TECHNICAL REPORT

ISO/TR 13637

First edition
1997-12-15

Petroleum and natural gas industries — Mooring of mobile offshore drilling units (MODUS) — Design and analysis

Industries du pétrole et du gaz naturel — Amarrage d'unités mobiles de forage en mer — Conception et analyse

Reference number
ISO/TR 13637:1997(E)

Foreword

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The main task of technical committees is to prepare International Standards, but in exceptional circumstances a technical committee may propose the publication of a Technical Report of one of the following types:

- type 1, when the required support cannot be obtained for the publication of an International Standard, despite repeated efforts;

- type 2, when the subject is still under technical development or where for any other reason there is the future but not immediate possibility of an agreement on an International Standard;

- type 3, when a technical committee has collected data of a different kind from that which is normally published as an International Standard ("state of the art", for example).

Technical Reports of types 1 and 2 are subject to review within three years of publication, to decide whether they can be transformed into International Standards. Technical Reports of type 3 do not necessarily have to be reviewed until the data they provide are considered to be no longer valid or useful.

ISO/TR 13637, which is a Technical Report of type 2, was prepared by the American Petroleum Institute (API) (as API Recommended Practice 2SK, 2nd edition) and was adopted by Technical Committee ISO/TC 67, *Materials, equipment and offshore structures for petroleum and natural gas industries*, Subcommittee SC 7, *Offshore structures*.

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Printed in Switzerland

This document is being issued in the Technical Report (type 2) series of publications (according to subclause G.3.2.2 of part 1 of the ISO/IEC Directives, 1995) as a “prospective standard for provisional application” in the field of offshore structures for the petroleum and natural gas industries because there is an urgent need for guidance on how standards in this field should be used to meet an identified need.

This document is not to be regarded as an “International Standard”. It is proposed for provisional application so that information and experience of its use in practice may be gathered. Comments on the content of this document should be sent to the ISO Central Secretariat.

A review of this Technical Report (type 2) will be carried out not later than three years after its publication with the options of: extension for another three years; conversion into an International Standard; or withdrawal.

Introduction

For the purposes of providing interim guidance on mooring/stationkeeping design, ISO TC 67/SC 7 has adopted API RP 2SK in recognition that it constitutes one of the most complete documents on the subject. API RP 2SK contains design guidelines which are based on experience in the offshore industry, results of several joint industry projects and many technical publications.

There are several issues that require further consideration and harmonization prior to this Technical Report being progressed further as part of ISO 13819-4. These include:

- environmental criteria in terms of return periods for temporary and permanent moorings;
- factors of safety for tensions, anchor load and fatigue;
- improving the definition and the methodology of the mooring analysis;
- improving the guidelines for thruster-assisted mooring systems;
- providing specific guidance in relation to corrosion protection of mooring lines;
- including IMO DP Guidelines (MSC Circ 645 "Guidance for vessel with dynamic positioning systems") and relevant industry standards.

Technical Report ISO/TR 13637 reproduces the content of API Recommended Practice 2SK, 2nd edition, 1996. ISO, in endorsing this API document, recognizes that in certain respects the latter does not comply with all current ISO rules on the presentation and content of a Technical Report. Therefore, the relevant technical body, within ISO/TC 67, will review ISO/TR 13637:1997 and reissue it, when practicable, in a form complying with these rules.

This Technical Report is not intended to obviate the need for sound engineering judgement as to when and where this Technical Report should be utilized and users of this document should be aware that additional or differing requirements may be needed to meet the needs for the particular service intended.

Standards referenced herein may be replaced by other international or national standards that can be shown to meet or exceed the requirements of the referenced standards.

Appendices A, B, C and D to this document should not be considered as requirements. They are included only as guidelines or information.

Petroleum and natural gas industries — Mooring of mobile offshore drilling units (MODUS) — Design and analysis

1 Scope

This Technical Report presents a rational method for analysing, designing or evaluating mooring systems used with offshore floating units for the petroleum and natural gas industries.

2 Requirements

Requirements are specified in:

> "API Recommended Practice 2SK, 2nd edition, December 1996 — *Recommended Practice for Design and Analysis of Stationkeeping Systems for Floating Structures*"

adopted as ISO/TR 13637.

For the purposes of international standardization, however, modifications shall apply to publication API RP 2SK as outlined below.

a) Information given in the SPECIAL NOTES and FOREWORD is relevant to the API publication only.

b) Throughout publication API RP 2SK, the conversion of English units shall be made in accordance with ISO 31. The content shall be replaced by the following.

LENGTH	1 inch (in) 1 foot (ft)	= 25,4 mm (exactly) = 304,8 mm
PRESSURE	1 pound-force per square inch (lbf/in^2) or psi NOTE 1 bar = 10^5 Pa	= 6 894,757 Pa
STRENGTH OR STRESS	1 pound-force per square inch (lbf/in^2)	= 6 894,757 Pa
IMPACT ENERGY	1 foot-pound force (ft·lbf)	= 1,355 818 J
TORQUE	1 foot-pound force (ft·lbf)	= 1,355 818 N·m
TEMPERATURE	The following formula was used to convert degrees Fahrenheit (°F) to degrees Celsius (°C): °C = 5/9 (°F – 32)	
VOLUME	1 cubic foot	= 0,028 316 8 m^3 or 28,316 8 dm^3
MASS	1 pound (lb)	= 0,453 592 37 kg (exactly)
FORCE	1 pound-force (lbf)	= 4,448 222 N

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Recommended Practice for Design and Analysis of Stationkeeping Systems for Floating Structures

Exploration and Production Department

API RECOMMENDED PRACTICE 2SK
SECOND EDITION, DECEMBER 1996

EFFECTIVE DATE: MARCH 1, 1997

American
Petroleum
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FOREWORD

The bar notations identify parts of this recommended practice that have been changed from the previous API edition. Note that all sections, paragraphs, figures, and tables have been renumbered.

This recommended practice is under the jurisdiction of the API Subcommittee on Offshore Structures.

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This standard shall become effective on the date printed on the cover but may be used voluntarily from the date of distribution.

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Recommended Practice for Design and Analysis of Stationkeeping Systems for Floating Structures

1 Scope

The purpose of this document is to present a rational method for analyzing, designing or evaluating mooring systems used with floating units. This method provides a uniform analysis tool which, when combined with an understanding of the environment at a particular location, the characteristics of the unit being moored, and other factors, can be used to determine the adequacy and safety of the mooring system. Some design guidelines for dynamic positioning systems are also included.

The technology of mooring floating units is growing rapidly. In those areas where the committee felt that adequate data were available, specific and detailed recommendations are given. In other areas general statements are used to indicate that consideration should be given to those particular points. Designers are encouraged to utilize all research advances available to them. As offshore knowledge continues to grow, this recommended practice will be revised. It is hoped that the general statements contained herein will gradually be replaced by detailed recommendations.

2 Basic Considerations

2.1 INTRODUCTION TO STATIONKEEPING SYSTEMS

The stationkeeping system for a floating structure can be either a single point mooring or a spread mooring. Single point moorings tend to be used more frequently for ship shaped vessels, while spread moorings are used mostly for semisubmersibles. A third type of stationkeeping system is dynamic positioning (DP). Dynamic positioning can be used as the sole source of stationkeeping or used to assist a catenary mooring. Dynamic positioning can be used with either tanker or semisubmersible based systems.

2.1.1 Spread Mooring

Figure 1 is an illustration of a catenary spread moored semisubmersible. This is a conventional mooring technique used in floating drilling operations. For floating production applications, spread moorings are used primarily with semisubmersibles. Since the environmental force on a semisubmersible is relatively insensitive to direction, a spread mooring system can be designed to hold the vessel on location regardless of the direction of the environment. However, this system can also be applied to ship-shaded vessels which are more sensitive to environmental directions. The mooring can be chain, wire rope, fiber rope, or a combination of the three. Either conventional drag anchors or anchor piles can be used to terminate the mooring lines.

A spread mooring offers some advantages to the semisubmersible based floating production system. Since it fixes the position of the vessel, drilling and completion operations can be carried out on subsea wells located immediately below the vessel. The same is true for workover operations. On the other hand, a spread mooring system has a fairly large mooring spread (on the order of several thousand feet). Anchors and suspended mooring lines are present within this spread. These must be considered in the installation or maintenance of pipelines, risers, or any other subsea equipment.

The combination of a spread mooring with vertical mooring tendons to restrain a tension leg platform (TLP) on location, as shown in Figure 2 enhances both the operability and reliability of the basic TLP concept. The spread mooring allows for adjustment of the surface vessel in a controlled manner and provides an independent parallel load path to react against the lateral environmental forces. With this concept it is possible to horizontally position drilling tools and production equipment packages to be landed and attached to seafloor structures. Otherwise, these equipment packages would have to be positioned by other means such as guidelines, thrusters, or skidding the derrick on the surface vessel. The configuration and design of this spread mooring will be very similar to a spread mooring system used to moor semisubmersible based floating production systems.

2.1.2 Single Point Mooring

Single point moorings are used primarily for tankers. They allow the vessel to weather vane. This is necessary to minimize environmental loads on the tanker by heading into the prevailing weather. There is wide variety in the design of single point moorings, but they all perform essentially the same function. Single point moorings interface with the production riser and the vessel. An introduction to typical single point mooring systems is as follows:

a. Turret mooring. A turret mooring system is defined as any mooring system where a number of catenary mooring legs are attached to a turret that is essentially part of the vessel to be moored. The turret includes bearings to allow the vessel to rotate around the anchor legs.

The turret can be mounted externally from the vessel bow or stern with appropriate reinforcements (see Figure 3—External Turret Mooring System) or internally within the vessel (see Figure 4—Internal Turret Mooring). The chain table can be above or below the waterline. The turret also could be integrated into a vertical riser system that is attached to the bow or stern of the vessel (or internally) through some kind of mechanism that allows articulation (gimballed table, "U" joint or chain connections). The base of the riser is often weighted through additional weight within the riser or suspended beneath the riser (counterweight). These items affect

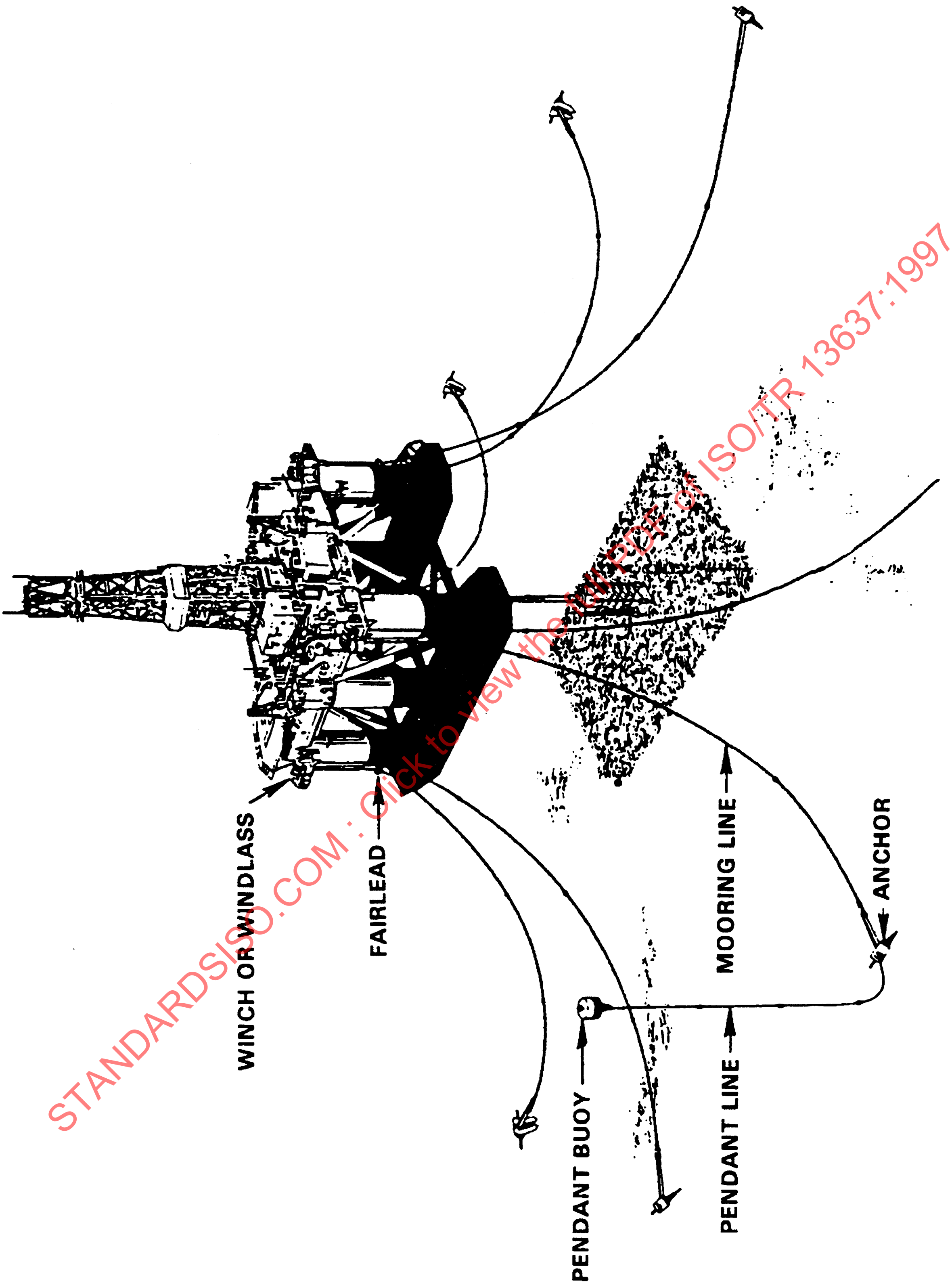


Figure 1—Spread Mooring

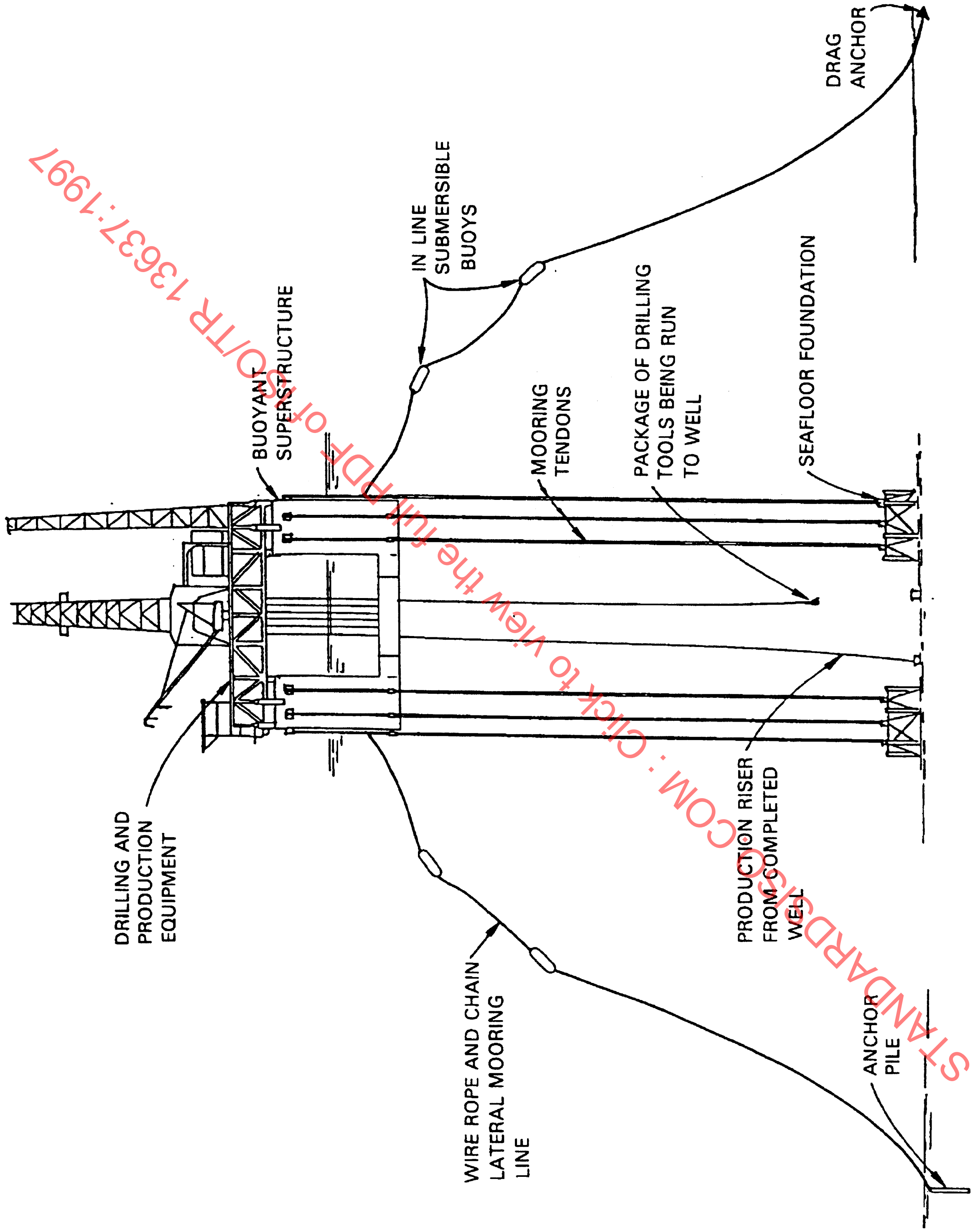


Figure 2—TLP Lateral Mooring System

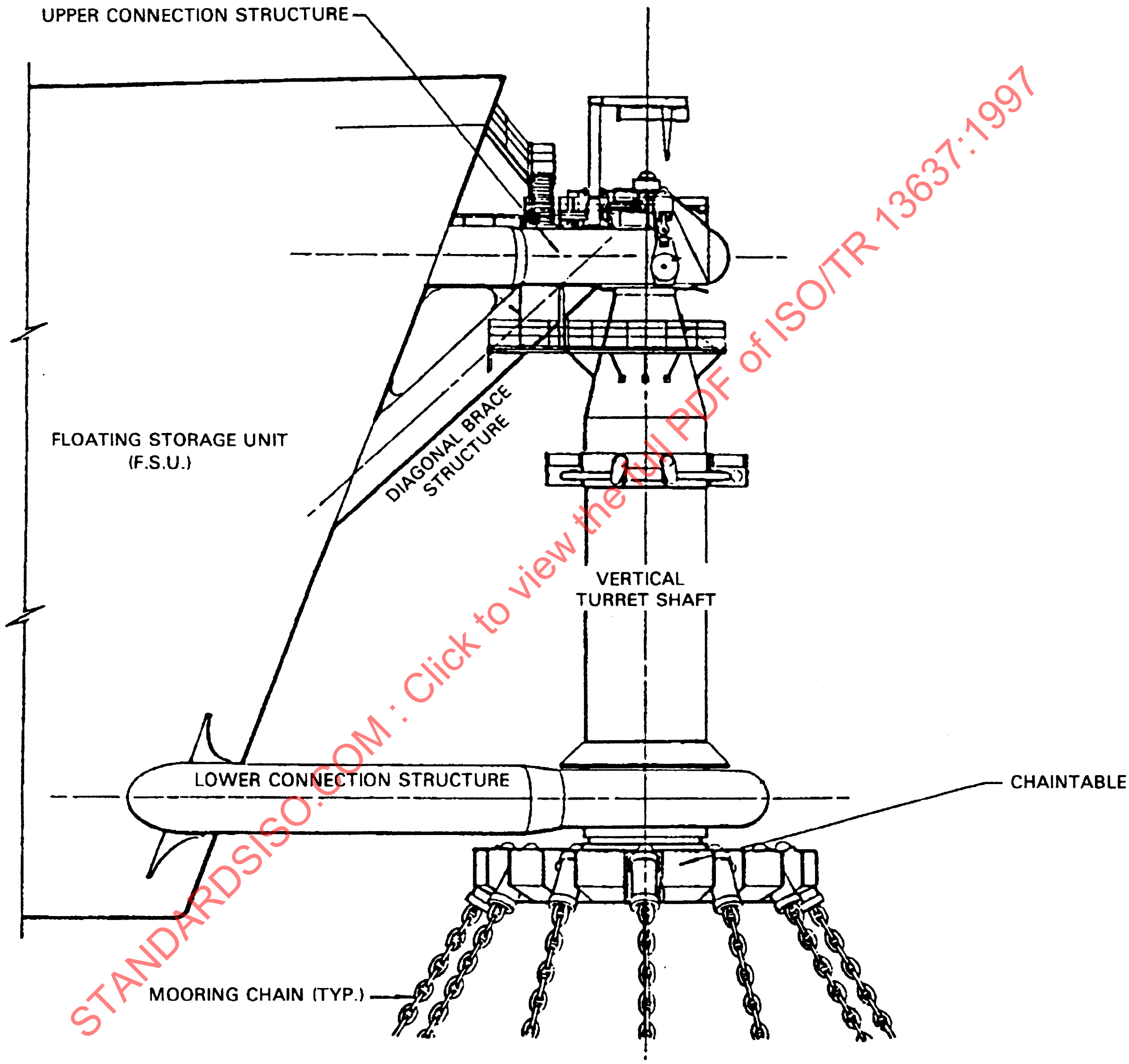


Figure 3—Typical External Turret Mooring Arrangement

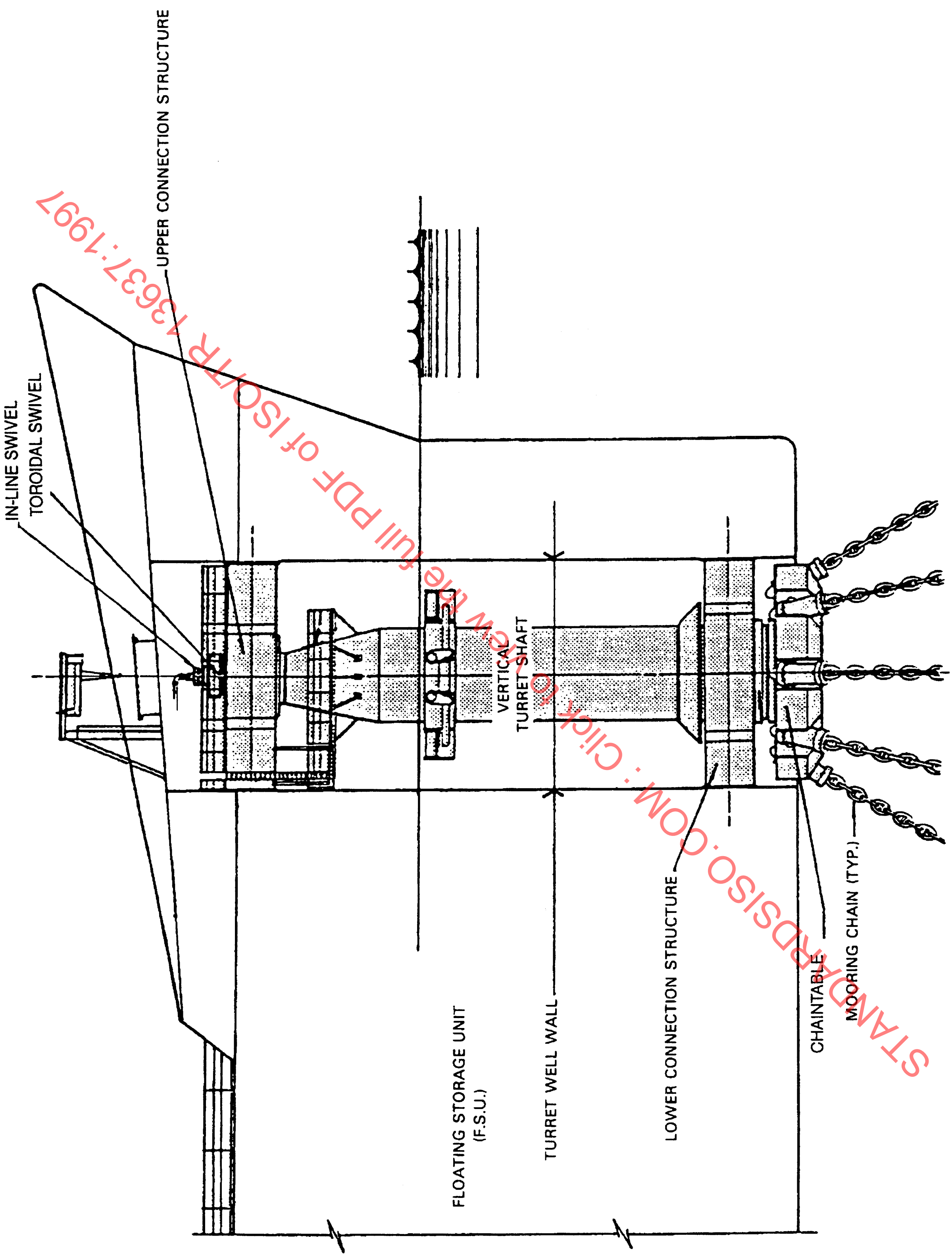


Figure 4—Typical Internal Turret Mooring Arrangement

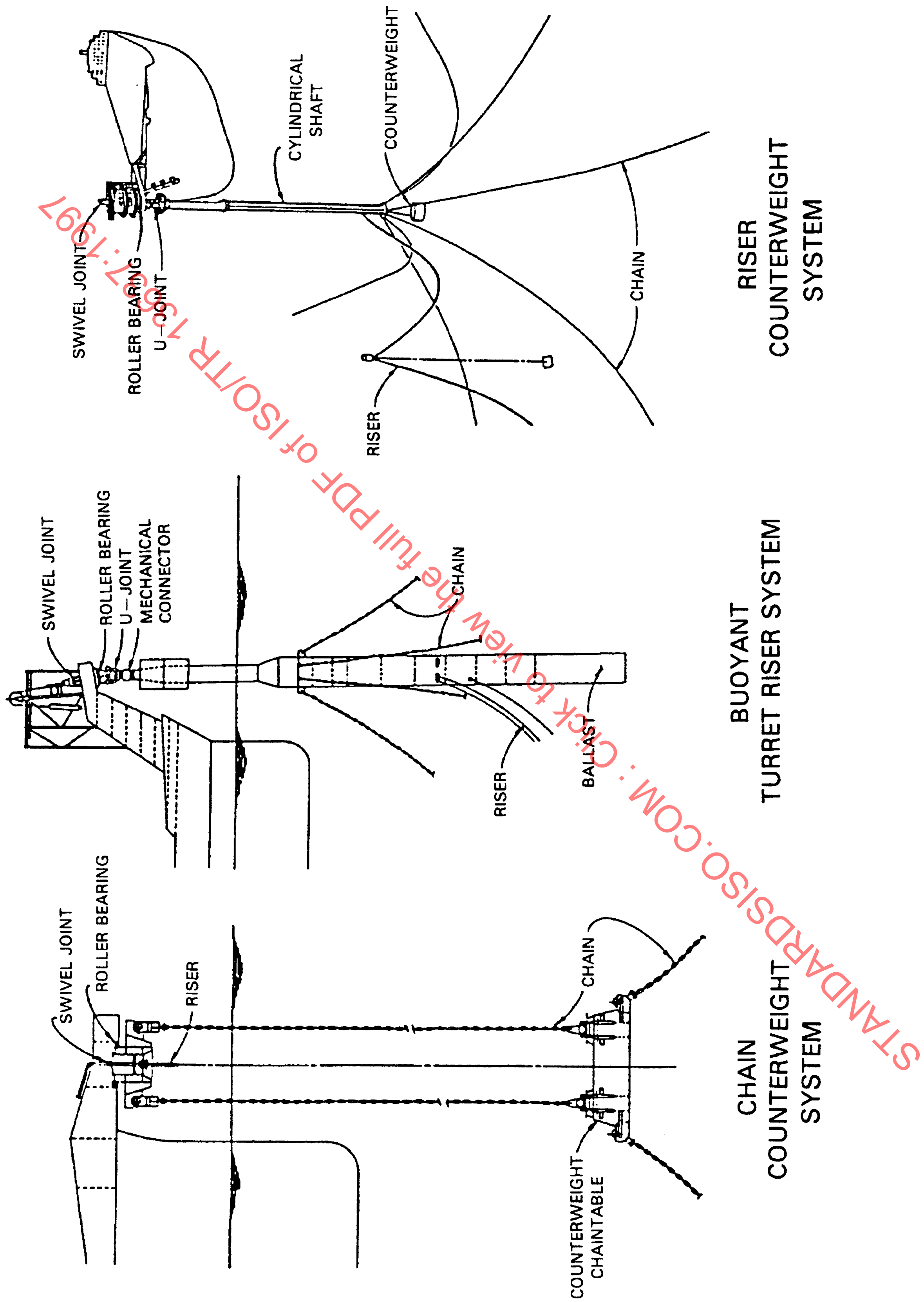


Figure 5—Variations on Turret/Riser Systems

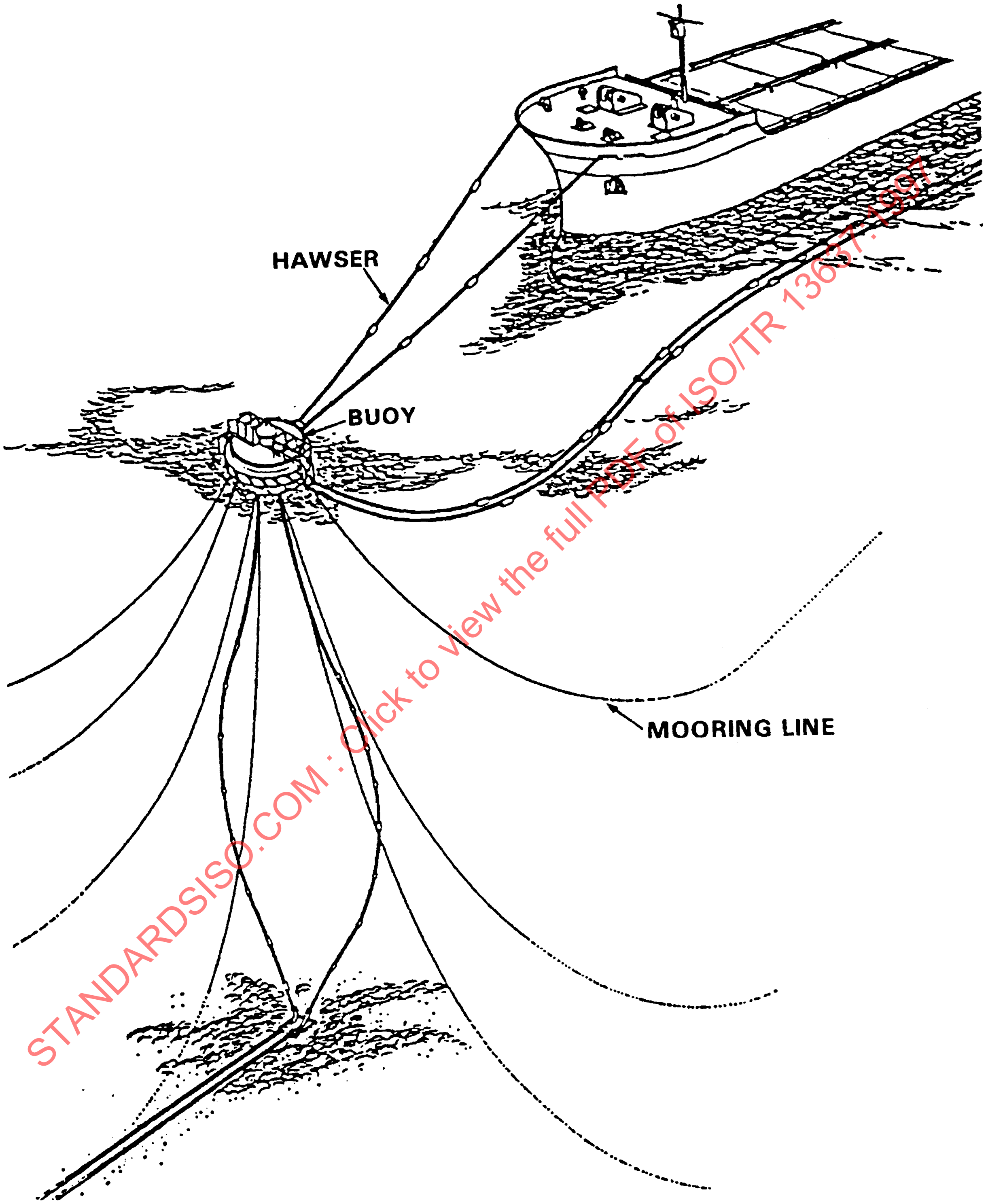


Figure 6—Catenary Anchor Leg Mooring (CALM) with Hawsers

the performance of the mooring system. The configuration of the riser could include steel tubular, chain or wire rope components and can vary considerably in diameter and length. The position of the chain table relative to the riser also can vary according to the design. Figure 5 shows some variations in the turret design offered by industry.

b. CALM (catenary anchor leg mooring). The CALM system consists of a large buoy that supports a number of catenary chain legs anchored to the sea floor (Figure 6). Riser systems or flow lines that emerge from the sea floor are attached to the underside of the CALM buoy. Some of the systems use a hawser, typically a synthetic rope, between the tanker and the buoy. Since the response of the CALM buoy is totally different than that of the tanker under the influence of waves, this system is limited in its ability to withstand environmental conditions. When sea states attain a certain magnitude it is necessary to cast the tanker off.

In order to overcome this limitation, rigid structural yokes with articulations are used in some newer designs to tie the ship to the top of the buoy. An example is shown in Figure 7. This rigid articulation virtually eliminates horizontal motions between the buoy and the tanker. A more recent development, shown in Figure 8, is a buoyant yoke with a soft mooring connection using chains attached to the yoke.

c. SALM (single anchor leg mooring). This system employs a vertical riser system that has a large amount of buoyancy near the surface, and sometimes on the surface, that is held by a pretensioned riser. The system typically employs a tubular, articulated riser with a fixed yoke (Figure 9). It is possible also to use a chain riser with soft mooring connections (Figure 10). The vertical buoyancy force acting on the top of the riser functions as an inverted pendulum. When the system is displaced to the side, the pendular action tends to restore the riser to the vertical position.

The tanker can be secured to the top of this SALM buoy with either a flexible hawser or a rigid yoke as discussed in the CALM description. The base of the riser is usually attached through a U-joint to a piled or deadweight concrete or steel structure on the sea floor. In deep water, the riser system usually has mid-span articulation.

2.1.3 Dynamic Positioning

Dynamic positioning (Figure 11) can be used as the sole source of stationkeeping or used to assist a catenary mooring system. Dynamic positioning consists of a position reference system, usually acoustic, coupled with computer controlled thrusters around the vessel. Dynamic positioning can be used in conjunction with a mooring which is called DP assisted mooring (or thruster assisted mooring if thrusters are manually controlled). Dynamic positioning is particularly well suited for a vessel designed to come onto and leave location frequently, such as an extended well test system.

2.2 DIFFERENCES BETWEEN PERMANENT AND MOBILE MOORING SYSTEMS

Permanent moorings are normally used for production operations with longer design lives. The mooring for a floating production system (FPS), for example, is a permanent mooring since FPSs typically have design lives of over 10 years. Mobile moorings often stay on one location for a short period. Examples of mobile moorings include those for mobile offshore drilling units (MODUs), and for tenders moored next to another platforms such as floatels, drilling tender, and service vessels. The division between mobile and permanent moorings may not be clear for operations with design lives of a few years. In this case, the user should make a judgment based on the risk of exposure to severe environments and the consequence of a mooring failure. Differences between permanent and mobile moorings are significant, as discussed below. The discussion can be used as a guideline to determine the category (permanent or mobile) to which the floating structure belongs.

2.2.1 Type of Mooring

A mobile vessel is normally equipped with a spread mooring, internal turret mooring, or dynamic positioning system. However, a permanent vessel has more choices of mooring design because mobility is normally not required.

2.2.2 Environmental Criteria

The design environments for mobile moorings are lower than those for permanent moorings. The lower design environment for mobile moorings is based on the consideration that the consequence of a mooring failure would generally be less severe. This can be illustrated by comparing a MODU with an FPS. In many instances, a MODU can at least disconnect and may even lay down its drilling riser. In the case of tropical storms, it may be possible to move the vessel before the arrival of a storm. By contrast, an FPS is unlikely to be removable from location, and may not even have quickly-retrievable risers.

2.2.3 Method of Analysis

A quasi-static analysis method is normally used for evaluating the performance of a mobile mooring system, and the effects of line dynamics are accommodated through the use of a relatively conservative safety factor. A more rigorous dynamic analysis is required for the final design of a permanent mooring system, and the factor of safety is relaxed to reflect that some uncertainty in line tension prediction is removed. Dynamic analysis should also be performed for mobile moorings if the consequence of a mooring failure is severe.

Also, a fatigue analysis is not required for mobile mooring systems. Because of abuse from frequent deployment and retrieval, many mooring components of a mobile mooring

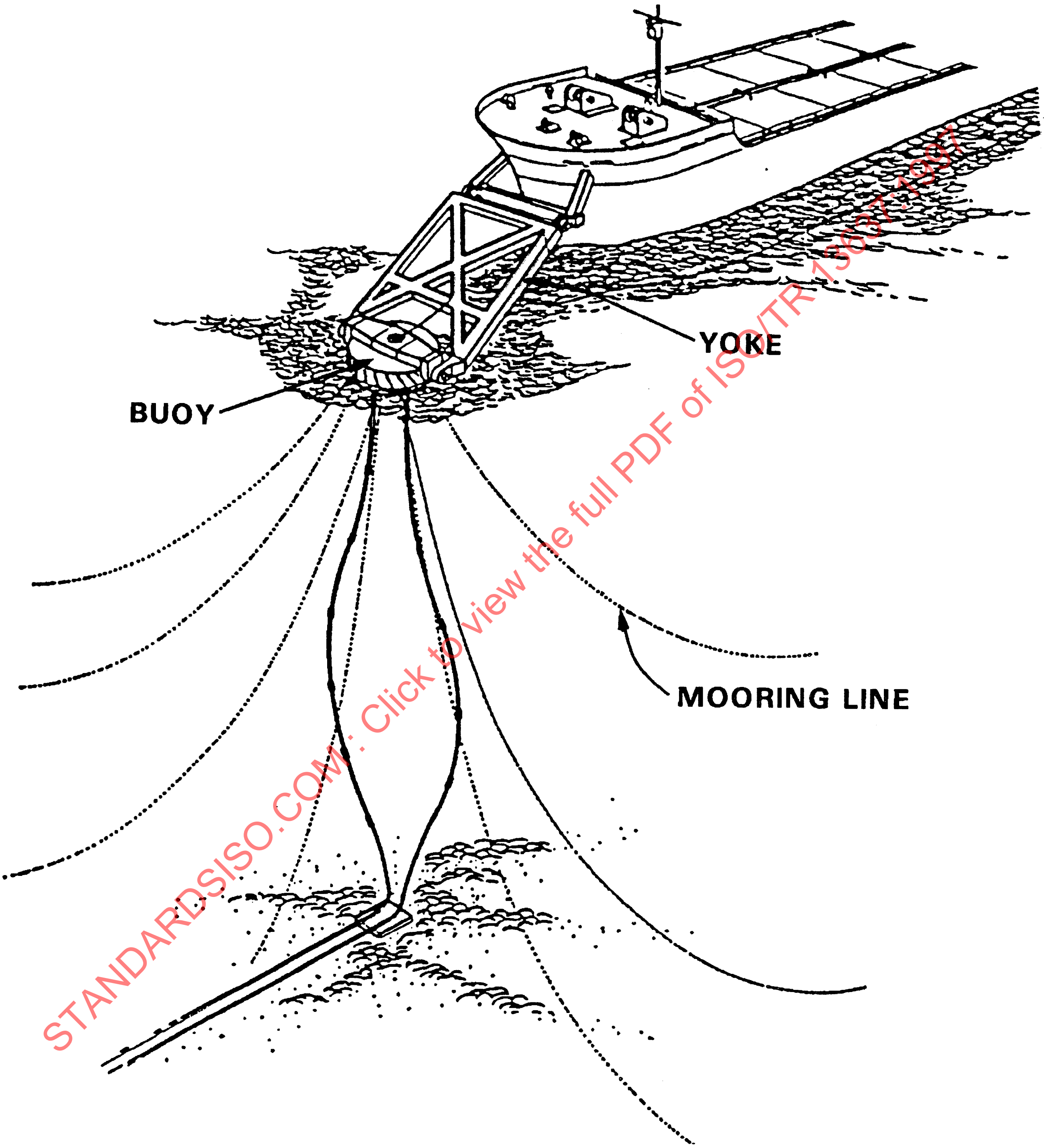


Figure 7—Catenary Anchor Leg Mooring (CALM) System with Fixed Yoke

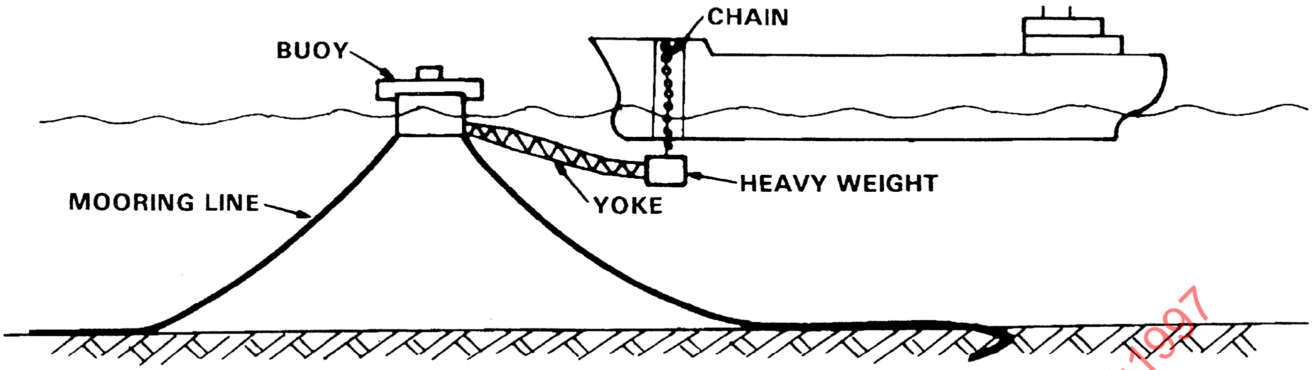


Figure 8—Catenary Anchor Leg Mooring (CALM) System with Soft Yoke

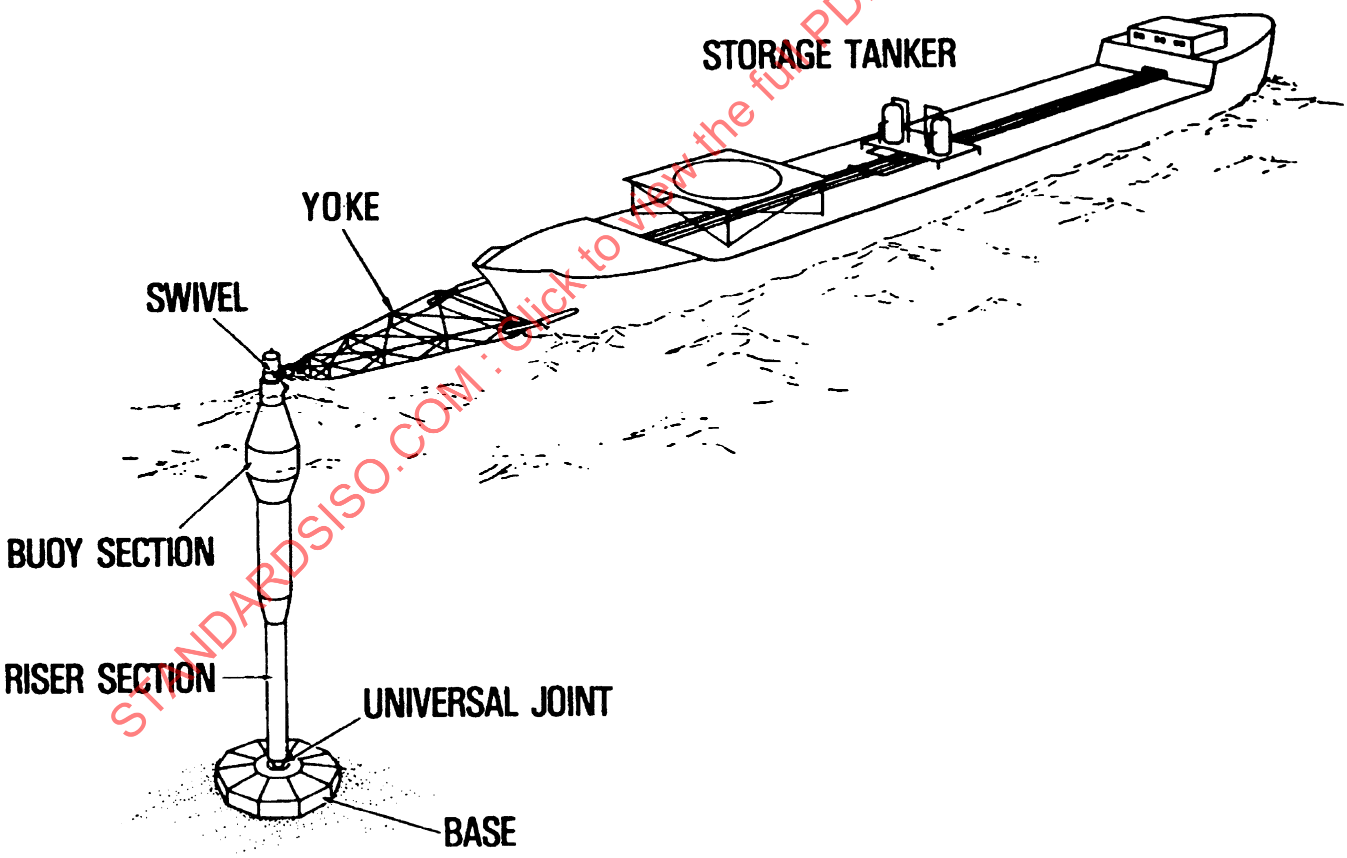


Figure 9—Single Anchor Leg Mooring (SALM) with Tubular Riser and Yoke

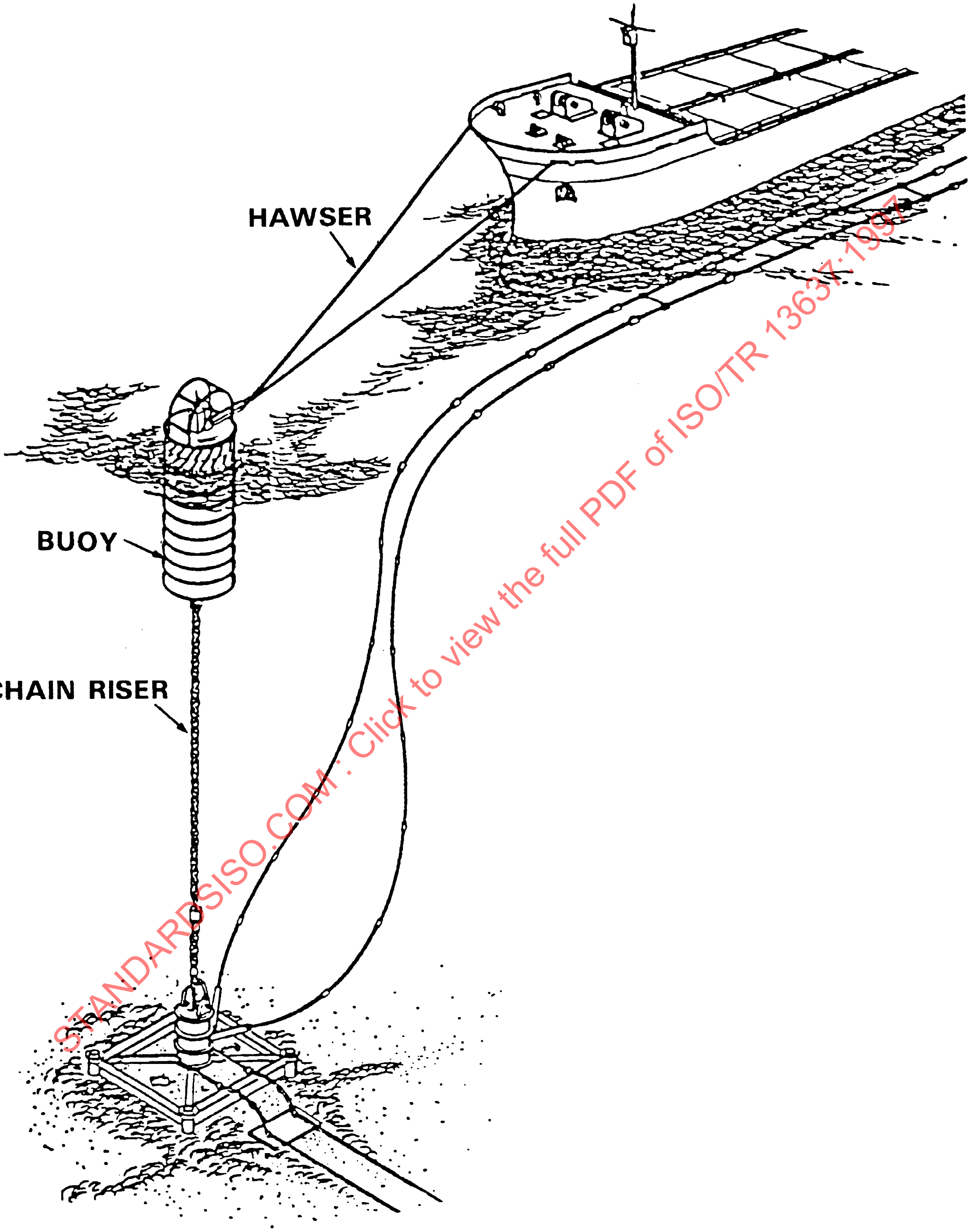


Figure 10—Single Anchor Leg Mooring (SALM) with Chain Riser and Hawser

system are replaced before they reach their fatigue limits. However, for permanent installations such as an FPS, fatigue is an important design factor, and a fatigue analysis should be performed.

