figure 1b. Typical Shafting for a Ship with a Low-Speed Diesel Engine

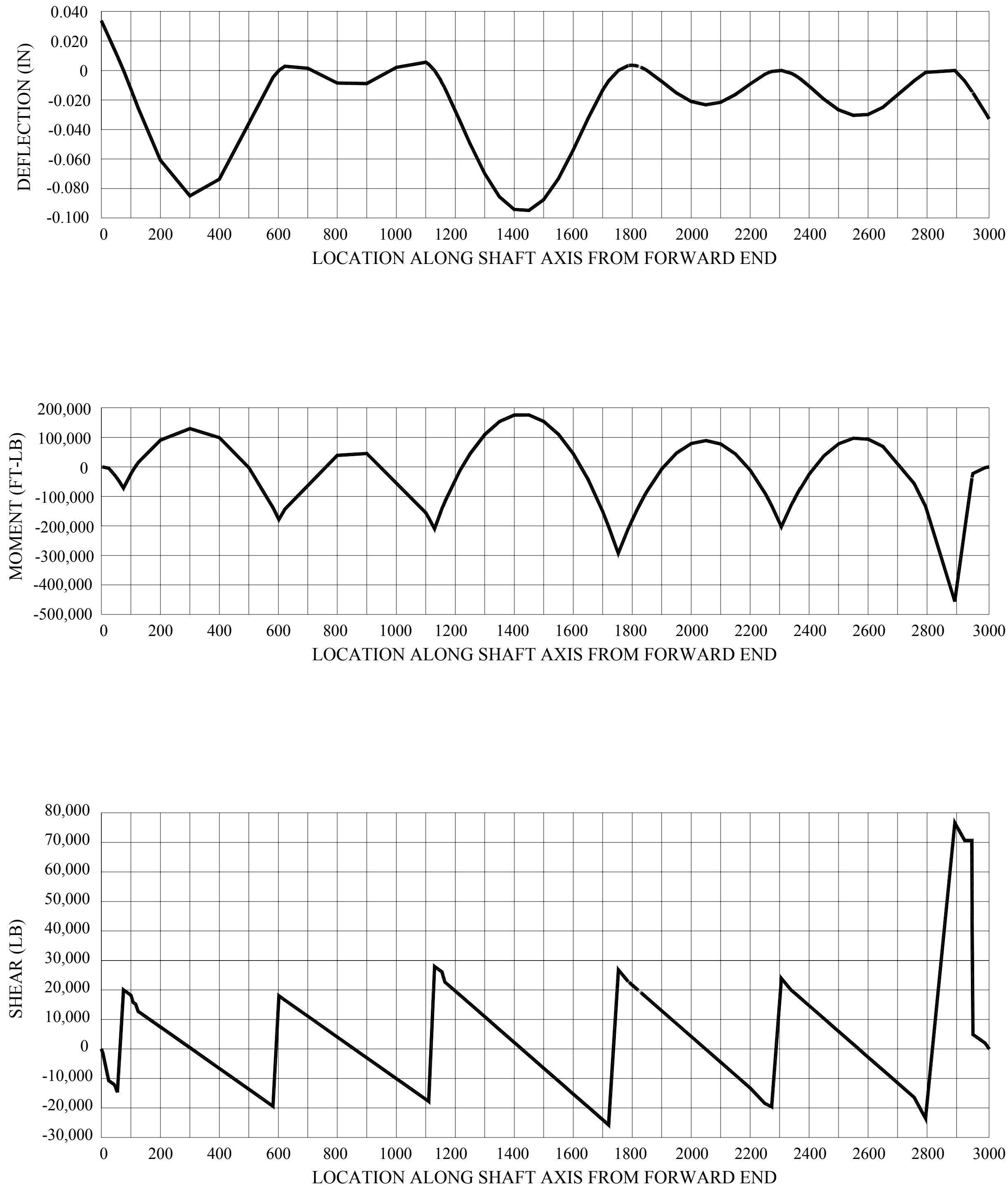


Figure 2. Example of Deflection, Moment, and Shear Diagrams.

