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GUIDELINES FOR DEEPWATER PORT SINGLE POINT MOORING DESIGN

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16. Abstract

This report provides guidelines for the establishing of design mooring loads for SPMs (single point moorings) and for the design of mooring components of SPMs. It first discusses the manners in which waves, winds, current, the tanker, and the mooring system influence mooring loads. It then describes how mooring loads should be determined through model testing and statistical analysis. Next, it discusses how the arrangement and strength of mooring fittings on tankers may limit the maximum allowable mooring loads. The report then discusses the properties of the synthetic ropes normally used as mooring hawsers and recommends factors of safety for SPM hawsers. It also discusses the rules and standards which should be used for mooring structures, anchor chains, bases, and piles. Next, the report discusses synthetic-rope testing and field inspection. The report ends with recommendations for further investigation of large-diameter synthetic rope properties, testing, and inspection. The report contains an extensive glossary of SPM terminology and a bibliography of other reports and articles in the field of SPMs.

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INTRODUCTION

1.1 PURPOSE

The "Deepwater Port Act of 1974" (Public Law 93-627, 1975) grants to the Secretary of Transportation authority to issue licenses for the construction and operation of deepwater ports off the coast of the United States. In the process of granting such licenses, the Secretary must determine "that the applicant has demonstrated that the deepwater port will be constructed and operated using best available technology, so as to prevent or minimize adverse impacts on the marine environment".

To implement this act, the Secretary of Transportation designated the U.S. Coast Guard as the agency to develop the rules and regulations under which a deepwater port may be constructed and operated. Pursuant to this goal, the Coast Guard has issued "Deepwater Port Regulations on Licensing Procedures and Design Construction, Equipment and Operations Requirements" (1975) and "Recommended Procedure for Developing Deepwater Ports Design Criteria" (1975).

One key element in the total safe operation of a deepwater port is the SPM (single point mooring). The Coast Guard Regulations require the applicant for a deepwater port license to submit "A description and the results of any design and evaluation studies performed by or for the applicant on a floating component". Further, the Coast Guard Recommended Design Criteria states, "The designer should submit data to support the operating mooring loads based on physical dynamic model tests and supporting calculations for similar SPMs and environments".

As the Deepwater Port Act requires that the best available technology be used to minimize risks, the Coast Guard is conducting research in a number of technical areas relating to deepwater port design, construction, and operation. Determination of the mooring loads at a SPM and design of a mooring system adequate for these mooring loads are among the most important aspects of the design of the deepwater port. The U.S. Coast Guard, in its role in granting permits for deepwater ports, will review the design of SPMs. As a part of this review, they will assess the manner in which mooring loads have been established, the compatability of the established mooring loads with various limiting criteria, and the adequate design of the mooring lines, structures and anchor chains for the established mooring loads.

1.2 SCOPE OF STUDY

The Coast Guard recognizes that the field of SPMs is an emerging technology and that the best available technology in this field is not found in standard text books and design manuals. In order to adequately prepare to perform reviews, the Coast Guard solicited proposals for a research study entitled "Deepwater Port Mooring Loads Determination Study". The objectives of this study were stated in the Coast Guard statement of work as follows (Coast Guard, 1976):

- Examine all parameters that affect the mooring loads to be experienced in all conditions of possible deepwater port operation.
- Provide guidance and direction to allow the thorough, meaningful review of the mooring segment of the deepwater port license application.
- Provide a basis for the continued Coast Guard review of the total mooring system adequacy.

In its response to the statement of work, Exxon Research and Engineering (ER&E) proposed to develop a set of guidelines which will enable the Coast Guard to evaluate the mooring loads submitted as part of deepwater port applcations and the manner in which they were determined, as well as the adequacy of the mooring system design and the operating and maintenance procedures established for the mooring portion of the deepwater port. The study was divided into eight tasks as follows:

Task 1 Background study.

Task 2 Develop guidelines for evaluating the influence of factors affecting mooring loads.

Task 3 Develop guidelines for evaluating the conducting of model tests and the analysis of model test data.

Task 4 Develop guidelines for evaluating factors influencing the maximum permissible environments and mooring loads.

Task 5 Develop guidelines for evaluating mooring system designs.

Task 6 Develop guidelines for evaluating mooring-line test procedures.

Task 7 Evaluate areas for further investigation.

Task 8 Prepare interim and final reports of study.

The following report presents the data and guidelines developed as a result of this study. Sections 2 through 7 of the report correspond to the above mentioned Tasks 2 through 7.

1.3 SINGLE POINT MOORINGS

In its broadest definition, the SPM for tankers consists of an integrated mooring and cargo transfer system which incorporates either a cargo swivel concentric with the mooring system or a mooring swivel concentric with the cargo system so the tanker can freely swing around the mooring in response to the environment while simultaneously transferring cargo. A general schematic plan view of an SPM is shown in Figure 1-1.

As the tanker is free to align itself into the environment at an SPM, mooring forces are minimized. Thus, the tanker can remain moored and continue transferring cargo in environments more severe than could be tolerated at moorings, such as piers and multiple buoy berths, where the tanker is held in a fixed heading. One of the principle advantages of the SPM is it can be located offshore in uncongested deepwater instead of in protected, crowded harbors and bays. (Flory, 1975).

The first type of SPM, known as the Catenary Anchor Leg Mooring (CALM), was developed simultaneously and independently by Shell Oil Company and IMODCO, then a Swedish company but now a U.S. company based in Los Angles. Shell installed a number of CALMs in the Far East in the early 1960's through contracts with IHC, a Netherlands shipbuilding company. IHC became a licensee of the Shell CALM and through a subsidiary, SBM Inc. of Monaco, has supplied over half of the CALMs in the world today. IMODCO developed CALMs for the Swedish and German navies and then became a major supplier to oil companies; the company also has furnished a number of CALMs for the U.S. Armed Forces in the Far East and for handling fluidized iron ore and liquid propane.

A typical CALM system is shown in Figure 1-2. The CALM consists of a large flat buoy, approximately 10 to 12 m (33 to 40 ft) in diameter and 3 to 5 m (10 to 16 ft) high, that is anchored by four or more chains extending in catenaries to anchor points on the sea floor, sometimes as far as 400 m (1,300 ft) from the buoy. The tanker is moored by bow hawsers to a turntable on the deck of the buoy. Floating cargo hose connects through piping on this turntable to a fluid swivel in the center of the buoy. Underbuoy cargo hose connects this swivel with a manifold at the end of the submarine pipeline.

A new type of SPM, the Single Anchor Leg Mooring (SALM), was developed in the late 1960's by ER&E. The first SALM was installed in Libya in 1969, and six more SALMs have been installed throughout the world to date. The SALM has been licensed to SBM Inc., IMODCO, and a new firm, SOFEC of Houston, Texas. As a result of extensive model testing, the SALM was selected by SEADOCK and LOOP for their proposed deepwater ports.

The SALM design for deepwater ports is shown in Figure 1-3. The SALM consists of a mooring buoy at the sea surface, which is attached to a base on the sea floor by a single anchor leg. The buoy is drawn down against it buoyancy by tension in the anchor leg. Tankers moor through lines to the buoy, and a swivel in the anchor leg or on the buoy allows the tanker to swing around the mooring point. A fluid swivel is mounted concentric about the anchor leg, either on top of the base or on top of a riser pivoted from the base and forming part of the anchor leg. Cargo hoses connect to an arm on the fluid swivel and rise to the surface, where they float and extend to the tanker manifold.

Another type of SPM consists of a fixed mooring tower with a mooring turntable on its deck. The first such mooring tower, which was installed by an Exxon affiliate in Libya in 1963, is shown in Figure 1-4. The tower features an underwater loading arm extending from the turntable to a riser positioned beside the tanker's midship manifold. Several mooring towers that employ floating loading hoses have been installed in Italy and Scotland.

A number of other SPM systems have been proposed, and examples of several of these have been installed. However; to date, the CALM, SALM, and tower-type SPM are the principal types of SPMs used as terminals in relatively shallow water. Most of the material in the following report is general enough to apply to any type of SPM, and some of the material will be of use in the design and evaluation of other mooring systems. A few subsections are particular only to the CALM and SALM.

The design of the cargo transfer components of SPMs is not covered in this report. Only a few remarks are given applicable to the effect of the environment on the design and operation of the cargo system. Separate Coast Guard sponsored research is being conducted by others on cargo transfer system design.