2.2.4 Mooring Hardware

Mobile moorings use the mooring hardware that can be rapidly deployed and retrieved. This limitation does not apply to permanent moorings. Many mooring components such as anchor piles, linear winches, buoys, and chain jacks that may not be suitable for mobile moorings can be used in a permanent mooring. Also permanent moorings often require heavier mooring hardware because of the more stringent design requirements.

2.2.5 Installation

The deployment of a mobile mooring is normally carried out with the assistance of work boats. This operation is simple and usually takes no more than a few days. The deployment of an FPS mooring often requires the assistance of much heavier vessels such as a derrick barge or a purposely built work boat. A portion of the mooring is usually preset. Sometimes special design features are incorporated in the mooring design to facilitate deployment.

2.2.6 Inspection and Maintenance

A mobile mooring can often be visually inspected during retrieval or deployment. Retrieving a permanent mooring for inspection can be very expensive. To inspect a permanent mooring, divers or ROVs are often used. Also, replacing faulty mooring components is easier for mobile moorings than for permanent moorings.

2.3 DESIGN CONSIDERATIONS

2.3.1 Primary Design Considerations

The primary design considerations associated with a mooring system are design criteria, design loads, design life, operation and maintenance considerations. These considerations are addressed in detail in the following sections. In addition, a designer must also pay attention to the riser and subsea equipment considerations.

2.3.2 Riser Considerations

Risers transfer fluids between the seabed and the production or drilling vessel, and constitute one of the primary design constraints of the mooring system. The riser system often places limitations on the allowable vessel offset. In the event of excessive vessel offsets, mooring line adjustments such as slackening the leeward lines are sometimes performed to avoid damage to the riser. An equally important consideration is interference between mooring lines and risers, during both operational and extreme weather conditions. The mooring system and riser system must therefore be designed to accommodate each other, and coordination of these two design efforts is essential.

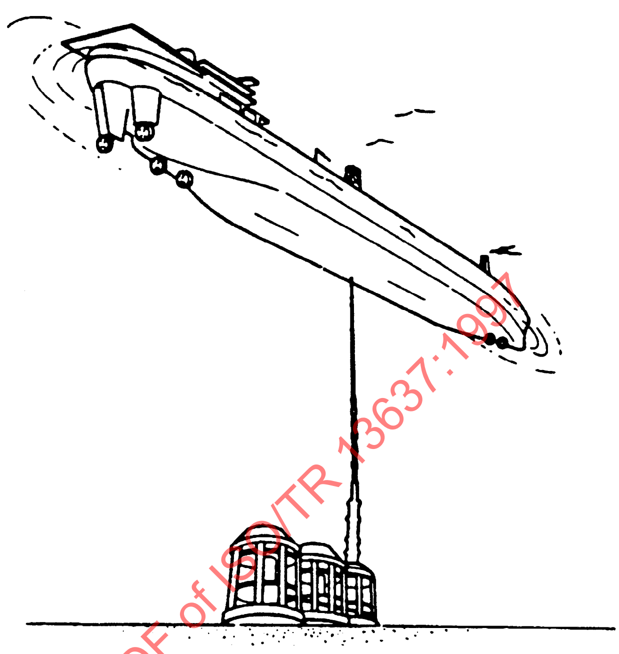


Figure 11—Dynamic Positioning

Design guidelines for riser systems can be found in API Recommended Practice 17A [5], API Recommended Practice 17B, [6], and API Recommended Practice 16Q [7].

2.3.3 Subsea Equipment Considerations

Subsea equipment such as templates, riser bases, satellite wells, and flowlines should be located clear of any potential mooring line interference. Any contact between mooring lines and subsea equipment during installation, operation or maintenance presents a high potential of damage to both the equipment and the mooring lines. If interference, or the potential for interference appears unavoidable, it may be possible to alter the layout and design of the mooring system through the use of an asymmetric arrangement of mooring lines, or the use of clump weights or spring buoys. Coordination of the mooring system design with the subsea equipment layout is essential.

Guidelines for the design of subsea equipment are given in API Recommended Practice 17A.

3 Mooring Components

3.1 MOORING LINE

3.1.1 Classification

Mooring lines for moored vessels may be made up of chain, wire rope, synthetic rope, or a combination of them. There are many possible combinations of line type, size, and location, and size of clump weights or buoys that can be used to achieve given mooring performance requirements. The following are typical systems used by the industry:

3.1.1.1 All-Wire Rope System.

Because wire rope is much lighter than chain, wire rope provides a greater restoring force for a given pretension. This becomes increasingly important as water depth increases. However, to prevent anchor uplift with an all-wire system, much longer line length is required. A disadvantage of an all-wire rope mooring system is wear due to long term abrasion where it contacts the seabed. For these reasons, all-wire rope mooring systems are seldom used for permanent moorings.

3.1.1.2 All-Chain System.

Chain has shown durability in offshore operations. It has better resistance to bottom abrasion and contributes significantly to anchor holding capacity. However, in deep water an all-chain system imposes an increasing weight penalty on the vessel's load carrying capacity by its own self weight and high initial tension requirements.

3.1.1.3 Chain/Wire Rope Combination.

In this system, a length of chain is typically connected to the anchor. This provides good abrasion resistance where the mooring line contacts the seabed and its weight contributes to anchor holding capacity. The choice of chain or wire rope at the vessel end and the type of termination also depend on the requirements for adjustment of line tensions during operations. By proper selection of the lengths of wire rope and chain, a combination system offers the advantages of reduced pretension requirements with higher restoring force, improved anchor holding capacity, and good resistance to bottom abrasion. These advantages make combination systems attractive for deep water mooring.

An alternative to the above system is the wire rope/chain/wire rope combination system where wire rope segments are connected to both the vessel and the anchor. A length of chain is used in the dip zone where the mooring line is in dynamic contact with the seafloor. This minimizes the amount of chain, which is costly and difficult to deploy at deepwater sites.

3.1.2 Chain

The choice of material and fabrication of large diameter chain for a moored vessel requires careful evaluation. It is desirable to have chain used for this application manufactured in continuous lengths for each mooring leg. This eliminates the need for chain connection links and the associated problems with fatigue. Otherwise, connecting links with sufficient fatigue life should be used.

Chain can be obtained in several grades with Grade 4 (K4) being the highest strength. Oil Rig Quality (ORQ) chain has been sold in large quantities to drilling contractors over the years and has generally performed well. Grade 3 chain has a catalogue break strength of approximately 93 percent of the equivalent ORQ chain, and K4 chain has a catalogue break strength of approximately 130 percent of the equivalent ORQ chain. Grade 2 chain is not recommended for major mooring operations.

A grade of chain somewhere between ORQ and K4, for example "ORQ + 20 percent" (breaking strength 20 percent higher than ORQ), is preferred by some designers since it is easier to manufacture than K4 chain. In any case it is recommended that considerable care is taken in establishing correct chemical composition of the bar stock, manufacturing techniques that incorporate precise quality control, and finally, comprehensive testing of samples of the final manufactured product. Detailed guidance in the specification, inspection, and testing of ORQ chain can be found in API Specification 2F [46].

Protection against chain corrosion and wear is normally provided by increasing chain diameter. The allowance in chain diameter for corrosion and wear is a complicated issue that still requires significant research and service experience to address. Current industry practice is to increase the chain diameter by 0.2 mm to 0.4 mm per service year in the spalsh zone where oxygenated water tends to accelerate corrosion and in the dip or thrust zone on hard bottom where heavy abrasion takes place. The diameter increase is reduced to 0.1 mm to 0.2 mm per service year in the remaining length. Galvanic protection for chain with anodes has been developed, but they have not been widely used because of their relatively short life (about 5 years), difficulty of attachment, and high cost.

3.1.3 Wire Mooring Line

The wire rope sections of the moorings can be of various constructions as shown in Figure 12. The wire rope construction type includes a number of strands wound in the same rotational direction around a center core to form the rope. The number of strands and wires in each strand (i.e., 6 by 36, 6 by 42, 6 by 54), core design and lay of strands are governed by required strength and bending fatigue considerations for the rope. This construction generates torque as tension increases.

The spin-resistant strand type constructions (spiral strand and multi-strand) are attractive for use with permanent moorings since they do not generate significant torque with tension changes. Both constructions use layers of wires (or bundles of wires) wound in opposing directions to obtain the spin resistance characteristics.

For corrosion resistance in permanent moorings, typically a polyethylene or polyurethane jacketing is employed. The jacketing material should be a high density type. Also all wires should be galvanized. Zinc filler wires are sometimes incorporated to provide additional corrosion protection. A filler material is used to block the inside spaces between the

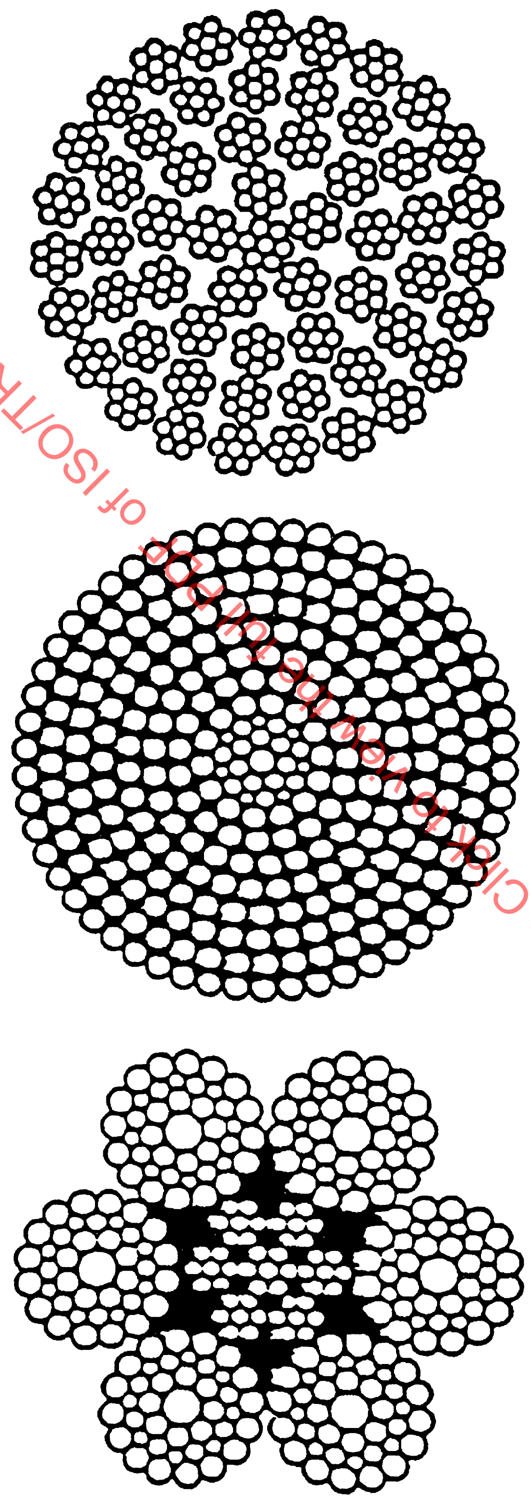


Figure 12—Different Wire Line Constructions

wires to minimize the spread of corrosion with ingress of salt water. It has not been a common practice to increase the diameter of wire mooring lines for corrosion and wear. However, in a few permanent moorings with unjacketed wire mooring lines, the rope diameters have been increased by 0.1 mm to 0.2 mm per service year for protection against corrosion and wear.

The ends of each mooring line section should be terminated with sockets. A resin material is preferred over zinc for pouring the sockets. For permanent moorings, the sockets should be provided with flex relieving boots (bend limiting devices) joined to the socket in a manner to seal out the ingress of water and limit free benching fatigue. Careful quality control and testing should be exercised prior to and during the fabrication of the rope to ensure that the rope meets design specifications and the final product produces guaranteed minimum break strength as specified.

3.1.4 Synthetic Mooring Line

Synthetic materials are generally not used in permanent mooring systems because of a lack of long term service experience. Research is underway to better understand the properties of various ropes made of high-performance synthetic materials, which may be in wide use in the future. An exception is the use of synthetic hawsers in CALM systems where high elasticity is an important property. These hawsers can be inspected frequently and replaced. Also, in very shallow water locations under mild design environment, a length of synthetic rope (typically, nylon rope) is often inserted in the mooring line to absorb the energy from vessel dynamics.

3.1.5 Clump Weight

Clump weights are sometimes incorporated in mooring legs to improve performance or reduce cost. By providing a concentrated weight to the mooring leg at a point close to the seabed, a clump weight can be used to replace a portion of chain and increase the restoring force of a mooring leg. Using clump weights in a mooring line design requires consideration of potentially adverse effects, such as increased use of connecting hardware, installation complexity, undesirable dynamic response of the mooring line, and embedment of the clump weight in the seabed.

3.1.6 Spring Buoy

Spring buoys are surface or subsurface buoys that are connected to a catenary mooring line. The benefits of spring buoys are the following:

a. Reduced weight of mooring lines that must be supported by the vessel hull; this is particularly advantageous to semisubmersibles moored in deep water.
b. Reduced effects of line dynamics in deepwater.
c. Reduced vessel offset for a given line size and pretension.

The adverse effects of spring buoys are the following:

a. Increased use of connecting hardware, and installation complexity.
b. Potential for increased design loads on the mooring lines due to dynamic response of the buoy in heavy seas.

Spring buoys used with permanent moorings could be constructed from steel or a combination of synthetic material surrounding a steel structure. A high density foam material (glass spheres encased in a high density foam) has been successfully used to provide buoyancy for deepwater drilling and production risers and floats for flexible risers. Steel buoys have been found to provide a cost competitive solution. The buoys can be built either spherical in shape, using unstiffened dished ends welded together, or with ring stiffened cylindrical bodies and ends. Buoys can be placed in line with the mooring (with a strength member through the buoy) or attached separately to the mooring through a tri-plate as shown in Figure 13. When using the in-line buoy approach, care must be taken to allow for rotation in the end connections.

The buoys should be designed to have adequate strength for maximum operating depth. During fabrication of the buoys, all welding should be tested with appropriate non-destructive testing. Also, corrosion protection should be adequately provided.

3.1.7 Connecting Hardware

Connecting hardware such as shackles, swivels, fishplates, and detachable links are used to connect the main mooring line components. Inspection and replacement of connecting hardware in a permanent mooring are difficult, therefore fatigue life and corrosion protection become important considerations. The design of all connecting hardware used in permanent mooring lines should be thoroughly evaluated to ensure that stress concentration factors are correctly identified, and that fatigue life and corrosion protection are adequate. Manufacturing of connecting hardware should be subject to an appropriate level of quality assurance.

Connecting links such as Kenter and Baldt links are often used in mobile moorings. They can pass through chain fairleads and windlasses and can be periodically inspected and replaced.

3.2 WINCHING EQUIPMENT

The type and design of winching equipment required in a particular mooring system depends on the type of mooring line to be handled, and whether or not the floating vessel itself must initially tension the mooring lines or test-load anchors. A floating vessel usually has the means of adjusting mooring line tension, pretensioning after anchor drag, and disconnecting individual mooring lines. Besides, a floating vessel is often used for combined drilling and production. This will require the capability for finite surface positioning

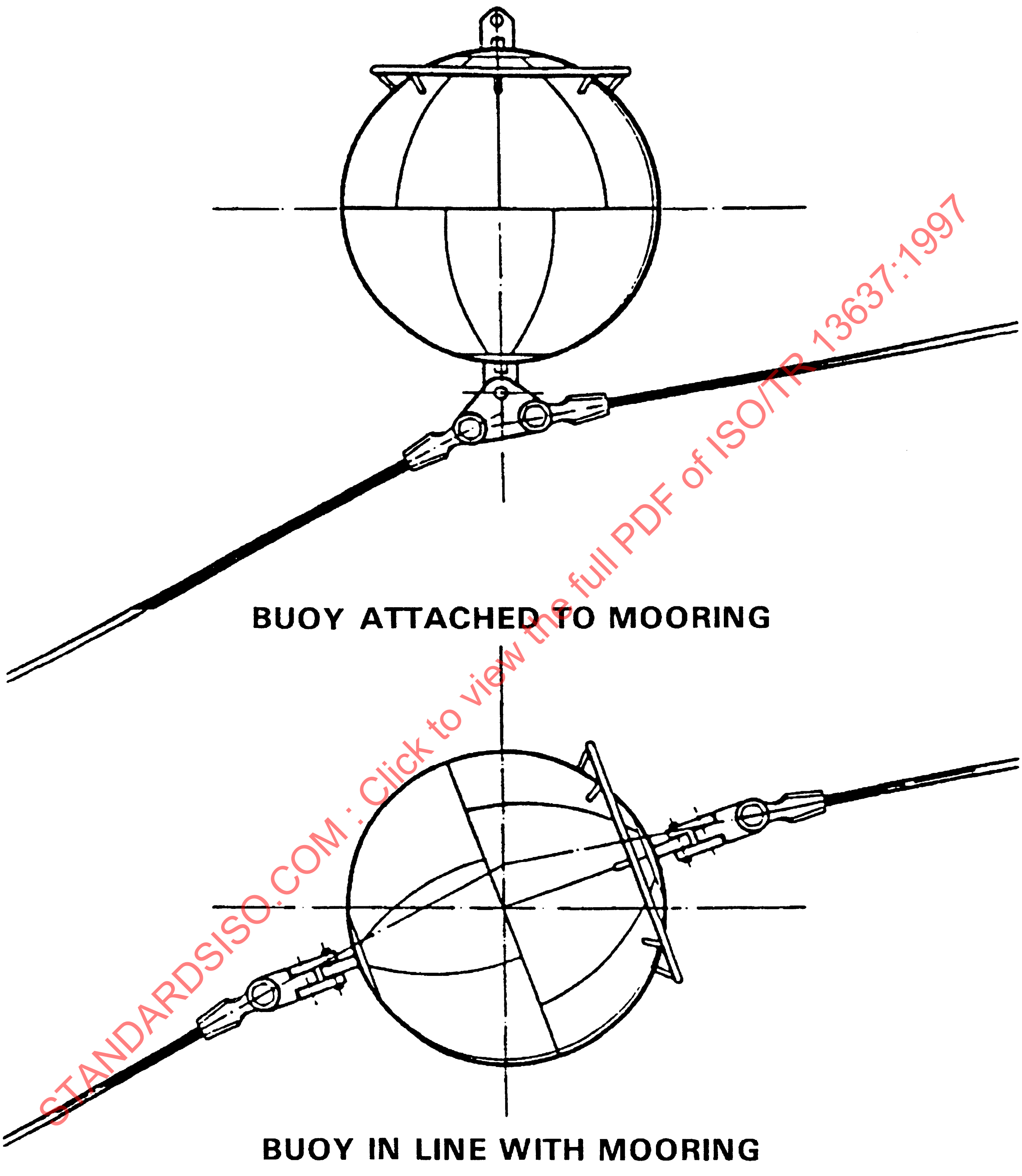


Figure 13—Submersible Buoy Configurations

for maneuvering the risers. This positioning can be achieved by paying-out and heaving-in mooring lines.

3.2.1 Windlass

The most common method of handling and tensioning chain is through the use of a windlass. The windlass consists of a slotted wildcat that is driven by a power source through a gear-reduction system. As the wildcat rotates, the chain meshes with the wildcat, is drawn over the top of the wildcat, and then lowered into the chain locker. Once the chain is hauled in and tensioned, a chain stopper is engaged to hold the chain. The windlass has proven to be a fast and reliable method for handling and tensioning chain.

3.2.2 Chain Jack

A chain jack is a device that reciprocates linearly to haul-in and tension chain. Usually powered by one or more hydraulic cylinders, the chain jack engages the chain, pulls in a short amount of the chain, engages a stop, retracts, and repeats the process. Although a chain jack can be a powerful means for tensioning chain, it is very slow and is recommended for applications that do not require frequent line manipulation.

3.2.3 Drum-Type Winch

The conventional drum-type winch is the most common method used for handling wire rope. Operation of a drum-type winch is fast and smooth. A drum-type winch consists of a large drum on which wire rope is wrapped. The base of the drum is often fitted with special grooves sized specifically to the size of wire rope being handled. The grooves control the positioning of the bottom layer of wire rope on the drum. For subsequent layers of wire rope, an external guidance mechanism such as a level-wind is often used to control positioning of the wire rope on the drum.

The drum-type winch can be a cumbersome method of handling wire rope for deepwater or high strength mooring systems. As the requirement for line sizes and lengths increases, the size of the winch can become impractical. In addition, when wire rope is under tension at an outer layer on the drum, spreading of preceding layers can occur causing damage to the wire rope.

3.2.4 Linear Winch

A linear winch is similar in principal to the chain jack. Two sets of grippers, one stationary and one translating, are used to haul-in and tension the wire rope. Linear winches are available in a single-acting form where the wire rope moves intermittently as the gripper is retracted to begin another stroke, and in a continuous double-acting form in which case two translating grippers are used alternately for continuous smooth motion of the wire rope. A linear winch is most applicable in a permanent application when high tension and large-diameter wire rope are required. A take-up reel is necessary in this case to coil the wire rope after it passes through the linear winch. A winching system using linear winches is illustrated in Figure 14.

3.2.5 Traction Winch

Traction winches have been developed for high tension mooring applications, and for handling combination mooring systems. They consist of a powered drum on which the wire rope makes just a few wraps (typically, 7). Tension in the wire rope causes the wire rope to grip the drum. The wire rope is coiled on a take-up reel that is required to maintain a nominal level of tension in the wire rope (typically 3 percent to 5 percent of working tension) to ensure the proper level of friction is maintained between the wire rope and the traction winch. This system has been favored for use in high tension applications due to its compact size, capability to provide constant torque, and ability to handle very long wire rope without reduced pull capacity.

3.2.6 Fairlead and Stopper

Mooring lines are subjected to high wear and stress at the fairlead and stopper arrangements. The long term service of a mooring system requires that fairlead and stopper arrangements be carefully designed to minimize wear and fatigue.

Mooring chain and wire rope are often stopped off at the vessel in order to take direct mooring loads off the winch. Chain stoppers and wire rope grips used for permanent mooring systems must be designed to keep the stress concentrations and wear within the chain or wire rope at acceptable levels.

Fairleads should provide sufficient sheave-to-rope diameter ratio to minimize tension-bending fatigue. Typically, 7-pocket wildcat sheaves are used for chain. Sheaves for wire rope have *D/d* ratios of 16/25 for mobile moorings, and 40/60 for permanent moorings. There are other devices that provide attractive alternatives for fairleading large diameter mooring lines. An example is the underwater swivelling bending shoe shown in Figure 14. This device incorporates a shoe-to-rope diameter ratio of more than 70 and a special high-density nylon bearing material secured to the bearing surface on the shoe. Replacement of the material is possible by slacking down the mooring line and removing the bearing material that is bolted to the bearing surface in sections.

3.3 ANCHORING SYSTEM

The options that are available for anchoring floating vessels include the following:

a. Drag embedment anchors.
b. Pile anchors (driven, jetted, drilled and grouted).

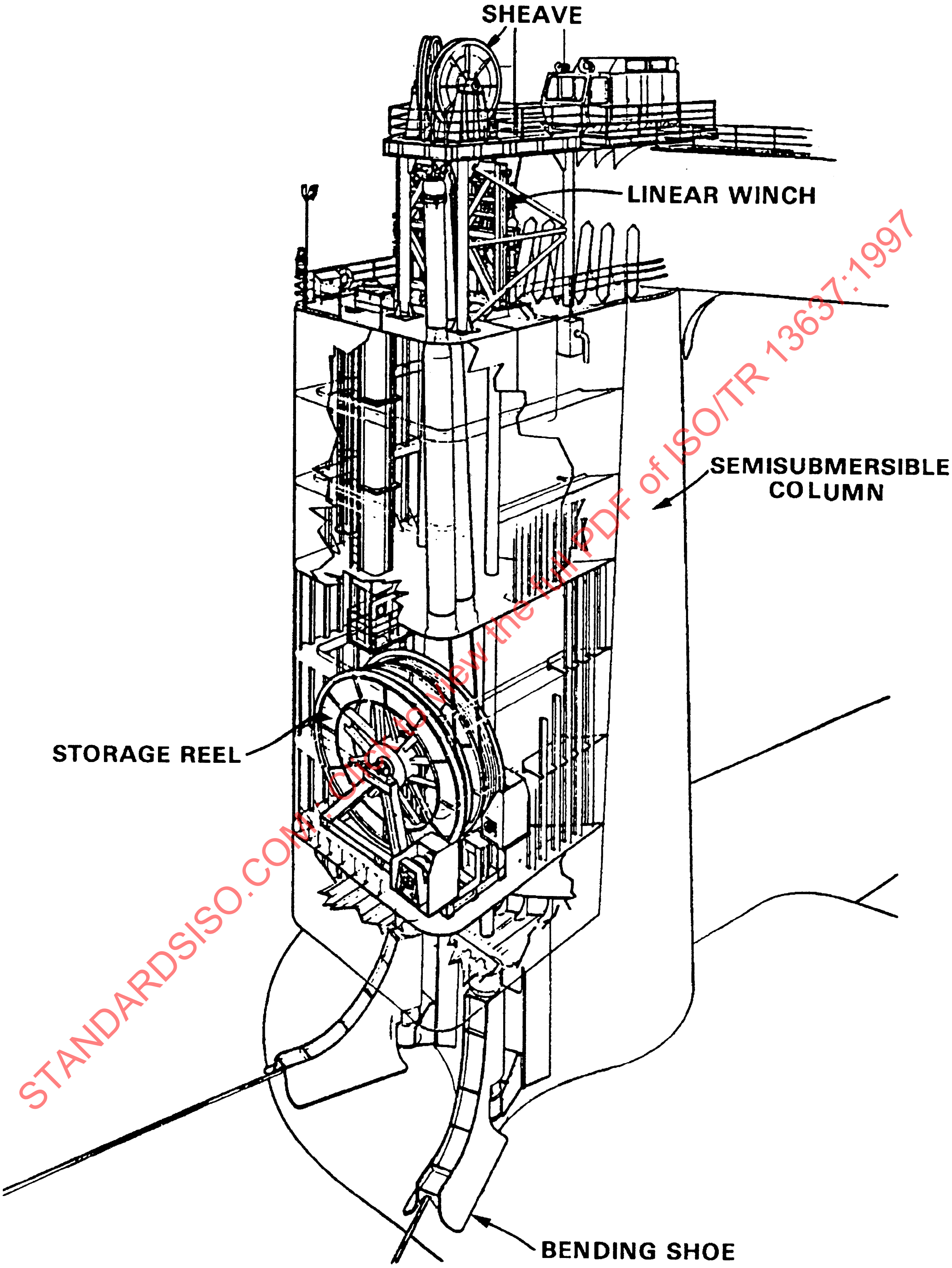


Figure 14—Winching System Using Linear Winch and Bending Shoe

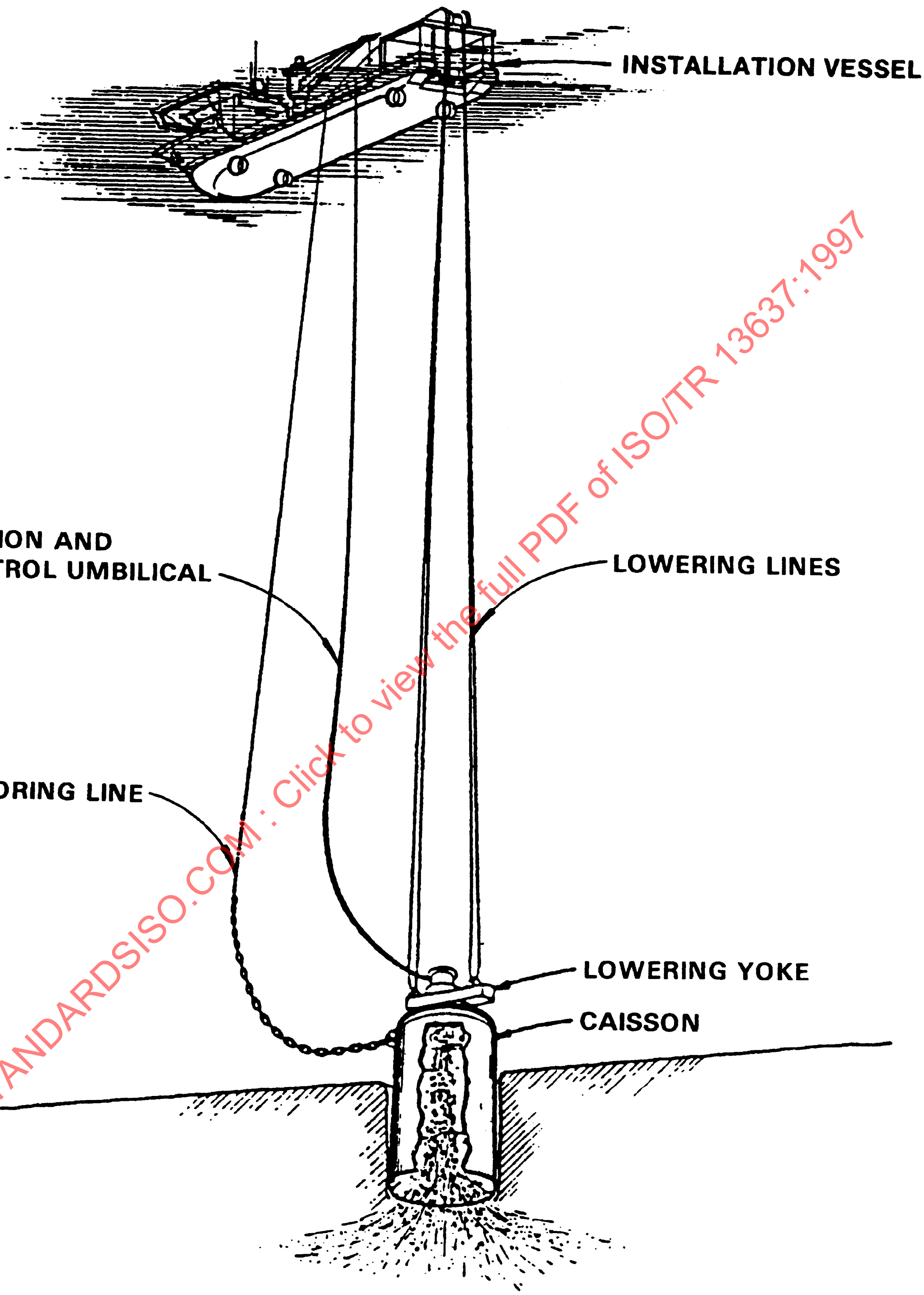


Figure 15—Installation of Caisson Foundation (Suction Anchor)

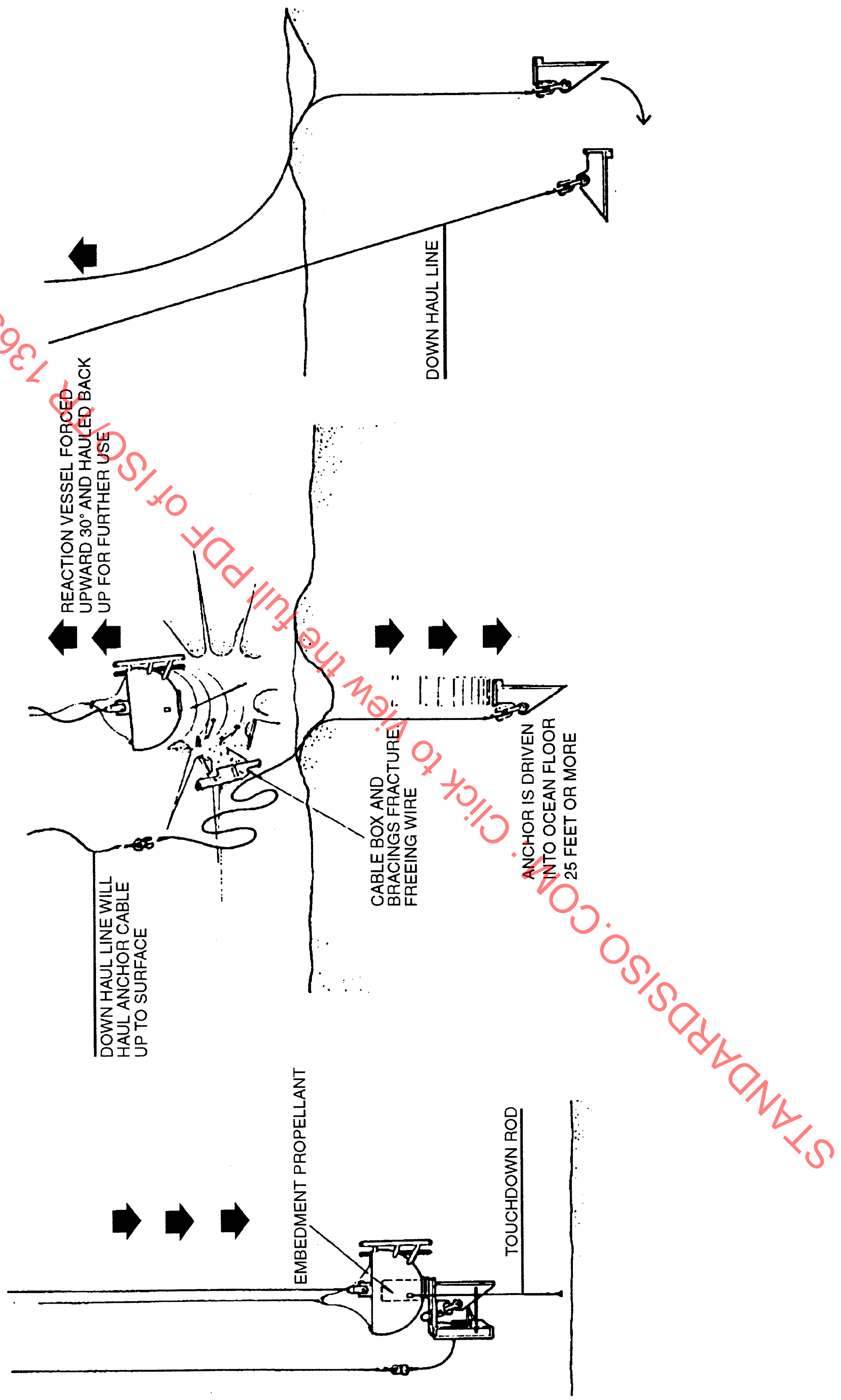


Figure 16—U.S. Navy Propellant Embedment Anchor

c. Caisson foundations (suction anchors).
d. Gravity anchors.
e. Propellant embedment anchors.

In selecting anchor options, considerations must be given to required system performance, soil conditions, reliability, installation, and proof loading.

3.3.1 Drag Embedment Anchor

Drag embedment anchor technology has advanced considerably in recent years. Engineering and testing indicate that the new generation of fixed fluke drag embedment anchors develop high holding power even in the soft soil conditions. High-efficiency drag embedment anchor is generally considered to be an attractive option for mooring applications because of its easy installation and proven performance. In fact, the majority of the existing permanent and mobile moorings use drag embedment anchors. The anchor section of a mooring line can be preinstalled and test loaded prior to platform installation.

3.3.2 Pile Anchor

A pile anchor's resistance to uplift and lateral loading is primarily a function of pile dimensions, the manner in which the pile is installed and loaded, and the type, stiffness, and strength of the soil adjacent to the pile. Horizontal capacity can be increased considerably by adding special elements such as skirts or wings to the pile top. Pile anchors can be designed to develop high lateral and vertical resistance, and be very stable over time. Piles are generally installed using driving hammers, although other methods such as jetting, and drilling and grouting have been used. Installation of jetted or drilled and grouted piles can be handled by a conventional drilling rig without major modifications. However, disturbance of soil during jetting and drilling operations should be carefully evaluated. API Recommended Practice 2A [2] provides detailed information on design and installation of driven piles.

3.3.3 Caisson Foundation (Suction Anchor)

A caisson foundation is installed by using a suction embedment technique and therefore is often called a suction anchor. A caisson foundation can take many forms, ranging from a gravity base with skirts to a no-ballast caisson that resists all applied loads by soil friction and lateral resistance.

Generally, a caisson is technically feasible for soft to medium hard soils. For very soft soils such as some Mississippi Delta areas, the caisson must extend so deeply into the soil to reach competent load bearing material that the structure becomes unwieldy and difficult to handle. For very hard soils, it may not be possible for the skirts to penetrate deeply enough to provide adequate in-place strength.

A caisson installation is illustrated in Figure 15. When lowered to the seafloor, the caisson will penetrate to a certain depth by its own weight and create a seal to allow the suction operation to commence. Water is evacuated from inside the caisson with a submersible or surface vacuum pump through an umbilical attached to the top of the caisson. This causes the pile to be anchored into the seabed. Embedment of the caisson is assumed to be completed prior to placing ballast material in the caisson's upper chambers.

3.3.4 Gravity Anchor

Gravity anchors are deadweight anchors that commonly consist of concrete or steel blocks, scrap metal, or other materials of high density. Design uplift capacity is dependent on the submerged weight of the anchor. Horizontal capacity is a function of the friction between the anchor and the soil, and shear strength of the soil beneath the anchor.

3.3.5 Propellant Embedment Anchor

The concept of a propellant embedment anchor was first developed by the U.S. Navy and used for relatively low capacity fleet moorings in shallow water. The system used by the Navy involves a fluke or plate that is lowered to the seabed with a mooring line. An explosive charge is then triggered from the surface forcing the plate a certain distance into the seabed. The mooring line is then pulled–tripping the plate into a horizontal position (Figure 16). This system could be advanced for high capacity mooring applications, and other concepts of propellant embedment anchor have been investigated. However, significant further development is required to advance this technology for high capacity mooring applications.