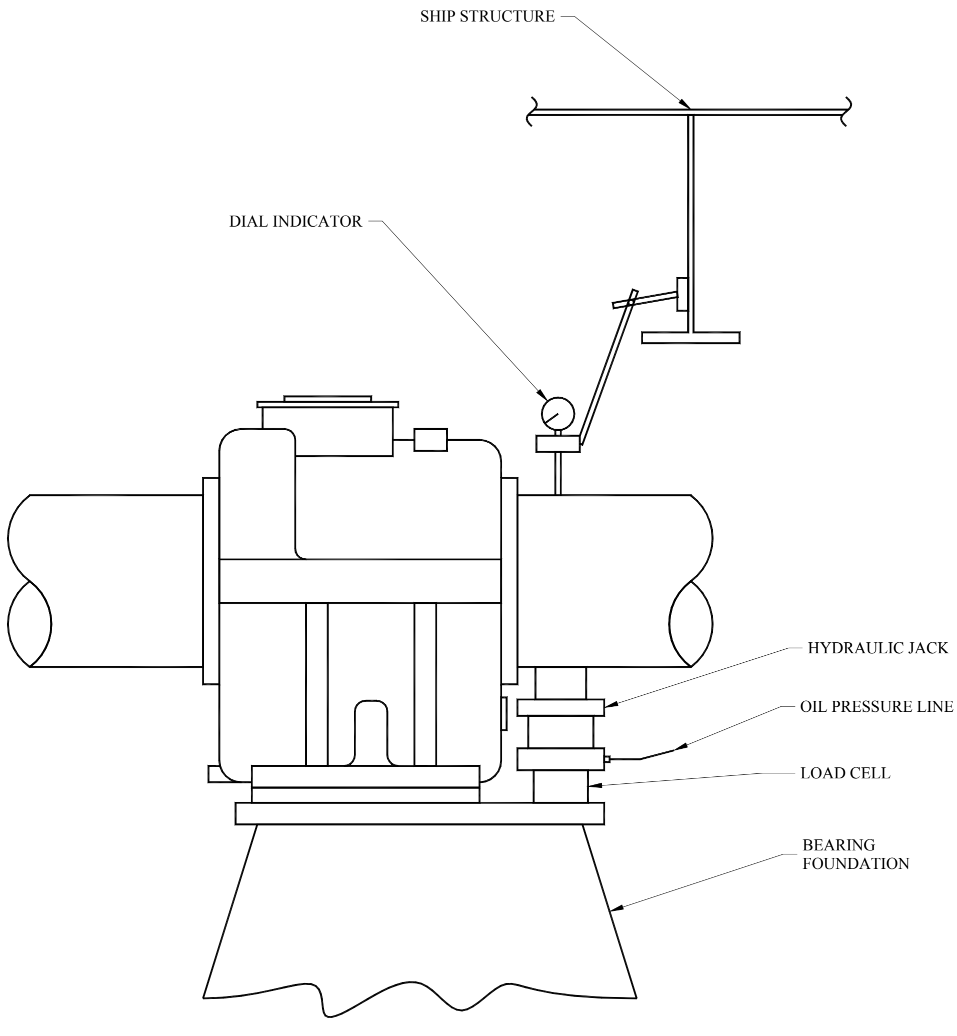


Figure 3. Hydraulic Jack Installation.