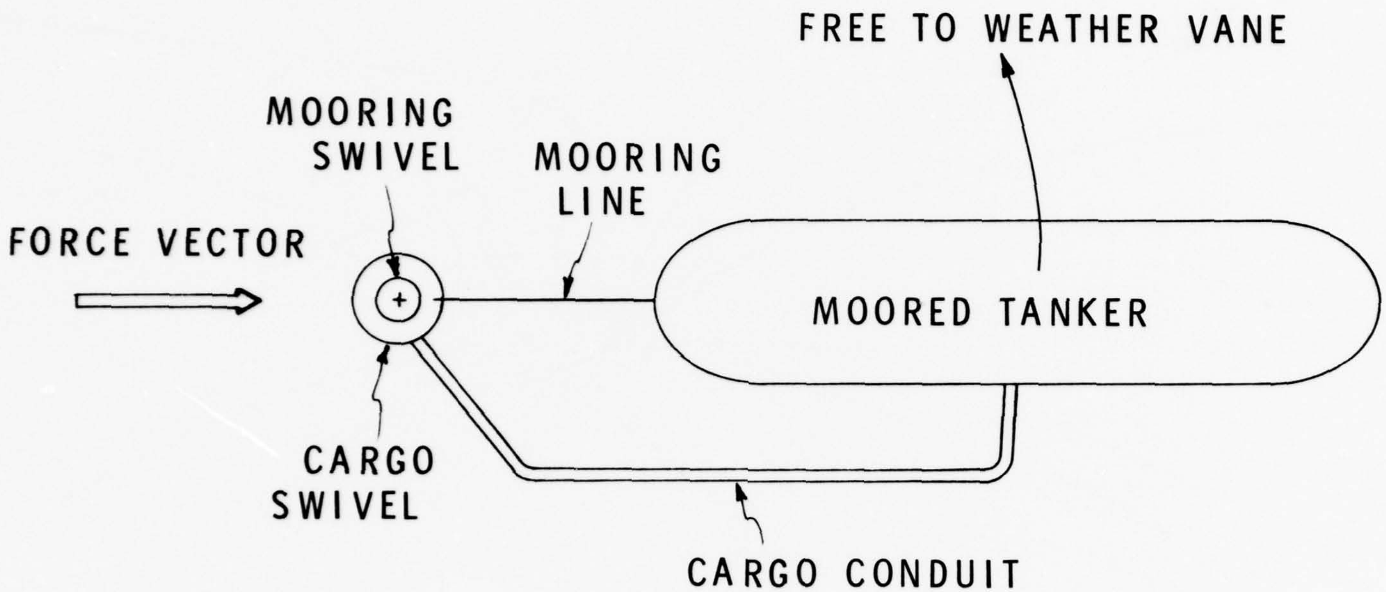


FIGURE 1-1 SCHEMATIC OF THE SINGLE POINT MOORING CONCEPT

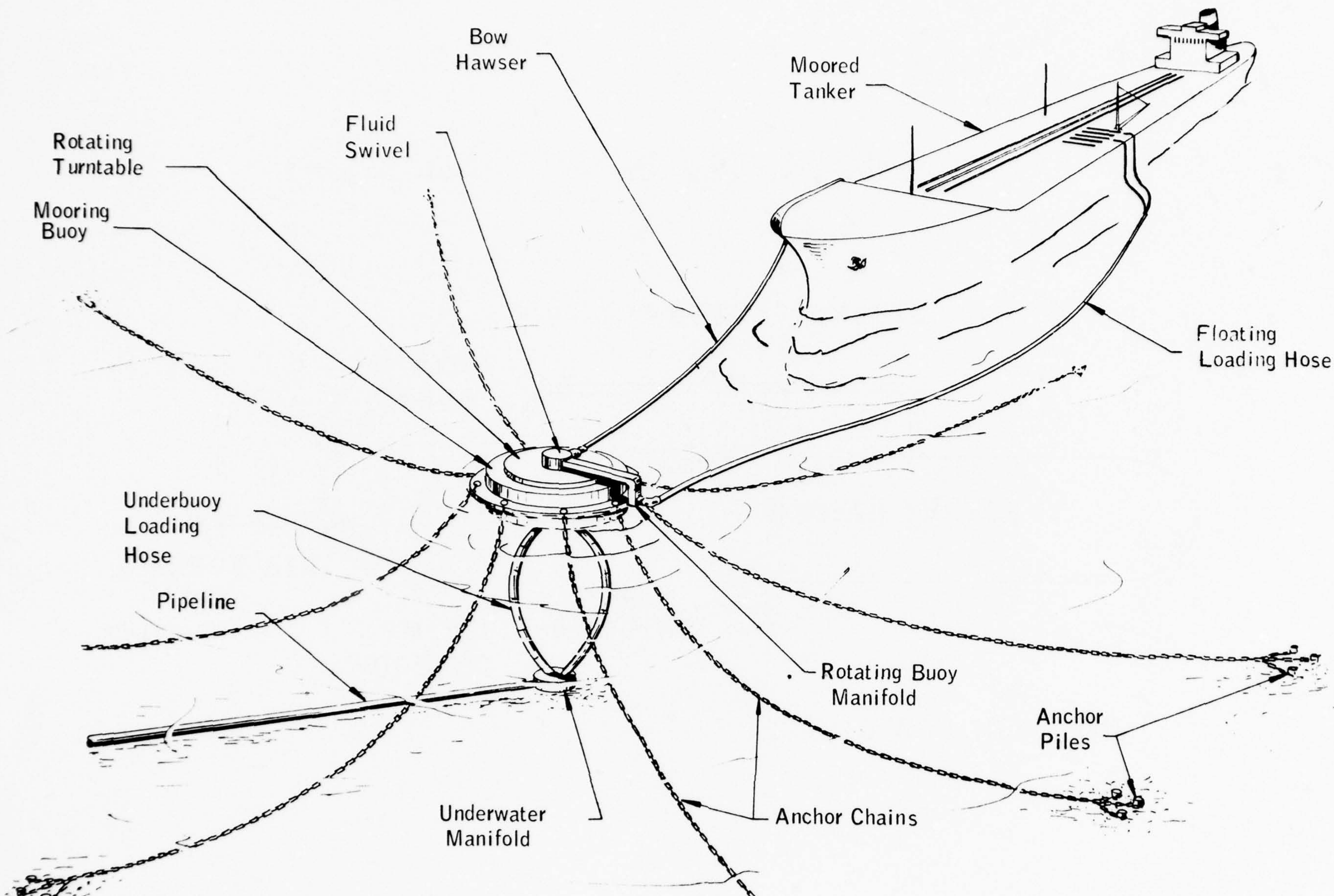


FIGURE 1-2 CATENARY ANCHOR LEG MOORING

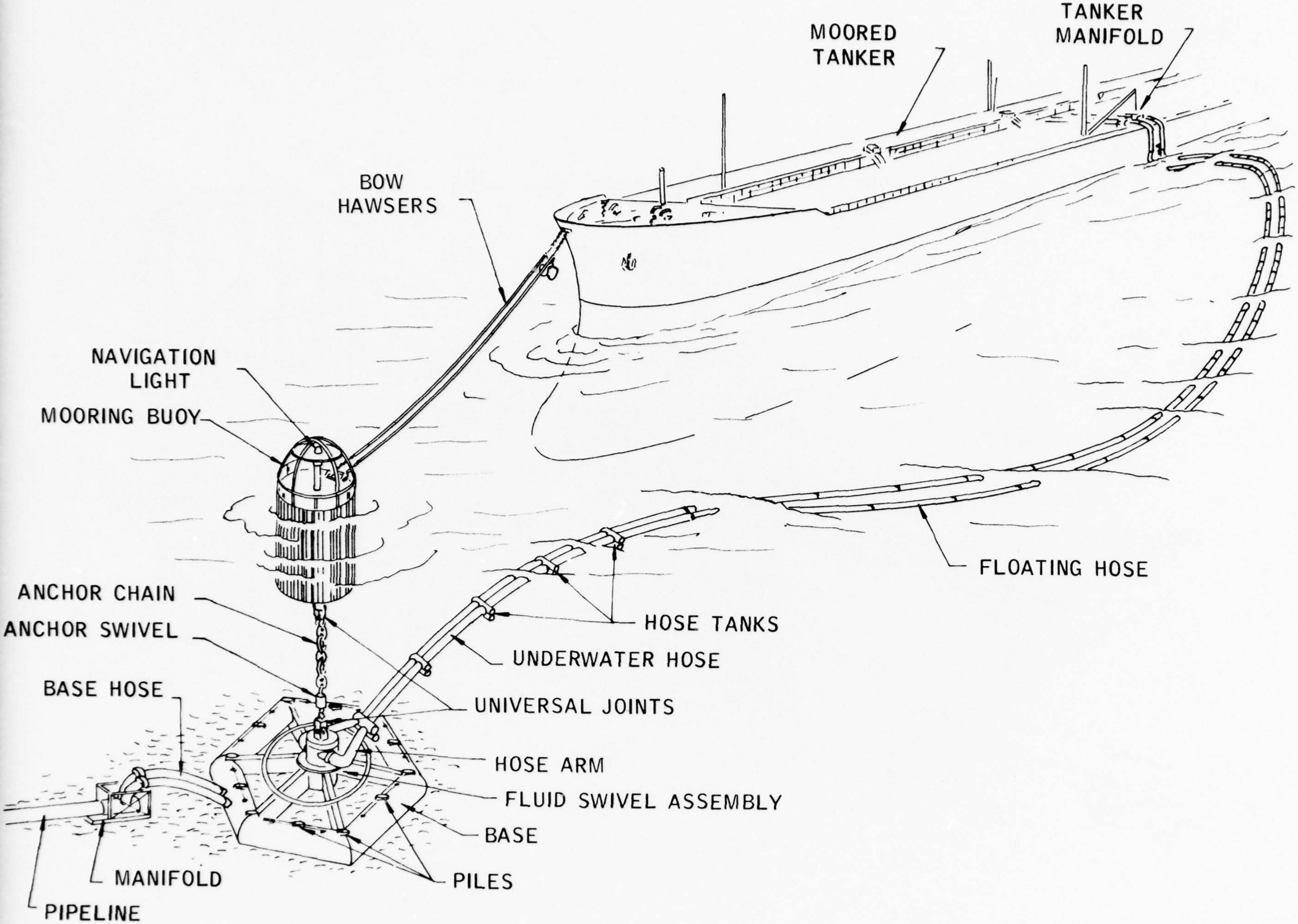


FIGURE 1-3 SINGLE ANCHOR LEG MOORING

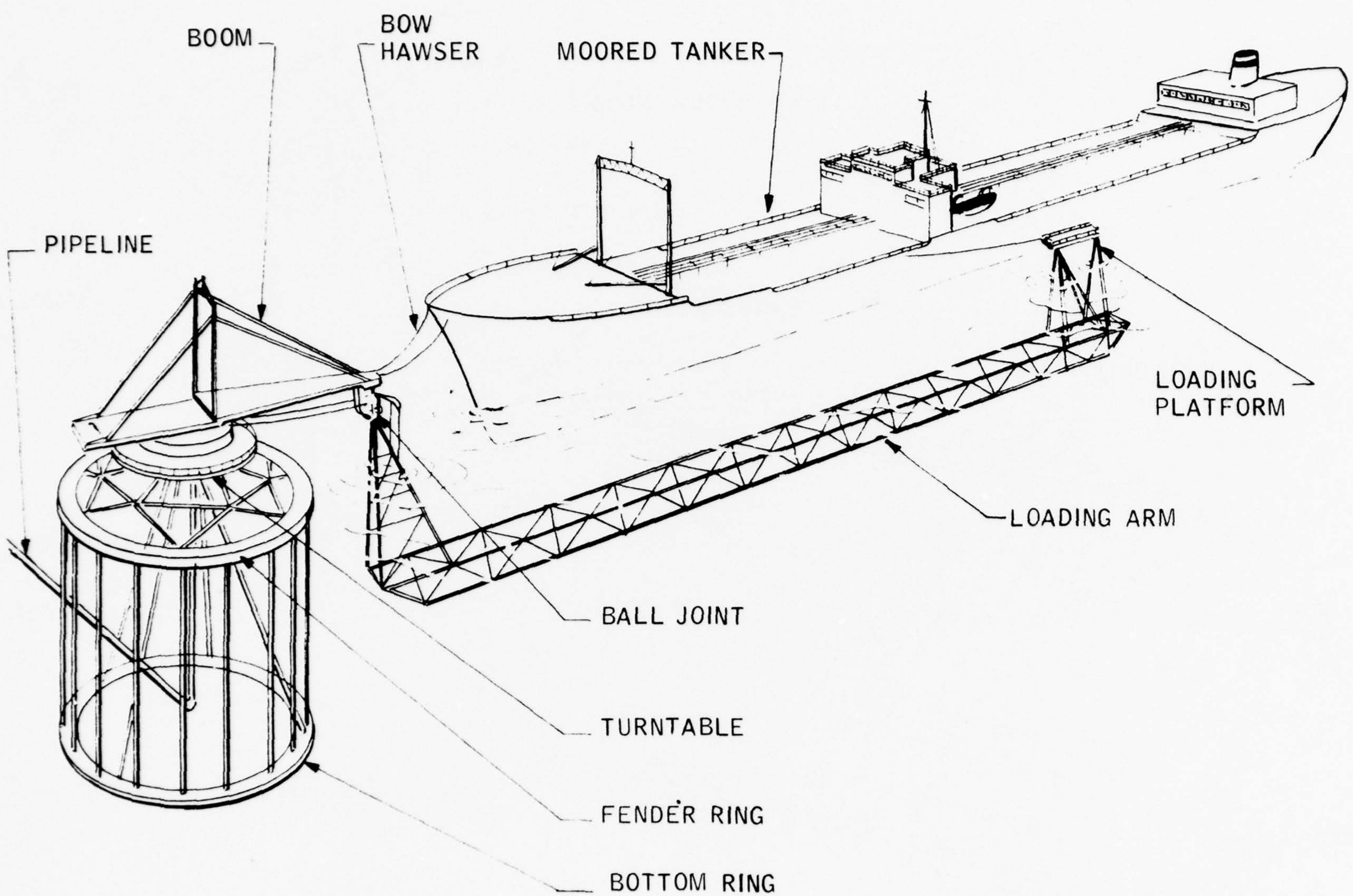


FIGURE 1-4 TOWER-TYPE SINGLE POINT MOORING

SECTION 2

FACTORS WHICH INFLUENCE MOORING LOADS AT SPMS

2.1 INTRODUCTION

In planning the design of an SPM, the designer must consider all pertinent design factors including the site, the environment, the tanker, mooring operation, and the physical design parameters of the mooring. Certain factors are very important in establishing design loads, while other factors have only minor effects on mooring loads. The designer must be aware of the significance of the various factors and the possible effect which a change in a parameter may have on mooring loads and mooring performance.

The Coast Guard, in evaluating the mooring system design aspects of a deepwater port license application, must be able to ascertain that all pertinent design factors have been considered by the designer. This section describes the various parameters which are known to influence mooring loads and mooring-system performance. The relative effects of parameters, and the manner in which variations in parameters may effect mooring loads are discussed. General quantitative assessments of the influences of variations in parameters are given where possible.

2.2 INFLUENCE OF WAVES

Wave height is generally the most important single parameter influencing SPM mooring loads. Defining waves by wave height alone is not, however, sufficient for design purposes. The distribution of wave energy with period, known as the wave spectrum, also has a significant influence on mooring loads.

2.2.1 Description Of Waves

Waves at typical deepwater port SPM sites, like ocean waves in general, can be expected to be irregular. They cannot be adequately represented in terms of a regular sinusoidal wave function having a single amplitude and a single frequency. It is difficult, if not impossible, to completely describe ocean waves. However, for most practical engineering purposes the concepts of significant wave height, statistical wave height distribution, and power or energy spectrum are used to define the important characteristics of irregular waves in the ocean. These concepts are described in some detail in Appendix A.

The significant wave height is the average of the highest one-third of the peak-to-trough waves in a wave record. Significant wave height is a convenient means of describing irregular waves. Throughout this report the term wave height will mean significant wave height unless otherwise stated. Significant wave height will be designated in the report by H_s. Significant wave height is frequently designated in other sources by $H_{1/3}$ or similar notation.

The term wave period applied to irregular waves is not as commonly agreed upon as significant wave height. In this report wave period will refer to the average of the periods of the highest one-third of the waves in a wave record unless otherwise stated. This wave period will be designated by T_s and will sometimes be referred to as significant wave period. Other definitions of period are also common in reference to irregular waves. In stating period, as well as wave height, the definition must be understood. In general this report deals with concepts and trends, and the conclusions drawn here with regard to trends will apply as long as the parameters being considered have consistent definitions.

2.2.2 Wave Height

Certain rules of thumb for the manner in which mooring loads vary with parameters have been developed from the analysis of model test data. These rules of thumb are not precise enough for design purposes, but they can serve to assess the effect which a change in a parameter will have on mooring load.

Mooring hawser load has been found to increase roughly in proportion to the square of wave height in the absence of wind and current.

$$F \approx k\, H_s^2 \qquad (2\text{-}1)$$

where F = significant mooring force

H_s = significant wave height

k = constant of proportionality

This rule of thumb assumes that tanker size and condition remain constant, the type of wave spectrum remains the same, and wave period changes commensurate with the wave height. In moderate and high waves with low to moderate wind and current acting nearly in line with the waves this rule of thumb may still apply. Where winds and currents are high and/or at large angles to waves, their influence will tend to overcome this rule, and mooring forces will probably not increase in proportion to the square of wave height. (Flory and Poranski, 1977).

2.2.3 Wave Period And Spectrum

Wave period and the type of wave spectrum can also influence mooring loads, though in general their influence is not as great as wave height. Recall that the wave period is not an independent parameter, but is related to wave height and the type of wave spectrum. Thus a change of wave period with constant wave height generally implies a change in wave spectrum, that is a change in the band of frequencies in which wave energy is concentrated.

For a constant significant wave height, mooring loads generally increase with a decrease in wave period. Mooring loads increase roughly in proportion to the inverse of the square root of the wave period. Thus from the wave-height rule of thumb and the wave-period rule of thumb, the following approximate relationship appears to apply:

$$F \approx k \frac{H_s^2}{\sqrt{T_s}} \tag{2-2}$$

where T_s = significant wave period

and F, H_s, and k are defined as before

Several different spectra may have nearly the same mean wave period for a given wave height. A narrow-band wave spectrum, where most of the energy is concentrated in a narrow frequency band, may influence the response of the SPM and the moored tanker differently than a wide-band wave spectrum. If the natural frequency of the buoy or of some mode of response of the moored tanker, for example roll, are very near the wave period, then response, and thus mooring loads will be higher in a narrow band wave spectrum. Conversely, if the natural frequencies of the buoy and tanker are near the wave period but far enough from that period that they are hardly influenced by a narrow band wave spectrum, those natural frequencies may be excited more by a wide-band wave frequency. (Flory and Poranski, 1977).

2.2.4 Long-Period Wave Phenomena

Long-period wave phenomena can have a major influence on the response of the moored tanker, and thus on mooring loads. These long-period phenomena, generally referred to as seiche or swell and wave grouping, are discussed in Appendix A.

If the period of the sieche or wave grouping phenomena corresponds to a natural period of the mooring system then peak mooring forces may be higher than would otherwise be experienced in waves of the same height. Wave grouping and seiche have been found in some wave records but it is not known how common they are in ocean waves. In planning an SPM terminal, an analysis should be performed to determine if these long period phenomena exist at the site. If they are expected at the site then they should be accounted for as much as possible in model testing. (Remery and Hermans, 1971; Remery and Kokkeel, 1976).

2.3 INFLUENCES OF WIND AND CURRENT

Unlike the loads imposed on the tanker by waves, the static loads imposed by wind and current can be fairly accurately determined. However, due to the nature of response of the tanker to these wind and current loads, the influences of wind and current on SPM loads may be difficult to determine. The term loads here is intended broadly to include both forces and moments.

Because moments are imposed when the tanker hull is at an angle to wind or current, the tanker may move to apparently unusual positions under the combined effects of waves, wind, and current. The loads imposed on the hull by wind or current alone and the possible response of the vessel to these loads are covered in this subsection. The influences of waves, wind, and current acting in combination are covered in subsection 2.4.

2.3.1 Wind Load On The Vessel

A number of model tests have been conducted in wind tunnels to determine wind loads on typical tankers. The Oil Companies International Marine Forum (OCIMF) recently commissioned a series of wind and current model tests on VLCCs. ER&E arranged for the model testing and analyzed the test data. This new data on wind and current loads on VLCCs has now been published by OCIMF. (OCIMF, 1977; Benham, Fang, and Fetters, 1977).

The prototype wind induced loads on the vessel are determined by model tests conducted in wind tunnels where vessel dimensions and wind velocity are scaled by Reynolds' law. The model is mounted near the floor of the wind tunnel in order to imerse it in the boundary layer. The velocity gradient in the boundary layer may be adjusted, for example by roughening the floor of the wind tunnel, to approximate the 1/7th power-law profile of the natural wind profile over water.

In wind-tunnel model tests forces and moments on the vessel are measured in the horizontal plane with the model set at various angles of wind attack. Non-dimensional wind force and moment coefficients are calculated from the wind tunnel test results. The most common method of defining these coefficients is as follows:

C_{Xw} = Longitudinal wind force coefficient

C_{Yw} = Lateral wind force coefficient

C_{XYw} = Yaw wind moment coefficient

Typical plots of C_{Xw}, C_{Yw}, and C_{XYw} as a function of angle of attack for a typical loaded and ballasted tanker are shown in Figures 2-1, 2-2, and 2-3.

The equations for calculating the longitudinal and lateral wind force and the wind moment acting on the vessel are as follows:

$$F_{Xw} = C_{Xw} \frac{k}{2} \rho a \; V_w^2 \; A_T \tag{2-3}$$

$$F_{Yw} = C_{Yw} \frac{k}{2} \rho a \; V_w^2 \; A_L \tag{2-4}$$

$$M_{XYw} = C_{XYw} \frac{k}{2} \rho a \; V_w^2 \; A_L \; L_{BP} \tag{2-5}$$

where;

F_{Xw} = Longitudinal wind force

F_{Yw} = Lateral wind force

M_{XYw} = Wind yaw moment

ρ_a = Density of air

V_w = Wind velocity

A_L = Longitudinal (broad-side) wind area

A_T = Transverse (head-on) wind area

L_{BP} = Length between perpendiculars of the vessel

k = conversion factors for units

In using wind load coefficient data and formulas, caution must be paid to the basis of the data, the form of equation it is to be used in, and the units to be used. Areas are sometimes based on projected areas, and sometimes based on total area, including that part of an aft house which is blocked from direct view by a midship house or the forecastle.

The above coefficients and equations are given in terms of forces acting along and transverse to the longitudinal centerline of the vessel and moment about the center of the vessel. In subsection 3.5.3 the coefficients are redefined in terms of lateral forces acting at the forward and aft perpendicular in lieu of a moment. Another method of representing wind loading which is sometimes used is to give the moment arm, either forward or aft, at which the lateral wind force must act to produce the wind-induced moment. Yet another convention is to define coefficients in terms of the forces acting parallel to and perpendicular to the direction of wind.

2.3.2 Effect Of Wind Gusts

The velocity of wind over the ocean varies with time, almost in a manner analogous to, though not as rapidly as, wave height. Wind velocity is expressed in many ways. The terms one-minute average and one-hour average do not need explanation. Fastest mile wind is the average velocity of a one nautical-mile length of wind as it passes the point of reference. At slow velocities a fastest mile wind may be close to the one-hour average while at high velocities the fastest mile will be close to the one-minute average. Gust velocities usually are averages over very short periods, such as a ten-second gust. Figure 2-4 shows the relationship of average wind velocities over various durations. (Vellozi, 1968).