4 Environmental Criteria

4.1 ENVIRONMENTAL CONDITION

The industry recognizes two classifications of environmental condition when evaluating mooring systems: maximum design condition and maximum operating condition.

4.1.1 Maximum Design Condition

The maximum design condition is defined as that combination of wind, waves, and current for which the mooring system is designed. Selection of the maximum design conditions should be the responsibility of the owner. The design environmental criteria should be developed from the environmental information described in 3.2 and may also include a risk analysis where prior experience is limited. The risk analysis may include: historical experience, the planned life and intended use of the mooring, the consideration of safety of operating personnel, prevention of pollution, the estimated cost of the mooring designed to environmental conditions for several average expected recurrence intervals, the probability of mooring damage or loss when subjected to environmental conditions with various recurrence intervals, and the financial loss due to mooring failure.

4.1.1.1 Maximum Design Condition for Permanent Moorings

Experience with major fixed platforms in the Gulf of Mexico supports the use of 100-year oceanographic design criteria. In the absence of extended experience with permanent moorings, this criteria can be used for permanent floating operations with design lives comparable to fixed platforms. Risk analysis may justify either longer or shorter recurrence intervals. However, no less than a 100-year oceanographic design criteria should be considered where the design event may occur without warning while the platform is manned, and/or when there are restrictions, such as great flying distances, on the speed of personnel removal.

If the design life of the platform is substantially lower than that of fixed platforms, a shorter recurrence interval may be justified. In this case, the recurrence interval should be determined by a risk analysis taking into account the consequence of failure.

For a permanent operation with a mooring system that permits rapid disconnection of the production vessel from the mooring, the maximum design condition is the maximum environment in which the production vessel remains moored. However, the mooring alone (without the production vessel) should be able to withstand the maximum design environment for permanent moorings.

Mooring systems should be designed for the combination of wind, wave, and current conditions causing the extreme load in the design environment. In practice, this is often approximated by the use of multiple sets of design criteria. For example, in the case of a 100-year design environment, two sets of criteria are investigated: (a) the 100-year waves with associated winds and currents, and (b) the 100-year wind with associated waves and currents. The most severe directional combination of wind, wave, and current forces should be specified for the permanent installation being considered, consistent with the site's environmental conditions.

4.1.1.2 Maximum Design Conditions for Mobile Moorings

Maximum design conditions including the following:

a. Operations away from other structures. In general, moorings for mobile floating units such as MODUs operating away from other structures should use a maximum design environment with a return period of at least 5 years. Special attention should be given to operations in areas of tropical cyclones such as the Gulf of Mexico (hurricane) and the South China Sea (typhoon). These areas are characterized by a generally mild environment combined with severe storms during the cyclonic storm season. For operations out of the cyclonic storm season, the 5-year environment can be determined using the environmental data excluding tropical cyclones. For operations during the cyclonic storm season, the tropical cyclone data should be included.

On some special occasions, the return period can be reduced for operations during the tropical cyclone season if the following conditions are met:

1. A risk analysis is conducted to evaluate the consequences of a mooring failure. Such an analysis examines various scenarios of mooring failure, probability of occurrence of each scenario, and their impact on safety and the environment.
2. Operational personnel evacuation is planned and executed before the arrival of a tropical cyclone.
3. A weather forecast system with local environment feedback is available to provide accurate forecasting.
4. For drilling operations, the drilling riser is pulled before the arrival of a tropical cyclone.
5. There is no other structure within a few miles of the operation.

The reduced return period should be determined by the risk analysis, but it should not be less than one year.

b. Operations in the vicinity of other structures. The maximum design environment for mobile units operating in the vicinity of other structures should be determined by a risk analysis that evaluates the consequence of a mooring failure. However, it should have a return period of at least 10 years. In an area of tropical cyclones, for operations out of the cyclonic storm season, the 10-year environment can be determined using the environmental data excluding tropical cyclones. For operations during the cyclonic storm season, the tropical data should be included. An example of such an operation is a MODU with mooring lines deployed over a pipeline. Damage to the pipeline may occur if the anchors are dragged into the pipeline. Other examples include a drilling tender, a floater, or a service vessel moored next to a platform.

The 5-year and 10-year return period environments discussed in 4.1.1.2, items a and b should generally be determined by annual statistics. However, if the operating season is well defined and seasonal environmental data are sufficient to provide meaningful statistics, these environments can be determined by seasonal statistics.

4.1.2 Maximum Operating Condition

The maximum operating condition is defined as that combination of maximum wind, waves, and current in which production and/or drilling operations can be conducted. The operating environmental criteria should be known to the people responsible for the drilling or production operations in order that timely plans to suspend operations can be performed.

Generally these criteria are less severe than those for the maximum design conditions. However, in some FPS operations where the system is designed to continue production during a severe storm, the maximum operating condition will be the same as the maximum design condition.

4.2 ENVIRONMENTAL DATA

Collection and selection of environmental data for a floating platform's mooring system is the responsibility of the owner. Experienced specialists should be consulted when defining the pertinent oceanographic and meteorological conditions for a site. The dynamic nature of a floating platform and its mooring system requires that the designer work closely with those specialists to develop the data and interpretations in the form needed for the particular design/ analysis to be used. Effects of directionality should, in particular, be considered.

Statistical models are essential for adequately describing environmental parameters. Recognized statistical methods and models should be applied to the assessment of maximum design and operating conditions. Models leading to the design responses of interest should consider the jointly distributed environmental phenomena. Environmental data, such as wind, wave, current, and tide, have site-specific relationships governing their interaction. The commonly used assumption of taking the combined maximum of each usually produces overly conservative designs. When collecting data, the various relationships should be included, if possible. Of particular importance are the wind/wave, wave height/period, and wave/current relationships and their relative directions. Design levels obtained from joint probability distributions of all relevant phenomena should be based on the design service life and risk analysis as described in 4.1.

The directions of the various environmental phenomena should also be considered. This is especially important at single point moorings. If all the environmental phenomena come from the same direction, and the moored vessel aligns with this direction, then the resulting force on the system is usually minimized. However, when waves act at high angles to winds or currents and the moored vessel is not aligned with the predominant environmental force, the resulting forces are generally higher.

4.3 WIND

Wind is a significant design factor. The wind conditions used in a design should be appropriately determined from collected wind data and should be consistent with other environmental parameters assumed to occur simultaneously.

Two methods are generally used to assess effects of wind for design. These include the following:

a. Wind forces are treated as constant and calculated on the basis of the 1-minute average velocity.
b. Fluctuating wind force is calculated on the basis of a steady component, based on the 1-hour average velocity, plus a time-varying component calculated from a suitable empirical wind gust spectrum.

The choice of the treatment depends on the system parameters and goals of the analysis. Either approach may give more severe loads, depending on the system moored and the wind spectrum used. The time varying component in method b is also known as low-frequency wind force. Although methods for predicting low-frequency wind force have been extensively studied, there is still a substantial degree of uncertainty in the estimation, particularly in the wind energy spectrum that is derived from measured wind data. Therefore, caution should be exercised in selecting the spectrum to ensure that it adequately represents wind energy at the low frequencies typically associated with natural frequencies of moored structures. A more detailed discussion on low-frequency wind forces is presented in A.4.

The design wind speed should refer to an elevation of 33 feet (10 meters) above still water level. Rapid changes in wind direction and the resulting dynamic loads should be considered in the design of single point mooring systems.

4.4 WAVES

Wind-driven waves are a major source of environmental forces on offshore facilities. Such waves are random in nature, vary in height and length, and may approach a platform from more than one direction simultaneously. For these reasons the intensity and distribution of the forces applied by waves are difficult to determine. Because of the complex nature of the technical factors that must be considered in developing wave-dependent criteria for the design of moored installations, specialized knowledge in the fields of meteorology, oceanography, and hydrodynamics should be applied.

Because of the random nature of the sea surface, the sea-state is usually described in terms of a few statistical wave parameters such as significant wave height (average of the highest one third wave heights), spectral peak or significant wave period, spectral shape and directionality. Other parameters of interest can be derived from these. Refer to API Recommended Practice 2A for a more detailed discussion on sea-state representation.

The design significant wave height should be determined based on the design recurrence interval and wave data. The wave data used to determine the design should include available measured data and storm hindcast data as well as ship observations. The wave height versus wave period relationships for the design sea-state should be accurately determined from oceanographic data for the area of operation. The period can significantly affect surge and sway amplitudes and mean drift forces. For cases where measured data are not available, Figure 17 provides significant wave period versus wave height relationships for wind-generated waves and for predominant swell conditions.

4.5 CURRENT

The most common categories of currents are: (a) tidal currents (associated with astronomical tides), (b) circulational currents (associated with oceanic-scale circulation patterns), (c) storm-generated currents, (d) loop and eddy currents, and (e) soliton currents. The vector sum of these currents is the total current, and the speed and direction of the current at specified elevations are represented by a current profile. The total current profile associated with the sea-state producing the extreme waves should be specified for mooring design. The frequency of occurrence of total current speed and direction at different depths for each month and/or season may be useful for planning operations. In certain geographic areas, current force can be the governing design load. Consequently, selection of appropriate current profiles requires careful consideration.

4.6 WATER DEPTH AND TIDE

Tidal components for design include astronomical, wind, and pressure-differential tides. Tides and slope and direction of the ocean floor should be included in the determination of the design water depth.

4.7 SOIL CONDITIONS

Bottom soil conditions should be determined for the intended site to provide data for the anchoring system design.

4.8 ATMOSPHERIC ICING

There are several sources of superstructure icing. Sea-spray icing can effect the structure to a height of 50 feet above the waterline. Ice formed from fog or rain can accumulate on any exposed surface. Superstructure icing should be considered in platform wind force calculations.

4.9 MARINE GROWTH

The type and accumulation rate of marine growth at the design site may affect weight, hydrodynamic diameters, and drag coefficients of vessel members. These should be taken into consideration in the design.

5 Environmental Forces and Vessel Motions

5.1 BASIC CONSIDERATIONS

The purpose of this section is to describe methods for calculating forces that act on floating vessel due to environmental effects, such as waves, winds, currents, and so forth. Forces due to platform motion responses are also significant and are discussed herein. Environmental parameters needed

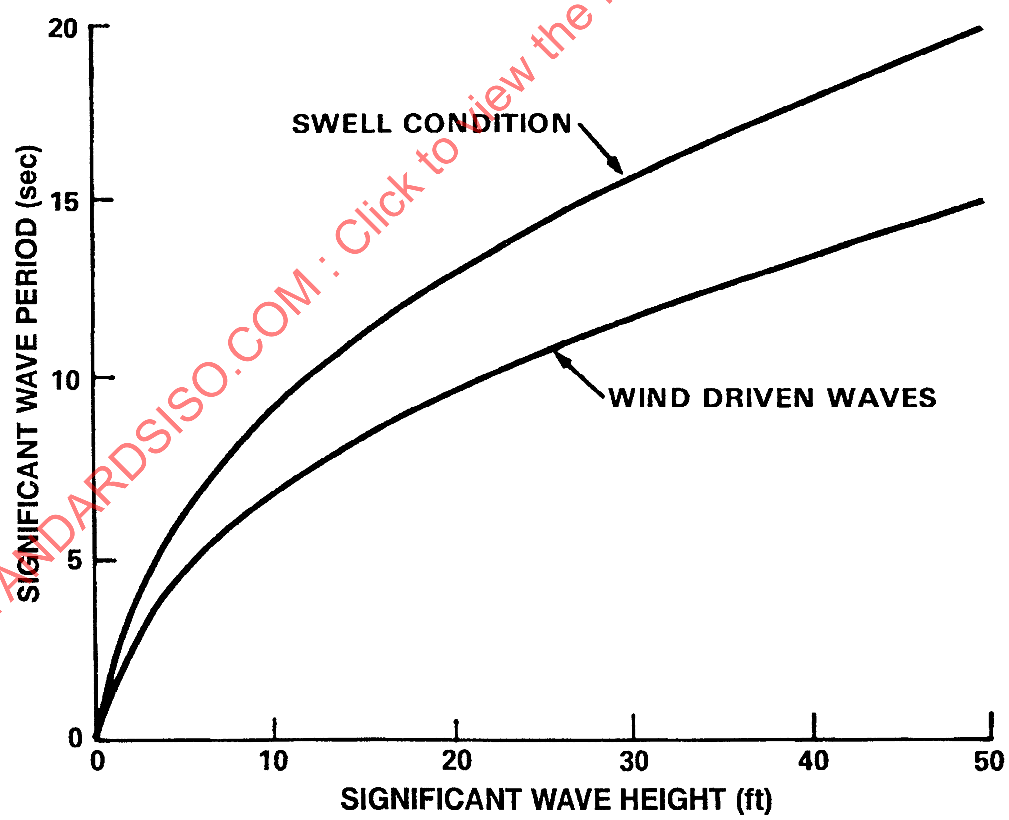


Figure 17—Wave Height/Wave Period Relationship

for these calculations are defined in Section 4. Methods for estimating platform motions and mooring system loads caused by these environmental forces are given in Section 7.

Environmental forces should be calculated at the following three distinct frequency bands to evaluate their effects on the system:

a. Steady forces such as wind, current, and wave drift are constant in magnitude and direction for the duration of interest.
b. Low-frequency cyclic loads can excite the platform at its natural periods in surge, sway, and yaw: typical natural periods range from 1 to 3 minutes.
c. Wave frequency cyclic loads are large in magnitude and are the major contributor to platform member forces and mooring system forces. Typical wave periods range from five to twenty seconds.

5.2 GUIDELINES FOR THE EVALUATION OF ENVIRONMENTAL FORCES AND VESSEL MOTIONS

Environmental forces and vessel motions can be determined either by model testing or calculation. Guidelines for evaluating these values by various approaches are provided in API Recommended Practice 2T [3]. The wind spectrum recommended for the evaluation of low-frequency wind forces can be found in API Recommended Practice 2A [2].

5.3 SIMPLIFIED METHODS

Design equations and curves for a quick evaluation of environmental forces and vessel motions are provided in Appendix A. These simplified analytical tools were developed primarily for the analysis of mobile moorings. They may be used for preliminary designs of permanent moorings if more accurate information is not available at the early stage of the design process and if the limits for these tools are not exceeded. For the final design of permanent moorings, however, the more rigorous approaches as outlined in 5.2 are recommended. Simplified methods are available for the following force components:

a. Current forces for ship-shaped and semisubmersible hulls.
b. Mean wave drift forces and low-frequency motions for ship-shaped and semisubmersible drilling vessels.
c. Steady wind force.
d. Wind and current forces for large tankers.
e. Forces due to oblique environment.

Appendix A also provides some general discussion on the evaluation of environmental forces and vessel motions.

6 Design Criteria

6.1 BASIC CONSIDERATIONS

In this section, design criteria are established for the following conditions:

a. Intact condition. This is the condition in which all mooring lines are intact.
b. Damaged condition. This is the condition in which the vessel settles at a new equilibrium position after a mooring line breakage.
c. Transient condition. This is the condition in which the vessel is subjected to transient motions (overshooting) after a mooring line breakage before it settles at the new equilibrium position.

Also, two mooring analysis methods, dynamic analysis and quasi-static analysis, can be used for mooring design. Different factors of safety were established for the two methods to account for the different levels of uncertainty of two approaches. Section 7 gives a more detailed discussion on the differences between dynamic analysis and quasi-static analysis.

Following are guidelines for the type of analysis to be conducted and conditions to be analyzed for various operations:

Type of Mooring	Analysis Method	Conditions to be Analyzed
Permanent Mooring		
Preliminary design	Quasi-static or dynamic	Intact/damaged
Final design	Dynamic	Intact/damaged/ transient
Fatigue Design	Dynamic	Intact
Mobile Mooring		
Away from other structures	Quasi-static or dynamic	Intact
Mooring lines over pipeline	Quasi-static or dynamic	Intact/damaged
Vessel next to a platform	Quasi-static or dynamic	Intact/damaged/ transient

In some operations, special operation procedures are used to reduce mooring line tensions and vessel offsets. For example, it is a common practice for MODUs to slacken a couple of leeward lines in a storm environment to reduce line tension. Such a practice may also be carried out in an operating environment with high currents to reduce vessel offset. Operational aspects of this nature can be taken into consideration in the analysis if such a practice is well defined in the operation manual and is routinely carried out by trained operation personnel. On the other hand, if the vessel operates in an area where a storm can come without much warning or can change direction suddenly, such a practice is not practical and therefore should not be incorporated into the mooring analysis.

In the areas of hurricanes or tropical cyclones, the drilling riser of a MODU is often pulled, and all the mooring lines are slackened before arrival of a hurricane or tropical cyclone. This operational practice can also be taken into consideration in the mooring analysis.

Another special operation procedure is to optimize the windward lines such that all the windward lines share approximately the same load. In general, this practice should not be considered in the analysis for several reasons. First, windward lines cannot be optimized in a confused sea as in an analysis. Second, it is dangerous to attempt optimizing windward lines in a severe storm where releasing the winch brakes or the chain stoppers can lead to serious consequences. Third, operation experience indicates that such a practice is seldom carried out even though it is stated in the operation manual. One exception is that for certain operations where storms develop slowly and maintain a fairly constant direction, a well trained crew will have sufficient time to establish favorable moorings for the impending storms. In this case some consideration can be given to windward line optimization in the mooring analysis.

6.2 OFFSET

6.2.1 Definition of Mean Offset

The mean offset is defined as the vessel displacement due to the combination of current, mean wave drift, and mean wind forces.

6.2.2 Definition of Maximum Offset

The maximum offset is the mean offset plus appropriately combined wave frequency and low-frequency vessel motions. Maximum offset can be determined by the following procedure.

Let:

S_{mean} = mean vessel offset.
S_{max} = maximum vessel offset.
S_{wfmax} = maximum wave frequency motion.
S_{wfsig} = significant wave frequency motion.
S_{lfmax} = maximum low-frequency motion.
S_{lfSig} = significant low-frequency motion.

(1) $S_{lfmax} > S_{wfmax}$, then

$$S_{max} = S_{mean} + S_{lfmax} + S_{wfsig} \tag{6.1}$$

(2) $S_{wfmax} > S_{lfmax}$, then

$$S_{max} = S_{mean} + S_{wfmax} + S_{lfsig} \tag{6.2}$$

A parametric study has been performed to assess the risk level associated with this method of combining wave and low-frequency motions [14]. The chance of exceeding the combined motions defined above was estimated using a probabilistic approach for different hull forms, water depths, environments, and types of mooring. The results of the study indicate that the combined low and wave frequency motions defined in this manner would be exceeded on the average of once in the specified storm duration.

An alternative to this approach is a time domain simulation of vessel motions or model testing for the specified storm duration. If this approach is used, the simulation or model testing should be of sufficient length to establish reasonable confidence bounds for the expected maximum response in the storm duration. Typically specified storm duration should be simulated several times, and statistical fitting techniques should be used to establish the expected maximum response.

The above discussion applies to the intact and damaged conditions. For the transient condition, maximum offset is defined in 7.3.

6.2.3 Offset Limits

The offset of the vessel from the sea floor well location must be controlled to prevent damage to drilling or production risers.

6.2.3.1 Drilling Operation

The mean offset should be controlled under the drilling operating condition because of its direct relevance to the mean ball joint angle of the drilling riser. The allowable mean offset should be determined by a drilling riser analysis, allowable mean offsets depend on many factors such as water depth, environment, and riser system. For the drilling operation to continue, allowable mean offsets normally fall in a range of 2 percent to 4 percent of water depth. Generally the lower bound applies to deepwater (2000-3000 feet) operations, and the upper bound applies to shallow water (below 300 feet) operations.

The maximum offset should be controlled under the maximum storm condition for the drilling operation to prevent damage to the mechanical stop in the ball joint below the drilling riser. The allowable maximum offset should be determined by a drilling riser analysis. Allowable maximum offsets depend on many factors such as water depth, environment, and riser system. They normally fall in a range of 8 percent to 12 percent of water depth. Generally the lower bound applies to deepwater (2000-3000 feet) operations, and the upper bound applies to shallow water (below 300 feet) operations. Analysis of vessel offset for drilling operations is required for the intact condition only. Guidelines for the analysis of marine drilling risers are given in API Recommended Practice 16Q [7].

6.2.3.2 Production Operations

There are basically four types of production risers:

a. Rigid riser.
b. Flexible riser.
c. Hybrid riser.
d. Riser integrated with single point mooring.

Rigid risers are tensioned from the vessel and can be either integral or nonintegral. An integral top tensioned riser is a multibore riser in which all fluid connections are made with a single coupling. A nonintegral top tensioned riser consists of individual stands of pipe with individual connections for each flow path. A flexible riser consists of flexible pipe that hangs in a catenary from the floating production vessel to the seafloor. Rigid and flexible risers can be combined into a hybrid production riser. A hybrid riser consists of a buoyant stand of rigid riser terminating at a point below the water surface. Flexible risers span the gap between the top of the rigid riser and the vessel. The fourth type of production riser includes those that integrate the production risers with the single point moorings such as a CALM or SALM system.

The offset limits for the vessel under the maximum design and operating conditions should be determined by a production riser analysis in conjunction with mooring analysis. Maximum allowable offsets for rigid risers normally fall in a range of 8 percent to 12 percent of water depth. Generally the lower bound applies to deepwater (2000-3000 feet) operations, and the upper bound applies to shallow water (below 300 feet) operations. This offset limitation often dictates that rigid production risers be disconnected during severe storms.

Maximum allowable offsets for deepwater (2000-3000 feet) flexible risers normally range from 10 percent to 15 percent of water depth, depending on the riser configuration. The maximum allowable offsets for shallow water (below 300 feet) flexible risers normally range from 15 percent to 30 percent of water depth. Flexible risers are usually designed to survive the maximum design environment while remaining connected to the vessel.

Design guidelines for production risers can be found in API Recommended Practice 17A and API Recommended Practice 17B.

6.2.3.3 Tender Operations

Offsets for vessels moored next to another installation are limited by the clearance between the two units. The offsets under the intact, damaged, and transient conditions should be limited to avoid contact of the vessel or its mooring with the nearby installation. Some margin of safety in the minimum clearance should be reserved.

6.3 LINE TENSION

6.3.1 Definition of Mean Tension

The mean tension is the line tension corresponding to the mean offset of the vessel.

6.3.2 Definition of Maximum Tension

The maximum tension is the mean tension plus appropriately combined wave frequency and low-frequency tensions. Maximum tension can be determined by the following procedure.

Let:

T_{max} = maximum tension.
T_{mean} = mean tension.
T_{wfmax} = maximum wave frequency tension.
T_{wfsig} = significant wave frequency tension.
T_{lfmax} = maximum low-frequency tension.
T_{lfsig} = significant low-frequency tension.

(1) $T_{lfmax} > T_{wfmax}$, then

$$T_{max} = T_{mean} + T_{lfmax} + T_{wfsig} \tag{6.3}$$

(2) $T_{wfmax} > T_{lfmax}$, then

$$T_{max} = T_{mean} + T_{wfmax} + T_{lfsig} \tag{6.4}$$

Similar to the case of vessel motions, the combined low and wave frequency tension defined in this manner would be exceeded on the average once in the specified storm period used in developing maximum low or wave frequency tensions.

The above discussion applies to the intact and damaged conditions. For the transient condition, the maximum tension is defined in 7.3. A tension limit can be expressed as a percentage of the nominal strength of the mooring component. The nominal strength of the wire rope can be taken as the catalog (or certified) break strength (CBS), provided it is new or in like-new condition. The nominal strength of chain may be taken as the break test load (BTL), provided the chain is new or in like-new condition. The breaking strength of a used chain or wire rope can be lower than its rated values and it should be determined by test or inspection.

Tension limits and equivalent factors of safety for various conditions and analysis methods are as follows:

Condition	Analysis Method	Tension Limit (Percent of Breaking Strength)	Equivalent Factor of Safety
Intact	Quasi-static	50	2.0
Intact	Dynamic	60	1.67
Damaged	Quasi-static	70	1.43
Damaged	Dynamic	80	1.25
Transient	Quasi-static	85	1.18
Transient	Dynamic	95	1.05

The above criteria apply to both maximum design and operating environments. This is a departure from the previous practice stated in API Recommended Practice 2P where a lower tension limit is recommended for the maximum operating environment. The rationale for the departure is as follows:

a. For operations such as drilling where the maximum operating environment is significantly lower than the maximum design environment, tensions need to be checked for the maximum design environment only. If the criteria are met, tension is not a concern for the milder maximum operating environment.

b. For operations such as certain floating production operations where production will continue under the maximum storm condition, the maximum design and operating environments are the same, and the same tension criteria should apply.

The tension limits stated above are intended for moorings that are properly maintained and periodically inspected, and have connecting hardware with breaking strengths equivalent to the mooring lines.

6.4 STATISTICS OF PEAK VALUES

In Equations 6.1 to 6.4, significant and maximum motion/tension values are used. These values can be calculated from the rms (root mean square) values using the following equations:

$$Sig.\ Value = 2\ (rms\ value) \tag{6.5}$$

$$Max.\ Value = \sqrt{2(\ln N)}\ (rms\ value) \tag{6.6}$$

$$N = T/T_a \tag{6.7}$$

Where:

T = specified storm period in seconds.
T_a = average zero crossing period in seconds.

For low-frequency components, T_a can be taken as the natural period of the vessel T_n which can be estimated by Equation 6.8.

$$T_n = 2\pi\sqrt{m / k} \tag{6.8}$$

Where:

m = vessel mass including added mass in slug.
k = mooring system stiffness in lbs/feet taken at the vessel's mean position.

A minimum of 3 hours should be specified for the storm period in Equation 6.7. The 3-hour period is typical for areas with short storms. For areas with long storm durations, for example the monsoon area, longer storm period should be specified.

Equation 6.6, which is based on a narrow band Gaussian process with Rayleigh distributed peaks, may not always yield conservative predictions of maximum value [45]. Catenary mooring line tensions, in particular, tend to be non-Gaussian and may have extreme values in excess of those predicted by this equation. An alternative approach is model testing or time domain simulation for the specified storm duration. If this approach is used, the simulation or model testing should be of sufficient length to establish reasonable confidence bounds for the expected maximum response in the storm duration. Typically specified storm duration should be simulated several times, and statistical fitting techniques should be used to establish the expected maximum response.

6.5 LINE LENGTH

If drag anchors are used, the outboard mooring line length should in general be sufficient to prevent anchor uplift under conditions as specified in 6.1. This requirement is especially important for anchors in sand and hard soil where anchor penetration is shallow. For certain modern high efficiency anchors in soft clay, however, shorter line lengths can be used if anchor tests and field experience indicate high vertical resistance of these anchors. Guidelines for the use of drag anchor to resist vertical loads are provided in Appendix B.

Shorter line lengths can be used for moorings with other anchoring systems such as pile anchor which can resist substantial vertical pulls.

6.6 HOLDING POWER OF ANCHORING SYSTEMS

6.6.1 Drag Anchor

The holding capacity of a drag anchor in a particular soil condition represents the maximum horizontal steady pull that can be resisted by the anchor at continuous drag. This load includes the resistance to the chain or wire rope in the soil for an embedded anchor, but excludes the friction of the chain or wire rope on the seabed.

Drag anchor holding capacity is a function of several factors, including the following:

a. Anchor type—Fluke area, fluke angle, fluke shape, anchor weight, tripping palms, stabilizer bars, etc. Figure 18 shows drag anchors commonly used by the offshore industry.
b. Anchor behavior during deployment—Opening of the flukes, penetration of the flukes, depth of burial of the anchor, stability of the anchor during dragging, soil behavior over the flukes, etc.

Due to the wide variation of these factors, the prediction of an anchor's holding power is difficult. Exact holding power can only be determined after the anchor is deployed and test loaded.

Anchor performance data for the specific anchor type and soil condition should be obtained if possible. In the absence of credible anchor performance data, Figures 19 and 20 may be used to estimate the holding power of anchors commonly used to moor floating vessels.

Figures 19 and 20 are reproduced from Techdata sheet 83-08R [12] except that the holding capacity curves for the Moorfast (or Offdrill II) and Stevpris anchor were upgraded. The upgrading of these two curves was based on model and field test data and field experience acquired in recent years. The design curves presented in these two figures represent in general the lower bounds of the test data. The tests used to develop the curves were performed at a limited number of sites. As a result, the curves are for use in generic soil types such as soft clay and sand. Recent studies indicate, however, that several parameters such as soil strength profile, lead line type (wire rope versus chain), cyclic loading, and anchor soaking may significantly influence anchor performance in soft clay. Also, some high efficiency anchors have demon-

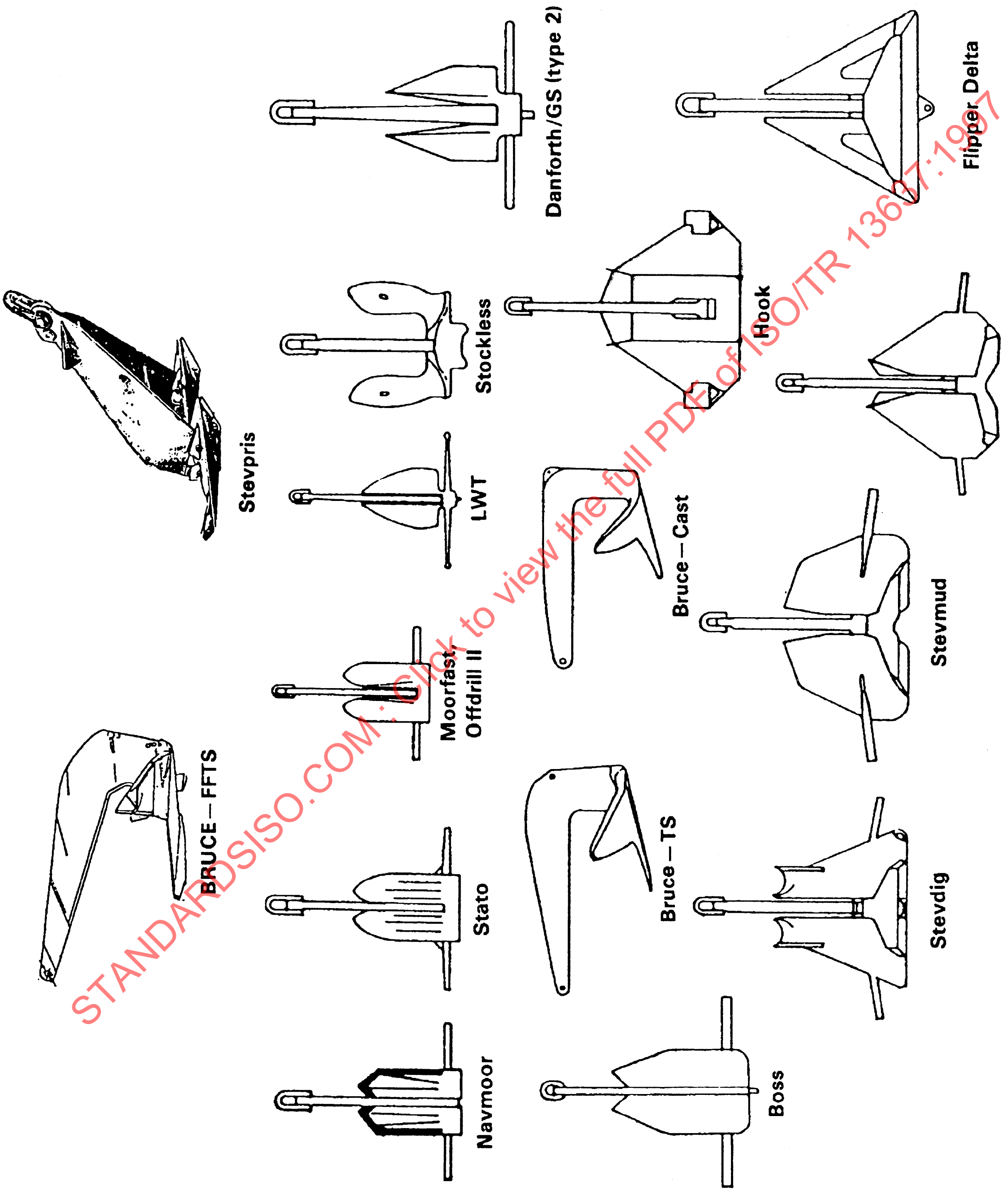
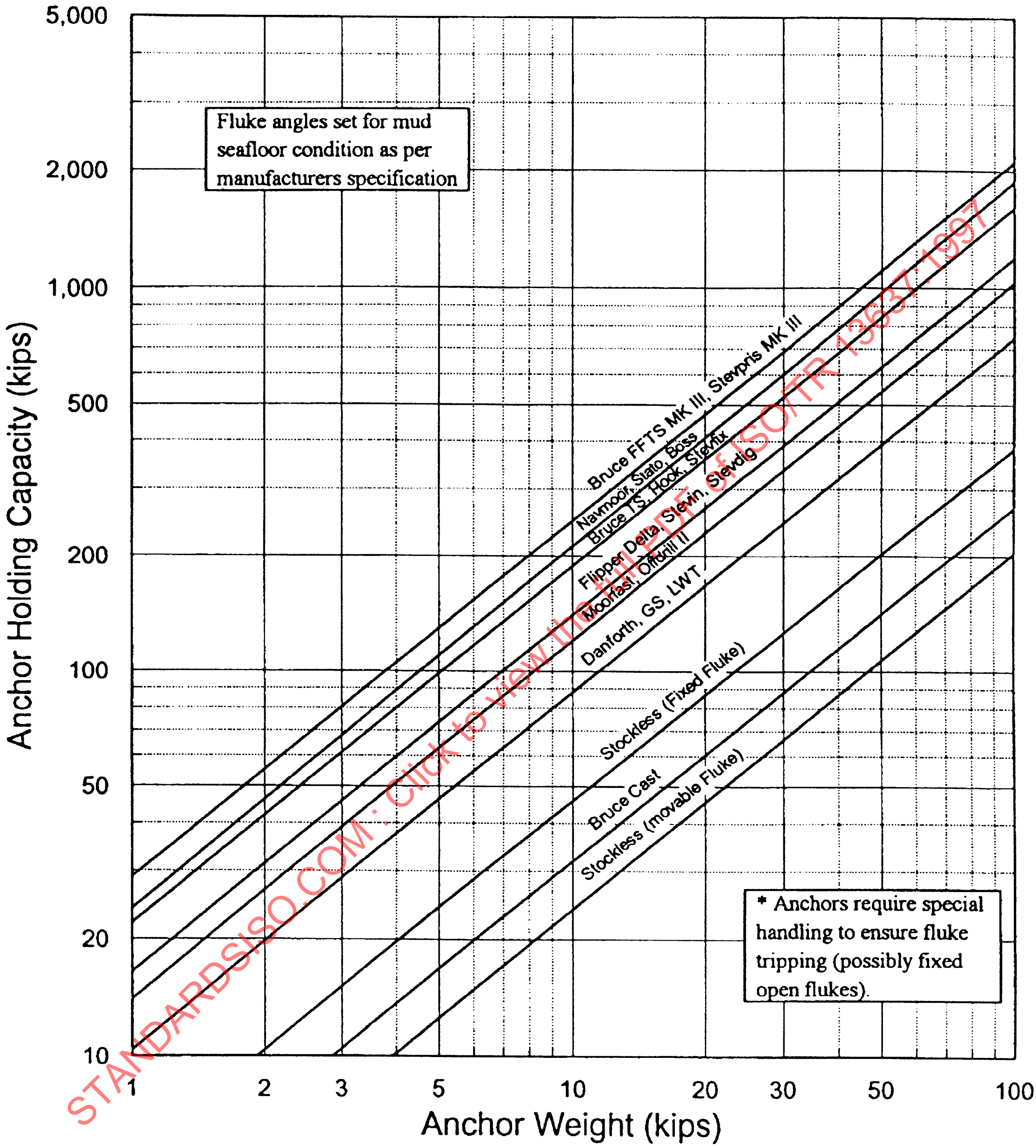
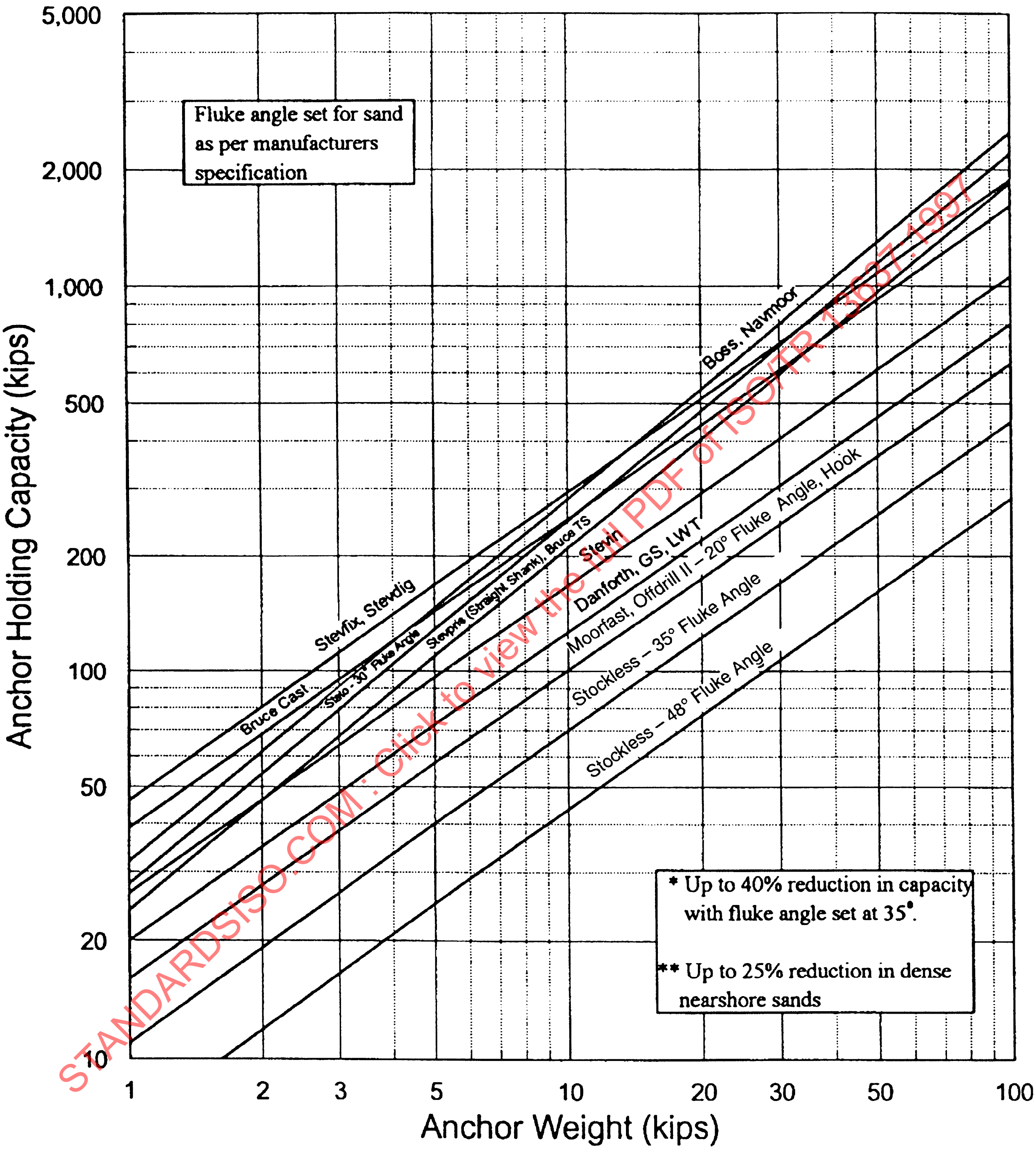


Figure 18—Drag Embedment Anchors



Note: This figure was reproduced from Techdata Sheet 83, "Drag Embedment Anchors for Navy Moorings," Naval Civil Engineering Laboratory, 1987, except that the holding capacity curves for the Moorfast (or Offdrill II) and the Stevpris anchor were upgraded. The upgrading of these two curves was based on model and field test data and field experience acquired in recent years. The design curves in the figure represent in general the lower bounds of the test data. They reflect data valid for anchor designs as of 1987. New anchor designs have since been developed. However, performance data for these new designs were insufficient and therefore their design curves were not included. Some guidelines for the performance evaluation of late anchor models are provided in B.8 of Appendix B. The design curves do not include a factor of safety.

Figure 19—Anchor System Holding Capacity in Soft Clay



Note: This figure was reproduced from Techdata Sheet 83, "Drag Embedment Anchors for Navy Moorings," Naval Civil Engineering Laboratory, 1987. The design curves in the figure represent in general the lower bounds of the test data. They reflect data valid for anchor designs as of 1987. New anchor designs have since been developed. However, performance data for these new designs were insufficient and therefore their design curves were not included. The design curves do not include a factor of safety.

Figure 20—Anchor System Holding Capacity in Sand

strated substantial resistance to vertical load in soft clay. Furthermore, there are new versions of high efficiency anchors that are not covered by these two figures. These issues are addressed in Appendix B.

Factors of safety for drag anchors are provided below.

	Quasi-Static Analysis	Dynamic Analysis
Permanent Mooring		
Intact condition	1.8	1.5
Damaged condition	1.2	1.0
Transient condition	not required	not required
Temporary Mooring		
Intact condition	1.0	0.8
Damaged condition	not required	not required
Transient condition	not required	not required

Note that the holding capacity curves in Figures 19 and 20 do not include a factor of safety. The factors of safety for anchor loads are substantially lower than those for line tensions, especially for temporary moorings. The rationale is to have the anchor moved instead of the mooring line broken in the event of mooring overload. Anchor movements of the most loaded lines would normally cause favorable redistribution of the mooring loads among the mooring lines resulting in lower line tensions and anchor loads for these lines. This would help the mooring system survive storm environments exceeding the maximum design environment.

6.6.2 Chain and Wire Rope

The holding capacity from friction of chain and wire rope on the seafloor may be estimated using Equation 6.9.

$$P_{cw} = fL_{cw}W_{cw} \tag{6.9}$$

Where:

P_{cw} = Chain or wire rope holding capacity, lb.

f = Coefficient of friction between chain or wire rope and the ocean bottom, dimensionless.

L_{cw} = Length of chain or wire rope in contact with the ocean bottom, feet.

W_{cw} = Submerged unit weight of chain or wire rope, lb/feet.

The coefficient of friction, f, depends upon the actual ocean bottom at the anchoring location and type of mooring line. Generalized friction factors for chain and wire rope are given in the following table. The starting friction factors are normally used to compute the holding power of the line and the forces on the line during deployment. These coefficients can be used for various bottom conditions such as soft mud, sand, and clay.

	Coefficient of Friction (f)	
	Starting	Sliding
Chain	1.0	0.7
Wire Rope	0.6	0.25

6.6.3 Pile Anchor

Anchor piles should account for pile bending stresses as well as ultimate lateral pile capacity. An analysis method capable of accounting for both aspects of behavior is to model the pile as a beam column on an inelastic foundation. The inelastic foundation can be modeled using soil resistance-deflection (d-y) curves, which are described for various soils in API Recommended Practice 2A [2].

Pile embedment should also be sufficient to develop the axial capacity to resist vertical loads with an appropriate factor of safety. Guidelines for the design of axially loaded piles are also provided in API Recommended Practice 2A.

6.6.4 Caisson Foundation and Gravity Anchor

Design criteria for caisson foundations and gravity anchor are still in a state of development and therefore are not addressed in this recommended practice. References 25-28 provide some useful information for the design of caisson foundations.

6.6.5 Mooring Test Load

After installation, the mooring should be test loaded to ensure adequate holding capacity of the anchoring system, eliminate slack in the grounded mooring lines, and detect damage to the mooring components during installation.

For permanent moorings, all the mooring lines should be test loaded to the maximum storm load determined by a dynamic mooring analysis for the intact condition, except for certain high efficiency anchors in soft clay as discussed in B.4 of Appendix B.