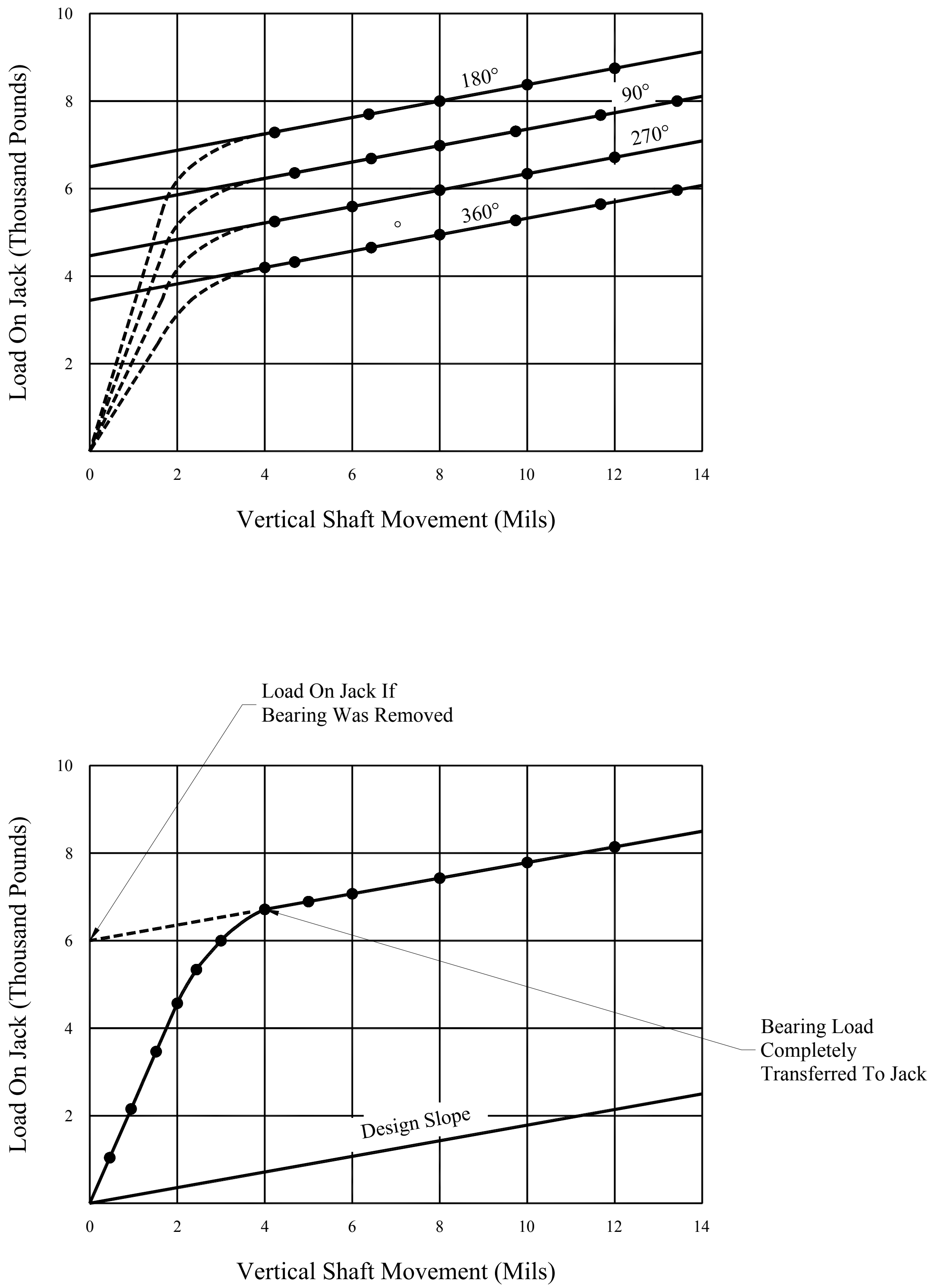
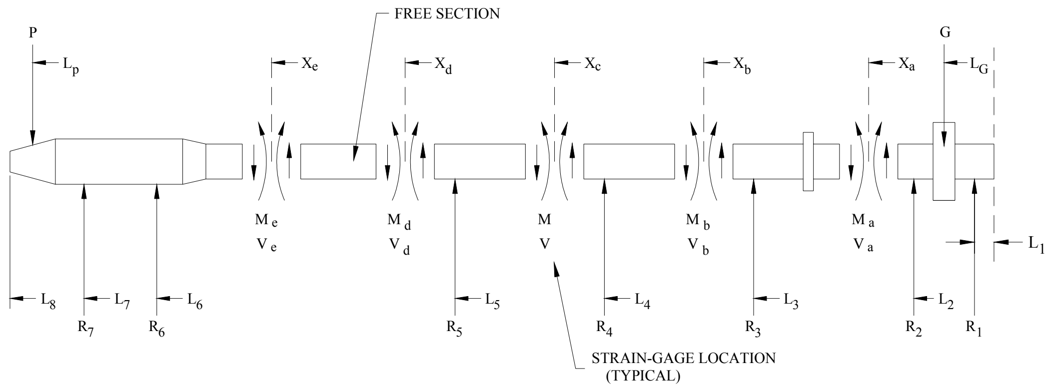