The statistics of wind velocities are not nearly as developed as those for waves. General relationships between the various gust and average wind velocities may be used to convert wind velocity measurements to different bases. However, only limited statistics on the recurrence or periodicity of wind gusts over the ocean are available, and randomly varying wind velocity corresponding to a wind spectrum have not been used in design and analysis. Wind gust spectral density variation used for structural design for winds over land is shown in Figure 2-5 (Vellozi, 1968). This wind spectrum shows that for a 30 knot mean hourly wind the maximum response will occur for a system with a natural period of about 40 seconds. However, the wind spectra is much broader than wave sectra, that is, wind-energy variation occurs over a very wide range of periods.

If wind spectra were available for use in the design of offshore moorings then the question would be raised; How should wind statistics be combined with wave statistics? It would be statistically extreme to assume the peak wind gust occurs simultaneously with the peak wave height. It would appear to be more valid to combine an irregular wind pattern with an irregular wave. However, superimposing two sets of random statistics in this manner may produce no different effect than that produced by only one of the sets. Furthermore, the statistics for the wind are simply not available in a useable form.

In one model test series ER&E used a varying wind. The wind was controlled to alternate between two different velocities with a duration at each veloctiy of 4 minutes and a transition time of 1 minute. The period of this varying wind was chosen to correspond with what was expected to be the natural period of the mooring. Results of this test series showed the peak mooring loads to be essentially the same as the peak mooring loads measured with a steady wind of the higher wind velocity. Choosing some other pattern of wind velocity variation might show varying wind to be a significant factor in determining vessel response and mooring loads. However, ER&E now uses a steady wind velocity in model testing.

The response of the moored vessel should be analogous to a large structure on land. The ANSI Building Code guidelines for wind gust response factors on buildings (Vellozi, 1968) provides a means of determining the gust duration which may influence the moored vessel. The wind gust response factor is a measure of the effective dynamic load produced by gusts and serves to relate the dynamic response phenomena to a simpler static design criteria. The ANSI design wind gust was calculated for a 190,000 dwt ballasted vessel moored at a typical SPM in a 45 knot wind. The analysis showed that the design wind gust duration was about one minute.

Based on that analysis, it appears a steady velocity corresponding to the one-minute gust velocity of the design wind is appropriate for use in model testing and design analysis of SPM systems. Using a one-minute gust velocity should produce essentially the same effect as a one-minute gust in an irregular wind record occurring simultaneously with the peak wave induced forces. The above analysis is not rigorous enough to firmly conclude this. However, the use of the one-minute gust velocity is recommended as an interim guideline until further investigations on this topic are performed. The one-minute gust velocity is approximately 1.24 times the mean hourly wind.

2.3.3 Influences On Wind Loads

From wind-load equations such as those given in subsection 2.3.1, it is obvious that wind forces and moments imposed on the vessel are proportional to the square of wind velocity. Also, the forces are proportional to the projected hull and superstructure areas, and the moment is proportional to the product of the longitudinal area and the length between perpendiculars. Although the areas would appear to be proportional to the two-thirds power of tanker size, expressed in dwt, and the length between perpendiculars would appear to be proportional to the one-third power, the superstructure, freeboard, and length do not increase by quite these proportions. As tankers grow in size, draft and beam are usually increased out of proportion. In general, the forces and moments due to wind increase roughly in proportion to the square root of tanker size.

The configurations of the vessel's hull and superstructure may influence the forces and moments. There are pronounced differences in the coefficients for mid-ship-house and bridge-aft tankers. The presence or absence of a forecastle, large differences in bow shapes, and large differences in superstructure shapes can also affect wind forces and moments. These influences are discussed further in subsection 2.5.3.

2.3.4 Current Loads On The Vessel

Current forces and moments on typical tanker hulls have been determined by a number of model tests. The recently published OCIMF report includes current load data for VLCCs, based on research conducted by ER&E for OCIMF under contract. (OCIMF, 1977, Benham, Fang, and Fetters, 1977).

Loads due to current flowing at a given velocity past the stationary hull of a vessel at a low water-depth to draft ratio will be higher than loads imposed on the hull as it moves at the same velocity though still water over the bottom of the body of water. Therefore, coefficients of drag measured on a moving vessel should not be used to represent current drag in shallow water. The loads in either case will be influenced by the magnitude of the water-depth to draft ratio. Only at water-depth to draft ratios greater than about 1.5 are the forces and moments in the two cases essentially the same. SPM terminal sites usually have water-depth to draft ratios of less than 1.5 for large vessels in loaded condition. Therefore, generally only data for the case of water flowing past the stationary vessel should be considered in deepwater port SPM studies.

In shallow water the predominent effects of current on the vessel are form drag and a difference in water level across the hull due to blockage. These gravity effects are realistically modeled by Froude's law. Viscous drag contributes only about 5% to the total force on the vessel when broadside to the current, but may contribute 70% of the force when the vessel is bow-on to the current. Adjustments are sometimes made in interpreting current-force model-test data to account for this inadequate modeling of viscous drag.

In model testing for stationary current-induced forces and moments, the hull dimensions and the velocity of current flow are modeled according to Froude's law. The vessel is constrained at various angles of current attack and the drag and lift forces and the moment due to the current are measured in the horizontal plane. Non-dimensional current force and moment coefficients are calculated from the measured results for the various angles. The most common method of defining these coefficients is as follows:

C_{Xc} = Longitudinal current-force coefficient

C_{Yc} = Lateral current-force coefficient

C_{XYc} = Yaw current-moment coefficient

Typical plots of C_{Xc}, C_{Yc}, and C_{XYc} as a function of angle and the water depth to draft ratio are given in Figures 2-6, 2-7, and 2-8.

The longitudinal current force, lateral current force, and current yaw moment acting on the vessel at any heading to the current can be calculated using the following equations:

$$F_{Xc} = C_{Xc} \frac{k}{2} \rho w \; V_c^2 \; D \; L_{BP} \tag{2-6}$$

$$F_{Yc} = C_{Yc} \frac{k}{2} \rho w \; V_c^2 \; D \; L_{BP} \tag{2-7}$$

$$M_{XYc} = C_{XYc} \frac{k}{2} \rho w \; V_c^2 \; D \; L_{BP}^2 \tag{2-8}$$

where

F_{Xc} = Longitudinal current force

F_{Yc} = Lateral current force

M_{XYc} = Current yaw moment

ρ_w = Density of sea water

V_c = Current velocity

D = Draft of vessel

L_{BP} = Length between perpendiculars of vessel

k = Conversion factor for units

As expressed earlier in the discussion of the wind load coefficients and equations (subsection 2.3.1), attention must be paid to the basis, form, and units of current load coefficients and equations. Like the equations for wind loads, those for current loads can take several different forms.

2.3.5 Influences On Current Loads

As with wind, current loads increase proportional to the square of the velocity of flow. Current loads will decrease with a decrease in draft. For a fixed draft, current loads will decrease with an increase in water-depth to draft ratio. This effect is very important up to a depth to draft ratio of about 1.5 and is less pronounced beyond that ratio.

As can be seen from the equations, current forces increase proportional to the product of the draft and the length between perpendiculars, and the moment increase by the product of the draft and the square of this length. Very roughly these proportions increase by the two-thirds power of vessel size expressed in dwt, however, because both the draft and the beam are easily defined variables, the effect of a change in tanker size on current loads is best worked out on the basis of the dimensions. The form of the bow of the hull, especially the presence or absence of a bulbous bow, can have significant effects on current moments. These effects are discussed in subsection 2.5.3.

2.4 RESPONSE TO WAVES, WIND, AND CURRENT

The motions of a vessel moored to an SPM in waves, wind, and current can be very complex. Many dynamic factors influence the response of the system. The mooring system is non-linear. The moored vessel is free to respond in three degrees of horizontal freedom; surge, sway, and yaw. Including the vertical modes of vessel response and all modes of SPM buoy response, there are a total of twelve degrees of freedom in the system.

In irregular waves the response of the tanker is caused by the variation in the wave energy with time. The wave induced motions of the tanker are easily understood. It is surprising to see large dynamic response of a tanker moored in constant wind or current alone. However, these responses observed in steady wind or current can be explained by analyzing the variations of moments and forces which occur as the tanker hull changes its heading to the direction of the steady wind or current field.

2.4.1 Vessel Response In Wind or Current

Until recently it had been assumed that wind or current acting on a vessel moored at an SPM produced an essentially constant force on the vessel, and that the wind or current flowing past the vessel would tend to stablize the weather-vaning action of the moored vessel. Yaw and sway of the vessel were believed to be caused primarily by wave action. When during model tests, the moored vessel was observed to yaw due to wind or current alone, this was attributed to vortex shedding which would not occur on the prototype.

Now a more complete understanding of the effects of wind and current has been reached through theoretical analysis, computer simulation, and model testing. It has been discovered that sustained combined yaw and sway motions can be caused by alternating wind or current induced lift forces acting on the vessel hull in resonance with the natural periods of the mooring. This phenomena can be explained, by referring to Figure 2-9, as follows.

The vessel may start in position 1, with a slack bow hawser and a slight yaw to the direction of wind. In this position the vessel is unsymetrical to the flow of air, and the hull acts like an airfoil or wing in the wind field. The lift created as air flows around the hull causes the hull to sway to starboard. The vessel will continue to sway until the bow hawser becomes taut.

As the tension in the bow hawser tightens, it retards the motion of the bow to starboard. The vessel then begins to yaw to port about its bow due to its inertia of movement as shown in the second position. The elastic bow hawser is tensioned and then relaxed as the vessel continues to yaw.

The rebound of the bow hawser pulls the vessel forward toward the mooring buoy and the bow is again unrestrained. The angle of the vessel hull to the wind field has now reversed, and the flow of air around the hull now lifts the hull to port, as shown in position 3.

The vessel will sway through the center position to the port side of the mooring until the bow hawser again becomes taut as shown in position 4. At this point, the bow is restrained and the inertia of sway is again transferred to yaw, causing the vessel to yaw to starboard. Once the vessel has yawed, reversing the angle of the wind lift on the hull, it will again sway to starboard, thus completing the cycle.

Substaining the action depends on the vessel yawing far enough while the bow hawser is taut to reverse the angle of the hull to the wind, and then on the vessel swaying to the other side of the mooring before the bow hawser becomes taut again. The action thus depends on proper phasing of the periods of surge, sway, and yaw of the vessel on the mooring. Under the proper combination of wind (or current) velocity, vessel size, mass and freeboard, bow hawser length and mooring system elasticity, the action will grow to a certain amplitude and continue indefinitely. (Wichers, 1976; Flory and Poranski, 1977; Muga and Freeman 1977).

2.4.2 Vessel Response In Waves

The vessel moored in waves alone tends to align itself into the waves and respond primarily in surge. In regular waves, the vessel will surge back on the mooring until the force in the hawser equals the mean wave-induced force on the vessel and will remain essentially in equilibrium at this position unless the waves are in resonance with a natural frequency or a harmonic of the mooring. The natural period in surge is normally far removed from any regular wave period. The moored vessel responds an impreceivable amount in surge to regular waves having periods less than about 30 seconds. Variations in mooring load records at the natural period of short-period regular waves are due to vessel heave, pitch and roll, and due to buoy motions.

In irregular waves, the mean wave force varies with time. Although the moored vessel does not respond in surge to each individual wave, it may respond to variations in the mean wave force. Long-period swell or seiche can cause long-period excitation of the moored vessel. Variations in mean wave force may also be caused by wave grouping. A group of low waves followed by a group of high waves will produce a variation in mean wave force. Wave grouping is discussed in Appendix A. The position in surge of the vessel on the mooring, and thus the hawser load, will change in response to the magnitudes of groups of waves. The response will be especially pronounced if the period of wave grouping or of seiche or swell corresponds to the natural period in surge of the mooring system.

Considerable attention is now being directed to wave grouping and its possible effects on mooring systems. (Remery and Hermans, 1971; Remery and van Oortmerssen, 1973). It is not known how common periodic wave grouping may be at typical mooring sites. As explained in subsection 2.2.4, if wave grouping is thought to occur at the site then it should be included in model testing and analysis of the mooring.

The moored vessel may experience a low-frequency subharmonic response in waves at a non-linear mooring system. This low-frequency subharmonic response is a property of the dynamics of a non-linear elastic system. Even with a linear mooring system the vessel may experience low-frequency response in waves due to non-linear coupling between the response modes. These subharmonic response phenomena are independent of the effects of wave grouping, seiche, and swell. Only limited analysis of these subharmonic resonances at mooring systems have been conducted to date. (Wilson and Awadalla, 1973).

2.4.3 Vessel Response In Waves, Wind, and Current

The response of a tanker moored to an SPM in an environment composed of waves, wind, and current can be rather complex, especially if the three environmental components are not colinear. Even if they are colinear, unexpected things can happen.

In some model tests with a constant wind or current acting colinear with irregular waves, the wind or current have been observed to have a steading effect on the response of the tanker. The peak mooring loads measured in some such tests have actually been lower than the peak mooring loads measured in the same wave environment without wind and current. In other tests with waves, wind and current in line, peak mooring loads much higher than those measured in waves alone have been measured. Sometimes these peak loads have been higher than would be predicted by superimposing the static wind or current force on the peak wave induced loads. One explanation for this is that the vessel is excited in yaw and sway by the wind and current induced moments and forces as explained in subsection 2.4.1. Not only do the wind or current-induced sway and yaw motions impose loads on the hawser, but also the yaw action increases wave induced loads.

When waves, wind, and current are not colinear, the response of the vessel is even more difficult to describe and explain. Figure 2-9 shows a mooring load record from a typical model test conducted with a ballasted 350,000 dwt tanker moored in 4.4 m (15 ft) waves with a 45 knot wind at 45^{o} to the waves and a 1 knot current at 90^{o} to the waves. Note the relative magnitude of the high-frequency variation in amplitude to the low-frequency variation in amplitude. The high-frequency load variation corresponds to the wave-induced buoy motions and the pitch, roll, and heave of the moored vessel. These high-frequency load variations are not negligible; they add to the magnitude of the peak mooring loads. However, most of the magnitude of the peak mooring loads are due to the slowly varying low-frequency motions of the moored vessel.

Figure 2-11 shows a continuation of the mooring load record in Figure 2-10. The 34 minute duration of the first record corresponds to the duration of a typical model test. The test recorded in the second figure consisted of a continuation of the first test in which the irregular waves were cycled over and over for a total of six cycles. The high-frequency component has been filtered from the record of the last five cycles.

The vessel motion observed in this test was very irregular. During most of the test, the vessel swayed and yawed in response to the wind and current and surged in response to waves. The vessel's bow sometimes traced a figure eight pattern, with peak loads being experienced at the top and bottom of the figure eight. At other times, particularly during the last cycle of the test the vessel motions appeared to damp out and the vessel responded very little.

This example test record shows the complex nature of the vessel response in waves, wind, and current, and shows the vessel motion will not always be the same even in response to the same wave, wind, and current environment. It is important that realistic wave, wind, and current environments be used in model testing and analysis. As explained in Appendix A, it is not so important that any particular wave spectrum be used, but that a realistic wave spectrum be used, and that long period phenomena such as seich or wave grouping also be reflected in the wave record if they exist in nature at the site. As explained in subsecton 2.3.2, it does not appear to be important that wind velocity be varied unless wind is the predominent forcing function. It is important, however, that the wind and current be accurately represented as to direction, velocity, and their moment producing effects on the vessel. Specific guidelines for producing wave, wind, and current effects in model tests are given in Section 3.

2.5 INFLUENCE OF THE TANKER

Variations in tanker size or shape do not affect mooring loads at SPMs as much as do variations in the environment. Nevertheless, where a large range of tanker sizes are to moor at a SPM, consideration must be given to the influence of tanker size on mooring loads. The larger tankers do not always impose the higher mooring loads.

The state of tanker loading, that is whether the tanker is loaded or light ballasted, can have a major influence on tanker response, and thus on mooring loads at SPMs. Consideration must be given to the loading condition in which tankers will most probably be exposed to severe environments.

2.5.1 Tanker Size

Tanker size will be refered to throughout this report in terms of dead weight tonnage (dwt). Dead weight tonnage represents not only the weight of cargo which can be carried by a vessel, but also the weight of all fuel, water, stores, and supplies which the vessel carries. For very approximate reference dwt is about ten percent greater than the weight of cargo, and about ten percent less than the total weight of water displaced by the vessel when loaded. Exact cargo capacity, total displacement, and dimensions for tankers can be found in Clarkson's Register. The figures for tanker size used in this report will be in dwt. However, as only trends with tanker size are discussed, the principals generally apply to other means of defining tanker size.