Test loads for temporary moorings are more difficult to determine and require substantial judgement because of operational and equipment restraints. For instance, the test load of a MODU is often limited by the existing winch capacity. In some areas with soft seafloor, a high efficient anchor may penetrate too deep if the mooring test load is high, and anchor retrieval becomes very difficult or impossible. On the other hand, a certain level of test load is necessary to ensure adequate anchor holding capacity.

Preferably the test load at winch for temporary moorings should be at the same level as the maximum line tension under the maximum design condition. If this cannot be achieved because of operational constraints, the mooring test load should not be less than the mean line tension under the maximum design condition, or the maximum line tension under the maximum operating condition, whichever is higher. The tensions mentioned above are for an intact mooring. Further reduction in test load may be considered provided a site-specific anchor performance analysis indicates that anchor performance will be satisfactory with the reduced test load.

6.7 THRUSTER ASSISTED MOORING

Some floating vessels are equipped with thruster or DP (dynamic positioning) assisted moorings. The use of thrusters to help resist the steady environmental forces acting on the unit for the maximum design condition can be considered using the following guidelines:

Mooring System Status	Manual Remote Control	Automatic Remote Control
All lines intact	70 percent of net thrust after failure of any one thruster	Net thrust after failure of any one thruster
One line broken	70 percent net thrust from all thrusters	Net thrust from all thrusters

The calculation of net thrust should be based on effective bollard pull at zero speed. It should also account for any directional restrictions, thruster/hull and thruster/thruster interference effects. Guidelines for determining net thrust from thrusters are provided in Appendix C.

The above criteria is based on redundant control systems for the automatic remote control case, and redundant power supply for both manual and automatic remote control cases. For floating vessels without redundant power supply, mooring line tensions for the intact condition should not exceed the tension limits as specified in 6.3 in a power blackout (no thrust from thrusters). For floating vessels with a single automatic control system, the criteria for the manual control case should apply.

6.8 FATIGUE LIFE

Fatigue design is required for permanent moorings only. A predicted mooring component fatigue life of three times the design service life is recommended. The factor of three accounts for uncertainties in lifetime load prediction, T-N curve data scatter, approximations in linear damage theory, and the effect of many mooring components being connected in series. This factor should be used in conjunction with the component T-N curve corresponding to a lower bound usually defined as the lower bound of a two-sided, 95 percent prediction interval (2.5 percent probability of fatigue resistance exceedance).

A nominal mooring component fatigue life can be predicted using the fatigue analysis procedure defined in 7.5. T-N curves for various mooring components should be based on fatigue test data for these components and a regression analysis. In the absence of more accurate data, T-N curves presented in Equation 6.10 can be used for calculating nominal tension fatigue lives of wire rope, chain, and connecting links.

$$NR^M = K \qquad (6.10)$$

Where:

N = number of cycles.

R = ratio of tension range (double amplitude) to nominal breaking strength.

M = slope of T-N curve.

K = intercept of T-N curve.

Let:

Lm = ratio of mean load to CBS (catalog breaking strength) for wire rope.

M and K values are provided below:

Component	M	K
Common chain link	3.36	370
Baldt and Kenter connecting link	3.36	90
Six/multi strand rope	4.09	$10^{(3.20-2.79\,Lm)}$
Six/multi strand rope (Lm = 0.3)	4.09	231
Spiral strand rope	5.05	$10^{(3.25-3.43\,Lm)}$
Spiral strand rope (Lm = 0.3)	5.05	166

Recent research indicates that mean load has a significant influence on wire rope fatigue life and therefore should be included in the design curve equations. A mean load of 0.3 CBS is considered to be representative for conventional mooring systems. For wire rope fatigue analysis, the following methods can be considered to account for the mean load effect:

a. For each sea-state, determine the mean load and the corresponding design curve that is then used to calculate the fatigue damage for that sea-state. This requires using different design curves for different sea-states.

b. Determine the average mean load for sea-states causing significant fatigue damage and use the design curve for the average mean load for all sea-states.

c. Use the design curve for a mean load of 0.3 CBS for conventional mooring systems. For a taut leg mooring or TLP tether system, method a or b should be used.

Among the three methods, method a is most accurate but requires more computational effort. If method b and c are used, a sensitivity study should be performed to ensure these simplified approaches produce conservative predictions.

The T-N curve for chain is based on test data for links of ORQ grade. Test data for links of higher grade (Grade 4, ORQ + 20 percent) are rare, and T-N curves for higher grade chains cannot be developed at this point. In the absence of better information, the T-N curves in Figure 21 can be used for higher grade chains. However, the nominal breaking strength should be taken as that of ORQ chain of the same size. This T-N curve is based on testing of sample links with tight studs for which fatigue failure usually occurs at the bend zone of the link. For chain links with loose studs, however, this curve may not apply since higher stresses may develop at the stud foot print resulting in fatigue failure at this location. In this case, the fatigue resistance can be substantially lower.

A new type of chain, studless chain, has gained wide acceptance in recent years. Results of finite element stress analysis for studless chain and limited data from studless chain fatigue testing in air indicate that studless chain has similar fatigue resistance as stud link chain. Therefore the T-N design curve for common chain link in Figure 5.4 may also be applicable to studless chain. This, however must be con-

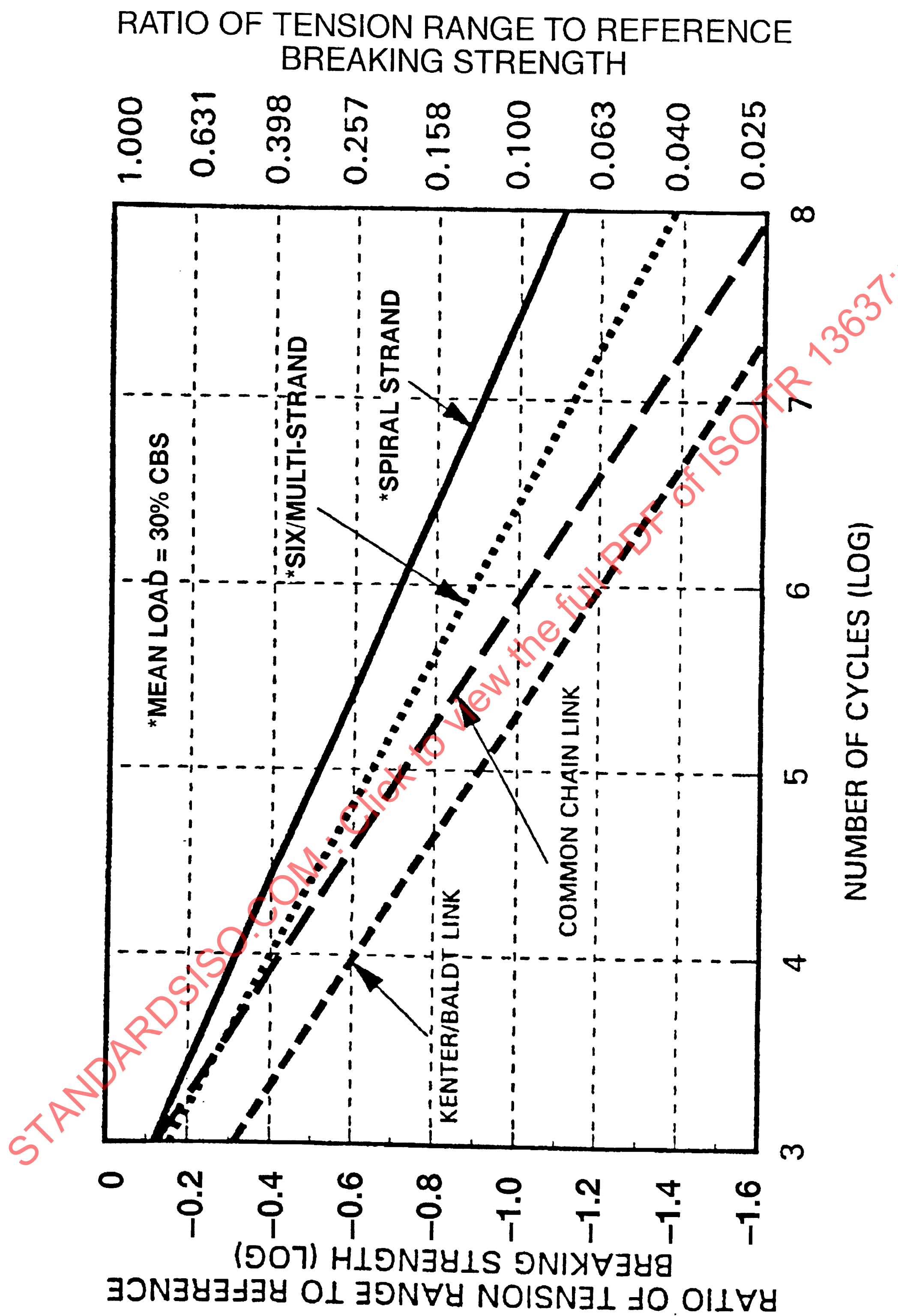


Figure 21—Fatigue Design Curves

firmed by additional studless chain fatigue testing and service experience in the future.

The T-N curves for wire rope are based on test data from various sources, including data from a joint industry project on wire rope endurance [48]. The data were obtained from tests conducted for six strand, multi-strand, and spiral strand ropes. Effects of corrosion have not been included. Therefore the T-N curves are only good for wire ropes protected from corrosion. Elements for corrosion protection include galvanizing, jacketing, blocking compound, and zinc filler wires. Careful investigation considering the design life, inspection, and change-out strategy should be carried out to determine the combination of these elements needed for a specific project.

The fatigue design curve for Kenter and Baldt connecting links is based on limited fatigue test data including 27 data for Kenter and 27 data for Baldt. The curve represents the lower bound of all test data except for one Kenter data, and it has the same slope of the curve for common chain link. The nominal breaking strength used for fatigue calculation of connecting links should be the same as that for the ORQ chain of the same size.

Data for other types of connecting links are insufficient for generating design curves. Limited data indicate that the fatigue life of D-shackles is comparable to that of common links of the same size and grade, provided the shackle is machined fit with close tolerance, no cotter pin is used through shackle body, and the shackle is the narrow throat type.

Figure 21 presents the fatigue design curves for chain, connecting links, six/multi-strand rope, and spiral strand rope. The two curves for wire ropes are for a mean tension equal to 30 percent of the nominal breaking strength.

Data for bending-tension fatigue of chain and wire rope are insufficient for generating design curves. In the absence of a fatigue design, precautionary measures should be taken to avoid mooring failure due to bending-tension fatigue. For example, the fairlead-to-line diameter ratio (*D/d*) should be large enough to avoid excessive bending. The portion of mooring line in direct contact with a fairlead should be regularly inspected. Also this portion should be periodically shifted to avoid constant bending in one area. A study comparing the bending-tension and tension-tension fatigue lives of mooring lines on a semisubmersible under the North Sea environment provided the following data that can be used as reference values for establishing operation policy to avoid excessive bending tension fatigue for wire ropes.

Rope Type	*D/d* Ratio	B-T Fatigue Life in Terms of Percentage of T-T Fatigue Life
Six strand	20	3
Six strand	70	8
Multi-strand	20	5
Multi-strand	70	15
Spiral strand	20	0.5
Spiral strand	70	1.5

Note that the analysis for bending-tension fatigue is very complex and the above values are rough estimates only. Some margin of safety is recommended when these data are used.

Free bending at the wire rope terminations can also significantly reduce the wire rope fatigue life. To avoid premature fatigue failure in permanent moorings, a bend limiting device should be incorporated at these locations. Such a device is designed to smoothly transfer the loads from the termination to the rope.

As for tension bending of chain, the portion of mooring line in direct contact with a fairlead should also be regularly inspected and shifted to avoid constant bending in one area. In general, the worst load case is to tension-bend a horizontal link over a shallow groove, that results in very high stress in the stud weld region. Therefore, fairleads must be shaped and sized to avoid this type of unfavorable bending of chain links. Limited fatigue T-N tests of chains over a five-pocket fairlead indicate 5 percent to 20 percent of T-B fatigue life in terms of T-T fatigue life. A seven-pocket fairlead design generally gives much improved T-B fatigue life.

Corrosion of wire at connections to sockets can be excessive due to the galvanized wire acting as an anode for adjacent components. For permanent systems, it is recommended that either the wire be electrically isolated from the socket or that the socket be isolated from the adjacent component. Additional corrosion protection can be achieved by adding sacrificial anodes to this area.

7 Mooring Analysis

7.1 BASIC CONSIDERATIONS

Permanent mooring systems should be designed for two primary considerations: system overloading and fatigue. Therefore, analysis for extreme response and fatigue damage should be performed. For mobile moorings, only analysis for extreme response is required.

The analysis procedure described in this section can be applied directly to spread mooring systems, as well as internal and external turret mooring systems. For systems where the mooring is connected to the vessel through a buoy (CALM system) or through a riser (turret-riser system), a similar analysis procedure will apply. However, evaluation of wave loadings on the buoy or riser and transformation of the vessel motions to the chain table through the buoy or riser require special consideration. Model testing or analyses with specialized computer programs are often required; however, these analyses are not covered in this recommended practice. Further research is required to develop recommended procedures for such analyses.

For a CALM system with hawsers, guidelines given in U.S. Coast Guard Report No. CG-49-77 [31] can be used for model testing, design, and analysis of the hawsers. The basis for the mooring analysis procedure presented in this section can be found in Appendix D [14, 30].

7.1.1 Extreme Response Analysis

Extreme responses normally govern the design of the FPS mooring. They include vessel offset, mooring line tension, anchor load, and suspended line length. The environmental events for extreme response are described in Section 4.

As stated in Section 5, environmental effects can be divided into three categories:

a. Steady state forces including current force, mean wind, and mean wave drift forces.
b. Low-frequency vessel motions due to wind and waves.
c. Wave frequency vessel motions.

The responses of a mooring system to mean forces are predicted by static catenary equations. Generally speaking, the responses to low-frequency motions can also be predicted by the same method because of the long periods of these motions. The responses to wave frequency vessel motions are usually predicted by one of the following two methods:

a. Quasi-static analysis. In this approach, the dynamic wave loads are taken into account by statically offsetting the vessel by an appropriately defined wave induced motion. Vertical fairlead motions and dynamic effects associated with mass, damping, and fluid acceleration are neglected. Research in mooring line dynamics has shown that the reliability of the mooring designs based on this method can vary widely depending on the vessel type, water depth and line configuration. Therefore, the quasi-static method is not recommended for the final design of a permanent mooring. However, because of its simplicity, this method can be used for temporary moorings and preliminary studies of permanent moorings with higher factors of safety.
b. Dynamic analysis. Dynamic analysis accounts for the time varying effects due to mass, damping, and fluid acceleration. In this approach, the time-varying fairlead motions are calculated from the vessel's surge, sway, heave, pitch, roll and yaw motions. Generally it is sufficient to account for only the vertical and horizontal fairlead motions in the plane of the mooring line. Dynamic models are used to predict mooring line responses to the fairlead motions. Several dynamic analysis techniques are available. The distinguishing feature among various dynamic analysis techniques is the degree to which nonlinearities are treated. There are four primary nonlinear effects that can have an important influence on mooring line behavior. They are the following:
   1. Nonlinear stretching behavior of the line. The strain or tangential stretch of the line is a function of the tension magnitude. Nonlinear behavior of this type typically occurs only in synthetic materials such as nylon. Chain and wire rope can be regarded as linear. In many cases the nonlinearity can be ignored and a linearized behavior assumed using a representative tangent or secant modulus.
   2. Changes in geometry. The geometric nonlinearity is associated with large changes in shape of the mooring line.
   3. Fluid loading. The Morison equation is most frequently used to represent fluid loading effects on mooring lines. The drag force on the line is proportional to the square of the relative velocity (between the fluid and the line) and is hence nonlinear.
   4. Bottom effects. In most mooring designs, a considerable portion of the line is in contact with the seafloor. The interaction between the line and the seafloor is usually considered to be a frictional process and is hence nonlinear. In addition, the length of grounded line constantly changes, causing an interaction between this nonlinearity and the geometric nonlinearity.

Two methods, frequency domain and time domain analyses, are commonly used for predicting dynamic mooring loads. In the time domain method, all of the nonlinear effects can be modeled. The elastic stretch is mathematically modeled, the full Morison equation is included, the position of the mooring line is updated at each time step, and the bottom interaction is included using a frictional model. The general analysis implies the recalculation of each mass term, dampening term, stiffness term, and load at each time step. Hence the computation can become complex and time consuming. The frequency domain method, on the other hand, is always linear as the linear principle of superposition is used. Hence, all nonlinearities must be eliminated, either by direct linearization or by an iterative linearization, as listed here:

a. Line stretching. The line stretching relationship must be linearized and a definite value of the modulus assumed at each point. The modulus cannot be a function of line tension but can vary along the line. This is usually not a bad assumption even in the case of synthetic material and, in most cases, a suitable linearization can be achieved.
b. Geometry change. In the frequency domain method it is assumed that the dynamic displacements are small perturbations about a static position. The static shape is fixed and all geometric quantities are computed based on this position. The mass, added mass, stiffness, etc. are computed only once. Changes in catenary shape due to the dynamic motion contribution are generally not severe. Hence, a linearization about the position under mean load is generally acceptable.
c. Fluid loads. The nonlinear term in the Morison equation must be linearized. The quadratic relationship in the relative velocity must be replaced by an equivalent linear relationship. The linearization should take into account the frequency content of the line motion spectrum.
d. Bottom effects. The frictional behavior between the grounded line and the seafloor cannot be represented exactly in the frequency domain. Only the average or equivalent behavior of the line can be postulated and included. This simplification should be adjusted to the design objective. Different models may be required for the fatigue and the extreme tension evaluations.

The relative influence of various nonlinearities is a function of numerous parameters—particularly water depth, line composition and motion magnitude. Methods to approximate nonlinearities in the frequency domain should reflect the importance of the various parameters.

7.1.2 Fatigue Analysis

Fatigue life estimates are made by comparing the long-term cyclic loading in a mooring component with the resistance of that component to fatigue damage. For mooring systems, a T-N approach is normally used. The T-N approach uses a T-N curve, that gives the number of cycles to failure for a specific mooring component as a function of constant tension range, based on the results of experiments.

The Miner's Rule is used to calculate the cumulative fatigue damage ratio, D, where:

$$D = \Sigma \frac{n_i}{N_i} \qquad (7.1)$$

Using Equation 7.1:

n_i = number of cycles within the tension range interval i.
N_i = number of cycles to failure at tension range i as given by the appropriated T-N curve.

D should not exceed unity for the design fatigue life which is the field service life multiplied by a factor of safety of 3, as defined in 6.8.

The quasi-static approach should not be used for calculating tension ranges due to its severe deficiency in estimating wave frequency tensions. Both time and frequency domain dynamic approaches may be used for tension range predictions. However, the computational effort for time domain analysis could be prohibitive because of the large number of tension range calculations required for a fatigue analysis. The frequency domain dynamic analysis approach appears to be the most suitable tool for tension range predictions, assuming that proper linearization techniques are incorporated into the frequency domain solutions. Alternatively, tension ranges can be obtained from model testing.

7.2 QUASI-STATIC AND DYNAMIC ANALYSIS

The procedure outlined below is recommended for the analysis of extreme responses using a quasi-static or dynamic approach. The calculated responses in accordance with this procedure should satisfy the design criteria as defined in Section 6. The use of this procedure is illustrated in the example in Section 11.

The analysis is normally performed with the following computer programs:

a. Hydrodynamic motion analysis programs. These programs are used to determine wave frequency and low-frequency vessel motions.
b. Static mooring analysis program. This program is used to analyze mooring line response to steady state environmental forces and low-frequency motions.
c. Dynamic mooring analysis program. This program is used to analyze mooring line response to wave frequency motions.

The recommended analysis procedure is described below:

a. Determine wind and current velocities, and significant wave heights and periods, for both the maximum design, and operating conditions in accordance with guidelines stated in Section 4.
b. Determine the mooring pattern, characteristics of chain and wire rope to be deployed, and initial tension.
c. Determine the steady state environmental forces acting on the hull using either model test data or the procedures described in Section 5.
d. Determine the vessel's mean offset due to the steady state environmental forces using the static mooring analysis program.
e. Determine the low-frequency motions using the data and procedures described in Section 5 or a hydrodynamic motion analysis computer program. Since calculation of low-frequency motions requires the knowledge of the mooring stiffness, the mooring stiffness at the mean offset should be determined first using a static mooring analysis computer program.

Recommended values for elasticity of mooring lines are given below. For chain, the elasticity, T/δ_c, in pounds of tension per foot of stretch is:

$$T/\delta_c = 1.2 \times 10^7 \, D_c^2/S_c \qquad (7.2)$$

Where:

T = mooring line tension (lbs).
δ_c = elastic stretch of chain (feet).
D_c = nominal chain diameter (in).
S_c = chain length (feet).

For wire rope:

$$T/\delta_w = 7.7 \times 10^6 \, D_w^2/S_w \qquad (7.3)$$

Where:

δ_w = elastic stretch of wire rope (feet).
D_w = nominal diameter of wire rope (in).
S_w = wire rope length (feet).

The stretch coefficient for wire rope in Equation 7.3 is appropriate for six strand wire ropes commonly used in mooring applications. For other types of wire rope such as spiral strand, the stretch coefficient may differ.

f. Determine the significant and maximum single amplitude wave frequency vessel motions using a hydrodynamic motion analysis program.
g. Determine the vessel's maximum offset, suspended line length, quasi-static tension, and anchor load using Equations 6.1 and 6.2 and the static mooring analysis program. Skip step h, if only a quasi-static solution is required.
h. Determine the maximum line tension and anchor load according to Equations 6.3 and 6.4. A frequency domain or time domain dynamic mooring analysis program should be

used. For the frequency domain approach, the analysis would involve the following steps:

1. Determine the significant and maximum low-frequency vessel motions using Equations 6.5 and 6.6.
2. Move the vessel to a position corresponding to the mean position plus significant low-frequency motion. Calculate mooring system responses at this position using the static mooring analysis program. Then impose wave frequency fairlead motions on the mooring line of interest and calculate the wave frequency responses using the dynamic mooring analysis program. For the frequency domain solution that yields rms values, Equations 6.4 and 6.6 are used to calculate the peak line tension under this condition.
3. Move the vessel to a position corresponding to the mean position plus maximum low-frequency motion. Calculate mooring system responses at this position using the static mooring analysis program. Then impose wave frequency fairlead motions on the mooring line of interest and calculate the wave frequency responses using the dynamic mooring analysis program. For the frequency domain solution that yields rms values, Equations 6.3 and 6.5 are used to calculate the peak line tension under this condition.
4. Compare the peak tensions obtained from steps 2 and 3, and select the maximum value. The maximum anchor load can be determined in the same manner. In many cases, it will be obvious which of steps 2 or 3 will yield the maximum value. In these circumstances only that case need be considered. It is not sufficient to compare maximum low and wave frequency motions. On many occasions the maximum wave frequency tension will be higher than the maximum low-frequency tensions even though the maximum low-frequency motion is higher than the maximum wave frequency motion.

For the time domain approach, similar steps can be followed to obtain the maximum line tensions and anchor load. However, determination of maximum wave frequency line tension in step 2, and significant wave frequency line tension in step 3 require special attention that is discussed below.

Maximum Wave Frequency Line Tension: To determine the maximum wave frequency line tension by a time domain approach, a wave frequency fairlead motion time record for the storm duration should be generated first. The tension response to this fairlead motion time history is then calculated using a time domain dynamic mooring analysis computer program. Finally, the maximum wave frequency tension response is selected and added to the mean and significant low-frequency tension to obtain the maximum line tension. Because of the random nature of the problem, special techniques are required to yield acceptable approximations for the maximum wave frequency tension. Two such techniques are listed below. Method A is more rigorous and will yield good approximations for various statistical peak distributions. However it requires much greater computational effort. The alternative method B yields acceptable approximations for tension peaks that can be represented by the Rayleigh distribution. It allows the user to obtain the expected maximum wave frequency tension by simulating a short tension history.

Method A—The objective of this method is to estimate the expected maximum tension in a storm of the specified duration. The maximum tension observed in any single time domain realization will vary about this expected value. In this approach, line tensions are firstly simulated several times for the specified storm duration. A statistical peak distribution (such as Weibull, Hermit/Gaussian, or Exponential), which is the best fit to all the peaks of the simulations, is then established. Finally the fitted distribution is used to determine the expected maximum tension.

Method B—In this approach, the maximum tension from a single computer run is modified by a Rayleigh factor. The procedure is as follows:

1. Generate a wave frequency fairlead motion time history for the specified storm period, and calculate the ratio of maximum to RMS tangential fairlead motion R.
2. Select a segment of the fairlead motion time history centering about the maximum tangential fairlead motion. This segment should be long enough to eliminate the transient effects in dynamic mooring analysis.
3. Input this segment of fairlead motion history to the dynamic mooring analysis program and obtain the maximum wave frequency tension.
4. Multiply the maximum wave frequency tension by a factor of $\sqrt{2(\ln N)}/R$, where N is the number of wave cycles in the storm period.

Significant Wave Frequency Line Tension: Similarly, a few techniques exist to yield acceptable approximations of significant wave frequency line tension. Two such techniques are the following:

Method A—The significant tension can be calculated from a single tension time history that is long enough to produce a stable significant value.

Method B—Same as method B for calculating maximum tension except that the multiplying factor is $2/R$.

i. Compare the maximum vessel offset and suspended line length from step g, and maximum line tension and anchor load from step g or h, with the design criteria stated in Section 6. If the criteria are not met, modify the mooring design and repeat the analysis.

7.3 TRANSIENT ANALYSIS

A moored vessel will experience transient motions after a mooring line breakage before it settles at a new equilibrium position. The transient condition can be an important consideration for certain operations, and analysis for this condition is required, as specified in 6.1. Transient analysis of a moored vessel under wind, wave, and current loadings is complex and requires a time domain solution. To simplify the analysis, a combination of time domain (transient motions) and frequency domain (vessel motions) approach is often used. The damping force on the moored vessel is most uncertain, and further research in this area is required. The recommended approaches for performing transient analysis are discussed below.

7.3.1 Time Domain Analysis

The basic equation governing transient motions of a moored vessel is:

$$(M + A)\ddot{X} + C\,|\dot{X}|\,\dot{X} + KX = F = Fm + Fw + F_l \quad (7.4)$$

Where:

X = displacement matrix.
M = mass matrix.
A = added mass matrix.
C = damping coefficient matrix.
K = stiffness matrix.
F = force matrix that includes wind, current, first and second order wave forces.
F_m = steady (mean) component of the total force F.
F_w = wave frequency component of the total force F.
F_l = low-frequency component of the total force F.

For dynamic mooring analysis, Equation 7.4 should be established for 6 degrees of freedom of the vessel including surge, sway, heave, roll, pitch, and yaw. For quasi-static mooring analysis, a model with three degrees of freedom including surge, sway, and yaw will be sufficient. Equation 7.4 can be solved in time domain, preferably with updated information for all parameters at each time step.

One major difficulty with this full time domain approach is the random nature of the problem. The time at which the mooring line breaks during a storm affects the boundary conditions under which the transient occurs. An acceptable approach is to repeat the calculation of maximum tension and offset for a number of wave force records, and for the break to occur at a number of times during each record. The times for the mooring line to break should center around the time for the peak in the motion/tension record. The maximum value of tension observed during these simulations, or a 90 percent confidence bound estimate of peak tension based on these simulations, should be used for the design.

7.3.2 Combination of Time and Frequency Domain Analysis

In this approach, the maximum transient motion is first determined using a time domain approach. Then vessel motions obtained from a frequency domain approach are superimposed on the transient motion. The recommended procedure is the following:

a. Compute the equilibrium position under mean environmental load for an intact mooring.
b. Break a line and compute the new equilibrium position under mean environmental load.
c. Compute the maximum transient motion (overshoot) in the time domain, using Equation 7.4 with $F = F_m$ (i.e., $F_l = F_w = 0$) and starting from the equilibrium positions from item a, above, with one line removed and the mooring system stiffness, K, updated at each time step. Generally a model with three degrees of freedom (surge, sway, and yaw) is required. Under certain conditions, a model with two degrees of freedom (surge and sway) or one degree of freedom (combination of surge and sway in the direction of the vessel traveling after a line breakage) may yield acceptable solutions. However, a sensitivity study should be conducted before such an approach is taken.
d. Determine maximum vessel offset and line tension.

Maximum vessel offset should be determined by Equation 7.5.

$$S_{max} = S_{mean} + S_t + S_{wfsig} + S_{lfsig} \quad (7.5)$$

Where:

S_{mean} = mean offset as calculated in step b.
S_t = maximum transient motion (overshoot) with respect to the equilibrium position from step b as determined in step c.
S_{wfsig} = significant wave frequency motion, calculated in the frequency domain.
S_{lfsig} = significant low-frequency motion, calculated in the frequency domain using the damaged mooring system stiffness.

Maximum line tension should be determined by Equation 7.6.

$$T_{max} = T_{mean} + T_t + T_{wfsig} + T_{lfsig} \quad (7.6)$$

Where:

T_{mean} = line tension due to mean environmental load as calculated in step b.
T_t = line tension due to maximum transient motion, S_t, determined in step c.
T_{wfsig} = significant wave frequency tension.
T_{lfsig} = significant low-frequency tension.

7.3.3 Damping Force

Among all the parameters in a transient analysis, the damping force on the moored vessel is most uncertain. This uncertainty can significantly affect the results of a transient analysis. The three major sources of damping include:

a. Viscous damping of the vessel.
b. Wave drift damping of the vessel.
c. Mooring system damping.

Technology to estimate viscous damping is well developed and therefore is always accounted for. Wave drift damping and mooring system damping received little attention previously and were often neglected. Recent research indicates however, that wave drift damping and mooring system damping can be significant. In some cases they can be higher than viscous damping. To include damping from various sources, the damping forces are often linearized, and Equation 7.4 becomes:

$$(M + A)\ddot{X} + C_l\dot{X} + KX = F = Fm + Fw + F_l \qquad (7.7)$$

Where:

C_l = linearized damping coefficient matrix.

Since the technology to evaluate and linearize damping from all major sources is still in a stage of development, no specific guidelines can be given at this point. As a guide, if the transient motions determined by including viscous damping only, are within acceptable limits, no further analysis is necessary. However, if the limits cannot be met, further analysis including also other sources of damping may be warranted. Information for such an analysis is provided in *Mooring Line Damping—Summary and Recommendations* [34] and *Full-Scale Measurements of Free-Oscillation Motions of Two Semisubmersibles* [44].

7.4 ANALYSIS FOR THRUSTER ASSISTED MOORING

In a thruster assisted mooring system, the thrusters can be manually or automatically controlled. The latter is often referred to as dynamic positioning (DP) system. A simple mean load reduction method can be used for the analysis of manual control case. However, the load sharing between the DP and the mooring in a DP assisted mooring system is complicated and can only be fully evaluated with a system dynamic analysis (DP simulation). Research has shown that the simple mean load reduction method will yield results that are close to those obtained from more complicated system dynamic analysis. Therefore, this method can be used for the preliminary design of a DP assisted mooring system. However, it is desirable to verify the final design with a system dynamic analysis, especially for cases where low-frequency motions dominate the design.

7.4.1 Mean Load Reduction Method

In this simplified approach the thrusters are assumed to counter only the mean environmental loads in the surge, sway, and yaw directions. Available thrusts from thrusters should be evaluated according to the guidelines presented in 6.7. The remainder of the total mean load, and the wave and low-frequency motions would be taken by the mooring system. The mooring system can be analyzed by the procedure outlined in 7.2.

7.4.2 System Dynamic Analysis

A system dynamic analysis is normally performed using a three axis (surge, sway, and yaw) DP simulator. This simulator generates the mean offset and low-frequency vessel motions and thruster responses corresponding to specific environmental force time records. In this analysis, constant wind, current, steady wave drift forces, and the low-frequency wind and wave drift forces are typically included. Wave frequency wave forces, which are not countered by the DP system, can be excluded in the simulation. The wave frequency motions are computed instead using a separate, frequency domain vessel motion program. The following describes the dynamic analysis procedure:

a. Select a range of set points. A set point (Figure 22) is a desired mean distance from the reference position, selected by the operator. Different set points will result in different levels of load sharing between the DP and mooring systems, and produce different vessel offsets. An optimum set point will result in minimum total vessel offset. Since the optimum set point is not known initially, a reasonable range of set points must first be selected.

b. Input a selected set point and the specific environmental force time record into the DP simulator and obtain a time history of the vessel offset representing selected storm durations. The mean, maximum, minimum, and significant vessel offsets are calculated from the time history. The beginning segments of the time histories are eliminated to minimize the effects of transient motions.

c. Repeat step b for all selected set points and select the optimum set point that yields minimum vessel effect.

d. Perform dynamic mooring analysis for the case with the optimum set point using similar procedures outlined in 7.2. Two analyses are required:

1. Move the vessel to a position corresponding to the combination of set point, mean offset, and significant low-frequency motion (Figure 22). Perform dynamic mooring analysis to determine the maximum wave frequency tension.
2. Move the vessel to a position corresponding to the combination of set point, mean offset, and maximum low-frequency motion. Perform dynamic mooring analysis to determine the significant wave frequency tension.

The peak line tensions obtained from the above two analyses are compared and the maximum value is selected.

System dynamic analysis is a time domain analysis. To obtain a proper maximum value from a time domain analysis, it may be necessary to generate a number of force and response records for the storm duration and calculate the expected maximum value using a statistical approach.

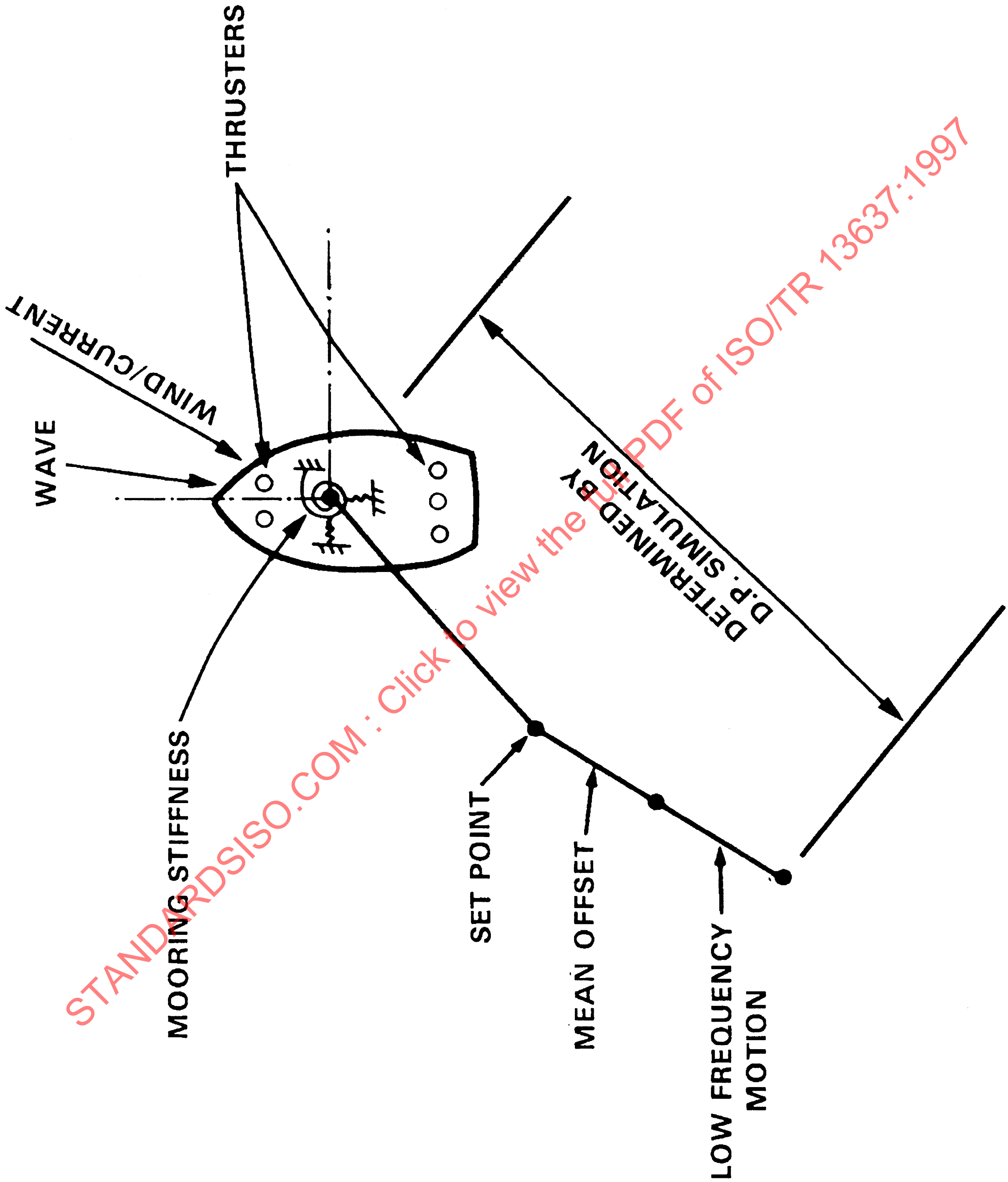


Figure 22—System Dynamic Analysis

7.5 FATIGUE ANALYSIS

The recommended procedure for a detailed fatigue analysis is described below:

a. The long term environmental events can be represented by a number of discrete design conditions. Each design condition consists of a reference direction and a reference sea-state characterized by significant wave height, peak spectral period (or equivalent), spectral shape, current velocity, and wind velocity. The probability of occurrence of this design condition must be specified. In general, 8 to 12 reference directions provide a good representation of the directional distribution of a long term environment. The required number of reference sea-states normally falls in a range of 10 to 50. Fatigue damage prediction can be fairly sensitive to this number for certain mooring systems, and therefore it is best determined by a sensitivity study.

b. Each design condition can be analyzed analogously to the procedure used for extreme conditions as described in 7.2. One simplification of this procedure can be used. The wave frequency tensions can be computed about the position of the mooring system under mean loading only. The following analysis procedure can be used:

1. Determine all loads and motions (low and wave frequency) as set out for an extreme analysis, in 7.2.
2. Compute mooring system responses under mean loading using a static analysis program. Then impose wave frequency motions and compute the rms wave frequency tensions from a dynamic analysis.
3. Add the low-frequency rms motions to the mean position and compute the corresponding low-frequency rms tensions. In situations where the mooring system is highly nonlinear, and where low-frequency effects dominate, a more conservative procedure is to add the maximum low-frequency motions to the mean position and compute the corresponding maximum low-frequency tensions. These can be scaled down to compute the rms low-frequency tensions using Equations 6.5 to 6.8. Alternatively, the rms wave and low-frequency tensions can be obtained from model testing.

c. Determine the T-N curve in the form of:

$$NR^M = K \tag{7.8}$$

Where:

N = number of cycles to failure.
R = ratio of tension range to a reference breaking strength.
M = slope of the T-N curve.
K = intercept of the T-N curve.

T-N curves for chain, wire rope, and connecting links can be found in 6.8.

d. Compute the annual fatigue damage from one environment (one sea-state in one direction) due to both the low-frequency and the wave frequency tension. Three methods can be considered for combining fatigue damages due to the low-frequency and wave frequency tensions, as follows:

1. Simple summation method. In this approach, low-frequency and wave frequency fatigue damages are calculated independently. The total damage is assumed to be the sum of the two.
2. Combined spectrum. In this approach, the combined low-frequency and wave frequency spectrum is first calculated. Fatigue damage is estimated using the combined rms tension range.
3. Time domain cycle counting. In this approach, the combined low-frequency and wave frequency tension spectrum is transformed into a tension time history. A special cycle counting method such as the RAINFLOW method is used to estimate the number of tension cycles and the expected value of the tension range from which fatigue damage is estimated.

Method 3 is generally considered to be more precise but is not an efficient approach for design. Method 1 will generally give an acceptable estimate of fatigue life. However, this method may underestimate fatigue damage in cases where typical low-frequency rms tension values are higher than typical wave frequency rms tension values in sea-states contributing significantly to fatigue damage. Method 2 is always conservative and may significantly overestimate the actual fatigue damage. As a guide, method 1 can be used for typical fatigue calculations. In cases where low-frequency rms tension values are higher than wave frequency rms tension values and the fatigue life calculated by method 1 is marginal, further investigations should be carried out using methods 2 and 3.

Analysis procedures for methods 1 and 2 are presented below. Analysis procedures for method 3 can be found in the July 1977 issue of *Journal of Engineering Materials* [20], and the March 1972 issue of the *Journal of Materials* [22].

Simple summation method:

Wave frequency and low-frequency fatigue damages are estimated by Equation 7.9, which is based on Rayleigh Distribution of tension peaks.

$$D = N_w(\sqrt{2}\,R_{wrms})^M \cdot \Gamma(1 + M/2)/K + N_l(\sqrt{2}\,R_{lrms})^M \cdot \Gamma(1 + M/2)/K \tag{7.9}$$

Where:

D = annual fatigue damage from wave frequency and low-frequency tensions.
N_w = number of wave frequency tension cycles per year.

This can be taken as:

$$N_w = \nu \times 3.15576 \times 10^7 \times P \tag{7.10}$$

Where:

v = the zero up-crossing frequency of the tension spectrum (hertz).

P = the total probability of occurrence of the environmental condition, generally computed as P_s (probability of the environmental occurrence, given a direction) times P_d (probability of directional occurrence).

R_{wrms} = ratio of rms wave frequency tension range to a reference breaking strength. The rms tension range should be taken as twice the rms tension.

Γ = Gamma function.

N_1 = number of low-frequency tension cycles per year.

This may be taken as:

$$N_1 = P \times 3.15576 \times 10^7/T_n \qquad (7.11)$$

Where:

T_n = the natural period of the vessel (Equation 6.8), computed at the appropriate mean position of the vessel. If available, the average zero up-crossing period for low-frequency effects should be substituted for T_n.

R_{lrms} = ratio of rms low-frequency tension range to a reference breaking strength. The rms tension range can be taken as twice the rms tension.

Combined spectrum method:

Compute the rms low and wave frequency tensions as before.

Compute the combined rms tension range as:

$$R_{rms} = \sqrt{R^2_{wrms} + R^2_{lrms}} \qquad (7.12)$$

Compute the annual fatigue damage as:

$$D = N(\sqrt{2}\, R_{rms})^M\, \Gamma(1 + M/2)/K \qquad (7.13)$$

Where:

N = the total number of cycles in the combined spectrum, per year. This may be computed as:

$$N = v_o \times 3.15576 \times 10^7 \times P \qquad (7.14)$$

Where:

v_o = the zero up-crossing frequency (hertz) of the combined spectrum.

P = the total probability of occurrence of this environmental condition. Equation 7.13 is based on Rayleigh Distribution of tension peaks.

e. Repeat step d for all sea-states and directions and compute the total annual fatigue damage, D_t.

$$D_t = \sum_{\text{dir}} \bullet \sum_{\text{seas}} D \qquad (7.15)$$

f. The calculated fatigue life of the mooring system is:

$$L = 1/D_t \text{ (years)} \qquad (7.16)$$

The service life of the mooring system should be less than the calculated fatigue life divided by a factor of safety of 3 as defined in 6.8.

8 Model Testing—Basic Considerations

During the final design stage, physical model experiments such as wave basin, towing basin, and wind tunnel tests are often performed to compensate for the limitations of analysis. Model tests may be used either as a self-contained program for determining the responses of a particular floating unit, or as a verification of analytical predictions of the responses. The primary objectives are the following:

a. To determine maximum responses for design purpose, for example, motions, line tensions, forces at mooring interfaces, and so forth.

b. To quantify important parameters and thereby to calibrate computer programs.

c. To confirm, using a physical model, that no important facet of the operation has been overlooked.

One of the values of model tests is that the results are obtained without requiring many assumptions about the nature of the responses. This is generally not true of numerical models. However, model testing has its limitations too. There are numerous sources that can cause errors in model test results. Therefore, numerical predictions and model experiment results are complimentary to each other. Through careful interpretation, each of these results can be used to partially circumvent the limitations of the other.