Figure 4. Example of Bearing Reaction Diagram – Plot of Load Versus Lift for Hydraulic Jack Method.



R_n = BEARING REACTION

L_n = DISTANCE FROM SHAFT FORWARD END

X_i = DISTANCE TO STRAIN-GAGE LOCATION

M_i = MOMENT IN SHAFT, ↑ POSITIVE

v_i = SHEAR IN SHAFT, ↑ POSITIVE

P = PROPELLER WEIGHT

G = GEAR WEIGHT

U(X) = SHAFT WEIGHT PER UNIT LENGTH

EXAMPLES OF EQUILIBRIUM EQUATIONS

REDUCTION GEAR SECTION

$$\sum M_{R1} = 0 = M_a + V_a(x_a - L_1) + R_2(L_2 - L_1) - G(L_G - L_1) - \int_{L_1}^{x_a} x\,u(x)\,dx + \int_{0}^{L_1} x\,u(x)\,dx$$

$$\sum F = 0 = V_a + R_2 - G - R_1 - \int_{0}^{X_a} u(x)\,dx$$

FREE SECTION

$$\sum M_d = 0 = M_e - M_d + V_e(x_e - x_d) - \int_{X_d}^{X_e} u(x)\,dx$$

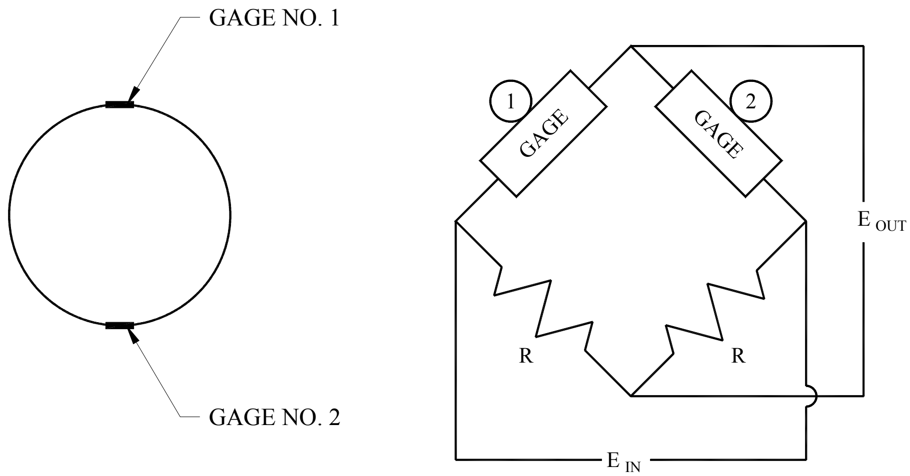
$$\sum F = 0 = V_e - V_d - \int_{X_d}^{X_e} u(x)\,dx$$