In conjunction with the rules of thumb relating mooring loads to wave height and wave period given in subsections 2.2.2 and 2.2.3, another rule of thumb has also been found through the analysis of model test data. In general the mooring forces will increase proportional to the square root of tanker size. The following general relationship then appears to relate wave height, wave period, and tanker size to mooring loads:

$$F_s \approx k \frac{H_s^2 \sqrt{D}}{\sqrt{T_s}} \qquad (2\text{-}9)$$

where D = tanker size (dwt) and F_s, H_s, T_s and k are defined as before. (Flory and Poranski, 1977).

Above a certain range of tanker sizes the above relationship has generally been found to be overly conservative. Mooring loads measured in model tests for tankers of 500,000 dwt and above have generally been no higher than, and in some cases have been lower than, those measured on tankers in the size range of 200,000 to 400,000 dwt. The 200,000 to 400,000 dwt tankers moved around more in response to the environment than the largest tankers. The situation is analogous to resonance response, although the periods of response seem to be much longer than the periods of the excitation. A mooring system which is optimum for a specific tanker size and environment may not be optimum for some other tanker size. Based on the above observations, not only the largest sized tanker, but also one or several smaller representative sized tankers should be considered in designing an SPM.

Another consideration is the probable strength of the mooring fittings on-board various sized tankers as discussed in subsection 4.4. A small tanker will probably not impose loads as high as those imposed by intermediate or large tankers, but its mooring fittings may not be as strong and thus, be inadequate for the mooring loads. Therefore, mooring loads on small tankers cannot be disregarded.

2.5.2 Tanker Loading Condition

For a given tanker size, different mooring loads will probably be experienced in the loaded condition than in the ballasted condition. Therefore, both the loaded and the ballasted tanker must be considered.

In most model tests in which both a loaded and a ballasted tanker of the same size have been moored to a SPM, the loaded tanker has been observed to respond less to the environment than the ballasted tanker. This appears to be a situation of resonance with the forcing function similar to that noted above in relation to tanker size. The significant forces measured with the loaded tanker may be equal to or higher than those measured with the ballasted tanker. However, because the ballasted tanker moves more at the mooring, it imposes higher peak mooring forces. The ratio of maximum to significant mooring forces to be used in design, as explained in subsection 3.11.2, are generally higher for the ballasted tanker than for the loaded tanker. Therefore, for moderate to long mooring durations, high loads may be more probable with a ballasted tanker than with a loaded tanker.

The more critical case of ballasted or loaded tanker cannot be assumed, therefore, both cases must be considered. There may be cases in which the loaded tanker exerts the higher loads on the mooring, depending on the tanker size, the mooring system, and the environment. It is not probable that an intermediate condition of tanker loading will exert loads significantly higher than the loaded or ballasted condition.

In determining the more critical condition, loaded or ballasted, the mode of operation of the terminal should be considered. Because of limitations on the environment in which launches can operate, tankers would generally not moor to an SPM in greater than about 2 m (6 ft) waves. Once moored, tankers may remain moored in much higher waves, possibly as high as 4.5 m (15 ft). If a tanker is to discharge cargo at the terminal, it is unlikely to experience high waves soon after mooring, when it is still nearly loaded. However, it is apt to remain moored in high waves in the nearly ballasted condition to complete discharge. Conversely, at a loading terminal the tanker is more apt to experience high waves when it is nearly loaded. Therefore, consideration of loaded tankers is more important at loading terminals and consideration of ballasted tankers is more important at discharge terminals.

2.5.3 Other Tanker Parameters

Tanker design parameters such as hull form and superstructure shape are generally not nearly as important as tanker size and loading condition as they affect mooring forces at SPMs. Two hull parameters which may have some influence are the length-to-beam ratio and the shape of the bow.

Wide-beam tankers are designed with relatively shallow loaded drafts in relation to their size, thus enabling them to enter shallower ports than conventional tankers of the same size. The volume lost in draft is made up primarily in beam. Such wide-beam tankers may typically have length-to-beam ratios of about 5 as compared to length-to-beam ratios of about 6.5 for conventional tankers. (In this ratio length refers to the length between perpendiculars).

Wide-beam tankers will experience greater loads than conventional tankers in near bow-on waves. This in turn will result in higher mooring forces. The magnitude of this effect is not known, but is believed to be small, or negligible, especially in conditions where conditions of cross wind or cross current will cause the tanker to lay at an angle to the waves.

Most large tankers have special bulbous bows which are designed to reduce wave-making resistance when the tanker is underway. There are many different bulbous-bow shapes, but the effect on mooring loads is not believed to be significantly influenced by bulbous-bow shape.

Some smaller tankers do not have bulbous bows, but have more conventional ship-shaped bows. Some new very large tankers have a new bow shape called a cylindrical bow, which has a rounded shape that varies little with depth and lacks a bulb-shaped projection. Current and wave loads on conventional-bow hulls and cylindrical-bow hulls are different than on bulbous-bow hulls. The magnitude of the change in effect due to waves is believed to be negligible. The magnitude of the change in effect due to current is generally small, however, where current is an important environmental factor, and especially where underkeel clearance is small and loaded tankers are the more important consideration, the effect of hull form should not be ignored.

Wind loads on the tanker hull are also influenced by hull form. Bow-on wind forces on wide-beam tankers will be higher than those on conventional tankers of the same displacement, but this influence would not generally be significant. Wind moments on cylindrical-bow hulls in the ballasted condition can be substantially higher than on conventional-bow hulls and bulbous-bow hulls. In circumstances where wind is an important environmental factor and ballasted tankers are the more important consideration, the effect of wind on the cylindrical-bow hull tankers should be considered if they are believed to constitute a significant portion of the tanker fleet to be moored. (OCIMF, 1977).

The arrangment and shape of the superstructure will influence wind loading as discussed in Section 2.3.1. Generally this is not an important consideration. However, there can be a significant difference in wind loading between tankers with midship houses and bridge-aft tankers. Almost all large tankers have bridge-aft designs. However, smaller and older tankers will have midship houses. Where a significant number of small tankers are to be moored and the mooring loads with these tankers is of concern, wind forces and moments due to the mid-ship house should be considered.

2.6 INFLUENCE OF THE MOORING SYSTEM

The design of the mooring system has a major influence on the mooring loads. The most important mooring system parameters are those which affect the elasticity of the mooring sytem. A very stiff mooring system will severely constrain the response of the tanker to waves and will result in very high mooring loads. Conversely, if the mooring system is very soft, the tanker may respond too freely, building up momentum as it moves under the influence of waves, wind, and current, and exerting high loads on the mooring as it comes to the limits of mooring system elasticity. The energy theory, which relates the energy stored in the mooring to the mooring loads, provides a tool for understanding the influence of mooring system elasticity.

2.6.1 Mooring System Elasticity

Analysis of results of model tests on various SPMs has shown that the elasticity of the mooring system must be within certain limits to provide near optimum mooring performance and minimize mooring loads. The mooring system must be elastic enough to allow the moored vessel to move under the influence of waves and other forces, but must be stiff enough to limit the extent of this motion.

In buoy-type SPMs, elasticity is generally composed of the hawser elasticity and the buoy-anchoring-system elasticity acting in series. In a tower-type SPM, the hawser is usually the only contributor to mooring system elasticity. The elasticity characteristics of synthetic rope hawsers are discussed in subsection 5.6. It is important that the proper hawser elasticity be represented in mooring system analyses. Hawsers which are broken-in or used, that is those which have experienced a number of cycles of loading, are significantly stiffer than new hawsers. The elasticity characteristics of a used synthetic hawser of the proper size, length, and material should be used in analysis.

Two common types of buoy-type SPMs now used at offshore terminals are the SALM (single anchor leg mooring) and the CALM (catenary anchor leg mooring). The SALM, shown schematically in Figure 2-12, consists of a buoy anchored to a base on the sea floor by a taut anchor leg. The SALM acts as an inverted pendelum to provide elasticity. The anchor leg is pretensioned, confining the buoy to move in an arc about the foundation. As the buoy is pulled to the side, the restoring moment exerted by its buoyancy increases.

Ignoring hawser elasticity and angle, and assuming the buoyancy is concentrated at a point, an approximate expression for the horizontal force-deflection relation for the simplified SALM shown in Figure 2-12 is

$$F \approx \frac{DB}{\sqrt{L^2 - D^2}} \qquad (2\text{-}10)$$

where F = the horizontal load on the mooring
D = the horizontal displacement of the buoy
B = the net buoyancy of the buoy
L = the length of the anchor leg

In an actual SALM, the mooring line exerts an upward force on the buoy, and elongation of the mooring line under load changes the geometry of the mooring as the load increases. Furthermore, the buoy is not a point and the mass and buoyancy of the buoy are not concentrated. Exact solution of the load-deflection characteristic of an acutal SALM is complex and requires computer analysis. The above equation is adequate, however, for understanding the parameters which influence mooring elasticity.

The SALM can be made stiffer by increasing the net buoyancy, B, of the buoy. Net buoyancy is defined as the difference between the gross displacement of the submerged buoy and the weight of the buoy and the anchor chain suspended beneath the buoy. Decreasing the length of the anchor leg, L, makes the mooring stiffer. Obviously, L cannot exceed the water depth, but can be made less than the water depth by submerging the buoy or by elevating the anchor point on the mooring base. (Flory, 1971; Flory, Mascenik, and Pedersen, 1972).

A schematic of the CALM system is shown in Figure 2-13. The elasticity of the CALM is the vector summation of the elasticities of the multiple catenary anchor chains. As the buoy deflects to the right, tension in the left anchor chain increases and more anchor chain is lifted from the sea floor. The tension in the anchor chain to the right decreases. For a given deflection of the buoy, the horizontal tension in all anchor chains must be added vectorially to determine the resultant force on the buoy. For a given horizontal hawser force, the buoy will displace until the resultant anchor chain forces are in equilibrium with this horizontal force.

Parameters which affect the elasticity of the CALM buoy anchoring system are the unit weight of the anchor chains, the number of anchor chains, the water depth, and the pretension in the anchor chains. Increasing the unit weight of the anchor chain, that is employing a heavier anchor chain, will increase the mooring system stiffness. Increasing the number of anchor chains has the same affect. Increasing the pretension in the anchor chain will increase the stiffness and will also decrease the amount of deflection which can be applied before the slope of the elasticity curve approaches a vertical asymptote. An increase in water depth will result in a softer elasticity curve. Buoy size has almost no influence on the elasticity of the CALM provided the buoy has sufficient buoyancy so it does not submerge under high mooring loads. (Flory, 1971; Maari, 1975).

2.6.2 The Energy Theory

From the analysis of data from many model tests, ER&E has developed a theory relating the significant mooring load to the energy stored in the mooring system. The principals of this theory are discussed in Appendix B. The theory is presented in this report not as a required or recommended method of predicting mooring loads, but as a tool to understanding certain interrelationships of the parameters which affect mooring loads at SPMs.

In brief, the energy theory relates the significant mooring load to the significant energy, that is the energy which is stored in the mooring system up to the significant mooring load. The significant mooring load is the average of the highest one-third peak mooring loads. By the theory, the significant energies determined at SPMs having different mooring elasticities will be the same for the same tanker and wave height. However, the energy-storing capabilities of different mooring systems are functions of the shape of the load-deflection curves for those mooring systems. If two mooring systems have different load-deflection curves, the significant mooring loads may be different at the two systems for the same tanker moored in the same environment.

Comparing the load-deflection characteristics of two SPM systems, SPM A and SPM B shown in Figure 2-14, the same amount of energy is stored in each system as represented by the areas under the curve. SPM A, which is stiffer than SPM B, requires a higher load to store the energy, and will, therefore, experience higher significant mooring loads. (Maddox, 1972; Flory and Poranski, 1977).

The energy theory provides some insight into the manner in which mooring forces vary with mooring system elasticity, wave height, and tanker size. The energy stored in the mooring system up to the significant mooring load will increase with an increase in tanker size or wave height. At different but generally similar mooring systems the increase in significant energy will be essentially the same for the same increase in tanker size or wave height. Thus, mooring loads will increase more in a mooring system having a steep load-deflection curve at the point of significant mooring load than at a mooring system having a more gradual slope. Also, the increase in significant mooring load in a mooring having a very non-linear load-deflection curve will not be proportional to the increase in a mooring system having a more linear load-deflection curve.

The characteristics of the load-deflection curve should be considered in determining mooring loads, especially in interpolating or extrapolating mooring loads for one mooring system based on those for another mooring system. The energy theory provides a tool for relating the effects of changes in tanker size and wave height to the characteristics of the mooring-system elasticity.

2.6.3 Other Mooring System Effects

The mooring system elasticity can have other effects on mooring loads than those demonstrated by the energy theory. A very soft mooring system, that is one which has a very gradual slope, can produce high mooring loads, especially if the system becomes stiff at relatively high loads. On a very soft mooring system the tanker may move about freely with little constraint under the influence of waves, wind, and current. Unrestrained movement of the tanker can result in large buildups of inertia, and high loads out of proportion to the environmental forces can result when the momentum of the tanker is converted to energy stored in the mooring system. Very non-linear mooring-system elasticity curves which permit large deflections before moderate loads are exerted and then rapidly become stiffer so that high loads are exerted by additional deflection may produce much higher loads than more linear mooring-system elasticity curves.

The length of the hawser can have a similar influence on mooring loads. A very long hawser can permit the moored tanker to move freely about the mooring with very little constraint. If the tanker moves forward on the mooring and then moves back under the influence of the environment, it can acquire considerable momentum before being constrained by a long hawser. However, a very short hawser will produce a very stiff mooring system. A rough rule of thumb is the hawser length should be approximately equal to the beam of the moored tanker.

The hawser elasticity can also affect the mooring loads imposed by the motion of the mooring buoy in response to waves. Buoy size and mass will also have an effect on these loads. As shown in Figure 2-7 and explained in subsection 2.4.3, the loads induced by buoy motions are generally lower in magnitude than those induced by tanker motions. However, these buoy induced loads do contribute to the peak mooring loads. Buoy-induced mooring loads will be higher for a large or heavy mooring buoy, either on the SALM or the CALM. Buoy induced mooring loads will be higher for a stiff hawser than for a soft hawser.

The degree of non-linearity of the mooring system will influence the subharmonic response of the moored vessel as explained briefly in subsection 2.4.2. A change in the elasticity characteristics of the SPM may change both the frequency and amplitude of this subharmonic response and thus influence mooring loads.

2.7 SUMMARY

There are many factors which influence mooring loads at SPMs. Wave height is generally the most important single parameter. As a rough rule of thumb, mooring loads vary in proportion to the square of wave height and to the inverse of the square root of wave period.

Wind and current effects can cause the vessel to move about on the mooring and can thus result in dynamic loads much higher than the static wind and current forces. Mooring analyses should account for this dynamic effect as well as the relative directions of waves, wind, and current.

Mooring loads vary approximately in proportion to the square root of tanker size. However, higher mooring loads may be experienced with intermediate-size tankers than with very-large tankers. The condition of loading of the tanker is important; generally higher peak loads will be experienced with light vessels than with loaded vessels.

The elasticity of the mooring system and the length of the hawser will influence mooring loads. Mooring systems which are very stiff may restrain the tanker such that wave forces will impose very high loads. On very soft mooring systems or systems with very long hawsers, the vessel may respond too freely, acquiring momentum which exerts high loads on the system. A change in mooring system design which affects elasticity may change the mooring loads.