In order to obtain realistic results from model tests, it is important to select the proper vessel and mooring pine model scales and testing facility (size and depth of model basin, wave, wind, and current generating capability). Often some of these parameters must be compromised to satisfy conflicting scaling laws and facility limitations. Because of this, scale errors should be estimated and minimized. Guidelines for wind tunnel tests can be found in SNAME T&R Bulletin 5-4 [50].

9 Single Anchor Leg Mooring Systems

9.1 BASIC CONSIDERATIONS

As discussed in 2.1.2, the SALM (single anchor leg mooring) system (Figures 9 and 10) is a single point mooring for production or storage vessels. Unlike a catanary mooring which relies on chains and wire ropes to provide restoring forces, a SALM system relies on the buoyancy of structural/mechanical components. This system, if properly designed, can effectively resist extreme environments and accommodate complex piping systems. However, industry experience indicates that this system is susceptible to single point failure and therefore requires special effort in detail design. Some of the design guidelines provided in Sections 4 to 8 can be used for the design of SALMs. They include:

a. Environmental criteria (Section 4, for intact conditions only).
b. Environmental loads on floating vessels (Section 5).
c. Design criteria (Section 6):
   1. Vessel offset (6.2).
   2. Statistics of peak values (6.4).
   3. Thruster assisted mooring (6.7).
d. Mooring analysis (Section 7, partially applicable).
e. Model testing (Section 8).

However, there are design aspects that are unique to SALMs. These aspects are addressed in the following sections.

9.2 SPECIAL DESIGN CONDITIONS

9.2.1 Damaged Condition

The damaged condition for SALMs is defined as damage to compartments resulting in compartment flooding. A SALM system should maintain its load carrying capability and stability under the one-year storm environment for the following conditions:

a. Any one compartment flooded anywhere in the system.
b. Any two adjacent compartments flooded in the collision zone. The collision zone should be established based on the dimensions of the vessels that will normally be operating near the system, such as work boats and shuttle tankers.

9.2.2 Installation Condition

Design criteria shall be established for all phases of the installation from load out through commissioning. All system components and installation equipment shall be designed for the established environmental limits. The design shall be verified for installation conditions by analysis or testing. An installation manual shall specify the limiting conditions for each phase and shall identify precautions to be taken to avoid exceeding the criteria.

Design wave and wind criteria for transoceanic tow require special attention because of the long tow duration. These criteria include a specified sea-state (significant wave height and period) and wind speed. The wave height and period are used in assessment of the strength of the barge and cargo or the structure itself in the case of a wet tow, and the wind speed is applied to ensure adequate stability. The return periods associated with the sea-state and wind speed criteria are normally up to ten years, depending on the exposure time, quality of weather forecasting and relative risk the operator is willing to assume. One acceptable approach is to select the twenty tow (not year) sea-state, so that the target sea-state is expected to occur on average once every twenty tows. This is equivalent to assuming five percent (one in twenty) risk that the specified sea-state will occur during transportation. A prudent method of estimating a rare sea-state for tow assessment should properly account for regional and seasonal weather changes. Wave and wind data bases are normally available that allow proper accounting for climatic variation with time and region. A rarer wind speed is normally selected due to the severe consequence of loss of stability. A one in one hundred tow wind speed, corresponding to one percent probability of occurring during the journey, is a reasonable criterion. A more detailed discussion on the tow criteria can be found in OTC 6684 [51].

9.3 EXTREME RESPONSE ANALYSIS

Dynamic analysis should be conducted for the design of SALMs. Either time domain or frequency domain analyses may be used, provided that the analysis techniques have been verified for systems similar to that being analyzed. For highly non-linear systems, a time domain analysis, confirmed by model tests, is more desirable. There are several special analysis considerations for SALMs, as discussed in 9.3.1-9.3.6.

9.3.1 Nonlinearity

The effects of system nonlinearity shall be considered in the analysis. Because the flexibility of the system is produced by the mechanical linkage of weights and buoyancies, the system shall be analyzed to ensure that, under the maximum design conditions, sufficient margin for additional deflection is available without reaching the limits of the mechanical linkage. (Refer to 9.4 for criteria.)

9.3.2 Vessel Induced Motions and Forces

Mechanical linkages in complex mooring systems will transform vessel motions into other motion patterns within the mooring system and may modify the induced motions of the mooring system. The effect of the vessel driving the mooring system shall be analyzed to determine the induced motions and forces.

9.3.3 Wave Forces

The calculation of wave forces on system components shall include the effects of vessel induced motions and their phasing. For systems with large mooring structures, the effect of wave loads on the mooring system shall be considered in the overall system analysis.

9.3.4 Directional Effects

The effects caused by noncollinear directions of wind, waves, and currents, and the effect of fishtailing of the system shall be considered in the analysis of the system.

9.3.5 Friction

The effects of friction shall be considered in determining component loads for all loading conditions, including fatigue. For journal bearings, a friction coefficient of not less that 0.25 shall be used, unless long term test data for similar conditions demonstrates that a lower value can be used. Lubrication of journal bearings is desirable, but the design values used should assume a failure of the lubrication system. For most systems, friction can be considered a local effect which does not affect the overall system loads and response.

9.3.6 Slamming

The calculation of slamming forces shall be based on the relative velocity of the member to the water particle, including the effect of vessel induced motions. Slamming pressures may be calculated based on a Morrison drag formulation using an appropriate drag coefficient. Procedures for such a calculation can be found in *Rules for the Design, Construction and Inspection of Offshore Structures* [32]. Because slamming is an impulse loading, a dynamic amplification factor of at least 2.0 should be used, unless an analysis, which considers the duration of the impulse loading and the natural frequency of the structure, demonstrates that a lesser value is appropriate.

9.4 DESIGN CRITERIA FOR EXTREME RESPONSES

It is not the intention of this document to provide detailed criteria for mechanical/structural component design since such criteria are available in other documents [2, 32]. A special consideration with SALM design is the mechanical clearance in articulations that is addressed below.

A SALM system shall be designed so that at maximum system deflections or articulations, there are sufficient reserve lateral and rotational clearances between the components. Clearances between components should be analyzed for all possible vessel draft conditions and the full range of tidal conditions for the maximum design conditions, damage conditions and installation conditions, unless operating procedures impose certain draft limitations for certain conditions. The following criteria shall be satisfied for all design conditions:

a. There shall be sufficient clearance between structural components for articulations to accommodate motions with a minimum safety factor of 1.3 on the most probable (63.2 percent probability of exceedance) relative displacements. For this calculation, 1.3 times the most probable variation from the static mean is to be added to the static mean value. The combined effect of high and low-frequency motions are to be considered for the storms of the specified duration.

b. As a second criteria, clearance shall be provided to accommodate a safety factor of 1.1 times the 10 percent probability of exceedance variation, added to the mean, for a storm of the specified duration.

9.5 FATIGUE ANALYSIS

In general, fatigue analysis procedures for SALMs are similar to those outlined in 7.5. However, design T-N curves for chain and wire rope and the associated factor of safety for fatigue life specified in 6.9 do not apply to the structural components in a SALM system. For fatigue evaluation of structural details, *Fatigue Strength Analysis for Mobile Offshore Units* [33] can be used. For selection of allowable damage based on redundancy, inspectability, and level of confidence in loading, *Rules for the Design, Construction, and Inspection of Offshore Structures* [32] can be used. In addition, attention should be paid to the following effects in a SALM fatigue analysis:

a. Effect of phasing of primary load components on the resulting cyclic stress at a point.

b. Effective static load and resulting friction loading in mechanical articulations for each sea-state.

c. Effect of nonsymmetrical stress pattern in pinned connections or articulations, where the stress pattern for tension loads is different from that for compression loads. Stress range should consider reversal around mean load.

9.6 SPECIAL DESIGN CONSIDERATIONS

Close attention should be paid to several areas of detail design that could potentially lead to performance problems if not handled properly. These include the following:

a. Hydrodynamic loading:
   1. Local strength of components to resist wave slamming.
   2. Vortex induced vibration of free spans of structural members and piping.

b. Design of mechanical components:
   1. Bearing foundations and retainers.
   2. Bearing clearance and its effect on induced shock loads.
   3. Provision for accessibility to monitor bearing wear rates.
   4. Effects of deflections of components and assemblies.
   5. Bolt preload and the grip length required to maintain pretension.
   6. Lubrication system effectiveness and reliability.
   7. Roller bearing sealing.
   8. Electrical bonding across articulations.
   9. Suitability of materials, including galvanic effects between dissimilar metals.
   10. Mechanical latches and locking mechanisms.
   11. Inspection and maintenance requirements and utility.
   12. Provision for monitoring onset of failure in rolling contact bearings.

c. Design of structural components:
   1. Recognition of the level of structural redundancy and appropriate selection of safety factor.
   2. Corrosion protection and selection of corrosion allowance.
   3. Requirement for through thickness properties for details with through thickness loading or weld restraint.
   4. Pressure relief of structural compartments in case of pressure piping leakage.
   5. Fatigue at pipe penetrations manways, transitions and attachments.
   6. Accessibility for fabrication and in-service inspection.
   7. Configuration of structure for installation requirements.
   8. Selection of weld details with high resistance to fatigue at transitions to mechanical articulation.
d. Design of piping components:
   1. Expansion of piping within, or attached to, structures.
   2. Induced piping deflections and loads resulting from clearance and tolerance caused eccentricities.
   3. Local loads, fatigue, and wear at riser connections.
   4. Location of emergency shut down and isolation valves.
   5. Ballast piping configuration to avoid multiple compartment flooding in case of damage.
   6. Compatibility of fluid swivel design and seal materials with fluid properties design conditions.

10 Dynamic Positioning System

10.1 BASIC CONSIDERATIONS

Dynamic positioning (DP) is a technique of automatically maintaining the position of a floating vessel within a specified tolerance by controlling onboard thrusters that generate thrust vectors to counter the wind, wave and current forces. Advances in control and position reference systems and improvements in component and overall system reliability in the past decade allows station keeping in severe environments and deepwater locations for extended periods of time.

Initially, DP systems were used mainly for deepwater drilling operations. Recently, the use of DP systems has been extended to assisting conventional mooring systems for drilling and floating production operations. Thruster assist can extend the water depth capabilities of a conventional mooring system for harsher environments. Additionally, automatic thruster assist will add significant damping to a conventional mooring system resulting in reduced vessel motions. DP systems are also used on offloading tankers, construction, pipelaying, and diving support vessels.

The guidelines in this section were derived mainly from many years of experience in designing and operating DP systems on MODUs. DP systems for operations other than MODUs may have different requirements that are not addressed in this section. It is hoped that this section can be

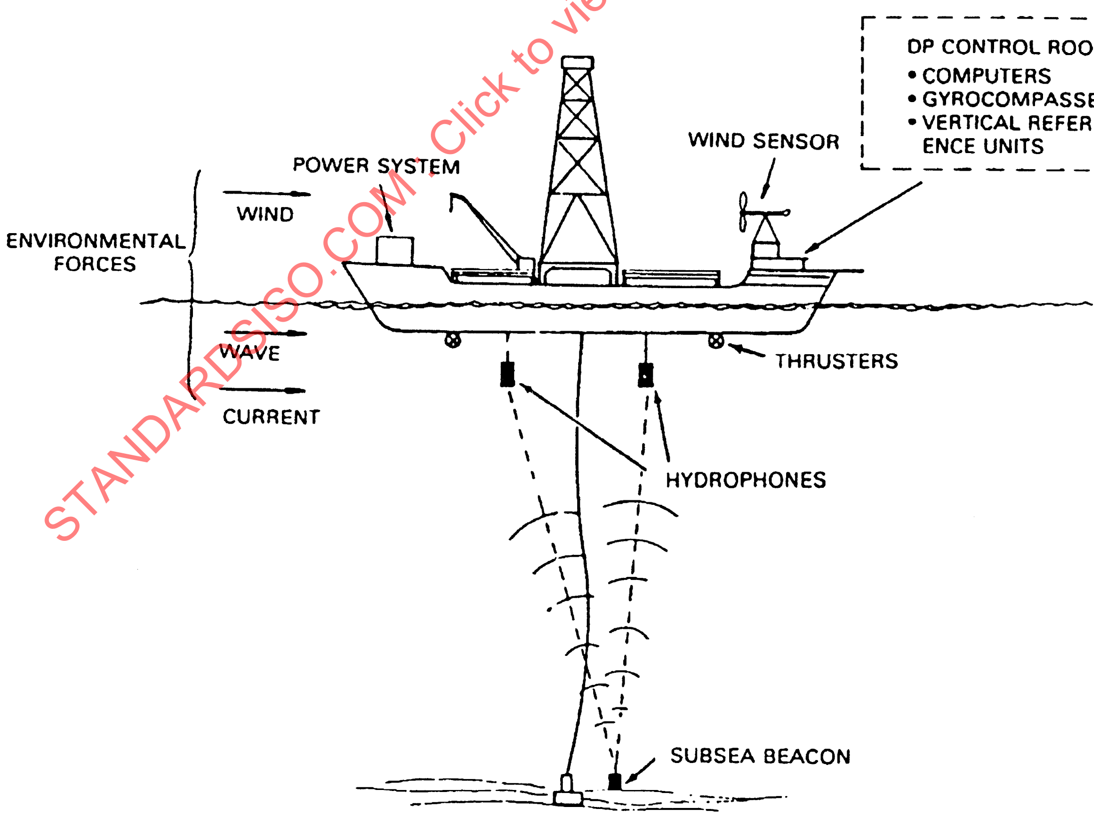


Figure 23—Major Elements of a Dynamic Positioning System

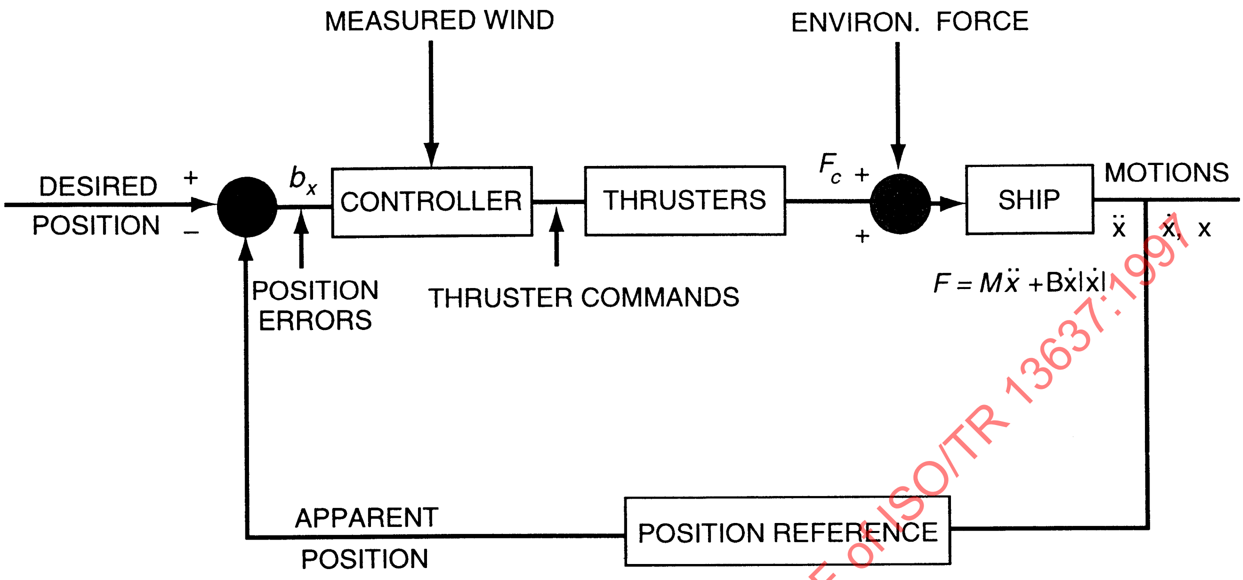


Figure 24—Dynamic Positioning Control Loop

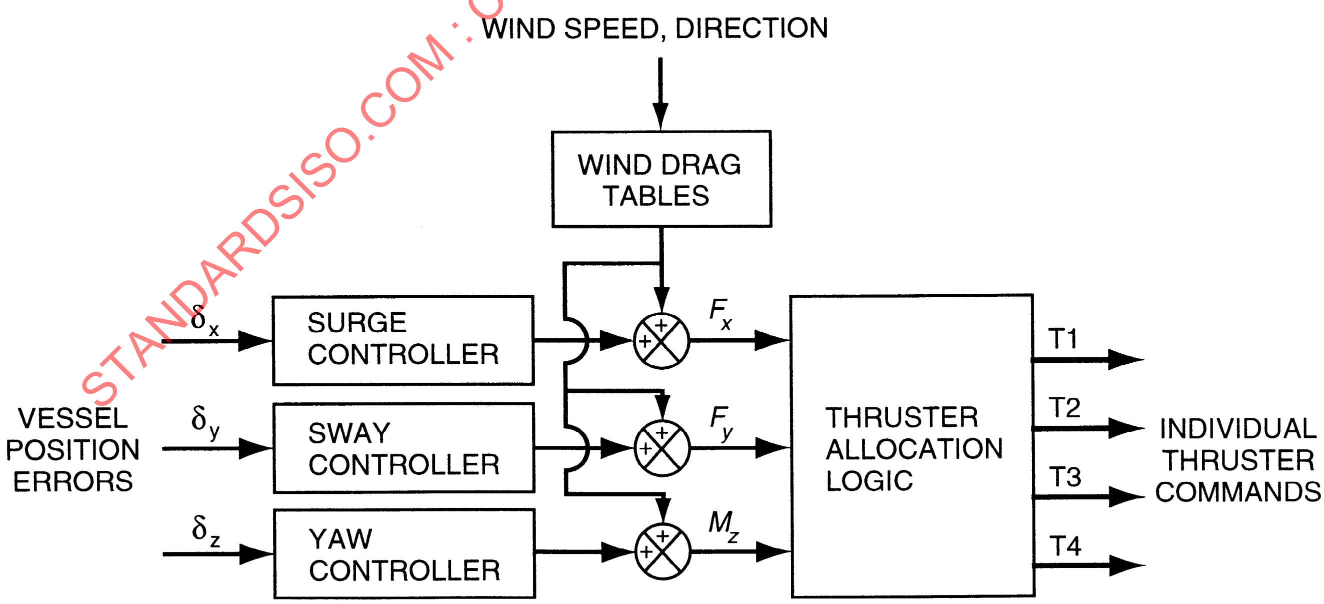


Figure 25—Three Axis Controller and Thruster Allocation Logic

expanded in the future to cover a wider range of operations.

Compared with conventional mooring systems, DP systems have advantages and disadvantages. The advantages of DP systems include:

a. Ability to operate in deep water (greater than 5000 feet).
b. Ability to quickly establish and leave location.
c. Rapid move-off capability during environmental hazards (such as, storms and icebergs)
d. Ability to turn vessel's heading to reduce environmental loads and vessel motions (only a turret-moored vessel can do likewise).
e. Ability to start up in higher sea states in which workboats would not be able to deploy a mooring system.

The disadvantages of DP systems include:

a. High initial equipment costs.
b. Higher operating costs due to higher fuel consumption, higher maintenance costs, and the need for specially trained personnel to operate the sophisticated equipment.
c. Decreased margin for safe operation in shallow depths say (less than 500 feet) where moored vessels operate regularly.
d. Shake-down time is required if the DP system is new or has just gone through a major modification.
e. Maintenance and periodic testing are more critical than for moored vessels.

This section does not address issues for detailed designs of DP systems. Instead, it provides guidelines for the design, test, maintenance, and operation of DP systems. More specific design requirements could be incorporated in the future.

10.2 BASIC CONCEPT AND MAJOR ELEMENTS

10.2.1 Basic Concept

The major elements of a DP system include the control system, the sensor system, the thruster system, and the power system. As shown in Figure 23, the computers in the control system process the position information provided by the sensors to compute the vessel position at regular time intervals, typically about once every second. Based on the position error (desired minus actual position), the computer calculates the control commands to the thruster system. The thrusters, powered by the power system, provide the necessary forces to counter the environments and maintain the vessel on location.

The major elements of the DP system, including the vessel, form a feedback control loop as shown in Figure 24. There are three independent control loops, one each for the surge, sway, and yaw axes of the vessel. The three control axes are coupled by the thruster allocation logic in the computer as shown in Figure 25.

10.2.2 Control System

The control system is an onboard digital computer. Its main functions are: (a) to process the sensor information and compute the instantaneous position of the vessel, (b) to calculate the force and moment required to counter the environment and minimize the position errors, and (c) to allocate the thruster forces according to some programmed logic to obtain the required total force and moment. All of the calculations have to be performed at high speed to ensure acceptable system performance. In most DP systems these calculations are done once per second.

a. PID controller—The conventional controller calculates the control thruster commands based on the position error, the rate of change in the position error, and time integral of the position error. This controller is commonly referred to as the PID (proportional-integral-derivative) controller. The proportional control provides the thrust that is analogous to the spring force generated by a mooring system when the vessel is offset from the equilibrium position. The derivative control provides the controlled damping, and the integral control is required to maintain a zero mean position error.
b. Kalman filter controller—A relatively new generation of controllers use what is known as Kalman filtering technology. Functionally, these modern controllers are analogous to the PID controller; the manner in which the proportional, integral, and derivative terms are computed is different. The Kalman controller has a much better position signal processing logic and results in improved system performance, especially in situations where the position signals are constantly contaminated by ambient noise that may result in excessive thruster modulation (fluctuation in thrust output). Another situation where the Kalman controller is superior is when the DP system is in a dead reckoning mode following a complete loss of all the on-line position sensors. Better dead reckoning performance allows the operating personnel more time to deal with the situation.
c. Feed-forward forces—In addition to the position error generated force commands, the controller also calculates the so-called feed-forward wind and/or current forces based on the on-line sensor information and the drag coefficients. These forces are then summed together to form the total thrust commands that are fed to the thrusters (Figure 25). The feed-forward feature in the control system helps minimize the vessel excursions in rapidly changing wind environments.
d. Stability gain margin—The basic performance characteristics of a DP closed loop control system are the system response time, the thruster modulations, and the stability of the system. The gain parameters in the controller are designed to provide acceptable system response time and minimal thruster modulation and to ensure that the closed-loop system is stable. These gain parameters are determined based on the vessel characteristics, thruster capacity and response, position sensor noise characteristics, and the expected operating conditions of the specific DP vessel. A comfortable margin in these gain parameters should be provided to accommodate the uncertainties and nonlinearities of the system. It is important to perform gain margin tests on each of the three control

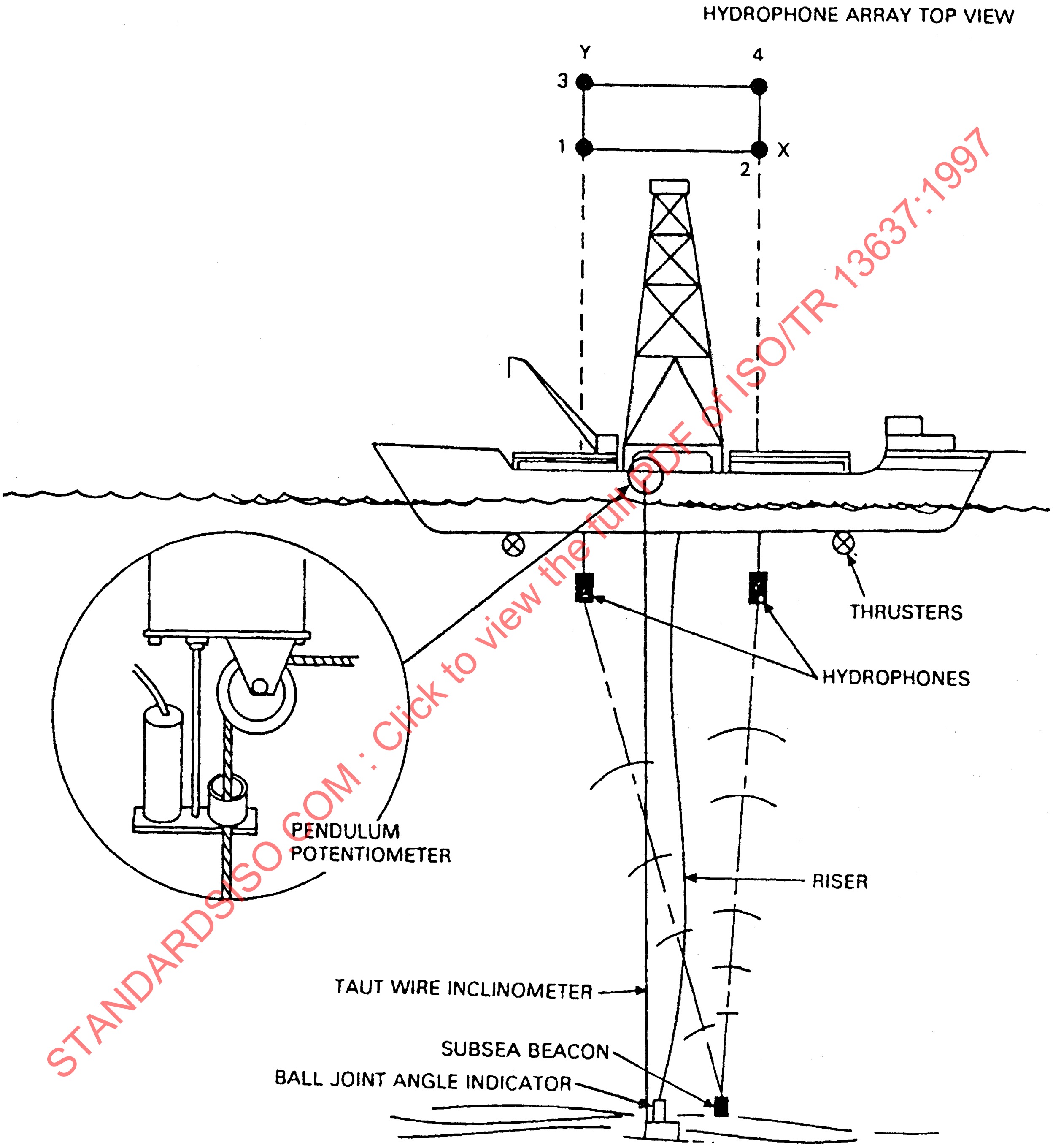


Figure 26—Position Sensing Systems

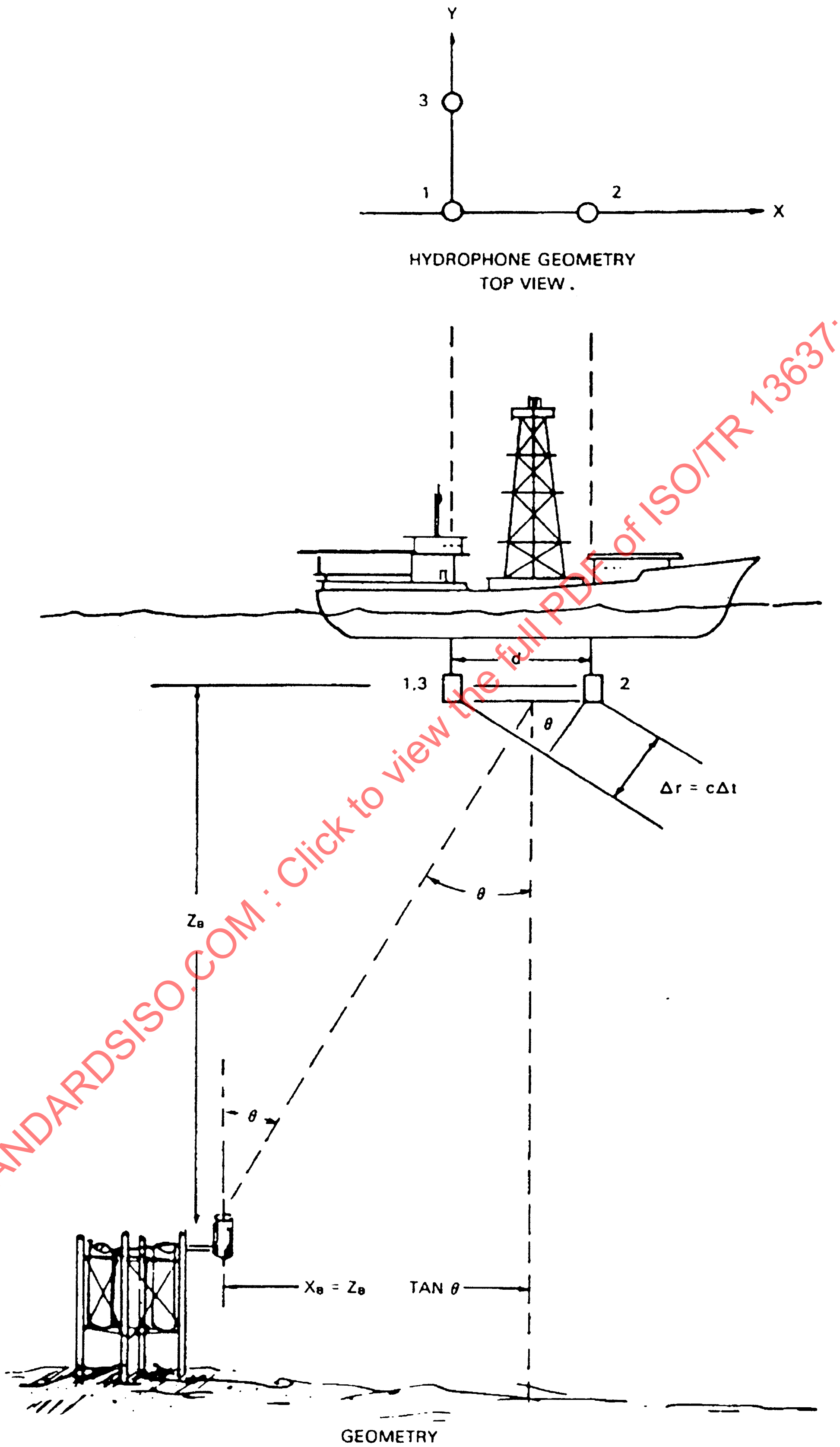


Figure 27—Principle of Operation of the SBS

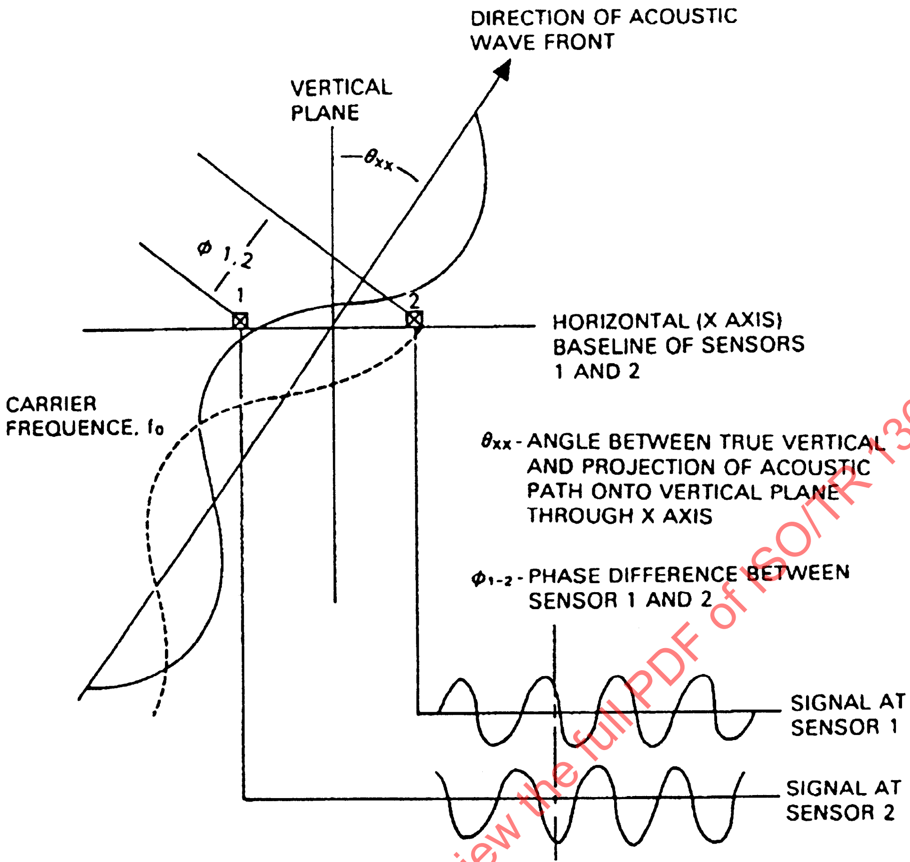


Figure 28—Acoustic Wave Phase Relationship

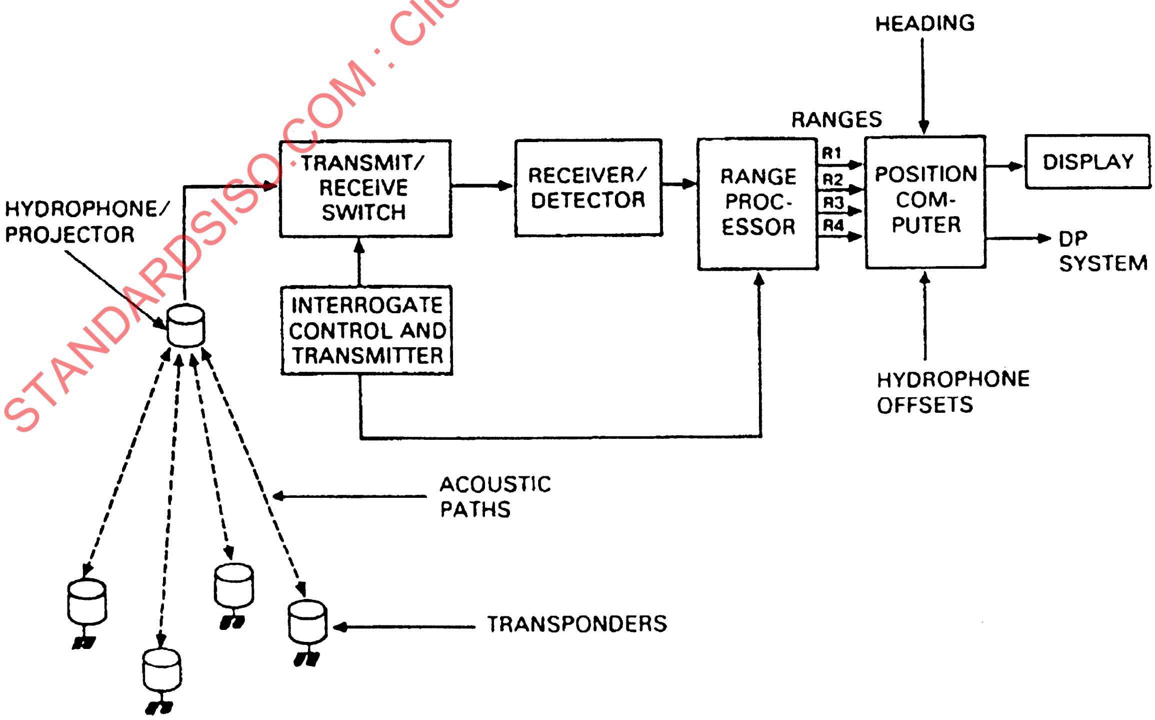


Figure 29—System Implementation of a Long Baseline System

axes for a new or modified DP vessel to ensure adequate margin of stability.

10.2.3 Sensor System

The position sensors continuously measure the position of the vessel for input to the controller. The measurement has to be highly accurate (less than 1 percent of water depth) and repeatable. For surge and sway measurements, the most commonly used sensor systems are the acoustic, the taut wire, and the riser angle systems (Figure 26). The acoustic system is normally the primary position sensor because of its reliability and accuracy under a wide range of environments. The taut wire and the riser angle systems are usually used as the backup sensors due to the angle signal noises induced by the cable and riser dynamics. Short range, microwave radar systems can also be used as the primary position sensor if the remote stations can be set up around the location. Recently satellite position reference systems became available in certain parts of the world, and they have been used successfully in a number of DP operations.

For yaw or heading measurement, gyrocompasses have a proven record of reliability and accuracy in marine environments. Pitch and roll measurements of the vessel are also required for the computer to perform accurate position calculations. These measurements are made by the VRU (vertical reference units) that come in different designs; however, in all designs, insensitivity to shock and ability to not respond to lateral accelerations are important considerations. Of the three major environmental forces (wind, wave, and current), normally only wind (speed and direction) is measured and input to the computer.

10.2.3.1 Acoustic Position Sensors

There are three types of acoustic sensors:

a. Short baseline system (SBS): This conventional system uses an array of receiving elements called hydrophones which are mounted on the vessel as shown in Figure 27. A minimum of three or more hydrophones are required. The time-of-arrival of an acoustic pulse transmitted from a subsea beacon is measured at these hydrophones, and the differences in time-of-arrival are computed. Once the difference in the time-of-arrival is computed for each vessel coordinate axis, the direction angles and the offset from the vessel to the beacon can be estimated. Figure 27 summarizes the geometry of the hydrophones and the subsea beacon.

b. Ultra short baseline system (USBS): This system uses phase difference measurements of the incoming acoustic pulse to estimate the direction angles and the offset from the vessel to the subsea beacon (Figure 28). The distance between the receiving elements for this phase measurement system is only 2 inches or less. Therefore, the entire array of receiving elements can be packaged in a single hydrophone assembly. That is, the ultra short baseline system requires only one hydrophone to provide the position estimates for dynamic positioning.

USBSs have degraded accuracy associated with transponders directly below the transducer, and for high offset transponders, where ray bending is a factor. The former limits USBS utility for applications such as DP drilling in deep water. The accuracy of this system is strongly determined by the performance of the vertical reference sensors.

c. Long baseline system (LBS): This system uses an array of transponders at the sea bottom and a reference interrogator/receiver on the vessel as shown in Figure 29. This is a purely range measuring system that provides the ranges from the vessel to each subsea transponder. The position computer calculates the vessel position using these range measurements and the coordinate system initially chosen based on the precalibrated locations of each transponder. This system provides excellent position accuracy and wide area coverage. It is not water depth sensitive within most water depths and is by far the most accurate accoustic position reference system. It requires a minimum of three transponders (for measured ranges) for the vessel position calculation.

d. Combination system: In addition to the basic types of acoustic position reference system systems (short baseline, ultra short baseline, and long baseline) there are now combinations of these systems, e.g., long short baseline systems (LSBS), that have been developed to provide certain applications with greater reliability, flexibility and accuracy than any single type of system can provide.

10.2.3.2 Taut Wire Systems

The taut wire position reference system (TWS) is basically a mechanical system in which the inclination of a line under constant tension is measured and transmitted electrically to the computer for vessel offset calculations. The major elements of a TWS include (a) the taut wire that includes the cable, anchor, and tensioner; (b) the fairlead follower and sensor gimbal mechanism; and (c) the angle sensor or inclinometer. Figure 30 illustrates the basic elements of a TWS and an example of TWS installation. The system provides satisfactory position estimates for dynamic positioning at up to 500 foot water depth. The position accuracy in deep water (beyond 1000 feet) is often less than satisfactory because of dynamics of the taut wire subjected to hydrodynamic drag and vessel motions. It is commonly used as a backup sensor to the primary acoustic position reference system in shallow water drilling operations.

10.2.3.3 Riser Angle System

The vertical angle of the marine riser at the lower ball joint (or flex joint) is critical to the drilling operation because the drill string may wear against the inside of the upper part of the BOP stack if the marine riser and the BOP stack are misaligned. In fact, the drill string could hang up in the BOP stack if the misalignment exceeds 5 or 6 degrees.

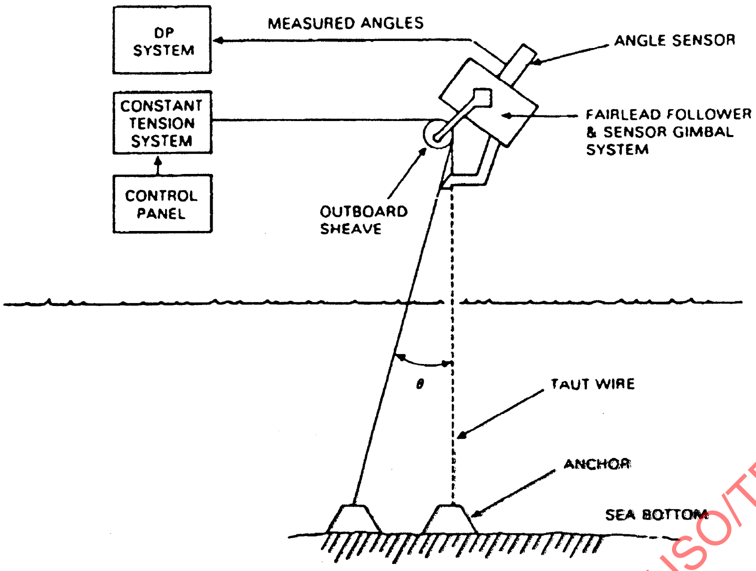


Figure 30—Taut Wire System

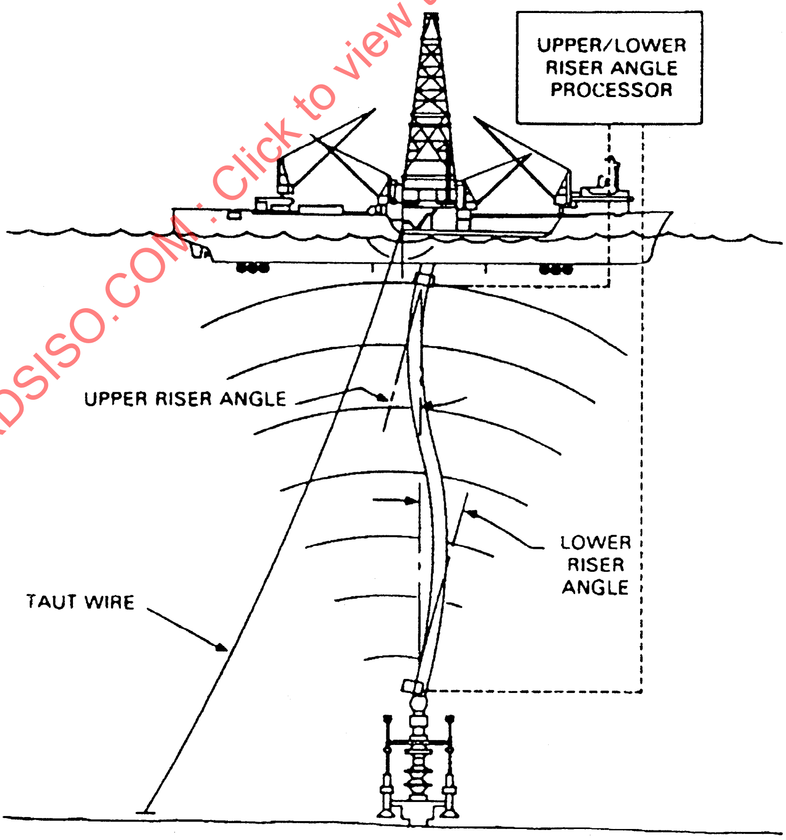


Figure 31—Adaptive Riser Angle Reference System (ARARS)

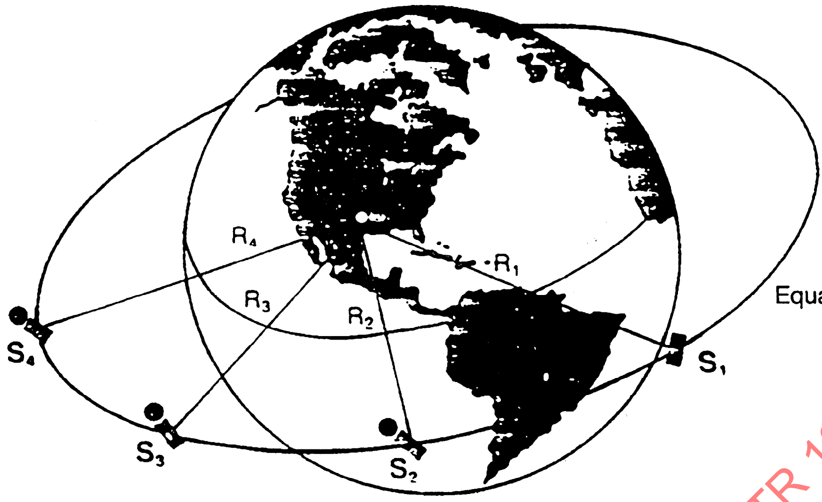


Figure 32—Communication Satellites for STARFIX System

Because of the importance of this riser angle, an angle sensor is installed on the riser just above the ball joint to monitor the ball joint angle continuously during drilling operations. A riser angle position reference system also uses this angle information to calculate the vessel offset from the wellhead. Like the taut wire system, the behavior of the marine riser in deep water is affected by the hydrodynamic drag on the riser. Moreover, the riser behavior is also affected by other parameters such as the top tension, mud weight, and buoyancy, which vary depending on the drilling operation. Therefore, the riser angle system is also commonly used as a backup position sensor to an acoustic system during drilling operations in deep water.