STERN TUBE SECTION

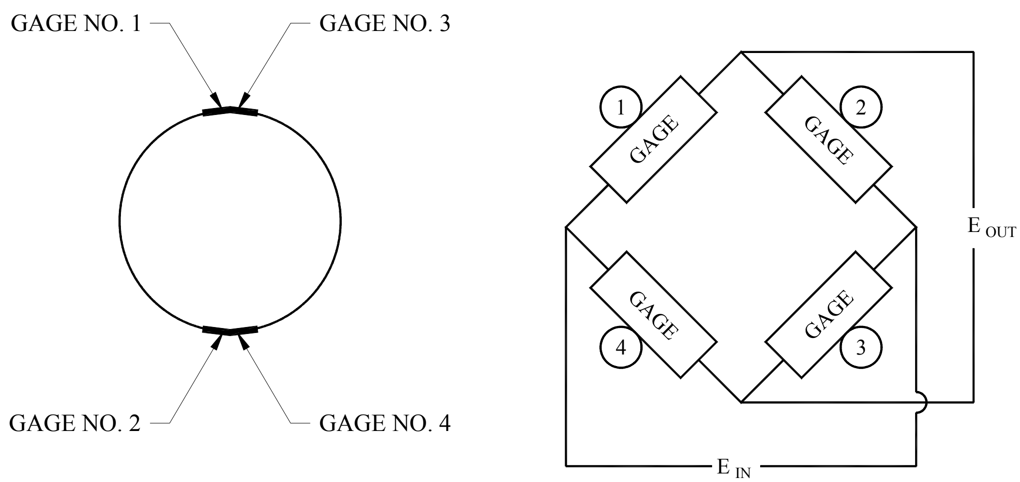
$$\sum M_6 = 0 = -P(L_p - L_6) + R_7(L_7 - L_6) + V_e(L_6 - x_e) - M_e + \int_{x_e}^{L_6} x\,u(x)\,dx - \int_{L_6}^{L_8} x\,u(x)\,dx$$

$$\sum F = 0 = -P + R_7 + R_6 - V_e - \int_{x_e}^{L_8} u(x)\,dx$$

Figure 5. Strain Gage Method Free Body Diagrams.

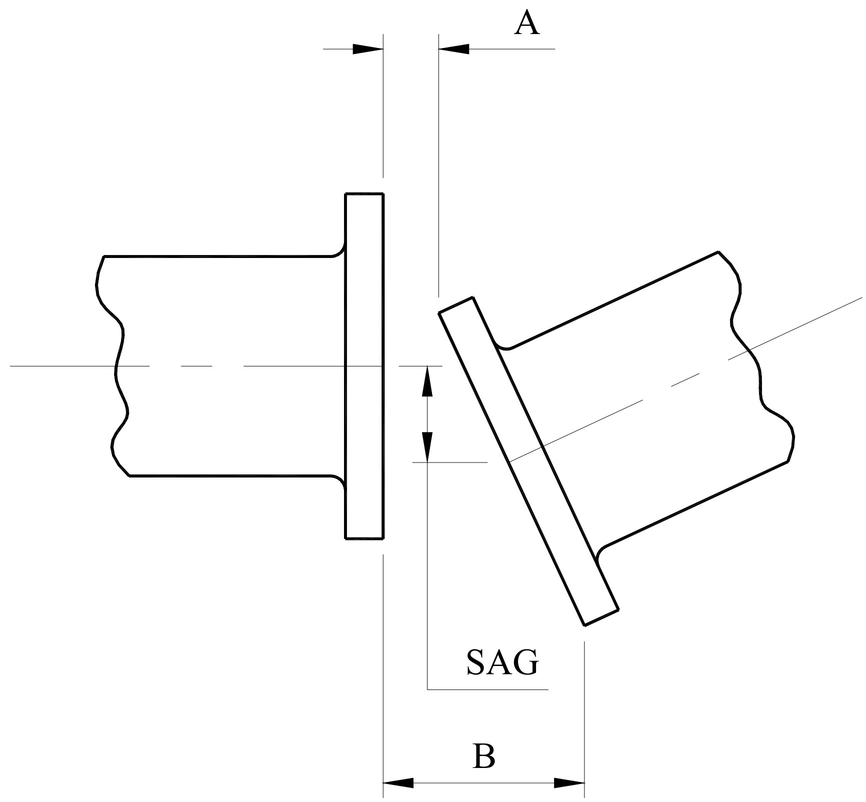


Two-Gage Configuration



Four-Gage Configuration

Figure 6. Two and Four Strain Gage Wheatstone Bridge Configurations.



$$GAP = B - A$$

Figure 7. Gap and Sag Measurement.

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