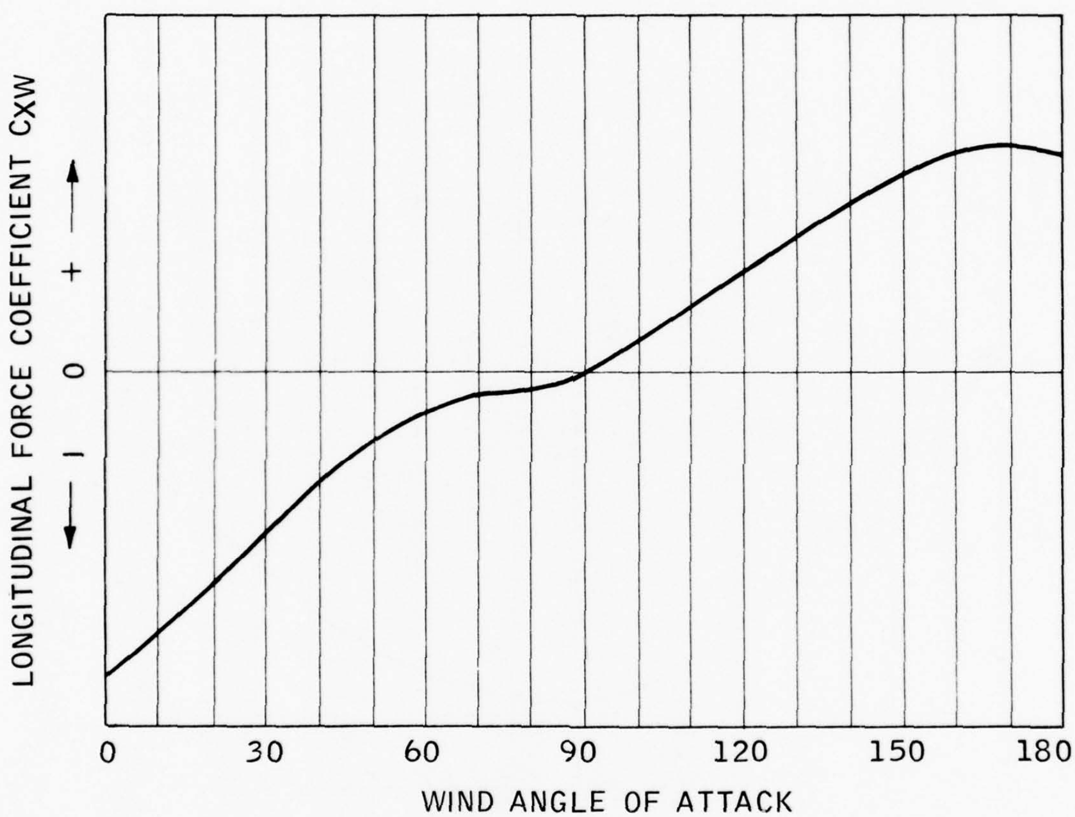


Figure 2-1 - WIND LONGITUDINAL FORCE COEFFICIENT AS FUNCTION OF TANKER ANGLE

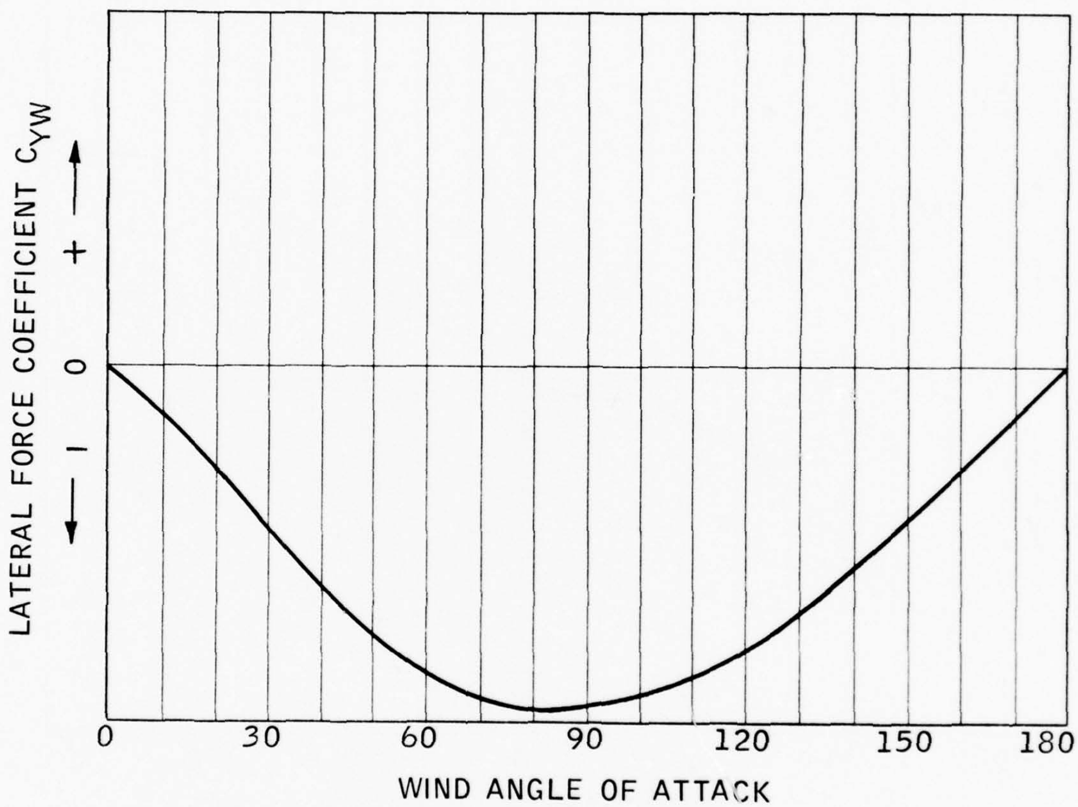


Figure 2-2 WIND LATERAL FORCE COEFFICIENT AS FUNCTION OF TANKER ANGLE

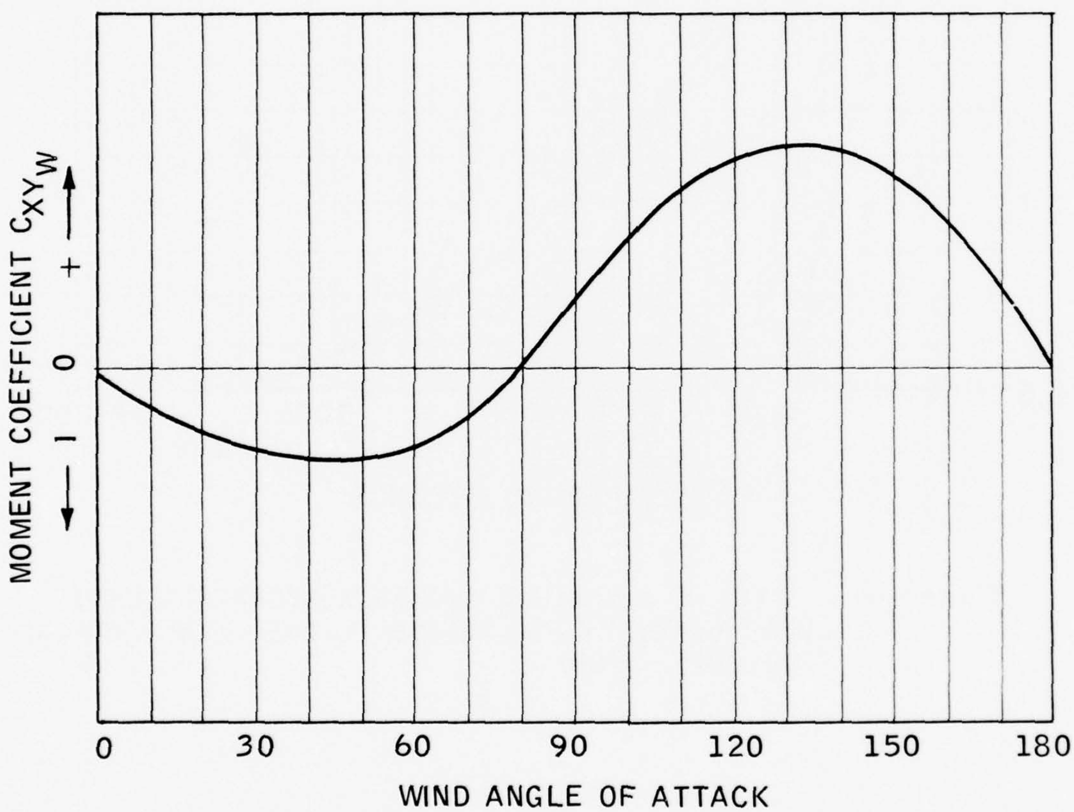


Figure 2-3 - WIND MOMENT COEFFICIENT AS FUNCTION OF TANKER ANGLE

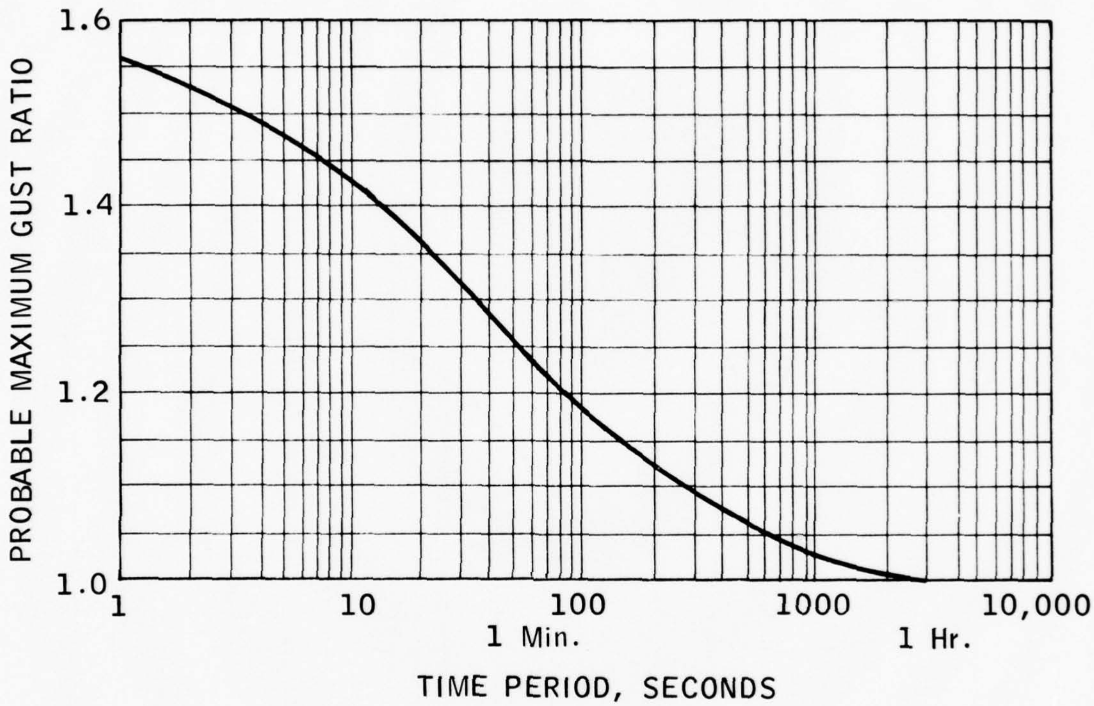


Figure 2-4 - RATIO OF PROBABLE MAXIMUM AVERAGE (GUST) WIND VELOCITY FOR PERIOD TO ONE HOUR AVERAGE (VELLOZI, 1968)

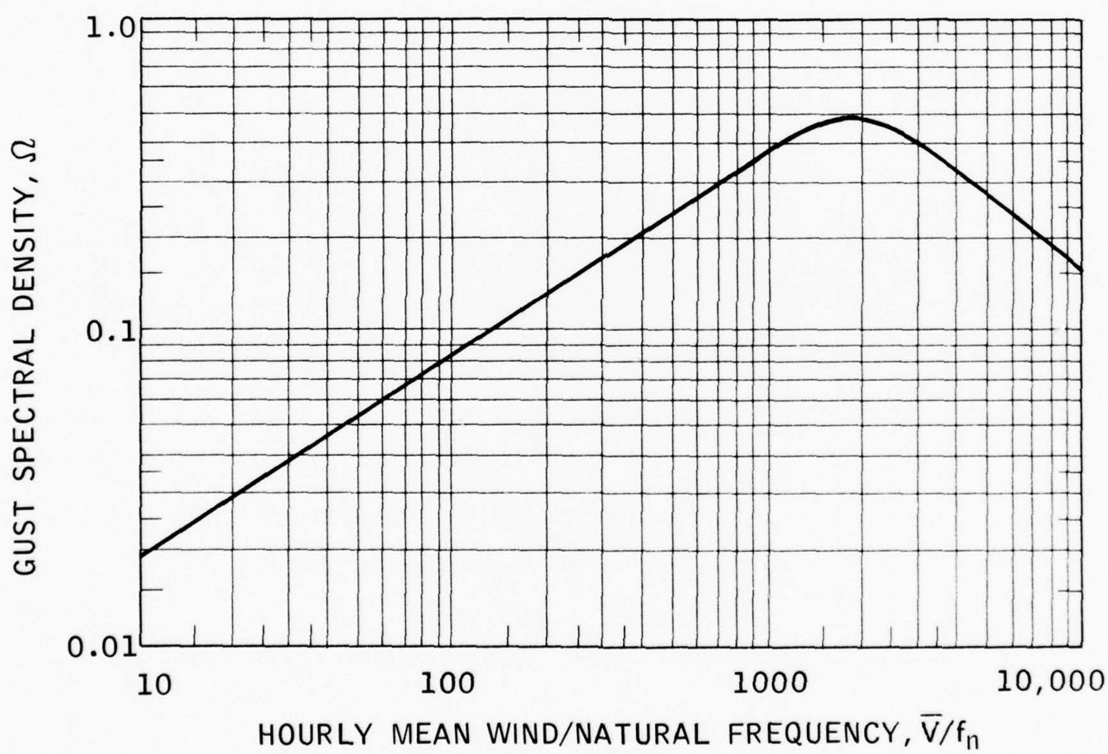


Figure 2-5 - WIND GUST SPECTRAL DENSITY VARIATION
(VELLOZI, 1968)