An improved riser angle position reference system is the adaptive riser angle reference system (ARARS) for deepwater applications (Figure 31). This system uses both the upper riser angle (at the slip joint) and the lower riser angle (at the ball joint) information and the pre-programmed riser characteristics to adaptively compensate for the riser dynamics and provide the vessel offset estimates. It performs explicit maximum likelihood identification of key riser dynamic parameters and adjusts the selected blending/shaping filters to achieve proper compensation.

10.2.3.4 Satellite Position Reference System

An example of a satellite position reference system that has been utilized as a DP position reference system in the Gulf of Mexico is the STARFIX system. STARFIX is a position reference system that uses up to eleven land stations to track and determine the precise location of four geosynchronous communication satellites located at approximately 75, 100, 120, and 140 degrees West longitude (Figure 32). One of the land sites, referred to as the master station, has up-link capabilities and transmits via the satellite's updated satellite position data. With known positions, the four satellites in essence become radio beacons that are tracked by the vessel's receiving equipment. Each satellite's position data is modulated on its 4 Ghz carrier signal and is updated on a 1.2 second interval. This information is used by the receiving equipment to determine the exact distance between the satellites and the azimuth stabilized receiving antenna.

As both the satellites and the receiving antenna are referenced to the center of the earth, the position of the receiving antenna in latitude and longitude is easily determined by the STARFIX system. This position information is output in the basic form to five decimal places as NN.nnnnn (latitude) and WW.wwwww (longitude) in decimal degrees. By comparing this position information with the actual latitude and longitude of the well head, position reference information can be input into the DP control system.

Recently a new satellite position reference system, the differential global positioning system (DGPS), is available to offshore operations. This system is available nearly everywhere worldwide and therefore is in wider use than STARFIX as a DP position reference.

10.2.3.5 Radar Position Reference Systems

When the desired position location is located close to land or an offshore fixed platform, a radar position reference system can be used. High resolution radar is utilized on board the vessel to fix the range to several radar reflecting targets placed at known locations. Using this range information and the known geometry of the reflectors, the system will give a highly accurate position of the vessel. The major drawback to this type of system is the fact that the targets must be stationary, necessitating the proximity of land or a fixed platform.

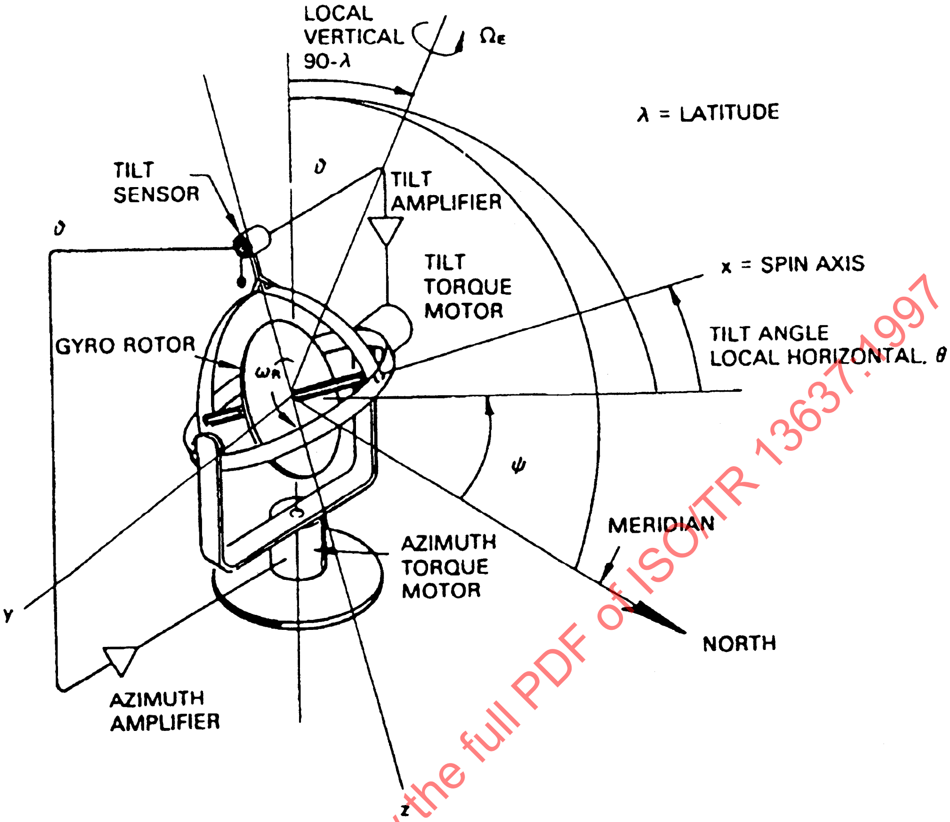


Figure 33—Gyrocompass Schematic

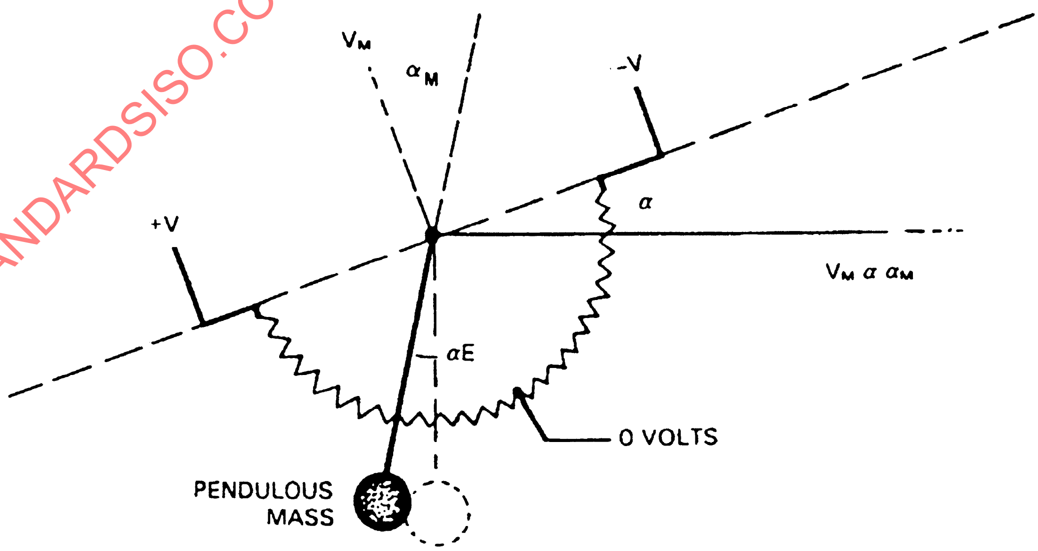


Figure 34—Pendulous Mass Vertical Reference Sensor Schematic

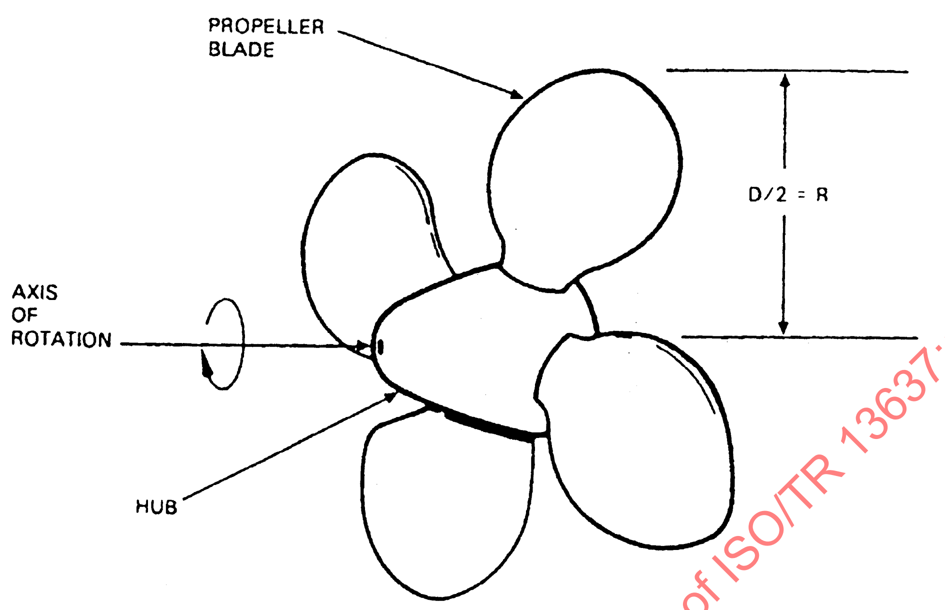


Figure 35—Open Propeller-Type Thruster

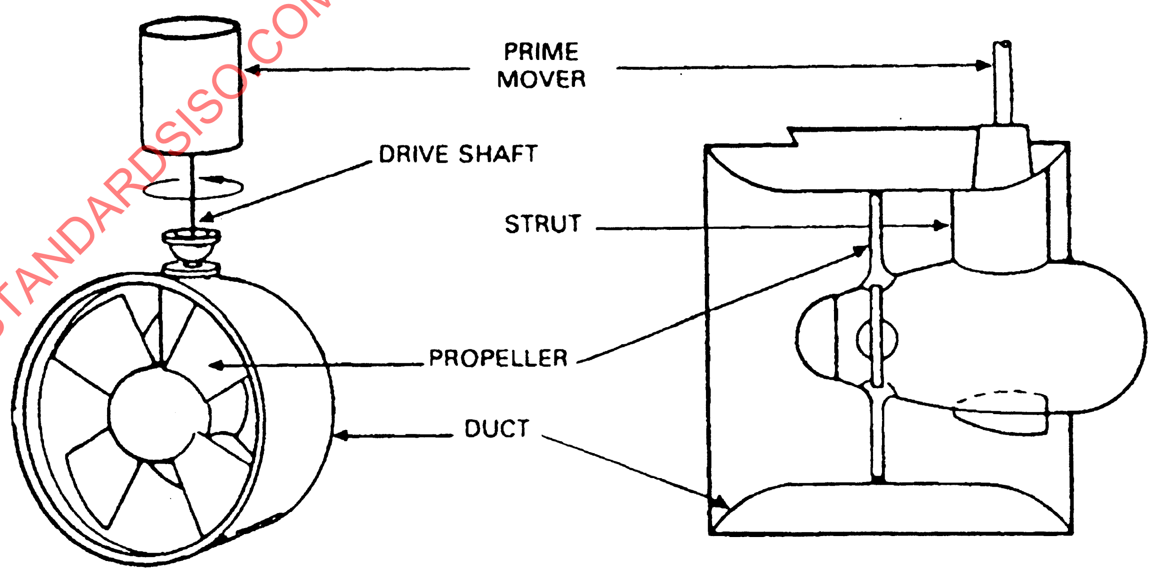


Figure 36—Typical Ducted Propeller

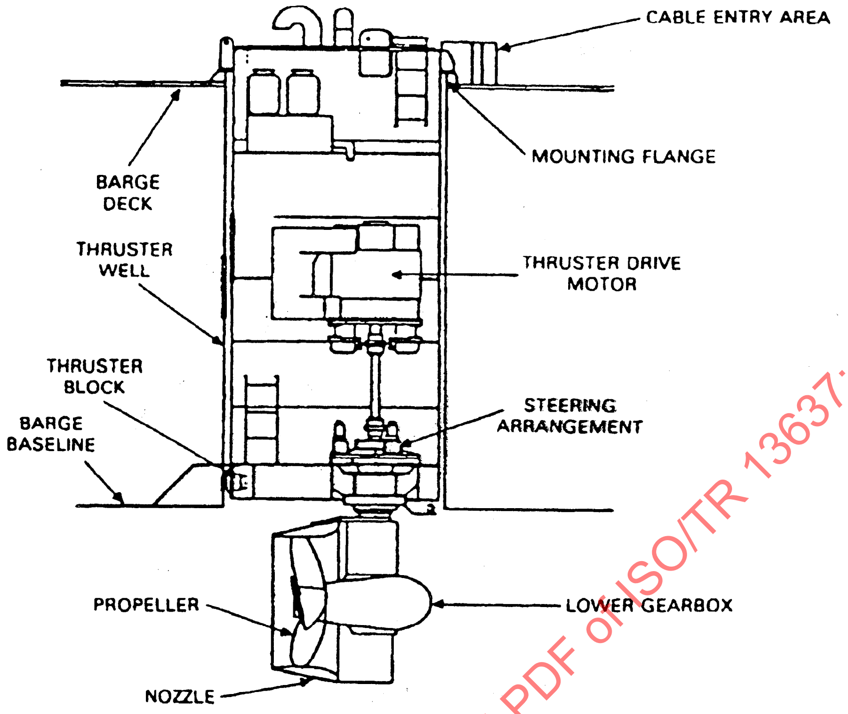


Figure 37—Typical Retractable Thruster Configuration

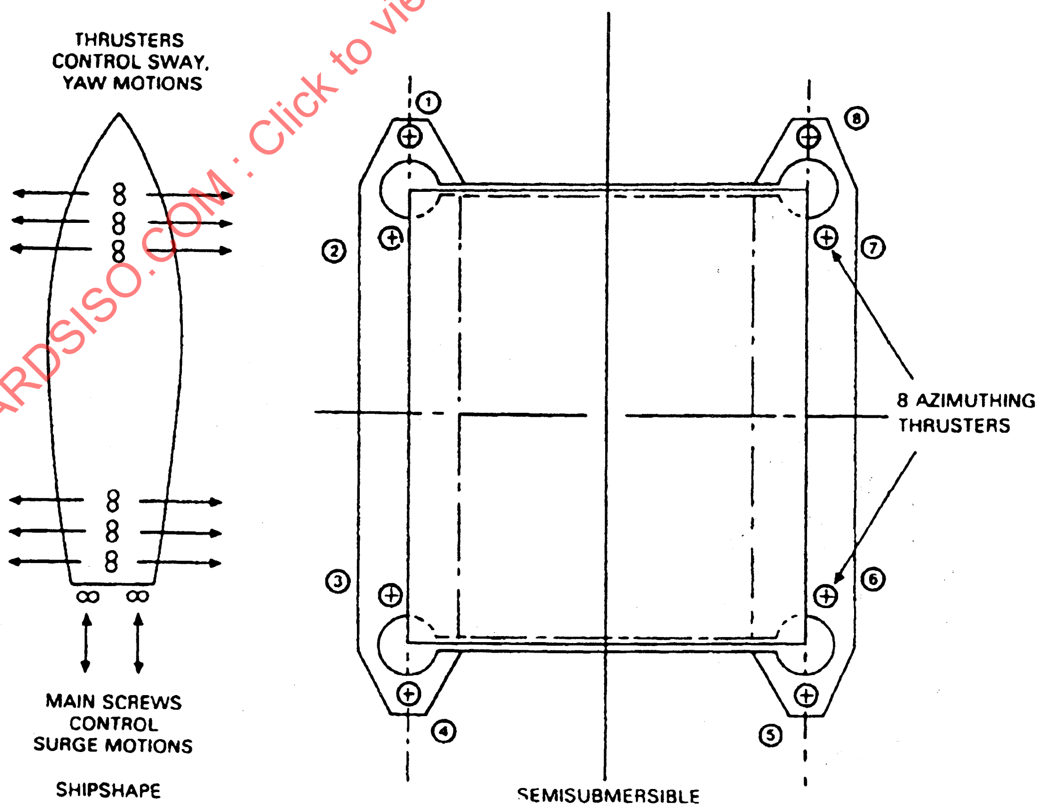


Figure 38—Typical Thruster Configuration

The system can also be affected by atmospheric conditions such as rain or fog.

10.2.3.6 Heading Sensors

Control of the vessel's heading is most often used to maintain the orientation of the vessel relative to the weather. A favorable heading with respect to the weather can reduce both the thruster loadings and the vessel motions that are desirable for stationkeeping and equipment handling during operation. The vessel's heading information is also necessary for coordinate transformations between vessel coordinates and earth coordinates in the horizontal plane.

The heading of a vessel can be measured either with a magnetic compass or a gyrocompass. The most commonly used heading sensor is the gyrocompass which has a proven record of reliability and accuracy in marine environment. The gyro compass is basically a two-axis gyro that senses the earth's rotation causing the gyro to align its spin axis in a northerly direction (Figure 33). The heading referenced to north is thus determined.

10.2.3.7 Vertical Reference Sensors

The vertical reference sensor is used to measure the pitch and roll angles between the local vertical and a reference plane on the vessel. These angles are used to compensate for the hydrophone plane variations in the SBS and USBS acoustic position reference system. The angles are also used for display to the shipboard personnel and for ballast control.

There are many types of vertical reference sensors. The pendulous mass type sensor is the simplest and least expensive, as shown in Figure 34. Its performance is only marginally acceptable due to the effects of lateral accelerations which cause an error in the position measurement. The other types of sensors include vertical gyros and the combination of linear and angular accelerometers which provide significant improvement over the pendulous mass type sensor. A combination of Global Positioning Systems can also be used as a vertical reference sensor on some types of floating vessels.

10.2.4 Thruster System

The thrusters provide the forces and moments to counter the environment and hold the vessel on station. There are many types and manufacturers of thrusters. One fundamental requirement in thruster design and selection is to have minimal aeration and cavitation that cause attenuation of the acoustic signals once they get into the transmission paths. Thruster noise can severely degrade the performance and reliability of the acoustic position reference system.

10.2.4.1 Basic Design

The most common thruster design is the open propeller design (Figure 35), which is generally called a screw. Another type of thruster design places the propeller in a specially shaped housing called a duct or a tunnel as shown in Figure 36. The primary reason for adding the ducting is to increase thruster efficiency or thrust output per unit horsepower. These ducted thrusters may also be retracted into the

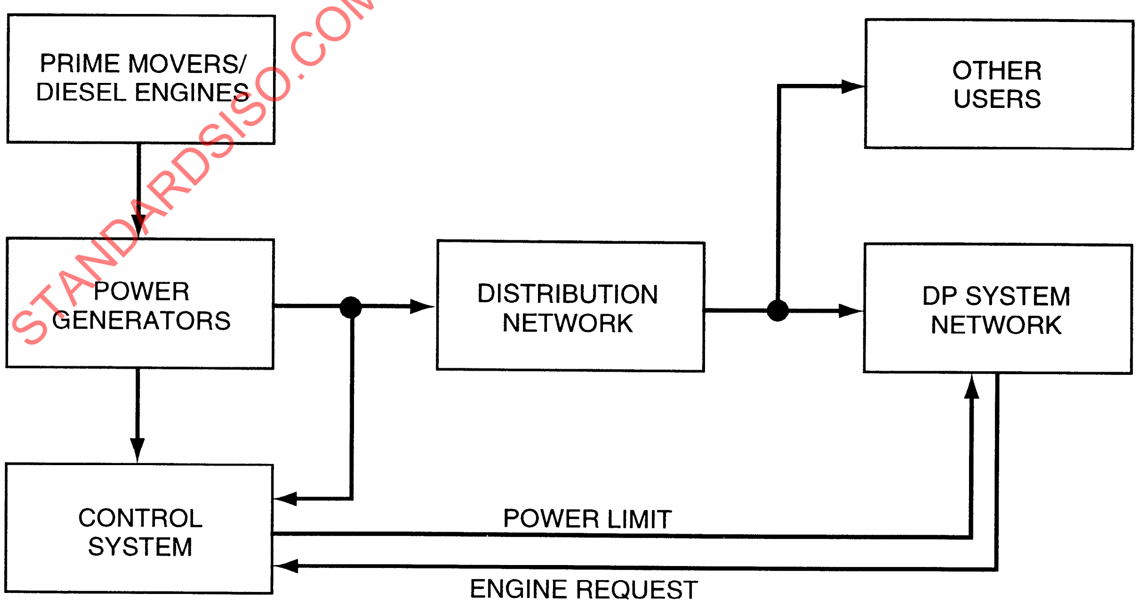


Figure 39—Basic Elements of a Power System

hull of the vessel during transit to reduce hydrodynamic drag (Figure 37).

10.2.4.2 Thrust Direction Control

In selecting the type of thrust direction control for a specific vessel, the basic choices are fixed axis where the thrust direction is fixed or azimuthing where the thrust direction of each thruster can be changed continuously. Figure 38 illustrates the typical thruster configurations on a shipshaped vessel with fixed axis thrusters and on a semisubmersible with azimuthing thrusters.

10.2.4.3 Thrust Level Control

There are two basic types of thrust level controls. Thrusters can be either controllable pitch with fixed speed or fixed pitch with controllable speed. The choice is usually made based on economics, experience, and the specific design environments. The availability or selection of the power system can also dictate the type of thrusters for the vessel. Controllable pitch thrusters with fixed speed are normally driven by constant speed AC motors. Fixed pitch thrusters are normally driven by DC motors with controllable speed or variable AC drives. In general, AC motors are smaller and lighter than the corresponding DC motor, and AC systems do not require additional SCR (rectification) equipment to convert AC to DC. An offsetting disadvantage of the AC pitch control system is the requirement for a separate hydraulic system for the pitch control. In shipshape vessels, the main screws may also be directly driven by diesel engines with pitch control for the thrust variations.

On a DP vessel there is generally a separate thruster control panel nearby the DP control system. It provides the ability to enable/disable thrusters for DP control, and it provides a manual control feature, as single or grouped thrusters. Often the Emergency Stop for thrusters is located in this panel.

10.2.5 Power System

The power system on a DP vessel is composed of the prime movers (usually diesel engines), the generators, and the control system, as well as the distribution/user system (Figure 39). Considering the amount of power required and the number of users, a centralized power system is the most cost and space effective approach. The power system is usually configured with multiple units and redundant features to handle power demands that range from very little in light weather to maximum in heavy weather.

To handle the required power distribution and usage quickly, efficiently, and safely, a few DP vessels also have a separate power management system that can monitor the system performance, maintain records (data logging), and make logical decisions in real time. This power management system may be co-resident with the DP system in the same computer or in a separate computer.

A power management system is configured to perform the following major functions:

a. Automatically start any generator unit.
b. Detect a malfunction in the power system.
c. Limit the selected user loads in critical situations.
d. Regulate the loading of the power system.
e. May provide data recording for system maintenance and failure analysis.

These functions provide the capability to react as fast as possible to emergency situations such as generator failures, or to limit the power usage to the maximum allowable level to prevent system blackout or brownout conditions during sudden power surges.

An uninterruptible power supply (UPS) unit (battery backup) is also required to ensure clean power for the DP system electronics and for computer memory protection during a power blackout condition.

10.3 DESIGN AND ANALYSIS

10.3.1 Basic Design Philosophy

A DP system should be able to keep a vessel in position within certain excursion limits under the design environment. Since the consequences of losing station can be serious, DP systems should be designed to have high reliability and a certain amount of built-in redundancy. As with any complex system, DP component failures are difficult to avoid. Components that are expected to fail occasionally are backed up by other similar or dissimilar components that can perform the necessary function when the primary equipment is either temporarily or permanently out of order. Temporary can mean from 10-20 seconds in the case of the prime reference system such as acoustic signal drop-outs, or it can mean the time between shipyard visits when, for example, major thruster repair work is required. To make best use of all the redundant equipment, there must be a method of automatically switching to the functioning piece of equipment. The onboard DP computer normally monitors the equipment status and performs most of the switching automatically. However, on some vessels certain switching must be done manually.

10.3.2 Guidelines for Design, Test, and Maintenance

It is not the intention of this document to specify detailed reliability and redundancy requirements for various DP systems. Such requirements should be determined by the DP system designers based on operation, cost, safety, and environment considerations. Instead, the following guidelines are provided for developing functional and reliable DP systems. The following guidelines were derived from many years of DP design and operation experience:

a. In the design process, a DP system must be considered as a complete system of its four major elements: the sensor system, the control system, the thruster system and the power system. Furthermore since the DP system can interact and influence every other system on board a vessel through a common element such as the power system and vice versa, a total system design of the vessel is necessary to obtain the highest possible dynamic positioning reliability.
b. Multiple thrusters and power generators are necessary to achieve the high degree of availability required for dynamic positioning. Each unit should be properly sized, highly reliable, and easy to repair and maintain.
c. Active or on-line redundancy is necessary in all elements of the DP system at all times coupled with high subsystem reliability. Additionally, standby reserve is required for mechanical systems such as thrusters and power generators so that they can be configured for various environmental loads, increased longevity, and easy repair and maintenance.
d. Increased reliability through redundancy and reliable subsystems is not sufficient to achieve the necessary availability required for dynamic positioning. The objective of fault tolerance should be achieved with features that include the following:

1. No single point failure.
2. Automatic fault detection.
3. Isolation of fault and recovery that is automatic and independent.
4. Automatic restoration of devices.
5. On line repair.

e. The DP control software should possess certain features such as active wind compensation, sensor data filtering, data reconciliation, dead reckoning, and so forth.
f. Each element of the DP system should be thoroughly tested prior to commissioning the vessel. Factory, dockside, and at-sea testing to a comprehensive test plan should be performed. DP control software should be thoroughly tested using dynamic simulators to verify normal performance and performance with system failures. On-going testing and training should be performed throughout the life of the system.
g. Careful design of the system is necessary to avoid any single points of failure and to make possible on-line repair, ease of maintenance, and on-line diagnostics.
h. Cabling and connector design and implementation are very critical to achieve long duration operation without failures. Cabling and connector failures can affect multiple elements of the system at the same time. Shipboard vibration must be taken into consideration.
i. Position measurement sensors that use independent methods of measurement are required so that a common failure does not affect all the systems at once. For example, an acoustic position measuring system should be used with a satellite navigation system instead of another acoustic system.

The guidelines given here only consider the DP system. Equally important, if not more important, is the reliability of the operator and repair people and the support system for repair and maintenance of the DP system.

10.3.3 Reliability Analysis

A DP system consists of various electrical and mechanical components which are susceptible to failures. To improve reliability of such a system, redundant components are incorporated. Since the failure rates for each component are different, and different DP systems incorporate different levels of redundancy, it is difficult to quantify the reliability a DP system. DP reliability analysis is one of the tools to yield objective estimate of DP system reliability.

10.3.3.1 Basic Reliability Model

The basic reliability model of DP system of a particular vessel is a series combination of the sensor, control, thruster and power systems. This is referred to as the Level 1 model (Figure 40).

For each element in the basic model there is a corresponding failure rate and average repair time. The corresponding MTTF (mean time to failure), MTTR (average repair time or mean time to repair) and steady-state availability of the DP system are computed as follows:

$$MTTF = 1/(F_1 + F_2 + F_3 + F_4) \tag{10.1}$$

$$MTTR = (F_1T_1 + F_2T_2 + F_3T_3 + F_4T_4)/(F_1 + F_2 + F_3 + F_4) \tag{10.2}$$

$$A = \frac{MTTF}{MTTR + MTTF} \tag{10.3}$$

$$D = T(1 - A) \tag{10.4}$$

Where:

F_1,F_2,F_3,F_4 = failure rates ($1/MTTF$) of the sensor, control, thruster, and power systems, respectively:

T_1,T_2,T_3,T_4 = repair time ($MTTR$) for the sensor, control, thruster, and power systems, respectively:

A = availability.
D = downtime.
T = operation period.

Following is an example of reliability analysis for a drillship. The *MTTR* for the thruster and power system is zero because standby reserve is available.

	MTTF (hours)	MTTR (hours)	Availability
Sensor system	8,311	2.4	0.999706
Control system	1,349	1.9	0.998594
Thruster system	572	0	1.0
Power system	242	0	1.0
Total system	148	0.3	0.998042

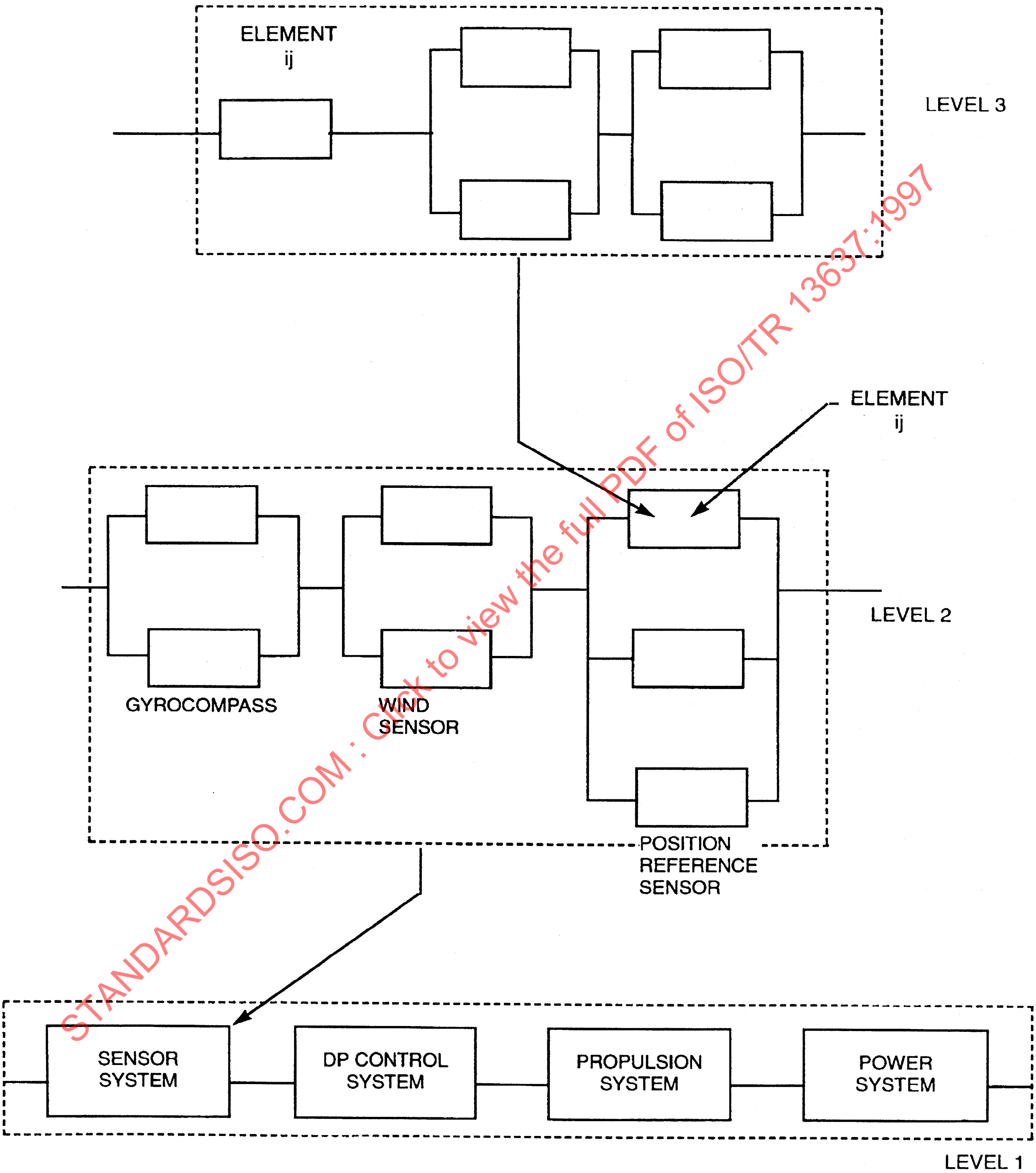


Figure 40—Generation of Reliability Measures at the Various Levels of the DP System

For a 100-day well, the expected downtime is:

$$D = 100 \times 24 \times (1 - 0.998042) = 4.7 \text{ hours}$$

10.3.3.2 Multileveled Reliability Model

Within each of the basic four elements there are various combinations of series and parallel elements that fulfill the primary functional requirements of the basic elements. These elements form Level 2 in the overall DP reliability model. At Level 2 a DP system may have multiple devices providing the same function. These multiple devices create parallel paths within the series structure of the four basic elements. An example of Level 2 model follows:

Sensor System:	Gyrocompass Position reference sensor Wind sensor
DP Control System:	Computer and input/output Operator console Uninterruptable power supply Operator
Thruster System:	Main screw Lateral tunnel thrusters Azimuthing thrusters
Power System:	Power generating system Power distribution system SCR unit

If failure data is available for parts of the Level 2 elements, then further refined models of the various Level 2 elements can be achieved. For example, if reliability data is available for the diesel engines, the electric generator, and the auxiliary support equipment, then a Level 3 model can be created for the thrusters. Within the Level 3 model parallel elements may exist and can be incorporated into the Level 3 model.

This approach can be carried out down to the most fundamental level of each part of the DP system. However, generally speaking there is not any reliability data available past the Level 3 for DP systems as they are operated and maintained today. A three-level reliability model is illustrated in Figure 40. The mathematics for a multilevelled reliability model consisting of multiple series and parallel elements is complex, and the calculation is normally carried out by a computer program.

10.3.3.3 Applications and Limitations

The reliability and downtime estimates computed by a reliability analysis can be used for comparison purposes, such as comparing in a relative sense the estimated downtime of two drilling vessels or the reliability of two DP design alternatives. They should not be regarded as realistic predictions for the following reasons:

a. The reliability model accounts for DP component failure only. Other factors such as environmental effects and human interactions are not included.
b. Because of a lack of automatic recording systems for failure logging, some of the DP reliability data can only be taken from people's recollections which are subjective and less precise.
c. Long term DP reliability data were available only from DP vessels built many years ago. Although their operational lives are sufficiently long to develop valid statistics, the DP systems utilized do not incorporate some of the latest advancements in the DP technology.

10.3.4 Determination of Stationkeeping Capability

A holding capability analysis should be performed to determine whether a DP system can maintain the position of a floating vessel within an acceptable watch circle under the operating environment. This analysis should be performed for new designs as well as for individual operations.

10.3.4.1 Operation Limits

In a drilling or production operation, a floating vessel is tied to the seafloor by a conduit which can tolerate only limited vessel movements. For example in a drilling operation, the vessel is tied to the seafloor by either a riser or a drill string, and several critical vessel motion limits exist. When these limits are exceeded, the drilling operation must be suspended. A vessel at sea has six degrees of freedom including surge, sway, and vessel heading (yaw). Some systems have roll and pitch compensation algorithms with a purpose of damping out thruster induced roll and pitch, but no active roll and pitch compensation is available. Accordingly, a vessel may reach its environmental limit due to excessive roll, pitch or heave motions. These motions can only be indirectly controlled by the DP system through heading control which is effective for ship shaped vessels. For semisubmersibles, however, vessel motions cannot be effectively controlled by vessel heading control.

Since surge and sway limits are determined by the angle at which the riser or drill string is bent from the vertical, these limits are expressed in terms of percent of water depths instead of actual excursions. Because of this, a DP system must respond faster and with more power to counter vessel position variations in shallow water then in deep water. In general, drilling operations are limited by lateral excursion in shallow water while they are more likely limited by roll/pitch or heave in deep water.

Thus, the stationkeeping characteristics of a DP vessel are dependent upon three criteria. First, sufficient thrust must be available to overcome the mean environmental forces, and second, additional thrust must be available to control the surge and sway motions of the vessel. In the third place, the given environment must not cause excessive roll/pitch or heave motions. The first criterion is simply a matter of balancing the mean environmental loading with available thruster capacity. The second, however, involves not only thrust capacity and stability of control system, but also the dynamic responses of the vessel as a function of the applied environmental and thruster forces. The third involves the roll/pitch and heave responses of the vessel and are not affected by the DP system.

10.3.4.2 Environmental Loads and Vessel Motions

As stated in Section 5, a floating vessel is subjected to the following loads and motions:

a. Steady loads induced by wind, waves, and current.
b. Wave frequency motions.
c. Low-frequency motions.

A DP system counters steady loads and damps out low-frequency motions. Wave frequency motions are not affected by the DP system and therefore can be excluded from the holding capacity analysis. However, they should be included in the determination of maximum offset.

Steady wind, wave, and current loads can be estimated by methods as described in Section 5. One exception is that the period for average wind speed is 5-minute instead of one minute. Also, the wind force associated with a 30-second wind gust should be calculated. If the calculated wind force due to gusts is larger than 1.25 times the calculated static wind force, then the average wind speed should be increased such that the static wind force is 80 percent of the maximum wind gust force.

Determination of low-frequency motions for a DP vessel required a frequency domain motion analysis computer program that includes a definition of the DP system or a time domain simulation of the vessel and the DP system.

10.3.4.3 Available Thrust

Guidelines for determining available thrust are provided in Appendix C which deals with typical propulsion devices and installation scenarios for DP vessels supporting offshore operations. These include:

a. Open and nozzle propellers installed in the stern of a ship-shaped vessel (conventional main propulsion arrangement).
b. Azimuthing or fixed-direction nozzle thrusters installed under the bottom of a hull.
c. Tunnel thrusters installed in a transverse tunnel in a hull.

Two methods of thrust evaluation are provided:

1. Tables and figures for quick and rough estimates that can be used for the design of thruster assisted mooring and preliminary design of a DP system.
2. References for more rigorous determination of available thrust. They can be used for the final design of a DP system.

The estimated available thrust as determined by Appendix C should be further reduced under certain conditions as specified in 6.7.

10.3.4.4 Holding Capability Analysis

Two methods can be used to analyze the holding capability of a DP system. A time domain system dynamic analysis is normally performed for new system designs and critical operations, especially those in shallow water. For routine operations in deepwater, say water depths greater than 2000 feet, a simplified method addressing only the mean environmental forces can be used.

a. System dynamic analysis: This analysis for a DP vessel is similar to that for a vessel with a DP assisted mooring as described in 7.4.2. The major differences are as follows:

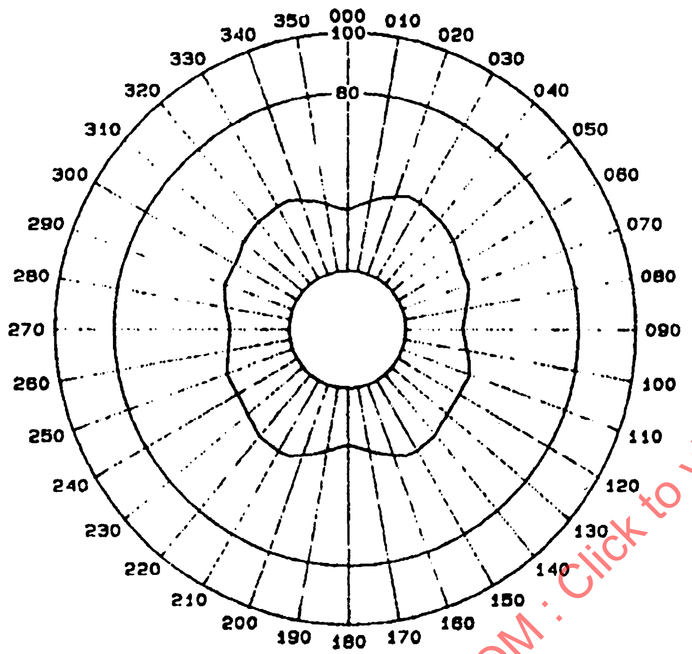
1. The mooring stiffness is not included in the analysis.
2. Step (4), Section 7, for the mooring analysis will not be performed. Instead the mean and low-frequency offsets from the system dynamic analysis will be combined with the wave frequency motions in a manner as described in 6.2 to yield the maximum offset.

b. Simplified method: In this approach, we assumed the DP holding capability is satisfactory if the required thrust output of the most loaded thruster to counter the mean environmental loads is less than 80 percent of the maximum available thrust. The 20 percent margin is reserved to damp out dynamic vessel motions. The analysis procedures are as follows:

1. Establish an operating environment and a vessel heading relative to the operating environment.
2. Calculate the mean surge and sway forces and the yaw moment due to wind, waves, and current.
3. Determine the required output of each individual thruster based on the DP system algorithm for thrust allocation.
4. Determine the available thrust for each thruster.
5. Calculate the required thruster output for the most loaded thruster in terms of percent of available thrust.
6. Repeat the previous steps for different headings or operating environments.

The analysis results are often plotted in rosette format of the required thruster output of the most loaded thruster in percent of maximum available versus vessel heading (Figure 41). The positioning window determined by this method includes those relative vessel headings to the design environment such that the most loaded thruster is at 80 percent of capacity or less. Examination of the total plot verifies that the installed thruster capacity is sufficient to yield the desired positioning window for the design environment. As seen in Figure 41, the positioning window is within 30 degrees either side of the vessel bow or stern for the drillship. In the case of the semisubmersible, the positioning window includes all headings relative to the environment.

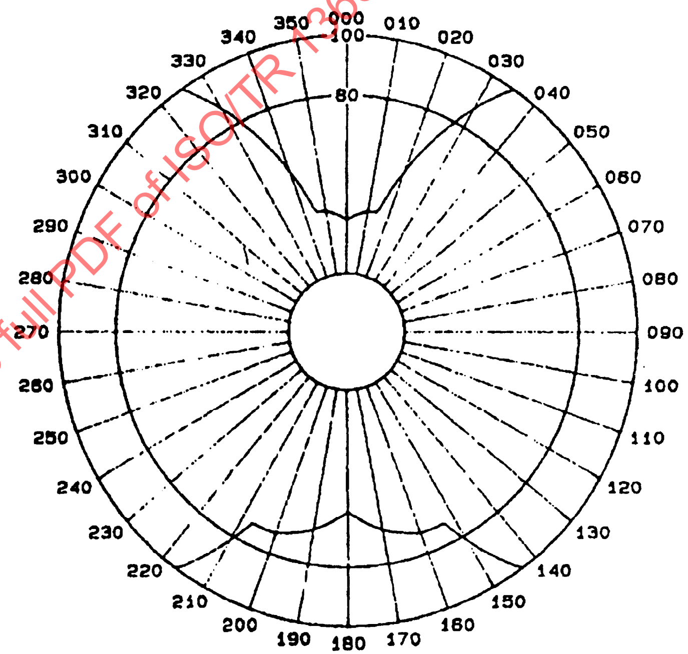
SEMISUBMERSIBLE



PERCENT OF AVAILABLE THRUSTER LOADING VERSUS VESSEL HEADING

NUMBER OF THRUSTERS AVAILABLE 8
VESSEL DRILLING DRAFT 70.0 Feet
WIND 54.0 Knots from 000 Deg
CURRENT .90 Knots from 000 Deg
WAVES 21.0 Feet from 000 Deg
RISER RESTORING FORCE 0 Kips

DRILLSHIP



PERCENT OF AVAILABLE THRUSTER LOADING VERSUS VESSEL HEADING

THRUSTERS AVAILABLE F-6 A-6 M-6
VESSEL DRILLING DRAFT 22.0 Feet
WIND 54.0 Knots from 000 Deg
CURRENT .90 Knots from 000 Deg
WAVES 21.0 Feet from 000 Deg
RISER RESTORING FORCE 0 Kips

Figure 41—Dynamic Positioning Holding Capability Rosettes

10.3.4.5 Example Analysis Using Simplified Method

To demonstrate the utility of the simplified method, the following example is presented for a typical drillship. The design environment for this exercise is as follows:

Wind (10 minute average)	54.0 knots from 000 degrees
Current (average)	0.90 knots from 000 degrees
Significant wave height	21.0 feet from 000 degrees
Peak wave period	8.5 seconds

Utilizing information in Section 4, the following environmental forces are calculated for the vessel on a heading of 020 degrees (20 degrees relative to the environment):

Environmental Force	Surge (Kips)	Sway (Kips)	Moment (Feet-Kip)
Wind	–48.4	59.9	2999.1
Current	–1.9	2.5	569.1
Waves	–12.0	33.7	4272.6
Subtotal	–62.3	96.1	7840.8
Cross-Coupling due to current sway into surge = 0.90	–8.6	N/A	N/A
TOTAL	–70.9	96.1	7840.8

The propulsion characteristics of the example drillship are as follows:

1. Six thrusters forward rated at 136.6 kips at an average lever arm of 136.26 feet.
2. Six thrusters aft rated at 135.8 kips at an average lever arm of 141.57 feet.
3. Two main shafts rated at 77.1 kips each in the ahead direction and 46.3 kips each astern.