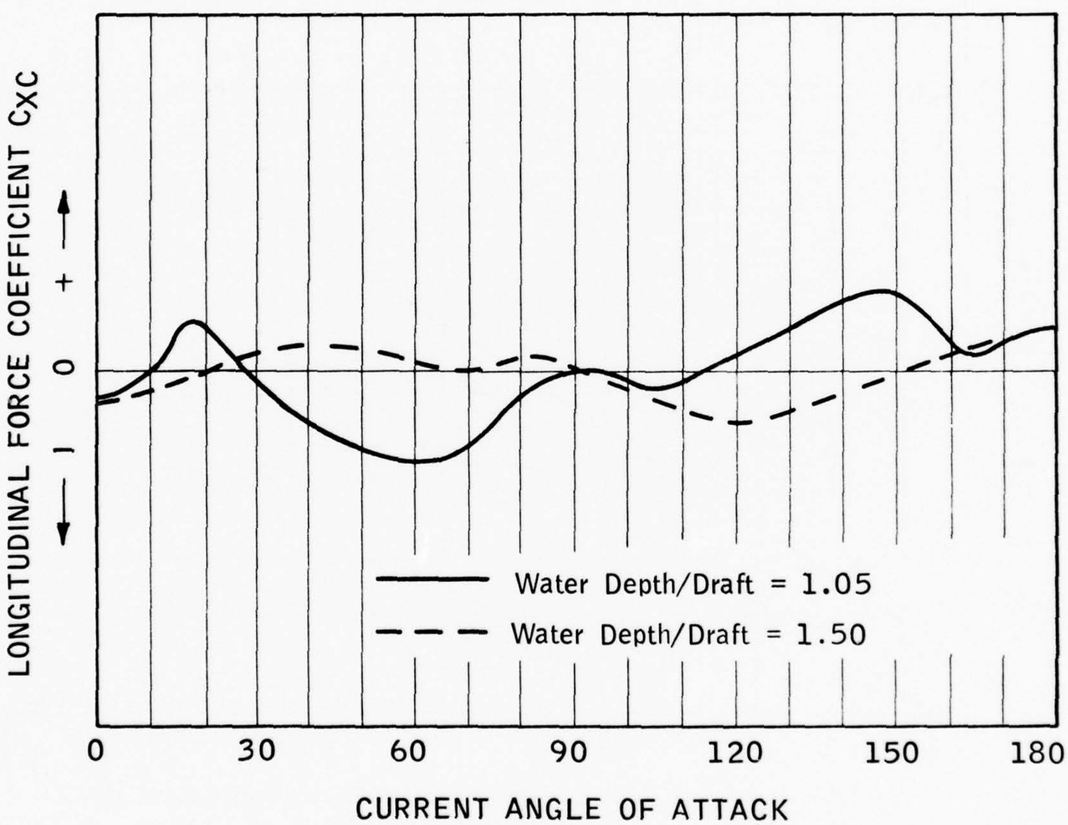


Figure 2-6 - CURRENT LONGITUDINAL FORCE COEFFICIENT AS FUNCTION OF TANKER ANGLE

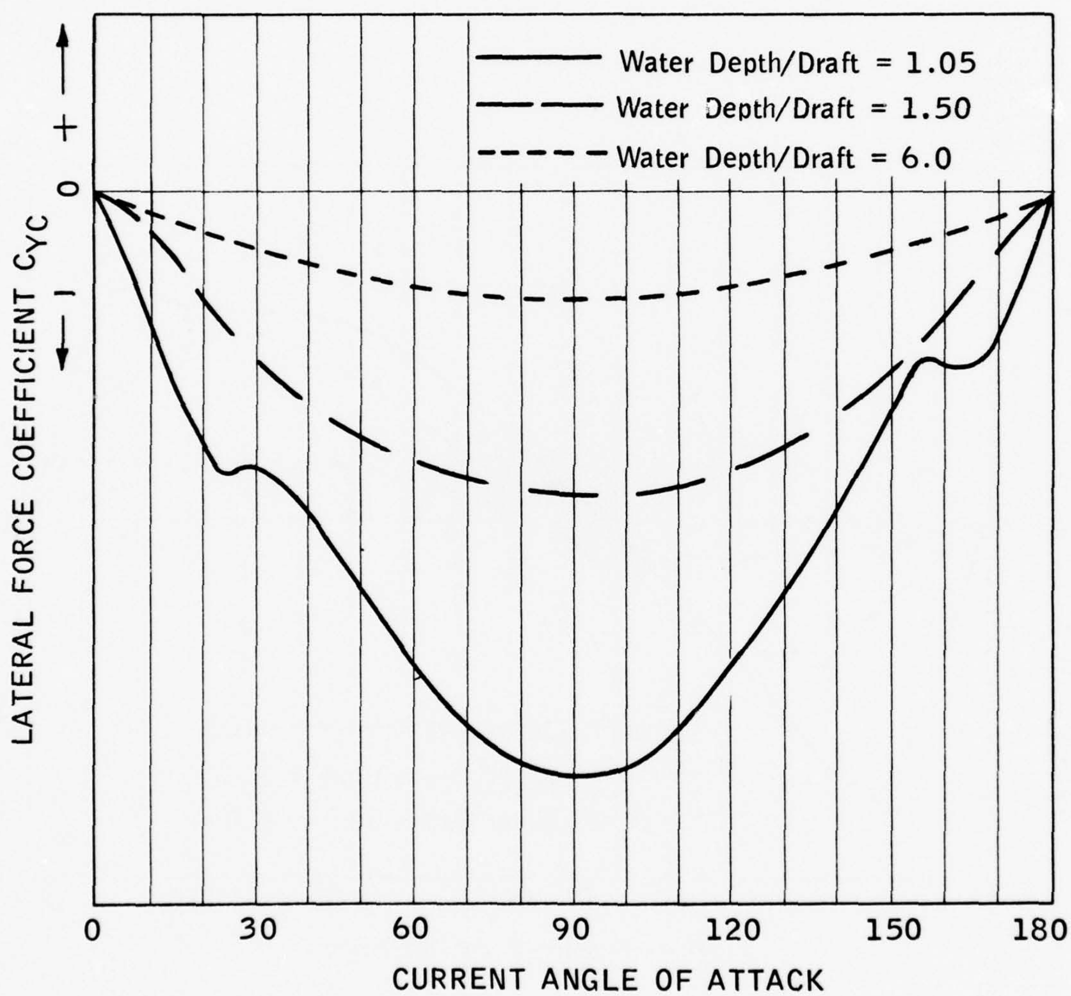


Figure 2-7 - CURRENT LATERAL FORCE COEFFICIENT AS FUNCTION OF TANKER ANGLE

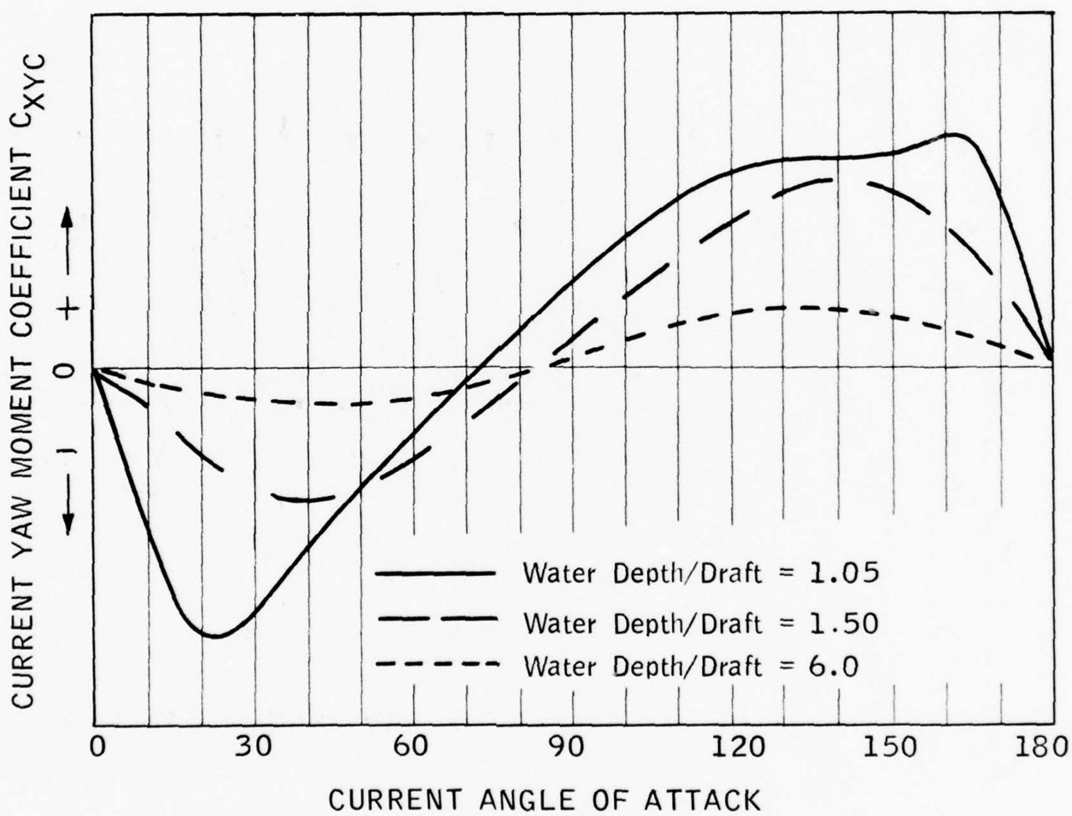


Figure 2-8 - CURRENT YAW MOMENT COEFFICIENT AS FUNCTION OF TANKER ANGLE

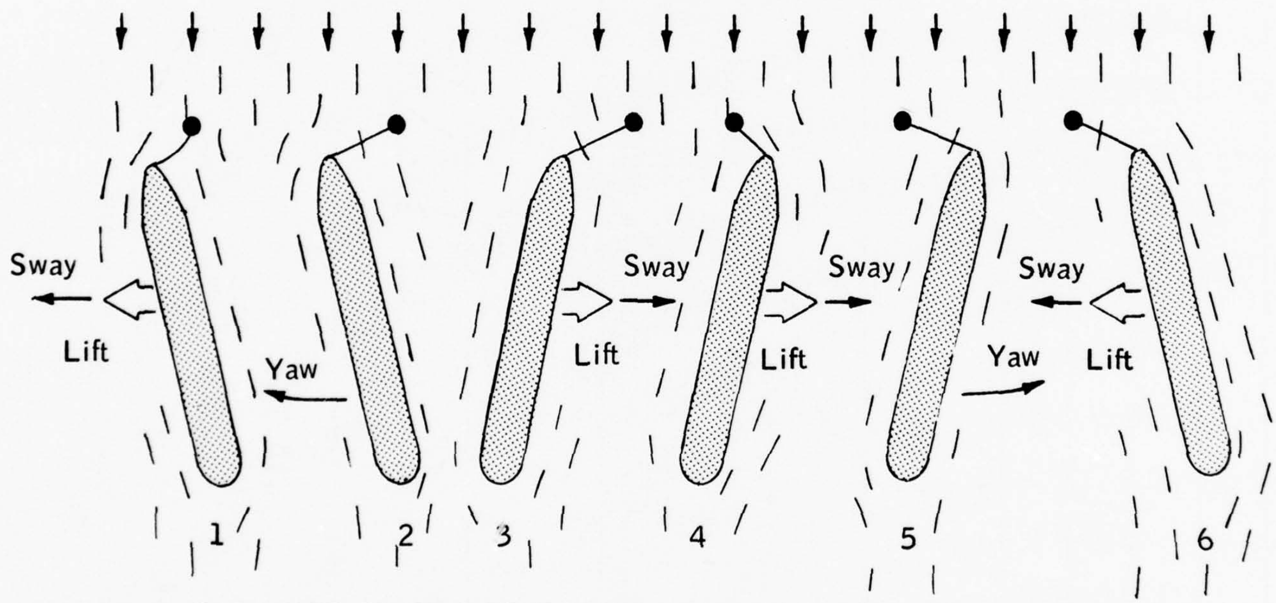


Figure 2-9 - MOTION OF VESSEL DUE TO WIND OR CURRENT AT SPM

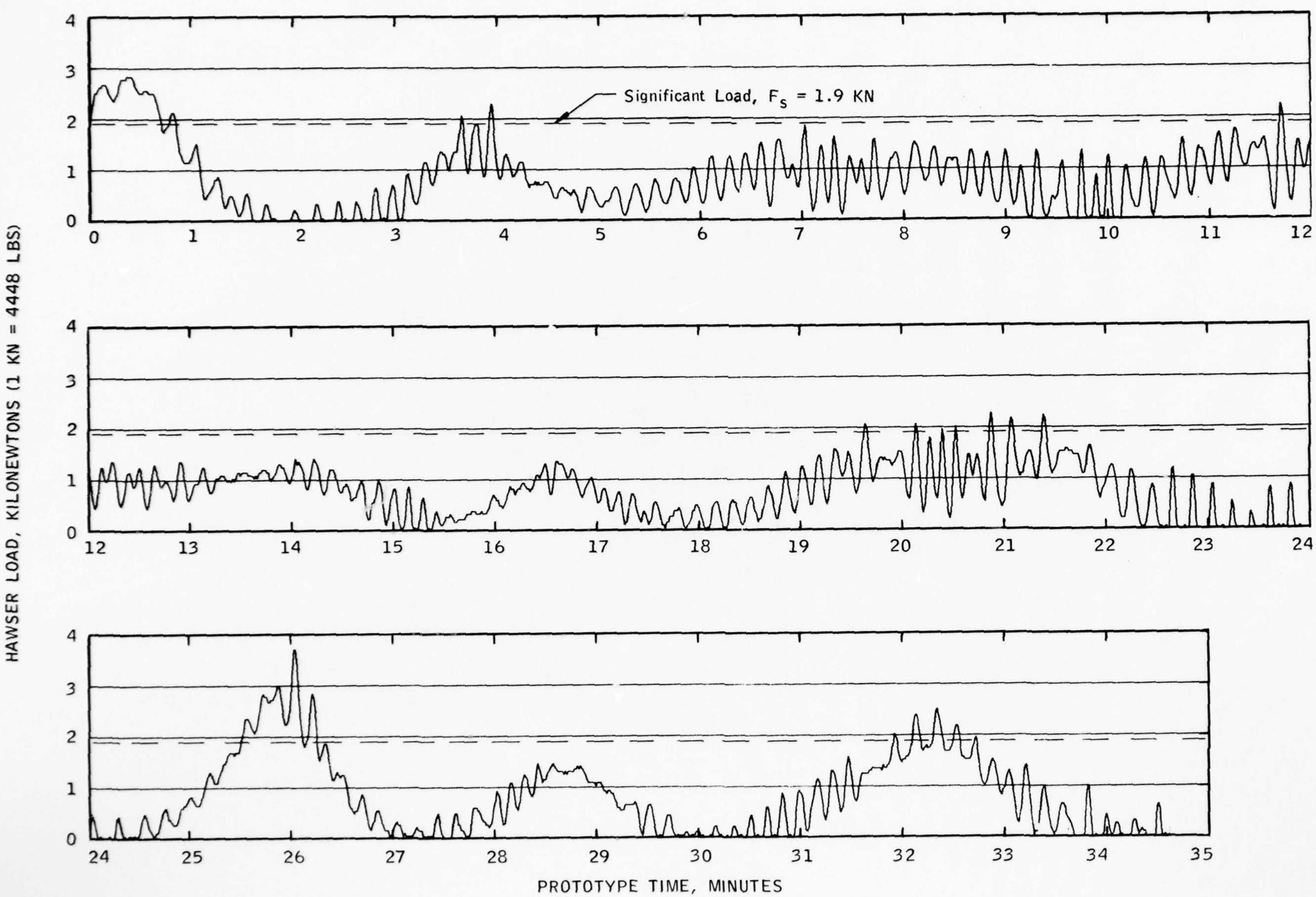


Figure 2-10 - SHORT MOORING-LOAD RECORD FROM MODEL TEST OF TANKER MOORED IN WAVES, WIND, AND CURRENT

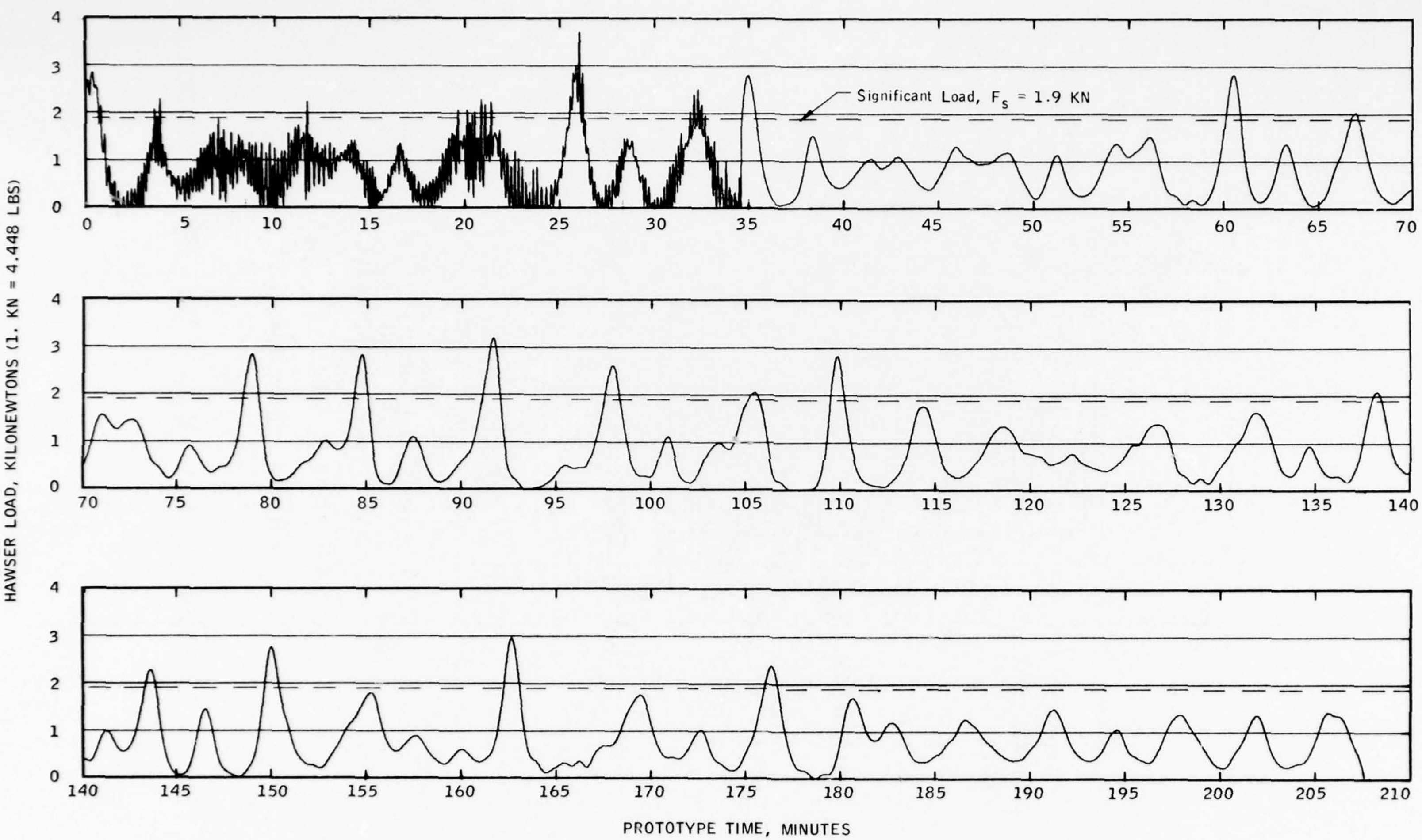
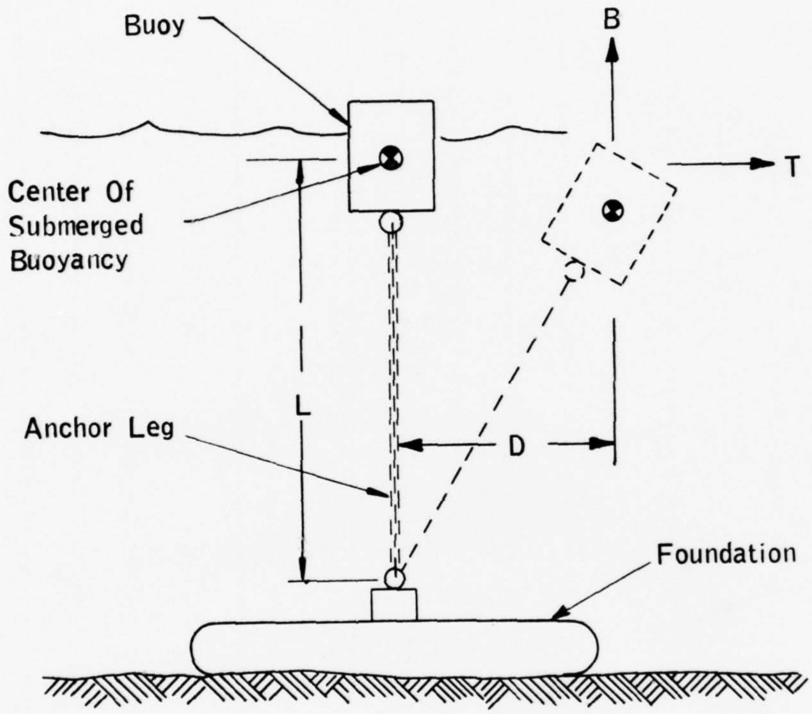


Figure 2-11 - LONG MOORING-LOAD RECORD FROM MODEL TEST OF TANKER MOORED IN WAVES, WIND, AND CURRENT



$$T = \frac{DB}{\sqrt{L^2 - D^2}}$$

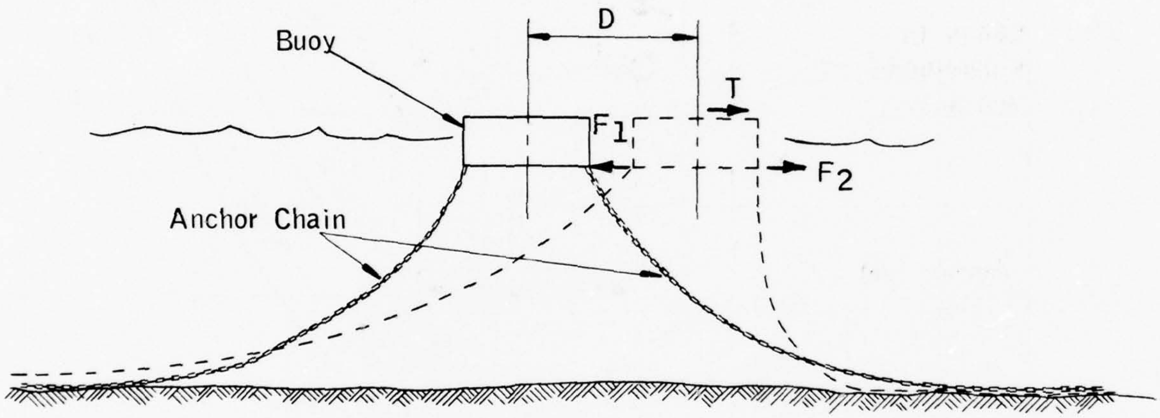
T = Horizontal Force

D = Horizontal Displacement

B = Net Buoyancy

L = Length Of Pendulum

Figure 2-12 - SCHEMATIC OF SINGLE ANCHOR LEG MOORING



T = Horizontal Mooring Force = $F_1 - F_2$

F = Horizontal Anchor Chain Force = WC

W = Unit Weight Of Chain

C = Property Of Shape Of Catenary Formed By Chain

Figure 2-13 - SCHEMATIC OF CATENARY ANCHOR LEG MOORING

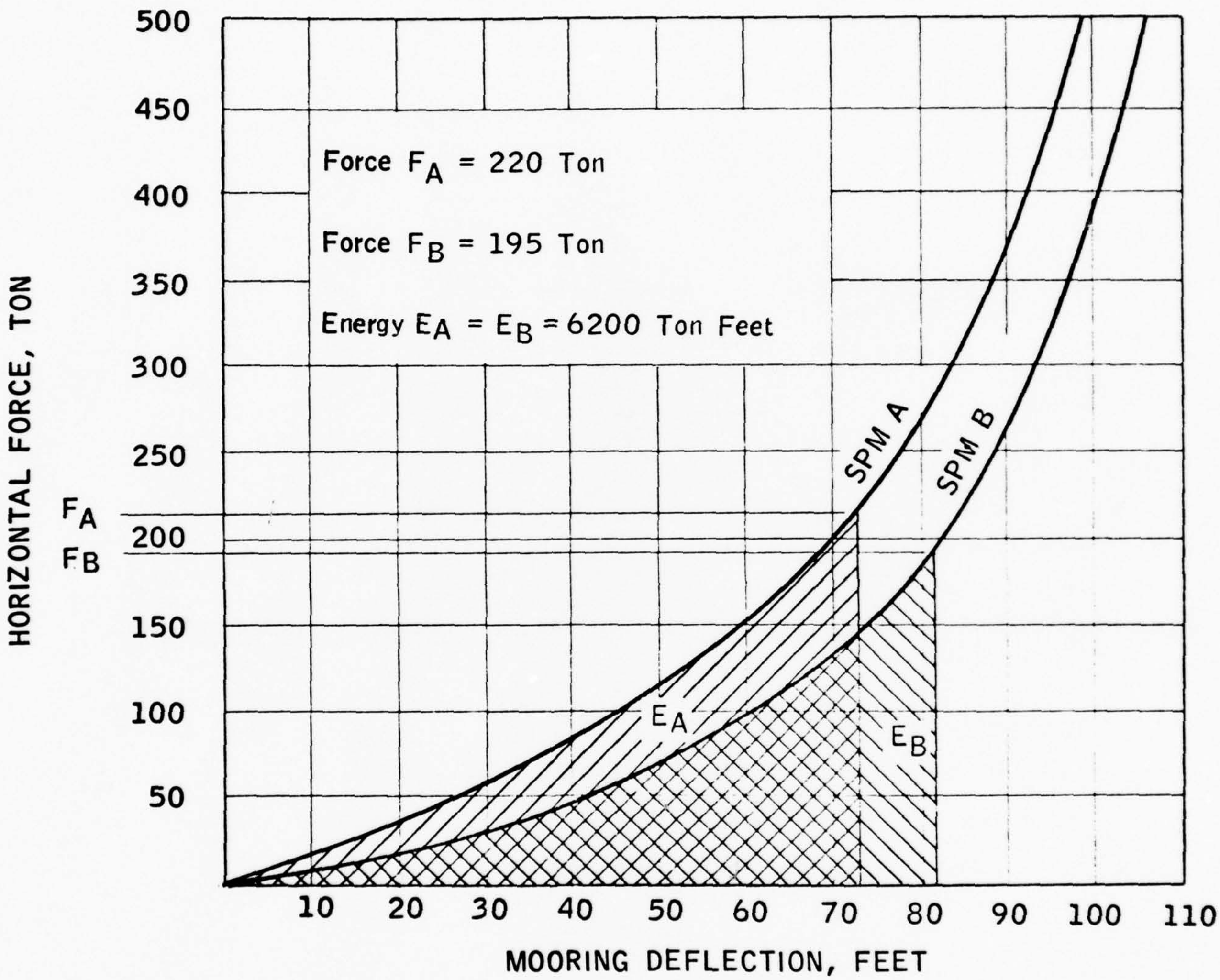


Figure 2-14 -SPM ELASICITY CURVES SHOWING APPLICATION OF ENERGY THEORY

SECTION 3

DETERMINATION OF SPM MOORING LOADS BY MODEL TESTING

3.1 INTRODUCTION

Model tests have been used for many years as a means of studying the behavior of vessels underway in a seaway. Procedures for designing and performing such tests and for analyzing and interpreting the data from such tests are highly developed. A high degree of reliability can be placed on the results of such tests. Although advanced computer programs are now available to analyze many of the moving vessel phenomena, the results of such computer analyses are still usually verified through model testing.

Analyzing the response of a moored vessel is similar to, but in some ways is more complex than, analyzing the response of a moving vessel. The stationary vessel is more responsive to waves, wind, and current. Together with the mooring system, the ship constitutes a many-degree-of-freedom system. The nonlinear characteristics of the mooring system further complicate the problem.

It is not surprising, therefore, that researchers and designers have relied on testing with scaled models to study and design SPM systems. Although less documented and publicized than the model testing of moving tankers, SPM model testing dates back almost 20 years. Shell conducted model tests during the design of their first SPM which was installed at Miri in 1959. Exxon conducted its first model tests of a tanker moored to an SPM in 1959.

This section is intended to serve as a guide for the evaluation of the manner in which SPM model tests are conducted and the manner in which the results are analyzed and interpreted. In general, the model testing and analysis methods presented are believed to be those now recognized as the best obtainable in the present state-of-the-art. Some of the methods presented are new, but are believed to be improvements upon past methods.

3.2 SCOPE OF MODEL TEST PROGRAM

The principal questions to be answered through SPM model tests are:

- Is the system properly designed to minimize mooring loads?
- What is the frequency of occurrence of loads of various magnitudes?
- What are the maximum loads for which the system should be designed?

The model test program must be broad enough in scope to test those combinations of environment and tanker size and condition which are apt to produce the highest loads, and to acquire sufficient data to permit statistical prediction of maximum loads. It may also be used to gather data on other aspects of the SPM design and performance, such as vessel motion and cargo hose loads.

The manners in which maximum mooring loads may be expected to vary with the conditions of the tanker and the environment are described in Section 2. The maximum loads on the mooring system will usually be induced by the largest vessel under study moored in the maximum operational environment. However, in some circumstances higher loads will be experienced with a smaller vessel or in a different environment. Maximum loads in anchor chains and the anchoring system may sometimes occur under survival conditions without a tanker at the mooring. Also, loads in the hose system will usually be highest in survival conditions.

3.2.1 Vessel Size and Condition

The model test program should be planned with the tanker size and state of loading in mind. If the terminal is to serve a wide range of tanker sizes, then not only the largest tanker, but also an intermediate-size tanker should be model tested. If the port is a discharge terminal, then testing should concentrate on tankers in ballast conditions. If it is a loading port then testing should concentrate on tankers in loaded conditions.

The maximum-size vessel will not necessarily induce the highest loads, as other intermediate-size vessels may move more in response to the environment and impose higher loads. In a single short test, peak mooring loads may be higher with a loaded vessel than with a ballasted vessel. However, statistical analysis of the maximum mooring loads measured in a series of model tests will probably show greater deviation in peak loads for the ballasted vessel. As a result the predicted maximum mooring load over a long duration of exposure may be higher for the ballasted vessel.

Consideration must be given to the state of loading which the vessel is more apt to be in when exposed to the maximum operational environment. The wave height in which the vessel will moor will typically be only moderate, probably no higher than about 2 m (6 ft), and the vessel will probably not moor if very high wind or waves are expected. The vessel is thus not apt to be exposed to severe environments in the condition of loading in which it will be moored. The vessel is much more apt to be exposed to severe environments many hours after being moored, as it nears completion of loading or discharging and is not yet ready to depart. At a loading port the vessel will more probably be exposed to the maximum operational environment in the nearly loaded condition. At a discharge port the vessel will more probably be exposed to the maximum operational environment in the nearly ballasted condition.

3.2.2 Waves, Wind, and Current

There is little reason to expect that maximum loads will be experienced in waves, wind, and current less than the maximum operational values provided the relative directions remain the same. However, some variability of loads will be experienced with variations in the relative directions of these factors. Loads will probably be higher when wind and current are not colinear with waves, but wind or current at obtuse angles to waves may produce lower loads. In some cases, wind or current in combination with waves have been found to stabilize motions of the vessel and higher mooring loads were measured in waves without wind or current. However, maximum waves will more generally occur in combination with higher wind and current.

It may be necessary to test several different combinations of wind and current angles to waves. These combinations should be limited to those which are realistic at the site.

3.3 MODEL TEST SCALE FACTOR

The choice of a scale factor to be used in the SPM model test is a compromise between the technical requirements for accurate similitude on the one hand and economics and model test facility limitations on the other hand. Model scaling rules are covered in detail in most texts on fluid mechanics and hydrodynamics, and thus they will not be discussed here (Langhaar, 1951; Streeter, 1966). A brief discussion of Froude's law scaling is given because this is the scaling law commonly used in SPM model testing.

In this report the term scale factor refers to the relationship of the dimension of the prototype compared to the dimension of the model. For example, if the prototype dimension is 60 times the dimension of the model, the scale factor, designated by λ, is 60. The term scale refers to the fractional ratio of the dimension of the model divided by the dimension of the prototype, that is, $1/\lambda$, the reciprocal of scale factor. A scale of 1/60 is the equivalent to a scale factor of 60. A large-scale model is closer to the size of the prototype than a small scale model. For example, 1/48 is a larger scale than 1/60. Unless otherwise indicated, the terms scale factor and scale will refer to the length dimension.

3.3.1 Froude's Law Scaling of Model Tests

Normally the principal effects to be modeled in SPM model testing are forces due to gravity waves. Therefore, Froude's law scaling is used for these model tests. Reasons for this choice are given in some of the following subsections.

By Froude's law scaling for dynamic similitude the following condition must be satisfied:

$$\left[\frac{V^2}{gL}\right]_p = \left[\frac{V^2}{gL}\right]_m \qquad (3\text{-}1)$$

where V = velocity
g = acceleration of gravity
L = characteristic length
p represents prototype
m represents model

Since the acceleration of gravity is the same on the model as it is on the prototype it follows that

$$\left(\frac{Vm}{Vp}\right)^2 = \frac{Lm}{Lp} = \lambda \tag{3-2}$$

where λ is the geometric on length scale factor. Thus, the ratio of the scale velocities is $\lambda^{1/2}$. The ratio for time scale is also $\lambda^{1/2}$ since

$$\frac{Tm}{Tp} = \frac{Lm/Vm}{Lp/Vp} = \left(\frac{Lm}{Lp}\right)\left(\frac{Vp}{Vm}\right) = \lambda\left(\frac{1}{\lambda^{1/2}}\right) = \lambda^{1/2} \tag{3-3}$$

Other scale factors between prototype and the model are summarized in Table 3-1 together with their values at a length scale factor of 50.

3.3.2 Scale and Accuracy

Testing at as large a scale as practical is desirable because the accuracy of the results will be better. The accuracies of the modeled environments and the dimensions of the model will be proportionately better at a larger scale. The ability to scale forces will also be better at a larger scale because, as discussed in subsections 3.5 and 3.6, certain phenomena scale better by one scaling rule than by another, and the deviations are greater at smaller scales.