Accordingly, simple thrust allocation logic yields the following required thrusts from the forward group (T_f), the after group (T_a) and the main screws (T_m).

Sway and moment requirements are defined as follows:

$$136.26\ T_f - 141.57\ T_a = 7840.8 \qquad (10.5)$$

$$T_f + T_a = 96.1 \qquad (10.6)$$

Solving the above two equations yields:

$$T_a = 18.9$$
$$T_f = 77.2$$

Surge thrust requirement is defined as follows:

$$T_m = 70.9$$

On a per thruster basis, T_f, T_a, and T_m are rewritten in terms of percent of capacity as:

$$T_f = 77.2 / 136.6 = 56.5 \text{ percent}$$
$$T_a = 18.9 / 135.8 = 13.9 \text{ percent}$$
$$T_m = 70.9 / 154.2 = 46.0 \text{ percent}$$

Thus at a vessel heading of 020 degrees, the most loaded thruster will be the forward group as seen in Figure 41. Note that at a vessel heading of 015 or less the most loaded thruster group will be the main screws. At 027 degrees, the forward thrusters are loaded to 80 percent of capacity defining the heading window with the environment on the bow as 027 degrees. Similarly, the heading window with the environment on the stern is 030 degrees.

10.4 DP SYSTEM OPERATION

Dynamic positioning systems by nature are highly complex and maintenance intensive systems. Although designed to be automatic, they must be continually monitored by an operator to ensure proper operation and coordinate system operation with other vessel operations. System failures that will be classified as stationkeeping problems can develop due to a large variety of reasons. Understanding and dealing with these failures are crucial for successful DP operations. Although the industry has gained significant experience in operating DP vessels for drilling, pipelaying, diving support, construction, and offloading, the most important lessons were learned from deepwater drilling operations. The following sections present operation guidelines developed mainly from the experience with drilling operations. However, some of these guidelines may apply to other operations.

10.4.1 DP System Failures

DP system failures can be generally grouped into the four categories described in 10.4.1.1-10.4.1.4:

10.4.1.1 Inability to Stay Within Watch Circle

When the environmental force exceeds the DP system's capacity, the vessel can be overpowered and blown off location. For example, this situation often occurs when squall wind shears in the Gulf of Mexico suddenly arrive from beam or quartering direction imposing significantly higher environmental loads on the vessel. The vessel can also be pushed outside the watch circle by a relatively mild environment in shallow water because the watch circle determined by the drilling riser is much smaller in shallow water. Another cause for loss of station is that the DP system response becomes unstable. In this case, instead of damping out vessel motions, the response of the system increases vessel motions causing the vessel to move outside the watch circle. Instability is a problem only for early DP systems.

10.4.1.2 System Equipment Failures

In general, DP systems are designed with sufficient redundancy such that single component failures result in loss of capacity and not in stationkeeping problems. Loss of power generating capacity in rough weather situations is probably the most common failure experienced of this type. If the system is equipped with an adequate power plant control system, such losses can easily be overcome by reduced capacity operation for the extremely short period of time it takes to automatically replace the failed skid. If there is no automatic power system controller, failures of this type quickly lead to a power plant blackout, ultimately causing a loss of station which is often referred to as a drift-off condition.

However, even in the most sophisticated systems, single point failures will still exist. A classic example of such a single point failure point that probably exist in most operating DP systems of today is in the regulated 120 volt power supply system feeding the DP system equipment. Most DP systems have dual uninterruptable power systems (UPS) to provide clean continuous 120 volt power to power the position reference and DP control systems. However, these dual UPS's are connected to the 120 volt bus through a static transfer switch. A failure in this switch can cause a complete loss of power to the processing components of the DP system thus causing a loss of control and loss of station.

10.4.1.3 Loss of Position Reference Information

DP systems need accurate and continuous position feed back information to operate properly. Since the normal update rate for digital DP processors is between $^1/_2$ to 2 seconds, loss of position information for a period exceeding acceptable limits can cause a loss of station (drive-off or drift-off condition).

Under normal operating conditions, a DP system should use more than one position reference system of different types. As noted above, acoustic systems are directly affected by the quality of the received signal. Classic problems with acoustic systems, called acoustic drop-outs, can be caused by mud dumping, or excessive bubbles in the water due to high thruster activity or high current. Accordingly, most DP systems that primarily depend upon acoustics have multiple separate acoustic channels that are independent of each other. However, to guard against a complete loss of acoustics, these systems also employ a taut wire, a riser angle, or satellite system for backup. The accuracy of taut wire and riser angle systems, however, is questionable in water depths greater than 1000 feet.

In the past several years for operation in the Gulf of Mexico, a satellite navigation system with sufficient accuracy and update rate has become available as a position reference system. However, even this system suffers from signal drop-outs due to loss of sync with one or more of the satellites because of shading by vessel structure during heading maneuvers and atmospheric conditions. Although these drop-outs last only a few minutes, this is too long to go without position reference information; and the use of a satellite system should be backed up with acoustics.

When the operating location permits the use of a radar position reference system, it should be considered. Again, however, an adequate backup position reference system should also be employed.

10.4.1.4 DP Operator Response Error

A DP operator often lives with hours upon hours of tedious boredom interspersed at rare and infrequent intervals by moments of emergency. Under normal operating conditions an operator's job is to monitor the operation of the system and coordinate vessel maneuvers with other operations. However, his primary function is to recognize impending stationkeeping problems and take appropriate action. Of primary concern, when operating with the riser connected to the BOP, is the timely notification of drilling personnel to activate an orderly disconnect should one be required. An unplanned disconnect can be the single most costly disaster as far as loss of time, money and equipment that a DP operation can suffer next to a catastrophic well control problem.

Probably more unplanned disconnects have occurred due to failure of the operator to recognize changes in environmental conditions than any other problem. This is especially true onboard shipshape DP vessels where the vessel heading must be adjusted to keep the bow into the impending environment. Rapidly deteriorating environmental conditions due to a squall in the Gulf of Mexico is a prime example of this type of problem. Failure to keep an adequate number of thrusters on-line can also lead to stationkeeping problems.

Many DP operator response errors are caused by poor equipment design, e.g., lack of information or poor presentation or organization of information, and by inadequate operator training.

Of these four categories, the fourth probably accounts for over half the stationkeeping problems that have been experienced. Loss of position reference information is probably the most common and can often contribute to improper operator responses. If the system has adequate redundancy and preventative maintenance, component failure should not cause stationkeeping problems. However such failures as loss of an operating thruster or an on-line skid, aggravated by improper operator responses, can lead to loss of station.

10.4.2 DP Operation Planning

Thorough planning is crucial to ensure safe and efficient DP operations. The following are important considerations:

a. Establish procedures for arrival and startup at a new location including system check out procedures.
b. Prepare written operating procedures and contingency plans including a clear line of authority not only for day-to-day operations but also for emergency situations.
c. Select and train DP operators and supervisors. Man-machine interaction is very important in DP operations. A competent and alert DP operator often can help avoid potentially disastrous consequences due to system component failures.
d. Prepare a startup seminar to familiarize the operating personnel with the intended drilling operation, the anticipated environmental conditions at the work location, and the stationkeeping capability of the DP vessel.

e. Understand the environmental limits for safe operation of the vessel. The limits of the vessel's capability to hold station in severe environment should be calculated for different combinations of wind, wave, and current environments and vessel heading as well as with assumed failure of one or two thrusters. Such data should be maintained on board and can be of help in operational decisions in rough weather.

10.4.3 Operating Personnel

Specially trained personnel are required to operate the DP system with its sophisticated electronic equipment. A competent and alert operator often can properly override the automatic control system and prevent potentially disastrous consequences due to system component failures.

In selecting DP operating personnel, it is desirable for the prospective operator to have a certain level of practical experience with electronic equipment. Such an operator is also likely to provide good maintenance while operating the DP system. Some of the better operators may develop the capability to identify malfunctioning components and provide troubleshooting at sea when the system fails. It is also desirable for the operator to have some knowledge and experience with marine operations which allows him to use better judgment in determining the safe operating conditions to support the drilling operations.

Special training is often required to familiarize operating personnel with the particular DP system and equipment and

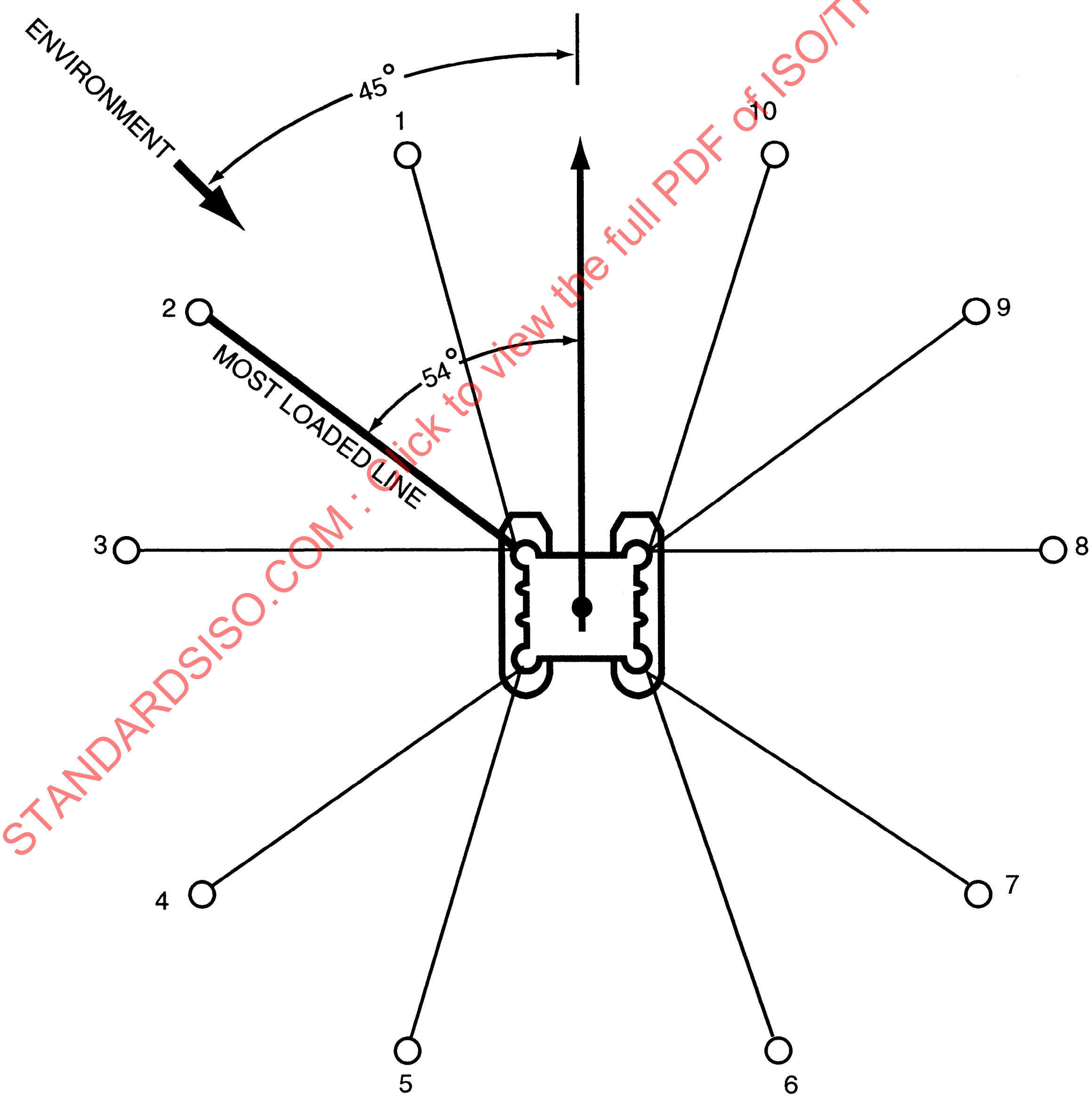


Figure 42—Mooring Configuration

provide a basic understanding of the theory of operation. The operators may also be trained to deal with simulated contingency conditions which would better prepare them to take proper control of the vessel in a real emergency condition.

10.4.4 Emergency Situations

A DP vessel may be driven off station involuntarily due to system malfunction or heavy weather. This may cause serious riser and BOP stack damage that could result in a significant amount of downtime. Therefore, the DP computer is often programmed to issue alarms to warn the DP operator and the driller when certain predetermined limits of offset from the wellhead have been exceeded. A yellow light warning alarm is used to indicate that the driller should start the hang off procedure and prepare to disconnect the marine riser from the BOP stack. When a red light disconnect alarm is indicated, the driller should punch the emergency disconnect button on the BOP control panel and start the disconnect sequence.

The yellow light setting (generally 2 percent to 3 percent of water depth) is based on past experience and water depth. This alarm should be set such that its limit is not frequently exceeded, but allows the drilling crew sufficient time to complete the riser hang-off procedure (approximately 30 seconds) prior to receiving a disconnect signal.

The criteria for setting the red light alarm (generally 4 percent to 6 percent of water depth) is based on the rig's ability to safely disconnect the riser prior to reaching the limiting value of any one of several critical parameters such as the ball joint angle, slip joint angle, tensioner stroke, riser and casing stresses. The time required to complete the emergency disconnect sequence (EDS) varies depending on the type of BOP control system. A typical multiplex BOP control system requires approximately 30 seconds to complete the EDS.

A detailed drive-off and drift-off analysis can also be conducted to calculate the time and distance traveled by the vessel under various power settings and environmental conditions. This is done to help establish the yellow and red light alarms at specific vessel offsets from the wellhead.

A well designed data recording system is required to record all of the critical data continuously and provide the relevant information for a post-event analysis. During an emergency situation, the operating personnel would be very busy in handling the emergency and in restoring control of the vessel. The operator would not have time to accurately record the details of the event which are usually very important for the post-event analysis. This analysis is necessary to diagnose the cause of the event so that corrective actions can be taken. This data logging system can also provide a continuous system performance monitoring capability which is very useful for the system maintenance and during system performance evaluations. The recorded data can also be used for reliability analysis.

Daily engineering records of all the critical parameters of the drilling operation including the DP system are also useful for maintaining a close surveillance of the operating conditions. This helps identify the malfunctioning equipment or the degrading performance of a certain operating condition. In some cases, operating personnel become more attentive to maintaining an efficient operation due to this reporting requirement. For example, the DP operator may help improve the efficiency of drill floor and equipment handling operations if he is attentive to maintaining the vessel heading into the weather to minimize the environmental loads and vessel motions. These daily records are also useful in the post event analysis and the documentation of the drilling operation.

11 Design Examples

11.1 EXTREME RESPONSE ANALYSIS EXAMPLE

The following problem illustrates the procedures for the analysis of a mooring system using three analysis methods:

a. Quasi-static analysis.
b. Time domain dynamic analysis.
c. Frequency domain dynamic analysis.

The example is intended to illustrate the principal steps in the analysis and to give guidance particularly on the dynamic analysis procedure. The method described here is not unique and it may be necessary to modify some of the steps in the procedure to accommodate specific software.

11.1.1 Mooring System Description

The system to be analyzed is:

a. A semi-submersible with a 10 point, 36 degree symmetric pattern, as shown in Figure 42.
b. $3\frac{1}{2}$ inch K4 chain 5.100 feet outboard, break test load = 1838 kips.
c. Initial tension = 280 kips.
d. Mooring line description.

1. Diameter = $3\frac{1}{2}$ inch.
2. Elastic stretch (AE) = 123.4 lbs/feet.
3. Weight in air = 107.2 lbs/feet.
4. Friction coefficient = 1.0 (with mudline).
5. Break test load = 1,838 kips.
6. Line mass = 3.84 slugs/foot.
7. Tangential added mass = 0.25 slugs/foot.
8. Normal added mass = 0.51 slugs/foot.
9. Drag coefficient = 1.2.
10. Drag diameter = 7 in. (2 5 nominal diameter).

The mooring properties for chain include provision for a tangential added mass. For wire or synthetic ropes, no tangential added mass is required. The nominal drag diameter is increased by a factor of 2 for chain. For wire rope or synthetic rope this is not required.

11.1.2 The Environment

The design example environment is as follows:

a. Water depth = 1,233 feet.
b. Significant wave height = 55.8 feet.
c. Peak spectral period = 17.49 sec.
d. JONSWAP spectrum with a peakedness factor 3.3.
e. Wind velocity (1 min.) = 100 kt.
f. Surface current velocity = 3.1 kt.
g. Quartering direction (See Figure 42).
h. Design storm duration = 3 hours.
i. Wind, wave, current colinear.

Wind loading on the vessel can be accounted for in two different procedures:

a. Wind load is considered to be applied statically to the vessel. In this procedure the one minute average wind speed is applied on the vessel as a static load.
b. The dynamic effects of wind are considered by combining a steady wind force with a fluctuating wind component. The one-hour average wind speed is applied statically to the vessel. A wind gust velocity spectrum is defined.

The approach described in item a has been used in the present example.

11.1.3 Mean Load Computation

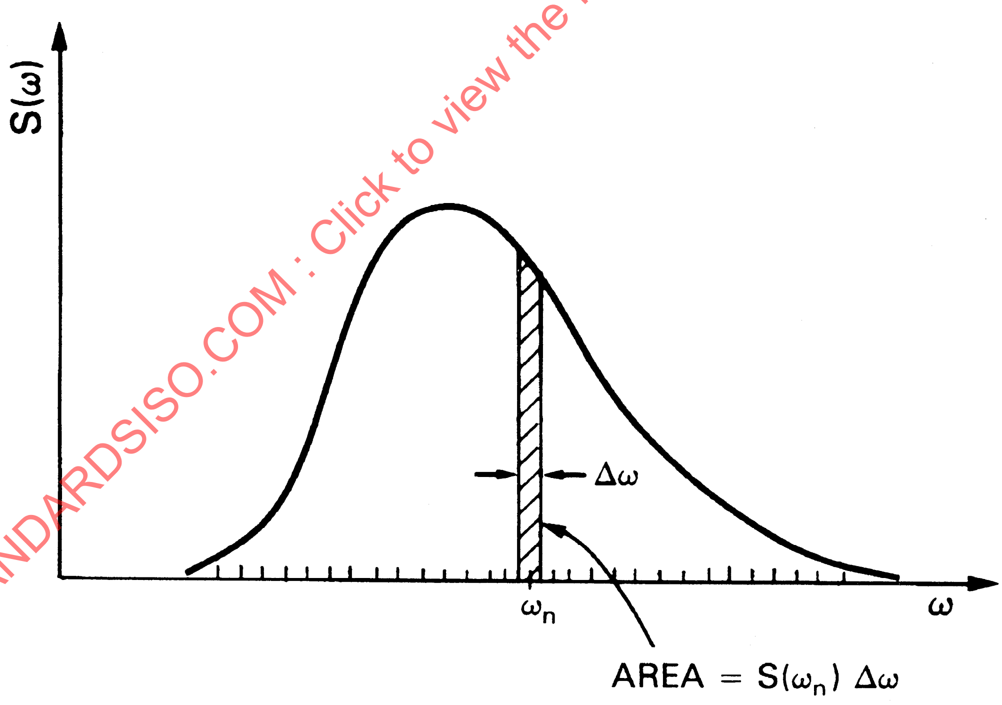
The mean loads can be derived from model test data or computed. The computed mean loads on the vessel are as follows:

Wind (1 minute average)	680 kips
Current	378 kips
Steady wave drift	70 kips
Total mean load	1,128 kips

11.1.4 Vessel/Fairlead Motions

Both wave frequency and low-frequency motions are required in all analyses. In general the response amplitude operators, and phases, in the three linear directions (heave, surge, sway) and the three angular directions (roll, pitch, yaw) must be derived. The derivation of these data requires hydrodynamic computer programs or model test data. Any suitable reference point, usually the vessel's center of gravity, can be used to define the motions.

The low-frequency motions can be computed from hydrodynamic computer programs, model test data, or design curves. The computed root mean square (rms) low-frequency motion for this example is 0.97 feet.



$$h(t) = \sum_{j=1}^{\omega} \sqrt{2\, S(\omega_j)\, \Delta\omega}\ \mathrm{Cos}\ (\omega_j t + \phi_j)$$

Figure 43—Spectral Decomposition

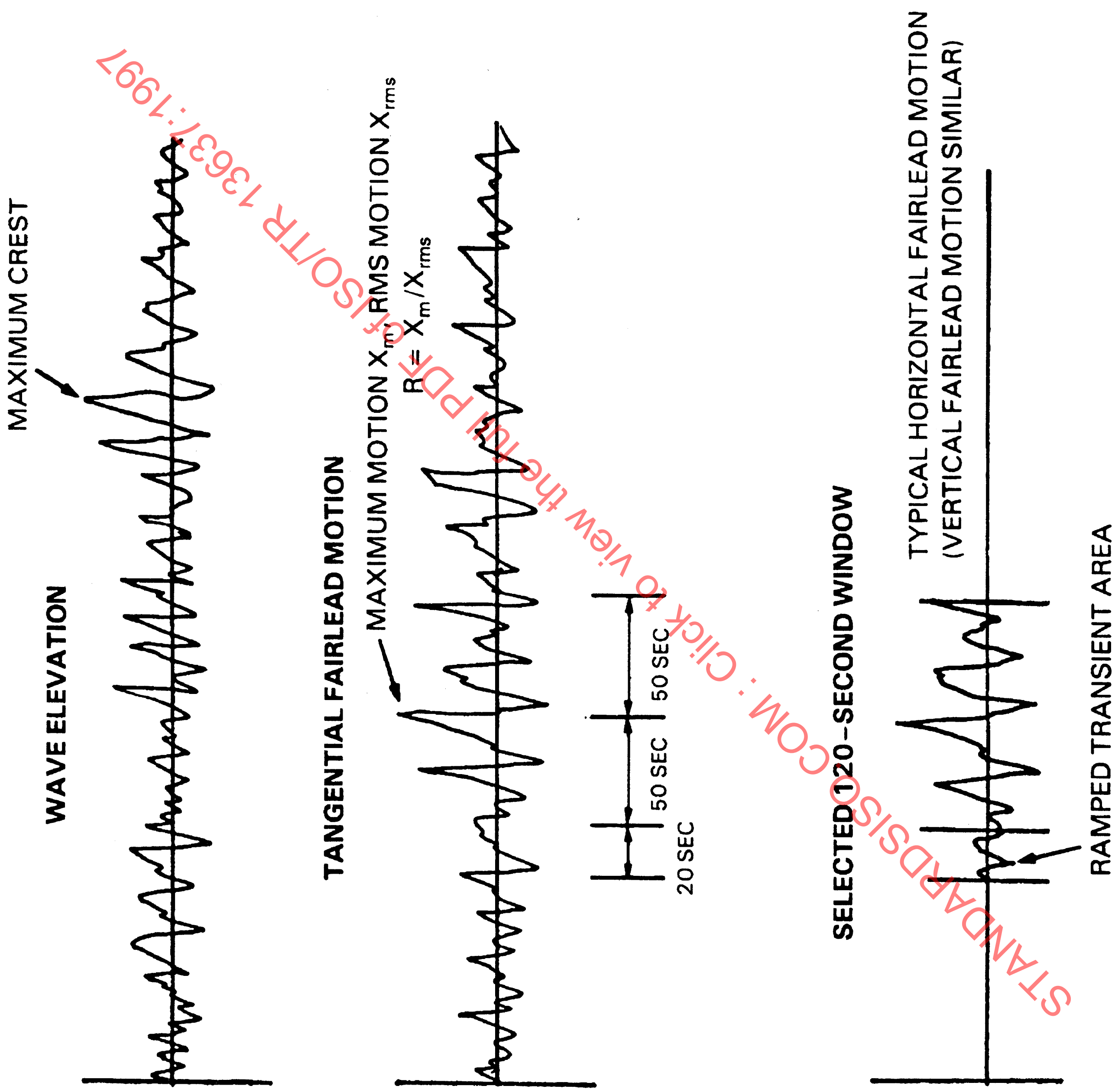


Figure 44—Selection of Input Motion

Table 1—Fairlead Motion Complex RAO

Line No. = 1
Line Heading = 54.00 Degrees
Wave Heading From +X = – 45.00 Degrees

Line End Motions with X in Line from Fairlead to Anchor and Z Vertical

		***Location (X = 110.00	Y = 104.00	Z = –35.00)***	
Wave Frequency (Rad/Sec)	Low Frequency (Sec)	X Real (AMP/AMP)	X Imag (AMP/AMP)	Z Real (AMP/AMP)	Z Imag (AMP/AMP)
0.20	31.42	–0.0036	0.9928	1.0976	0.0518
0.26	24.32	–0.0053	0.9085	0.7092	–0.2909
0.32	19.84	–0.0061	0.8496	0.5965	0.2052
0.37	16.76	–0.0092	0.7840	0.5820	0.3278
0.43	14.50	–0.0120	0.7032	0.5158	0.3992
0.49	12.78	–0.0141	0.6080	0.4273	0.4386
0.55	11.42	–0.0152	0.5014	0.3289	0.4419
0.61	10.33	–0.0153	0.3899	0.2315	0.4110
0.67	9.42	–0.0140	0.2815	0.1452	0.3489
0.72	8.67	–0.1115	0.1849	0.0773	0.2685
0.78	8.02	–0.0083	0.0997	0.0249	0.1788
0.84	7.47	–0.0048	0.0542	0.0090	0.1050
0.90	6.98	–0.0014	0.0121	–0.0047	0.0340
0.96	6.56	–0.0004	0.0143	0.0025	0.0145
1.02	6.18	0.0006	0.0165	0.0096	–0.0051
1.07	5.84	0.0000	0.0179	0.0114	–0.0120
1.13	5.54	–0.0023	0.0186	0.0073	–0.0051
1.19	5.27	–0.0047	0.0193	0.0032	0.0018
1.25	5.03	–0.0071	0.0200	–0.0009	0.0087
1.31	4.80	–0.0073	0.0173	–0.0013	0.0079
1.37	4.60	–0.0073	0.0142	–0.0013	0.0061
1.42	4.41	–0.0074	0.0111	–0.0012	0.0044
1.48	4.24	–0.0074	0.0079	–0.0011	0.0026
1.54	4.08	–0.0074	0.0048	–0.0011	0.0008
1.60	3.93	–0.0075	0.0033	–0.0011	–0.0001

The vessel motions at the reference point must be transformed to the end or fairlead of the line to be analyzed. The procedure required varies with the type of analysis.

11.1.4.1 Quasi-Static Analysis

In a quasi-static analysis, only surge in the quartering direction is considered. Heave is ignored. The vessel motion RAOs and phases can be transformed into the quartering direction and convoluted over the sea-state spectrum to produce rms line end motion. The computed wave frequency rms motion in this example was 8.6 feet. To establish the dominant frequency response, compare the maximum wave frequency motion with the maximum low-frequency motion as follows:

maximum wave frequency motion:

$$3.72 \times 8.6 = 32.0 \text{ feet}$$

maximum low-frequency motion:

$$3.03 \times 0.97 = 2.9 \text{ feet}$$

where the factors 3.72, 3.03 represent a 1 in 1,000 and 1 in 100 wave maximum respectively, typical for 3-hour storm conditions. Wave frequency dominates. Hence the design condition is:

maximum wave frequency motion + significant low-frequency motion = 32.0 + 2 × 0.97 = 33.94 feet

Note the factors 3.72 and 3.03 are approximate values typically used for a 3-hour storm. More precise values can be obtained by Equations 6.5 to 6.8.

11.1.4.2 Frequency Domain Analysis

To compute line end motions in the frequency domain, the vessel RAOs and phases in the six degrees of freedom must be translated to the fairlead location. In general, only the motions in the plane of the line are of interest. The line end motions in the horizontal and vertical directions in the plane of the line, and the phases, between them are computed for each frequency. It is extremely important to retain all phase information to this point as the dynamic behavior of the line is heavily influenced by the tangential motion or stretching of the line. The line end motions used in this example are given in complex form in Table 1 for each frequency. The standard RAO value is the square root of the sum of the squares of the real and imaginary points, in the case of each motion.

11.1.4.3 Time Domain Analysis

Time domain analysis requires a further step beyond frequency domain analysis. A time history of the motions is required. The following procedure was used in this example:

a. A three hour time series, representing the wave elevation was generated from the sea-state spectrum. The procedure is illustrated in Figure 43.

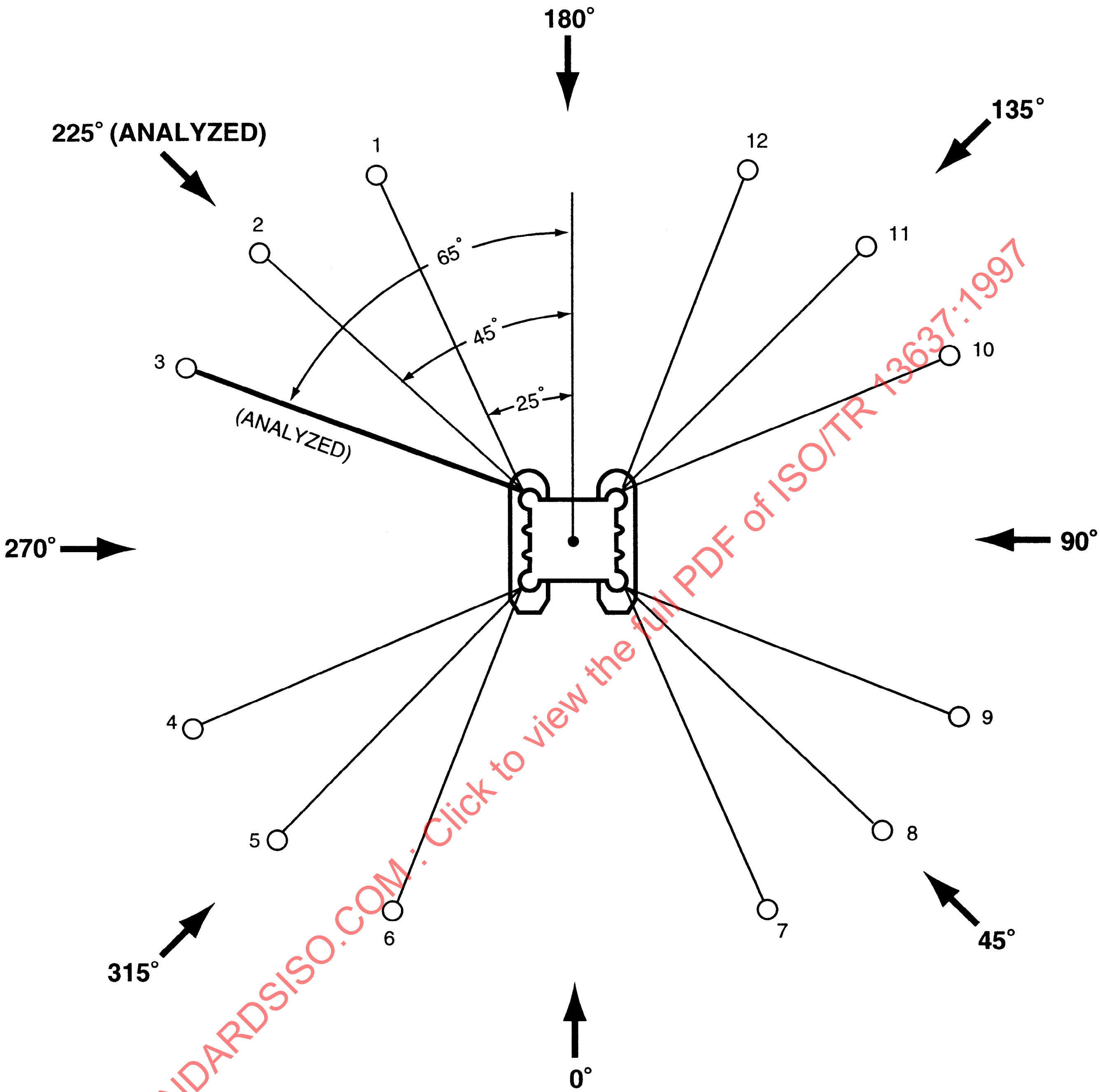


Figure 45—Mooring System and Environmental Directions

$$h(t) = \sum_{j=i}^{n} A_j \cos(\omega_j t + \phi_j)$$

Where:

A_j = $\sqrt{2S(\omega_j)\,\Delta\,\omega}$
ϕ_j = Random Phase $[0, 2\pi]$.
$S(\omega_j)$ = Spectral Density at Frequency ω_j.

At least 50 frequencies are required to develop a realistic time series. Care should be taken in generating the wave elevation, that the time series does not repeat itself prematurely. This is normally achieved by using varied frequency spacing. The fairlead motion RAOs of Table 1 are used to transform the wave elevation time series into a horizontal and vertical motion at the line end, in the plane of the line. Because phasing has been properly maintained in Table 1, the resulting horizontal and vertical motions are correctly phased.

The analysis of a full three hour time series is generally not practical in standard design practice. A simplified procedure is illustrated in Figure 44. The maximum tangential motion is computed. A 120-second segment is selected centered on the maximum tangential motion. In addition, the root mean square tangential motion is computed for the full three hour time history, the ratio of maximum tangential motion to the root mean square value R is computed. The maximum line tension is then calculated using Method B in 7.2.

The tangential motion is used because tangential motion greatly influences the line stretch. The use of a limited time segment based on the maximum wave elevation is not in general recommended. The above procedure, although simple, can be used to ensure an acceptable approximation for time domain analysis. Alternative methods, which ensure a statistically realistic response level can be used. The 120 seconds of data is adequate and will give results of comparable magnitude to those developed in a full three hour simulation for the computer programs used in this analysis. However, the length of the data should be determined after a sensitivity study for a particular computer program has been conducted.

11.1.5 Mooring Analysis Results

11.1.5.1 Quasi-Static Analysis

The mooring system was analyzed using a computer program based on catenary equations modified for elastic stretch and bottom friction. The computed line tensions were for the most loaded line No. 2 (54 degree direction).

Mean Tension = 643 kips (35 percent BTL).
Mean + Low-Frequency = 650 kips (35 percent BTL).
Maximum Tension = 779 kips (42 percent BTL).

Other derived parameters are:

Maximum Anchor Load = 353 kips.
Maximum Suspended Line Length = 3,986 feet.

11.1.5.2 Frequency Domain Analysis

The most loaded line No. 2 was analyzed. Initially, a quasi-static analysis is required under mean plus low-frequency tension. From a review of the data it was concluded that wave frequency tensions would dominate. Hence, the line is initially analyzed under mean plus significant low-frequency conditions:

Mean Tension = 643 kips.
Mean + Low Frequency = 650 kips (35 percent BTL).

The line was analyzed under a 650 kip tension with the specified end motions. The output tension spectrum is given in Table 10.2. The computed rms tension was 111.5 kips. The maximum wave frequency tension was 3.72 × 111.5 = 415 kips. The total tension is:

Maximum Design Tension = 1,065 kips (58 percent BTL).

The other derived parameters are:

Maximum Anchor Load = 870 kips.
Maximum Suspended Line Length = 4,284 feet.

Table 2—Tension Spectrum

Frequency Rad/Sec	Tension Spectrum
0.200	0.1067002E-09
0.258	0.3302278
0.317	80.27989
0.375	586.9768
0.433	2172.754
0.492	1969.006
0.550	510.8987
0.608	187.6015
0.667	60.40662
0.725	21.31249
0.783	13.28411
0.842	5.786316
0.900	1.439414
0.958	0.1947693E-01
1.017	2.250558
1.075	4.270370
1.133	2.902684
1.192	1.725605
1.250	0.8388341
1.308	0.5401919
1.367	0.3427716
1.425	0.2073264
1.483	0.1237301
1.542	0.7565913E-01
1.600	0.5461835E-01

The line tensions are acceptable (below 60 percent BTL) and suspended line length ensures adequate grounded length. Anchor loads are substantially higher then those produced by quasi-static analysis.

11.1.5.3 Time Domain Analysis

Only the most loaded line, No. 2, was analyzed. The procedure is similar to the frequency domain in that a quasi-static analysis was first carried out under mean plus significant low-frequency tension. The applied end motions were then combined with the resulting tensions.

Mean Tension = 643 kips (35 percent BTL).
Mean + Low-Frequency = 650 kips (35 percent BTL).
Maximum Tension = 1,101 kips (60 percent BTL).

Other derived parameters are:

Maximum Anchor Load = 886 kips.
Maximum Suspended Line Length = 3,903 feet.

The peak quantities are compared to allowables as before. Tensions are acceptable (60 percent limit); suspended line length ensures adequate grounded length; and a suitable anchor must be designed.

11.2 FATIGUE ANALYSIS EXAMPLE

The following example illustrates the computation of lifetime fatigue damage on a mooring line and the estimation of allowable fatigue life. A frequency domain method of dynamic analysis is used.

11.2.1 Mooring System Description

11.2.1.1 A semisubmersible system with a 12 line, 25 degrees/45 degrees/65 degrees mooring is considered. The system is shown in Figure 45.

11.2.1.2 Mooring parameters used in the analysis for wire rope:

Elastic stiffness (EA)	94,355 kips
Air weight	22.7 lbs/feet
Submerged weight	19.3 lbs/feet
Mass	0.705 slugs/foot
Normal added mass	0.133 slugs/foot
Drag diameter	3.5 inch
Drag coefficient	1.2
T-N curve	$NR^{4.09} = 731$ (For example only)
Reference breaking strength	1,110 kips

11.2.1.3 Mooring parameters used in the analysis for chain:

Elastic stiffness (EA)	147,074 kips
Air weight	123 lbs/feet
Submerged weight	107 lbs/feet
Mass	3.73 slugs/foot
Tangential added mass	0.25 slugs/foot
Normal added mass	0.50 slugs/foot
Drag diameter	7 inch
Drag coefficient	1.2
T-N curve	$NR^{3.36} = 370$
Reference breaking strength	1,383 kips

Fatigue analysis are provided here for the wire rope at the fairlead and for the chain at the chain/wire rope intersection.

11.2.2 Environmental Conditions

A fatigue analysis consists of multiple individual line tension analyses as described in 7.5. The environment is defined as a set of the following:

a. Directions.
b. Wind speeds.
c. Current speeds.
d. Wave heights, spectral shapes, spectral peak periods or equivalent.
e. Probability of occurrence of the above.

The mean forces associated with wind, wave and current are computed by the procedures in Section 5. The low-frequency motions associated with each environmental condition can be obtained from computer programs, model tests, or design curves in Figures A.3 to A.17 of Appendix A.

In the usual case, eight directions at 45 degree intervals are sufficient to define the environment. In this example we will consider one such direction, 225 degrees, in detail. The environmental conditions to be analyzed are given in Table 3. The sea-states, cumulative probability of occurrence, mean loads and rms low-frequency motions are given. The following additional parameters are used:

Table 3—Environmental Condition, Mean Loads and Low-Frequency Motions for the Analyzed Direction

Sig. Wave (feet)	Peak Period (sec)	Probability (percent)	Mean Loads (kips)	Low Freq. rms[a] (feet)
3.60	8.4	16.96	28.7	0.40
8.30	9.2	36.29	61.4	0.66
13.01	10.4	26.07	131.4	1.07
17.71	11.6	13.05	212.2	1.22
22.42	12.7	5.31	309.3	1.35
27.12	13.6	1.64	418.8	1.43
31.83	14.4	0.52	552.1	1.50
36.53	15.3	0.13	714.9	1.60
41.24	16.1	0.02	891.7	1.68
45.94	17.7	0.01	1239.1	1.75

[a] Low frequency motions based on a reference stiffness of 18 kips/foot.

Table 4—Wire Rope Damage

Annual Accumulated Fatigue Damage by Sea-State for Direction 6, 225.0
(All Periods and #'s of Cycles Refer to Tensions)
All Data is Per Year

Mean Ten.	Sig. Height	RMS Tension (Wave)	RMS Tension (Low)	# of Cycles (Wave)	# of Cycles (Low)	Zero Cross. Period (Wave)	Zero Cross. Period (Low)	Wave Frequency Damage	Low Frequency Damage
153.0	3.60	2.2	0.4	0.121E+06	0.780E+04	7.10	109.76	0.196E–06	0.172E–10
162.0	8.30	4.4	1.4	0.243E+06	0.168E+05	7.56	109.14	0.748E–05	0.526E–08
175.7	13.01	7.2	2.6	0.163E+06	0.122E+05	8.09	108.22	0.037E–04	0.438E–07
192.0	17.71	10.8	3.3	0.723E+05	0.616E+04	9.11	106.94	0.863E–04	0.599E–07
215.5	22.42	16.3	4.2	0.263E+05	0.258E+04	10.19	103.97	0.169E–03	0.619E–07
242.4	27.12	24.0	4.9	0.734E+04	0.816E+03	11.23	100.89	0.226E–03	0.380E–07
278.2	31.83	34.7	5.5	0.212E+04	0.265E+03	12.25	97.79	0.296E–03	0.200E–07
324.8	36.53	49.3	6.5	0.442E+03	0.629E+02	13.34	93.75	0.258E–03	0.900E–08
376.9	41.24	67.7	7.3	0.799E+02	0.130E+02	14.60	89.52	0.170E–03	0.300E–08
480.2	45.94	96.1	8.4	0.325E+02	0.633E+01	16.08	82.44	0.290E–03	0.300E–08
							Total	0.154E–02	0.243E–06

Total damage measure for this direction:

Direction (6)	225.0
Probability percent for direction	0.160E+02
First order damage	0.154E–02
Second order damage	0.243E–06

Total number of tension cycles per year for this direction:

Wave frequency tension cycles	0.634E+06
Low frequency tension cycles	0.466E+05
Average wave tension period	0.795E+01 (zero crossing)
Average low tension period	0.108E+03

Table 5—Chain Damage

Annual Accumulated Fatigue Damage by Sea-state for Direction 6, 225.0
(All Periods and #'s of Cycles Refer to Tensions)
All Data is Per Year

Mean Ten.	Sig. Height	RMS Tension (Wave)	RMS Tension (Low)	# of Cycles (Wave)	# of Cycles (Low)	Zero Cross. Period (Wave)	Zero Cross. Period (Low)	Wave Frequency Damage	Low Frequency Damage
153.0	3.60	2.0	0.4	0.121E+06	0.780E+04	7.08	109.76	0.507E–05	0.182E–08
162.0	8.30	4.3	1.3	0.243E+06	0.168E+05	7.55	109.14	0.119E–03	0.229E–06
175.7	13.01	7.1	2.7	0.162E+06	0.122E+05	8.11	108.22	0.437E–03	0.123E–05
192.0	17.71	10.8	3.4	0.721E+05	0.616E+04	9.14	106.94	0.789E–02	0.143E–05
215.5	22.42	16.4	4.2	0.262E+05	0.258E+04	10.22	103.97	0.120E–02	0.122E–05
242.4	27.12	24.2	4.9	0.733E+04	0.816E+03	11.24	100.89	0.123E–02	0.670E–06
278.2	31.83	35.0	5.6	0.211E+04	0.265E+03	12.26	97.79	0.123E–02	0.330E–06
324.8	36.53	49.8	8.5	0.442E+03	0.629E+02	13.33	93.75	0.850E–03	0.130E–06
376.9	41.24	68.4	7.4	0.801E+02	0.130E+02	14.57	89.52	0.440E–03	0.400E–07
480.2	45.94	97.1	8.5	0.326E+02	0.633E+01	16.04	82.44	0.590E–03	0.300E–07
							Total	0.689E–02	0.531E–05

Total damage measure for this direction:

Direction (6)	225.0
Probability percent for direction	0.160+02
First order damage	0.689E–02
Second order damage	0.531E–05

Total number of tension cycles per year for this direction:

Wave frequency tension cycles	0.634E+06
Low frequency tension cycles	0.466E+05
Average wave tension period	0.795E+01 (zero crossing)
Average low tension period	0.108E+03

a. Water depth 1,476 feet.
b. Pierson Moskowitz Spectral form.
c. Environment is in the analyzed direction 16 percent of the time.

In this example, wind, wave, and current are assumed co-linear. All low-frequency motions are assumed to be from wave effects and are based on a reference stiffness of 18 kips/foot. Wind effects are included in the mean load. For a mooring system with a stiffness k kips/foot at the mean position, the rms low-frequency motions are computed as:

$$rms\ (k) = rms\ (18) \times \sqrt{18 / k}$$

Where:

$rms\ (k)$ = the motion at stiffness k.
$rms\ (18)$ = the motion at the reference stiffness 18 kips/foot.

If low-frequency wind effects were included, a square root relationship between the actual and reference stiffness values cannot be used. It is necessary to define the rms motions for the actual mooring stiffness.

11.2.3 Fatigue Analysis

The fatigue damage is computed as follows. The annual fatigue damage is initially computed. Low-frequency effects are minor but are included. A year is assumed to have 3.15576 × 10 seconds. By Narrow Band Theory, the fatigue damage in a given sea-state is:

$$D = N_w (\sqrt{2}\, R_{wrms})^M \bullet \Gamma\ (1 + M/2)/K + N_l (\sqrt{2}\, R_{lrms})^M \bullet \Gamma\ (1 + M/2)/K \qquad (11.1)$$

Where:

D = annual fatigue damage.
K = the intercept parameter of the T/N curve (731 for wire rope and 370 for chain, these values are for example only).
M = the slope of the T/N curve (4.09 in this example for wire rope and 3.36 for chain, these values are for example only).
R_{wrms} and R_{lrms} = the ratios of wave frequency and low-frequency rms tension range(twice the single amplitude value) to reference breaking strength.
N_w and N_l = the numbers of wave frequency and low-frequency tension cycles per year.

N_w can be computed as:

$$N_w = v \times 3.15576 \times 10^7 \times P_d \times P_s \qquad (11.2)$$

Where:

v = zero up-crossing frequency of the tension spectrum (hz).
P_d = probability of the direction (16 percent for this example).
P_s = probability of the sea-state given the direction. (Table 3).

Low-frequency fatigue damage and wave frequency fatigue damage are computed independently. The low-frequency zero crossing period is estimated to be the natural period of the vessel/mooring system as a function of applied mean load. The appropriate number of low-frequency cycles per year can be computed as:

$$N_l = (3.15576 \times 10^7 \times P_d \times P_s)/T_n$$

Where:

T_n = vessel/mooring system natural period and the remaining quantities are as previously defined.

If procedures are used that incorporate a general low-frequency (from wind or wave or both) tension spectrum, the actual zero up-crossing period can be used, rather than the natural period. In the usual case, low-frequency effects are minor and the natural period is adequate. The wave frequency zero up-crossing period or frequency should be computed directly from the tension spectrum generated in a line dynamic analysis, as follows:

$v = \sqrt{M_2 / M_o}$ hz
$M_2 = \int_o^\infty f^2\, S(f) df$
$M_o = \int_o^\infty S(f)$
f = Frequency (hz).
$S(f)$ = Tension spectrum (kips²/hz).

11.2.4 Fatigue Damage

A detailed computation of the fatigue damage per year associated with a 225 degree direction (wave heading towards), for line 3 (65 degree spreading angle) is given in Table 4 for wire rope and Table 5 for chain. The total annual accumulated fatigue damage is:

$D = 0.689 \times 10^{-2}$ (chain).
$= 0.155 \times 10^{-2}$ (wire rope).

In this example, low-frequency motions have essentially no effect on the fatigue life, being about three orders of magnitude less severe than wave frequency effects. This, however, may not be true for the cases where low-frequency motions are dominant. The fatigue damages for chain and wire rope associated with all directions are given in Table 6. The total annual fatigue damage on the line, counting all directions is:

$D = 0.218 \times 10^{-1}$ (chain).
$D = 0.418 \times 10^{-2}$ (wire rope).

The useful fatigue life, computed as:

Life = 1/(3D) years.
= 15 years (chain).
= 80 years (wire rope).

The safety factor applied is 3, based on 6.8.

Table 6—Annual Fatigue Damage as a Function of the Environment

Line 3 (65 degrees)			
Direction	Probability of Direction (percent)	Annual Fatigue Damage Chain	Wire Rope
0	6.0	0.426×10^{-3}	0.412×10^{-4}
45	8.0	0.535×10^{-3}	0.436×10^{-4}
90	13.0	0.125×10^{-2}	0.109×10^{-3}
135	12.5	0.375×10^{-3}	0.337×10^{-4}
180	14.0	0.133×10^{-2}	0.151×10^{-3}
225	16.0	0.689×10^{-2}	0.155×10^{-2}
270	18.0	0.100×10^{-1}	0.213×10^{-2}
315	12.5	0.983×10^{-3}	0.131×10^{-3}
Total	100.0	0.218×10^{-1}	0.418×10^{-2}

APPENDIX A—SIMPLIFIED METHODS FOR THE EVALUATION OF ENVIRONMENTAL FORCES AND VESSEL MOTIONS

A.1 Basic Considerations

Design equations and curves for a quick evaluation of environmental forces and vessel motions are provided in this appendix. These simplified analytical tools were developed primarily for the analysis of mobile moorings. They may be used for preliminary designs of permanent moorings if more accurate information is not available at the early stage of the design process and if the limits for these tools are not exceeded. For the final design of permanent moorings, however, the more rigorous approaches as outlined in 5.2 are recommended.

A.2 Current

Current forces are normally treated as steady state forces in a mooring analysis. They can be estimated by model tests or calculations.

A.2.1 Model Tests

Model test data from towing tank or wind tunnel tests may be used to predict current loads for mooring system design provided that a representative underwater model for the unit is tested and that the contribution to current load made by thrusters, anchor bolsters, bilge keels, and other appendages is accounted for. Care should be taken to assure that the character of the flow in the model test is the same as the character of the flow for the full-scale unit.

A.2.2 Current Force Calculations

If current forces are to be calculated, the following equations should be used:

a. Current force due to bow or stern current on ship shaped hulls:

$$F_{cx} = C_{cx} S V_c^2 \qquad \text{(A.1)}$$

Where:

F_{cx} = current force on the bow, lb (N).
C_{cx} = current force coefficient on the bow.
= 0.016 lb/(ft² • kt²) (2.89 Nsec²/m⁴).
S = wetted surface area of the hull including appendages, ft²(m²).
V_c = design current speed, kts (m/sec).

b. Current force due to beam current on ship-shaped hulls:

$$F_{cy} = C_{cy} S V_c^2 \qquad \text{(A.2)}$$

Where:

F_{cy} = current force on the beam, lb (N).
C_{cy} = current force coefficient on the beam.
= 0.40 lb/(ft² • kt²) (72.37 Nsec²/m⁴).

Note: Equations A.1 and A.2 were developed for estimating current forces on drillships. They are applicable only to production vessels with similar hull form and size.

c. Current and wind forces for large tankers: Current and wind forces for large tankers can be estimated using the report *Prediction of Wind and Current Loads on VLCCs* published by Oil Company International Marine Forum [11]. This report presents coefficients and procedures for computing wind and current loads on very large crude carriers (VLCCs), namely, tankers in the 150,000 to 500,000 dwt class. Wind/current force and moment coefficients are presented in nondimensional form for a moored vessel in various draft and under keel clearance conditions. While the analysis of mooring restraint has not been addressed, coefficients are provided for use with either computer oriented or hand calculation techniques for design of tanker/terminal mooring equipment.

d. Current force on semisubmersible hulls:

$$F_{cs} = C_{SS}(C_d A_c + C_d A_f)\, V_c^2 \qquad \text{(A.3)}$$

Where:

F_{cs} = current force, lb (N).
C_{ss} = current force coefficient for semisubmersible hulls.
= 2.85 lb/(ft² • kt²) (515.62 Nsec²/m⁴).
C_d = drag coefficient (dimensionless).
= 0.50 for circular members. (See Figure A-1 for members having flat surfaces.)
A_c = summation of total projected areas of all cylindrical members below the waterline, ft² (m²).
A_f = summation of projected areas of all members having flat surfaces below the waterline, ft² (m²).

e. Current force on mooring lines and risers: The effect of current loads on mooring lines and risers on the overall mooring design should be evaluated. This is particularly important for deepwater locations with high currents. Current loads on mooring lines and risers can be calculated using appropriate current profiles and drag coefficients. In high currents, drag coefficients should be adjusted for the presence of vortex-induced vibrations.

A.3 Waves

Interactions between ocean waves and a floating vessel results in forces acting on the vessel that can be conveniently split into three categories (Figure A-2): (a) first order forces that oscillate at the wave frequencies inducing first order motions known as high frequency or wave frequency motions; (b) second order forces with frequencies below wave frequencies inducing second order motions known as low frequency motions; and (c) steady component of the second

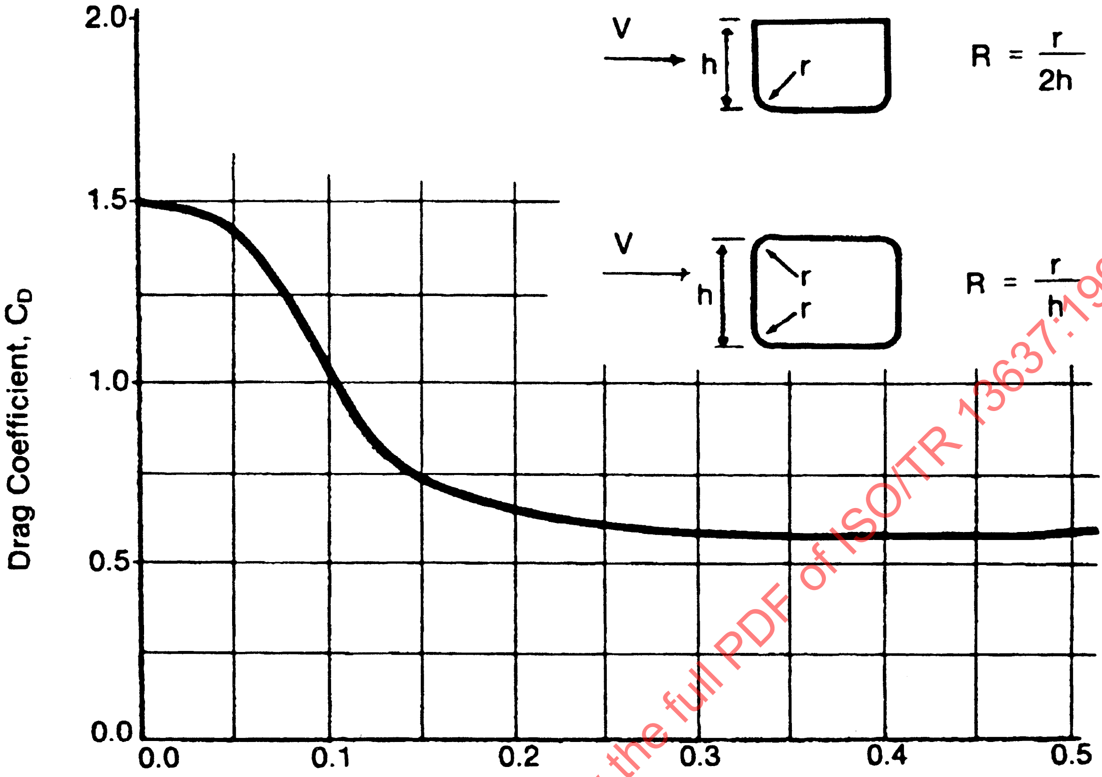


Figure A-1—Semisubmersible Current Drag Coefficient for Members Having Flat Surfaces

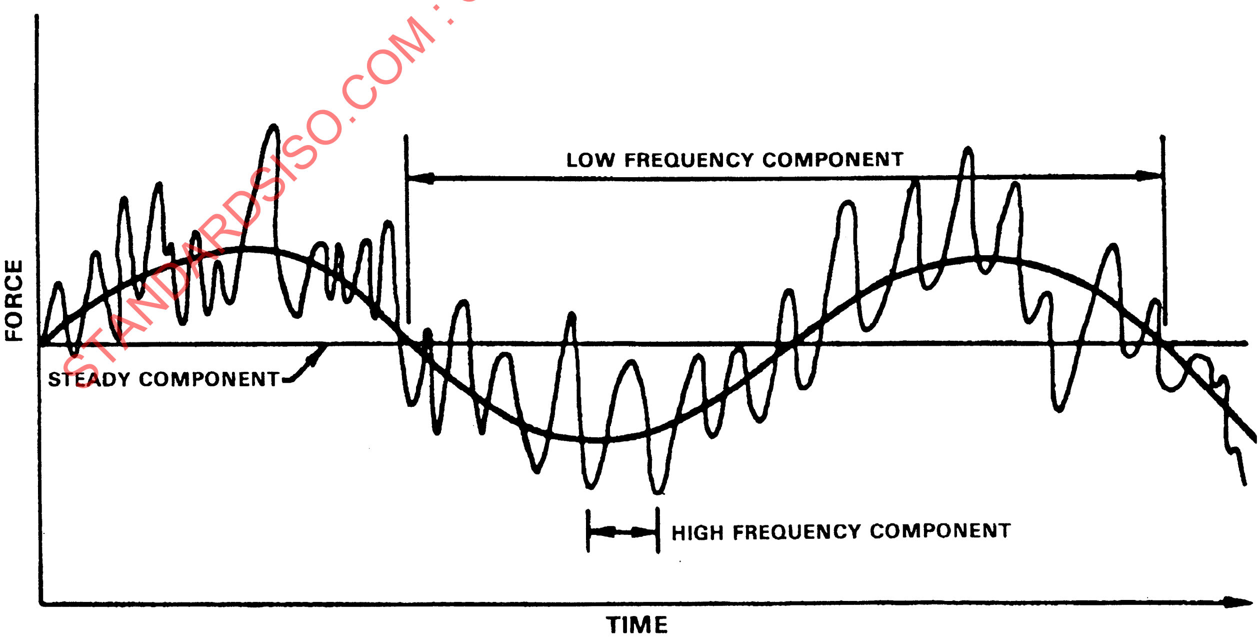


Figure A-2—Wave Force Components

order forces known as mean wave drift forces which can be estimated by model test or calculation.

A.3.1 Model Tests

Model test data may be used to predict wave forces for mooring system design provided that a representative underwater model of the unit is tested. Care should be taken to assure that the character of the flow in the model test is the same as the character of the flow for the full-scale unit.

A.3.2 Wave Frequency Vessel Motions

The motions of the vessel at the frequency of the waves is an important contribution to the total mooring system loads, particularly in shallow water. These wave frequency motions can be obtained from regular or random wave model test data or computer analysis using either time or frequency domain techniques.

Wave frequency motions have six degrees of freedom: surge, sway, heave, pitch, roll, and yaw. They are normally considered to be independent of mooring stiffness except for floating systems with natural periods less than 30 seconds.

A.3.3 Mean Wave Drift Force

The mean wave drift force is induced by the steady component of the second order wave forces. The determination of mean drift force requires motions analysis computer programs or model tests. Design curves for estimating mean wave drift forces for drillships and semisubmersibles are provided in Figures A-3 to A-17. The curves are applicable to typical MODU type vessels. However, for large drilling and production semisubmersibles (with displacements over 30,000 short tons) and large tankers, the use of these curves is not recommended.

The curves for semisubmersibles represent the upper bound of the mean wave drift forces generated by a motions analysis computer program for four semisubmersible designs including typical 4, 6, and 8 circular column twin hull designs and the pentagon design. The curves for drillships were generated for ship lengths of 400 feet to 540 feet. For drillships which are outside this length range, the mean drift force can be estimated by extrapolation. However, extrapolation for ship lengths below 350 feet or above 600 feet is not recommended.

The data presented are appropriate for ship-shape vessels with normal hull form. Care should be used in applying this data to vessels with blunt bows or sterns or other unusual hull features.

A.3.4 Low Frequency Vessel Motions

Low frequency motions are induced by the low frequency component of the second order wave forces, which in general are quite small compared to the first order forces. Because of this, the low frequency forces do not play a significant role in the motions in the vertical plane (i.e., roll, pitch, and heave motions) where large hydrostatic restoring forces are present. However, in the horizontal plane (i.e., surge, sway, and yaw motions), where the only restoring forces present are due to mooring or dynamic positioning systems and production risers, the motions produced by the low frequency forces can be substantial. This is particularly true at frequencies near the natural frequency of the mooring. Therefore, in general, only low frequency surge, sway, and yaw motions are included in a mooring analysis.

Low frequency motion of a moored vessel is narrow banded in frequency since it is dominated by the resonant response at the natural frequency of the moored vessel. The motion amplitude is highly dependent on the stiffness of the mooring system. The motion amplitude is also highly dependent on the system damping so that a good estimate of damping is critical in computing low frequency motions. Methods for predicting the low frequency motions are still in a state of development. There is a substantial degree of uncertainty in the estimation, particularly in damping.

There are three sources of damping:

a. Viscous damping of the vessel.
b. Wave drift damping of the vessel.
c. Mooring system damping.

The technology to estimate viscous damping has been well established, and viscous damping is normally included in the low frequency motion calculations. Wave drift damping and mooring system damping, however, are more complex and are often neglected because of a lack of understanding in these damping components. Recent research indicates that wave drift damping and mooring system damping can be significant. They can even be higher than viscous damping under certain conditions, and neglecting them may lead to significant over-estimation of low frequency motions. In applications where low frequency motions are an important design factor, such as for large tankers, it may be warranted to evaluate damping from all these sources either by analytical approach or model testing. (See Appendix D, References 34 through 44 for sources providing information for this evaluation.)

The determination of low frequency motions requires motions analysis computer programs or model tests. Design curves for estimating low frequency motions for drill ships and semisubmersibles are also provided in Figures A-3 to A-17. These curves are applicable to typical MODU type vessels. However, for large production and drilling semisubmersibles (with displacements over 30,000 short tons) and large tankers, the use of these curves is not recommended.

The curves presented are appropriate for mooring spring stiffness of 18 kips per foot of vessel offset. For other mooring stiffnesses, the results from Figures A-3 to A-17 should be adjusted by Equations A.4 and A.5:

$$X_S = (X_S)_{REF}\left(\frac{18}{k}\right)^{1/2} \quad (A.4)$$

$$Y_S = (Y_S)_{REF}\left(\frac{18}{k}\right)^{1/2} \quad (A.5)$$

Where:

$(X_S)_{REF}$ = rms single amplitude low frequency surge from Figures A-3 to A-17.

k = mooring system spring stiffness in kips/ft taken at the vessel's mean position.

$(Y_S)_{REF}$ = rms single amplitude low frequency sway from Figures A-3 to A-17.

The drillship curves in these figures are for drillships of 400 feet to 540 feet in length. For drillships that are outside this length range, the low frequency motions can be estimated by extrapolation. However, extrapolation for ship lengths below 350 feet or above 600 feet is not recommended.

A.4 Wind

The force due to wind may be determined by using wind tunnel or towing tank model test data or Equation A.6. The wind speed used is defined in 4.3.

A.4.1 Model Tests

Model test data may be used to predict wind loads for mooring system design provided that a representative model of the unit is tested, and that the condition of the model in the tests, such as draft and deck cargo arrangement, closely matches the expected conditions that the unit will see in service. Care should also be taken to assure that the character of the flow in the model test is the same as the character of flow for the full scale unit.

A.4.2 Wind Force Calculation

A.4.2.1 Constant Wind Force

The steady state force due to wind acting on a moored floating unit can be determined using Equation A.6.

$$F_w = C_W \Sigma (C_s C_h A) V_w^2 \quad (A.6)$$

Where:

F_w = wind force, lbs (N).
C_w = 0.0034 lb/(ft²•kt²) (0.615 Nsec²/m⁴).
C_s = shape coefficient.
C_h = height coefficient.
A = vertical projected area of each surface exposed to the wind, ft²(m²).
V_w = design wind speed, knots (m/sec).

The projected area exposed to the wind should include all columns, deck members, deck houses, trusses, crane booms, derrick substructure, and drilling derrick as well as that portion of the hull above the waterline. Wind shielding in accordance with acceptable methods may be considered.

In calculating wind areas, the following procedures can be followed:

a. The projected area of all columns should be included.
b. The blocked-in projected area of several deck houses may be used instead of calculating the area of each individual unit. However, when this is done, a shape factor, C_S, of 1.10 should be used.
c. Isolated structures such as derricks and cranes should be calculated individually.
d. Open truss work commonly used for derrick mast and booms may be approximated by taking 60 percent of the projected block area of one face.
e. Areas should be calculated for the appropriate hull draft for the given operating condition.
f. The shape coefficients, C_S, of Table A-1 can be used.

Table A-1—Wind Force Shape Coefficients

Exposed Area	C_s
Cylindrical shapes	0.50
Hull (surface above waterline)	1.00
Deck house	1.00
Isolated structural shapes (cranes, channels, beams, angles)	1.50
Under deck areas (smooth surfaces)	1.00
Under deck areas (exposed beams and girders)	1.30
Rig derrick	1.25

g. Wind velocity increases with height above the water. In order to account for this change, a wind force height coefficient, C_h, is included. The height coefficients, C_h, of Table A-2 can be used. This table applies to the approach using 1-minute constant wind (see 4.3).

Table A-2—Wind Force Height Coefficients (For 1-Minute Wind)

Height of Area Centroid Above Water Level				
Feet Over	Feet Not Exceeding	Meters Over	Meters Not Exceeding	C_h
0	50	0	15.3	1.00
50	100	15.3	30.5	1.18
100	150	30.5	46.0	1.31
150	200	46.0	61.1	1.40
200	250	61.0	76.0	1.47

Note: This table applies to the approach using 1-minute constant wind (4.3). It is based on the following equation for wind velocity:

$$\frac{Vz}{V_{10}} = \left(\frac{Z}{10}\right)^{1/10}$$

Where:

Z = Height of area centroid above water level (m).
Vz = Wind velocity at z.
V_{10} = Wind velocity at 10 m height.

h. Equation A.7 may be used to adjust the wind velocities of various average time intervals.

$$V_t = \alpha V_{hr} \tag{A.7}$$

Where:

V_t = wind velocity for the average time internal t.
α = time factor from Table A-3.
V_{hr} = 1 hour average wind velocity.

Table A-3—Wind Velocity Time Factor

Average Time Period t	Time Factor α
1 hour	1.000
10 min.	1.060
1 min.	1.180
15 sec.	1.260
5 sec.	1.310
3 sec.	1.330

A.4.2.2 Low Frequency Wind Force

As stated in 4.3, wind force can be treated as constant or a combination of a steady component and a time varying component. The time varying component is also known as low-frequency wind force. Similar to the low frequency second order wave forces, low-frequency wind force also induces low frequency resonant surge, sway, and yaw motions. Low-frequency wind forces are normally computed from an empirical wind energy spectrum. Low-frequency wind and wave forces are normally combined to yield low frequency vessel motions due to both effects.

Due to the large variability in measured wind spectra, there is no universally accepted spectral shape. In the absence of data indicating otherwise, the wind spectrum presented in API Recommended Practice 2A [2] can be used.

A.4.2.3 Wind Force for Large Tankers

Steady wind forces for large tankers can be estimated using Equation A.1, as discussed in A.2.2, item a.

A.5 Oblique Environment

The equations presented are convenient for calculating wind and current forces for bow and beam environments. For environments approaching from an oblique direction, Equation A.8 can be used to evaluate wind and current forces if more accurate predictions are not available.

$$F_\phi = F_X\left[\frac{2cos^2\phi}{1 + cos^2\phi}\right] + F_Y\left[\frac{2sin^2\phi}{1 + sin^2\phi}\right] \tag{A.8}$$

Where:

F_ϕ = force due to oblique environment, lbs (N).
F_x = force on the bow due to a bow environment, lbs (N).
F_y = force on the beam due to a beam environment, lbs (N).
ϕ = direction of approaching environment (degrees off bow).

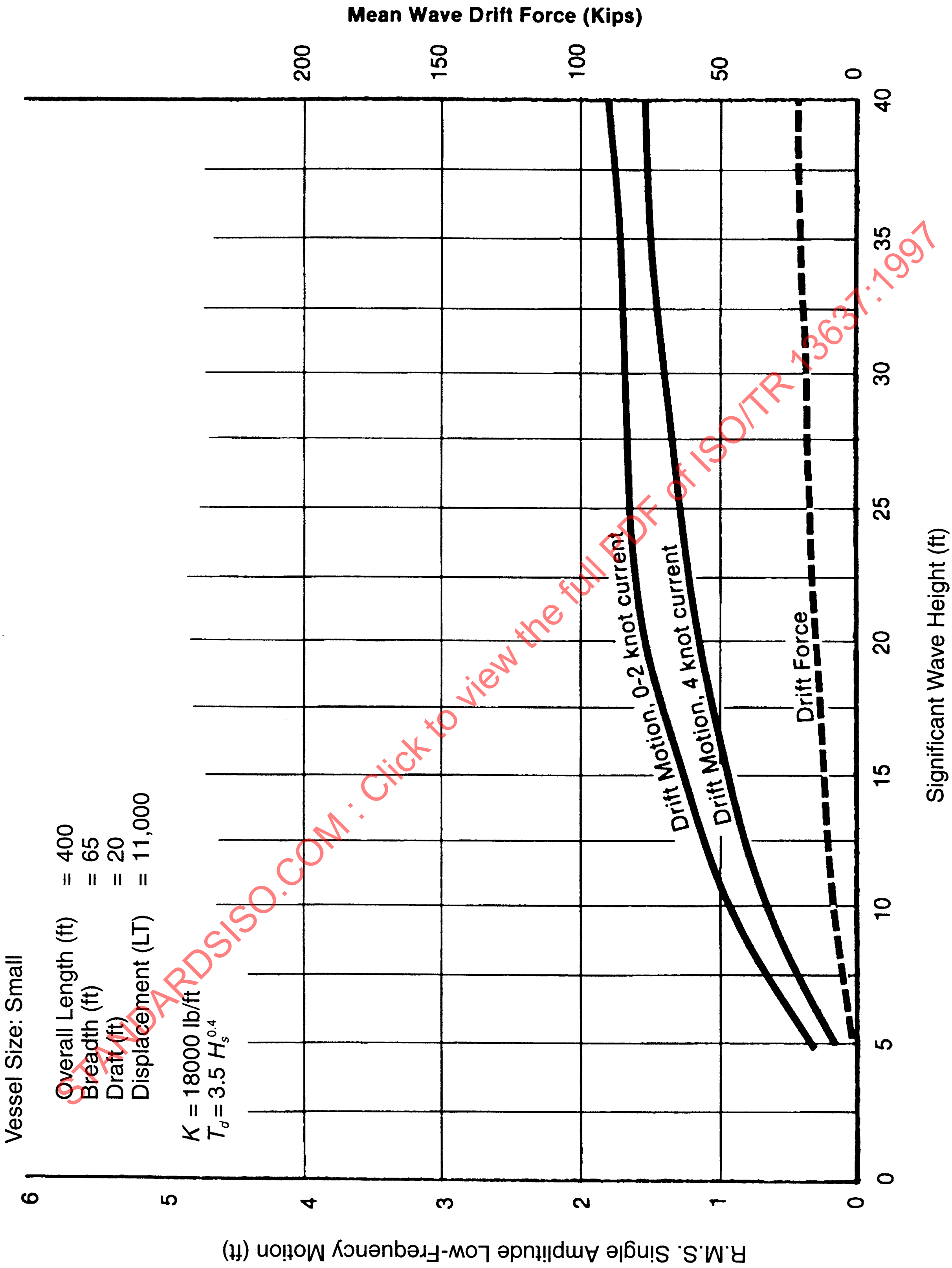


Figure A-3—Wave Drift Force and Motion for Drillships Bow Seas

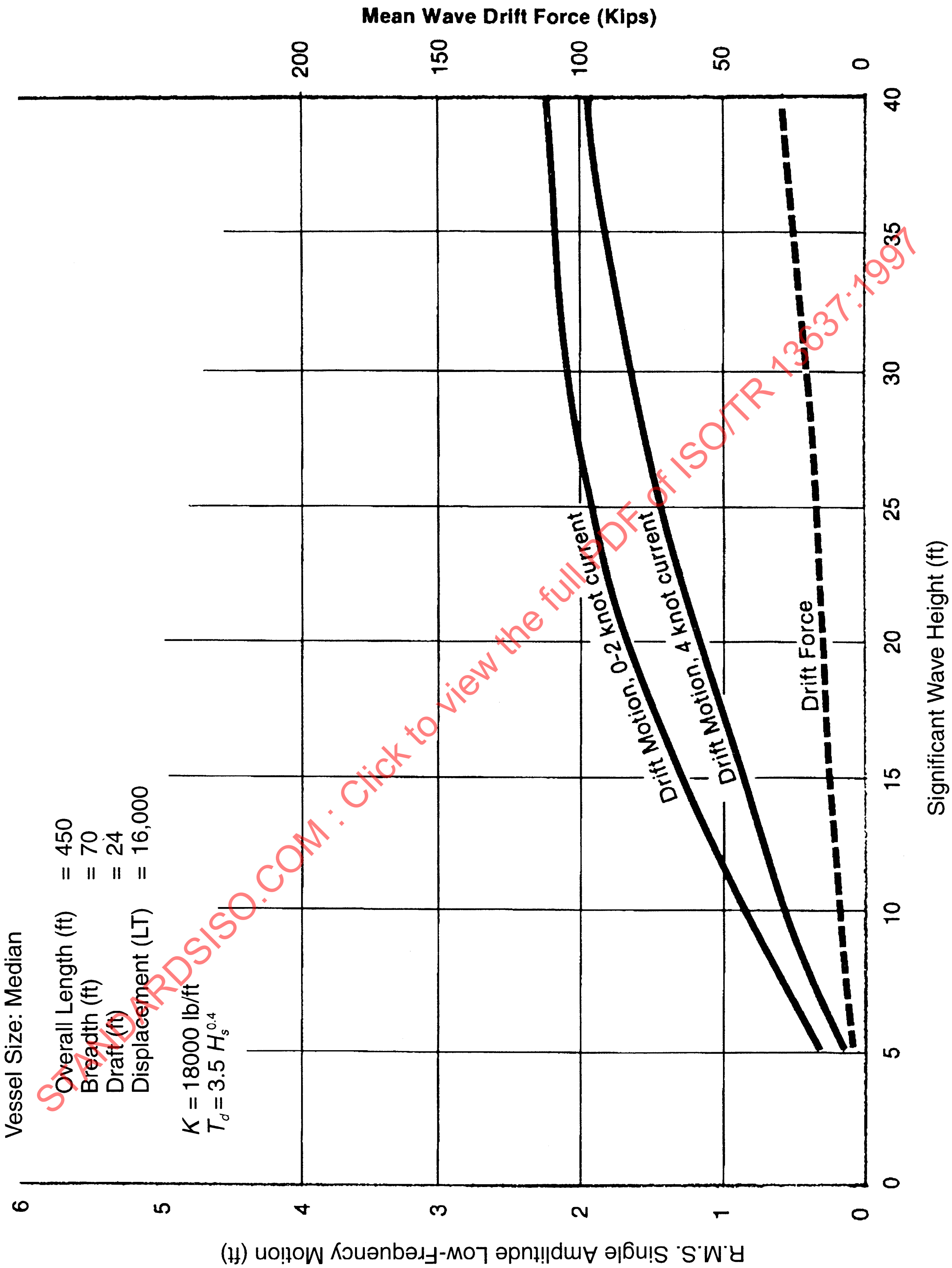


Figure A-4—Wave Drift Force and Motion for Drillships Bow Seas

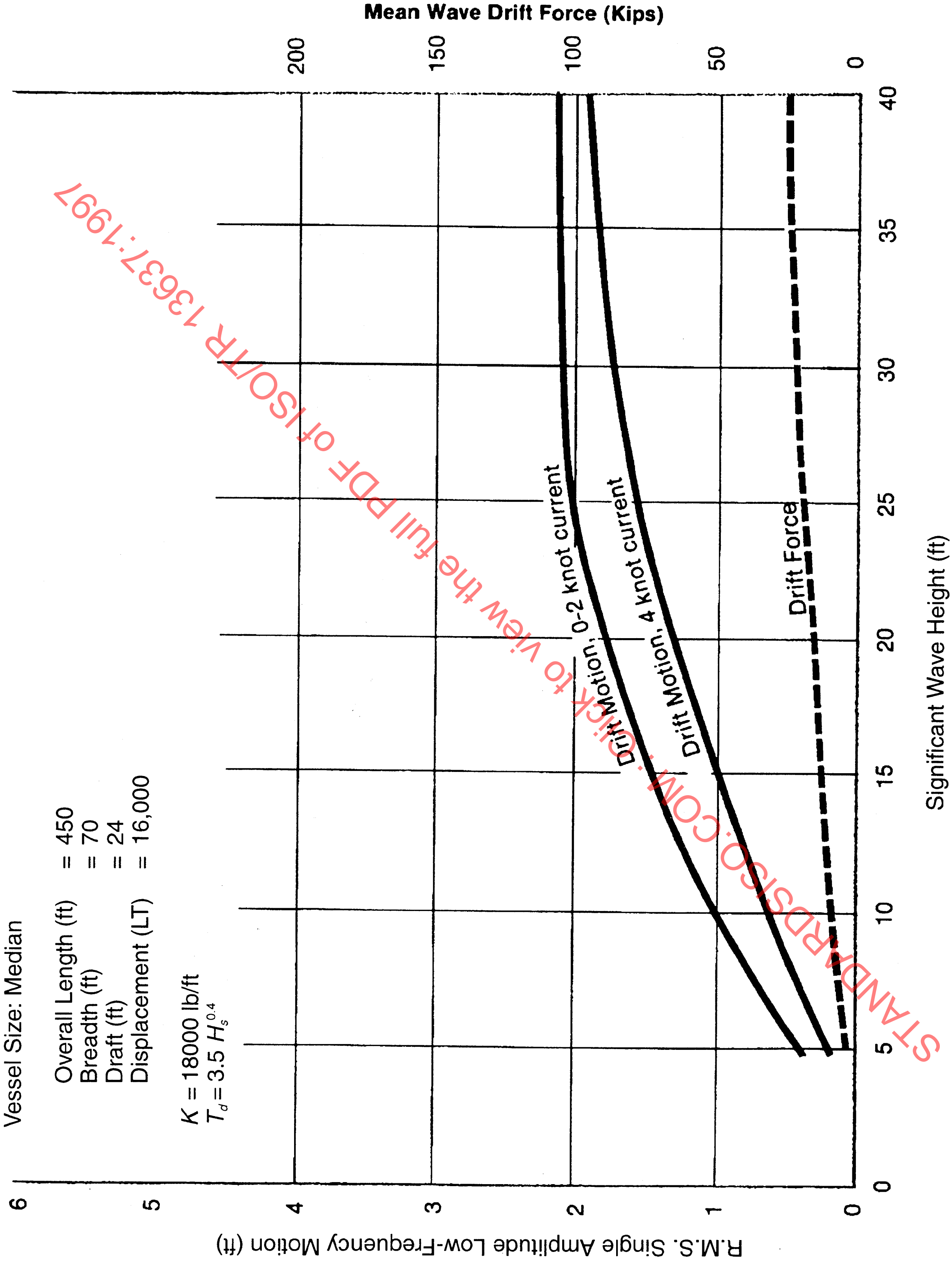


Figure A-5—Wave Drift Force and Motion for Drillships Bow Seas

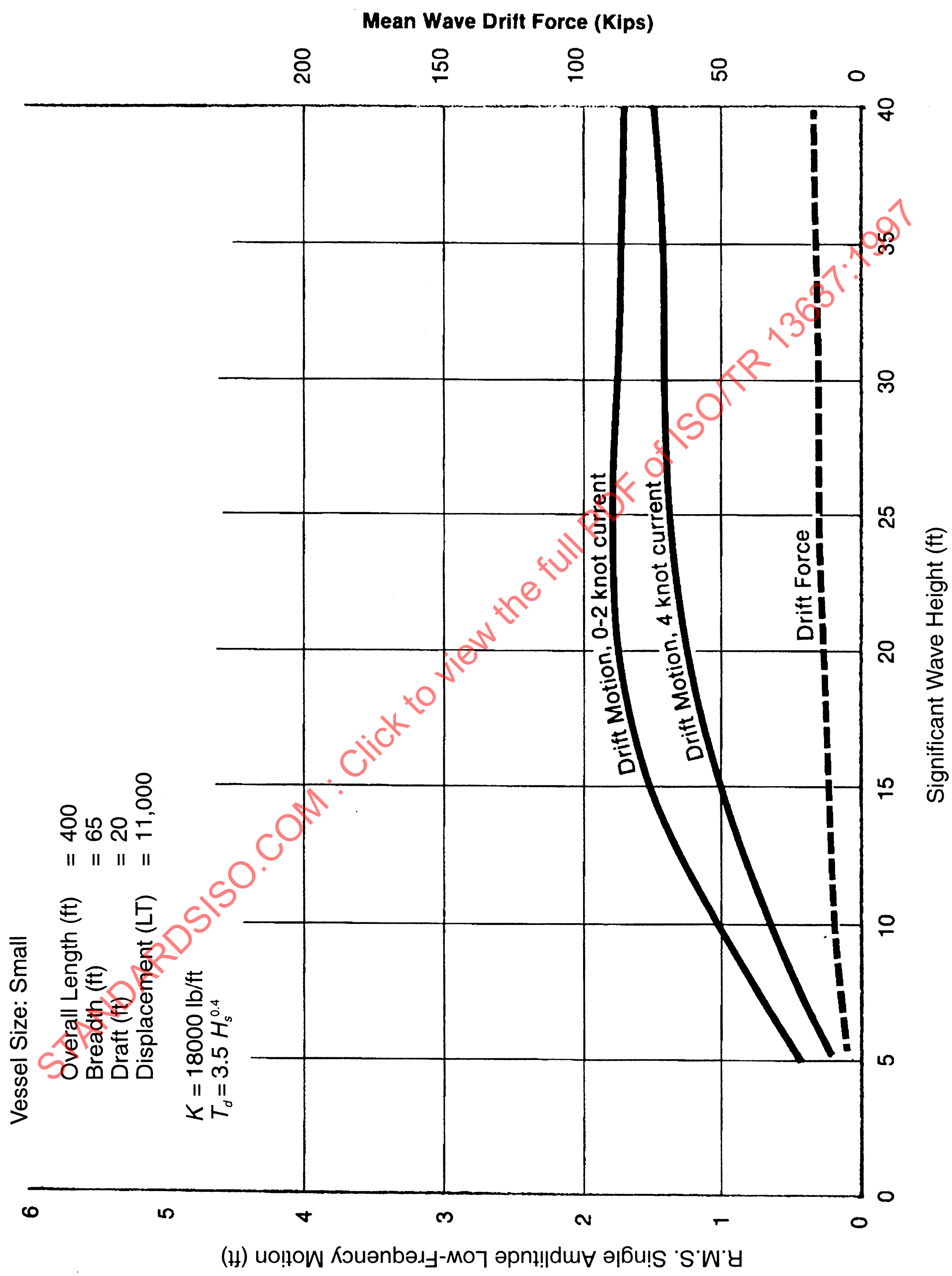


Figure A-6—Wave Drift Force and Motion for Drillships Quartering Seas, Surge

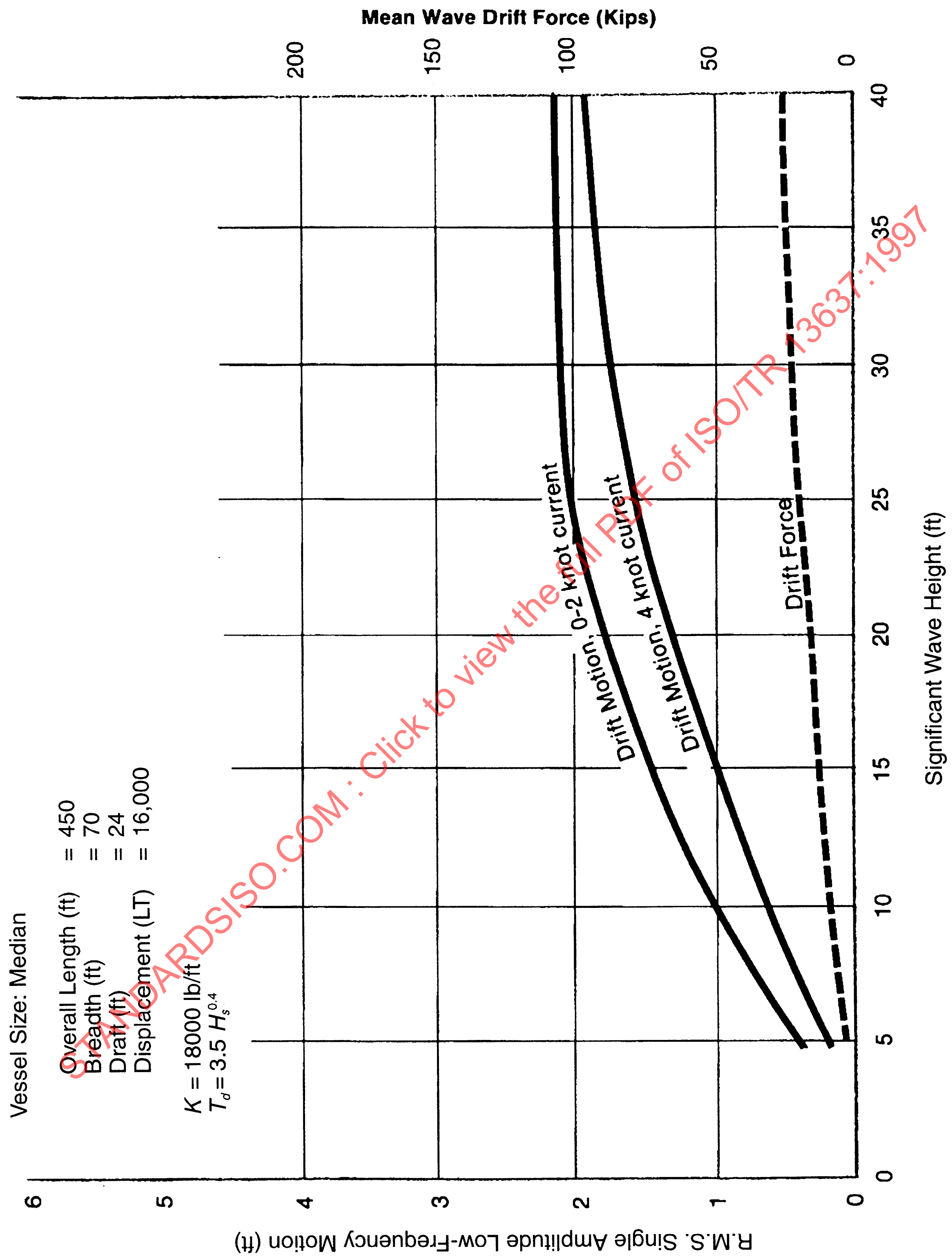


Figure A-7—Wave Drift Force and Motion for Drillships Quartering Seas, Surge