The effect of scale is more important in modeling small objects, such as cargo hoses, than it is in modeling very large objects, such as tanker hulls. For example, above a critical Reynolds number viscous-drag coefficients are essentially independent of velocity, but small objects are more apt to have Reynolds numbers below the critical range, and thus not scale properly. Also, vortex shedding, a function of Reynolds number, Kuelega-Carpenter number, system stiffness, and surface roughness, may be produced by models of small objects at conditions in which it would not occur on the prototype. Furthermore, the frequency of vortex shedding, a function of Strouhal number, may be different on the model than on the prototype (Batchelor, 1967).

It is very difficult to construct accurate models of some key components, such as swivels and cargo hoses, at small model scales. Such components must be modeled not only with respect to size and weight, but also with respect to friction, as in the case of fluid swivels, or to bending rigidity, as in the case of hoses. Certain small components become very fragile when modeled at too small a scale, and distortion can cause inaccuracies.

Force and moment measurements are much more accurate at larger model scales. There is a practical limit to how small and how accurate force transducers can be fabricated and calibrated. A one-Newton error in a transducer produces a much larger error when scaled up from a small model. Furthermore, at small scales the presence of a relatively large or heavy transducer can influence forces on the model or the response of the model, and in turn produce errors in the results.

3.3.3 Scale and Cost

The cost of an SPM model test program increases roughly in proportion to the scale. There are several reasons for this. The volume of the model test facility which is required to run tests, and also the volume and weight of the models which must be prepared, increase approximately in proportion to the cube of the scale. Also, the duration for which a model test must be run to represent a given prototype duration increases, though only by the square root of scale. This rough rule of thumb for the cost of a model test does not apply for very small scales because very fragile and accurately constructed models may be required, and precise miniature instrumentation may be necessary.

3.3.4 Scale and Test Facility Limitations

Limitations of available model test facilities and equipment will govern more than cost. The length, width, and depth of the model basin, the capacity of the wave generators, and the pumps and fans used to produce current and wind will limit the size of the model which can be tested. There may also be a limiting relationship between the depth of water in the basin and the height of wave or velocity of current which can be produced.

Basin width may limit model motions and produce edge effects. Limited basin length may produce reflected waves. The distance between the test site and reflecting boundaries in combination with the scale factor dictates the prototype time during which tests can be conducted without contamination of the incident waves by reflected or refracted waves.

Limitations may also be imposed by the size of model which can be manufactured in the shop or transferred into the basin. Availability of existing models may also influence the choice of scale, as a large tanker model may cost as much as $10,000 to construct, while another model simulating the tanker on a smaller scale may already be available.

Primarily because of the above limitations, most SPM model tests with very large crude carriers (VLCCs) have been run at scales of from 1/50 to 1/60. At a scale of 1/50 a model of a 550,000 dwt tanker will be approximately 8.3 m (27 ft) long and will weigh approximately 50 kN (11,000 lb) at loaded draft. Some SPM model tests on relatively small VLCCs have been run at a scale of 1/38, but no noticeable improvement in results was noted. However, the results of model tests run at a scale of 1/100 or smaller have not always been satisfactory.

3.4 MODELING OF WAVES

The characteristics of the waves are commonly defined in terms of the following:

- Significant wave height
- Significant wave period
- Wave spectrum

For model testing of an SPM, several operational waves may be defined, associated with the operational limits of several sizes of vessels. A maximum or survival wave may also be defined for testing the survival of the SPM system in severe storms when no tanker is moored. Peak or maximum wave heights might also be defined. However, since peak values are statistical extreme phenomena, they would not normally be duplicated in a model test of limited duration. As discussed in subsection 2.2.1, waves are preferably defined in terms of significant wave height and amplitude spectrum.

3.4.1 Limitations Of Wave Generators

Several types of mechanisms are used to generate waves in the various model basins. The principal types of wave generators are mechanical flaps or paddles, mechanical plungers, and pneumatic generators. Although the characteristics of these types of generators are different, there appears to be no reason to favor any type for generating waves for SPM model testing. Limitations of stroke amplitude and speed may, however, determine that the wave generator at a particular basin is inadequate for the purpose. Although pneumatic generators are claimed to have faster response time, the depth of the baffle between the pneumatic chamber and the wave basin, the mass of water between the free surfaces inside and outside of the chamber, and the compressibility of the air all impede effective transfer of the higher frequency components to the free waves.

Wave generators generally are limited in their ability to produce short period (high frequency) waves. For example, if a wave generator cannot cycle faster than 1.5 hertz, it will not be able to produce waves with periods shorter than about 5 seconds at a length scale factor of 60. (In Froude's law scaling, time scales by the square root of the length scale). This inability to produce the shorter period components of wave spectra is common to all wave basins, as far as we are aware.

3.4.2 Problems Of Matching Wave Spectrum

The significant wave height corresponds to the area under the wave spectrum curve as explained in Appendix A. When a model basin is asked to model a given significant wave height, and to match a defined wave spectrum, it must produce a model wave spectrum which envelopes an area equivalent to that corresponding to significant wave height. The defined wave spectrum may have some of its energy (area) at frequencies higher than those which can be produced in the model basin. To compensate for this lack of high frequency energy, extra energy will be produced at medium or low frequencies, thus distorting the shape of the spectrum in order to produce the desired significant wave height.

This problem is most evident at low significant wave heights, which have more high frequency energy. Figures 3-1, 3-2, and 3-3 show typical comparisons of theoretical wave spectra and model basin produced spectra. These spectra were produced by a modern flap generator having variable speed and variable stroke.

For the 2 m (6.5 ft) significant-wave-height spectrum, Figure 3-1 the model spectrum falls below the theoretical spectrum at frequencies above 0.16 hertz, and there is almost no actual energy above 0.2 hertz. This lack of energy, or area under the curve, must be made up by adding more energy at medium frequencies, resulting in a high peak at 13.5 hertz to make up the significant wave height. Figure 3-2 shows a 3 m (10 ft) significant-wave-height spectrum. The model spectrum falls far below the theoretical spectrum at frequencies above 0.15 hertz. However, a much larger proportion of the energy in the 3 m theoretical spectrum lies below 0.15 hertz, and thus it is easier to match than the 2 m spectrum. Still it was necessary to add more energy at the mid frequencies to produce the significant wave height.

The match for the 4 m (13.5 ft) significant-wave-height spectrum, Figure 3-3, is better. Although the model spectrum falls below the theortical spectrum beyond about 0.14 hertz, enough energy has been produced to fill in most of this portion of the spectrum. Note the 4 m theoretical spectrum extends down to 0.06 hertz and peaks about 0.085 hertz, while the 2 m theoretical spectrum extends down to about 0.08 hertz and peaks at about 13.5 hertz. Approximately 40% of the energy in the 2 m energy in the theoretical spectrum lies above 0.18 hertz while only about 10% of the 4 m theoretical spectrum lies above 0.18 hertz.

3.4.3 Proposed Redefinition Of Model Wave Spectrum

In modeling a wave spectrum for SPM model tests, it would be preferable to attempt to more accurately model the medium and low-frequency portions of the spectrum, than to distort these portions of the spectrum to make up for the deficiency in high-frequency energy. A frequency cut-off point should be defined commensurate with the requirements of the test program and the capabilities of the wave generators.

Typically the fundamental period of response of a SPM is greater than 10 seconds, although it may have secondary frequencies of 5 seconds or less. The natural periods of roll, pitch, and heave of large tankers are typically 10 seconds or more, and, when moored at a SPM, the periods of tanker surge, sway, and yaw are measured in minutes. Thus, wave periods less than about 5 seconds normally have little effect on the mooring system and moored tanker.

As an example, consider that 0.18 hertz (prototype time) is defined as the spectrum analysis cut-off point for a proposed SPM model test program. This corresponds to a minimum period of 5.5 seconds, much below the periods of response of interest. By this frequency cut-off method the modeled spectral curve would be required to match the theoretical spectrum curve for frequencies below 0.18 hertz. That is, the area under the spectrum curve below the cut-off period of 0.18 hertz should be equal. The significant wave height would be defined as that corresponding to the total area under the theoretical spectrum.

When testing in survival conditions with cargo hoses, high-frequency response characteristics may be of major concern. In such cases it will be necessary to match the high-frequency portion of the spectrum more precisely. This may necessitate modeling the buoy and cargo hose system (without a tanker moored) at a larger scale.

3.4.4 Criteria For Model Wave Spectra

A point-by-point matching of the wave spectrum cannot be expected, nor would it be completely realistic. A wave spectrum as defined by a formula or smooth curve is only a general description of typical wave frequency distributions at the site. Definitions of such wave spectra are discussed in Appendix A. Spectra differing from a spectrum measured during a specific storm at the site will probably be measured during other storms.

Some tolerance in matching the spectrum must be allowed, but a practical limit must be placed on this tolerance. The tolerance may be stated in terms of the percent of deviation of the ordinate of the model spectrum from the specified spectrum. From discussions with model basins and a review of wave spectra modeled in past tests by these basins, ER&E believes it is generally realistic to expect the ordinate of the model spectrum to deviate from that of the specified spectrum at any frequency by not more than ±30%. The application of this criteria to several typical spectrum are shown in Figures 3-4 and 3-5.

Through successive trials skillful model-basin technicians can usually improve the model spectrum to meet this criteria in a moderate amount of time. Meeting this criteria may prove troublesome in the case of a peculiar complex spectrum, such as one having several peaks or one very sharp peak. Also, meeting this criteria may be difficult at low wave heights.

The principal difficulty in matching a spectrum is that of attempting to match the total energy (area under the curve) corresponding to the significant wave height in the absence of high-frequency components in the model spectrum. As discussed in subsection 3.4.3, the energy or area missing from the high frequency portion of the spectrum must be made up in the intermediate frequencies, thus creating undesired peaks of energy. Redefining the significant wave height of the model spectrum as proposed in subsection 3.4.3 will eliminate this difficulty. It may then be possible to match the ordinate of the spectrum within a closer tolerance than the $\pm 30\%$ proposed above. This cannot be determined until attempts are made to apply these criteria.

3.5 MODELING OF WIND

Modeling wind in SPM model tests requires special methods of scaling. The effect of wind on the moored vessel is a viscous drag phenomena. Viscous drag effects are accurately modeled by Reynolds' law scaling. However, modeling wave effects requires Froude's law scaling, and viscous drag effects are distorted using Froude's law scaling.

Modeling wind velocity by Reynolds' law scaling is impractical in SPM model testing because a wind-velocity scale factor inversely proportional to the length scale factor would have to be produced. Furthermore, by true Reynolds' law scaling, the force produced would be independent of the scale factor and thus the force on the model would be the same as that on the prototype. In order to overcome these problems, the forces and moments produced by the wind may be scaled instead of scaling the velocity of the wind.

Because Froude's law scaling is used to model wave effects, the forces on the model are proportional to the cube root of prototype forces. For compatability, it is desirable that wind forces produced on the model also scale by Froude's law. If wind velocity is scaled by Froude's law, then the wind velocity produced should be proportional to the square root of the scale factor and the resulting wind forces proportional to the cube root of the scale factor, provided the Reynold's number produced is above the critical range. However, experience has shown that scaled forces are not necessarily produced by this means. Discrepancies may be caused by the variation of Reynolds' number, and by such factors as the variations of model wind velocity with elevation and spacial variation of the wind field in the model tests.

3.5.1 Prototype Wind Forces

Instead of attempting to accurately model the velocity of the wind by scaling laws, it is better to generate a wind velocity which produces properly scaled forces and moments on the model by Froude's law scaling.

From force and moment coefficient curves generated from wind tunnel tests, the longitudinal wind force, lateral wind force, and wind yaw moment acting on a vessel at any heading to the wind can be calculated using equations 2-3, 2-4, and 2-5 given in subsection 2.3.1.

3.5.2 Modeling Wind Forces By Wind

The most generally accepted method of modeling wind during SPM model testing is to produce the forces and moments on the vessel by means of a generated wind field. A bank of fans is arranged perpendicular to the desired wind direction at some distance from the model such that a uniform wind field is developed over the area occupied by the vessel at the SPM. Figure 3-6 shows a typical fan-generated wind-field test set-up.

At the beginning of a test series in which wind will be modeled, a series of constrained-vessel wind-force tests should be conducted to verify that Froude's-law-scaled forces and moments are produced. The longitudinal wind force, lateral wind force, and wind yaw moment are measured on a restrained vessel for a number of different wind directions. A complete rotation of the wind field around the vessel, with associated wind-force and -moment measurements, is not necessary because during an actual model test under the influence of wind, wave, and current the vessel does not present itself to the wind at all angles. Because of this and symetry of the vessel, normally it is sufficient to conduct the constrained wind test at wind angles from 0 to 90 degrees off the bow of the vessel.

During the constrained tanker wind-force tests, the forces and moments that are measured are compared to those forces and moments that have been determined in wind-tunnel model tests. The velocity of the wind field generated by the bank of fans is adjusted until a reasonable match is obtained.

It is difficult to match the data obtained from the wind tunnel tests exactly. One reason for the difficulty is the wind field in the model basin is not as uniform as that generated in the wind tunnel. Another factor is the vessel modeled in the SPM tests may not be identical to the vessel modeled in the wind tunnel tests.

A typical good wind-field distributed from an actual SPM model test is shown for reference in Figure 3-7. In general, a good correlation should be assumed to exist between the wind tunnel test data and the constrained vessel test data if the lateral and longitudinal forces compare within ± 10 percent and the yaw moment compares within ± 20 percent.

By using a real wind field modeled by Froude's law scaling of forces, the vessel response at an SPM will be accurately simulated. This method has been used satisfactorily by ER&E in a number of model test programs.

3.5.3 Indirect Means Of Modeling Wind Forces

An early method of modeling the effect of wind during SPM model tests was to pull on the model with a string run to a weight over a pulley. The weight simulates a constant wind load and the string transmits this load horizontally to the model tanker at a point which corresponds to the center of pressure of the vessel. A typical set-up of this modeling technique is shown in Figure 3-8. The forces thus generated on the vessel are constant in magnitude but change in direction of application depending on the angle of the vessel. The magnitude of the force applied is scaled to repre3ent the wind force expected on the vessel and depends on the size and ballast condition of the vessel, and on the velocity of the wind.

Modeling wind by weights over pulleys is unrealistic, and the results from model tests that model wind in this manner are thought to be non-conservative. The constant force that is applied to the vessel restrains the vessel from normal motion. Increases in forces and moments due to yaw of the vessel are not modeled. Maximum wind-produced forces and moments due to lift on the hull are not developed. This method of modeling wind bears little relationship to the prototype wind environment or to the effects it produces. The importance of modeling the changing wind forces and moments as the vessel moves in relation to the wind field has been demonstrated in subsection 2.4.1.

A newly proposed method of modeling wind for SPM model tests is to place fans on the model vessel to simulate the forces and moments on the vessel due to a uniform wind field. This method could use C_{Xw}, C_{Yw}, and C_{XYw} coefficient curves generated from wind tunnel model tests, such as those given in subsection 2.3.1. However, these curves can be reduced into equivalent forces and a force couple which act at the forward and aft perpendiculars of the vessel. The following equations illustrate this relationship:

Lateral Force at the Aft Perpendicular

$$F_{YAw} = 1/2\ F_{Yw} - \frac{M_{XYw}}{L_{BP}} \qquad (3\text{-}4)$$

Lateral Force at the Forward Perpendicular

$$F_{YFw} = 1/2\ F_{Yw} + \frac{M_{XYw}}{L_{BP}} \qquad (3\text{-}5)$$

where F_{Yw} and M_{XYw} are the force and moment at the point of intersection of the transverse and longitudinal centerlines.

From the force coefficient curves the longitudinal wind force, lateral wind force at the aft perpendicular, and the lateral wind force at the forward perpendicular may be calculated using the following equations:

Longitudinal Wind Force $$F_{Xw} = C_{Xw} \frac{k}{2} \rho a V_w^2 A_T \qquad (3\text{-}6)$$

Lateral Wind Force at Aft Perpendicular $$F_{YAw} = C_{YAw} \frac{k}{2} \rho a V_w^2 A_L \qquad (3\text{-}7)$$

Lateral Wind Force at Forward Perpendicular $$F_{YFw} = C_{YFw} \frac{k}{2} \rho a V_w^2 A_L \qquad (3\text{-}8)$$

where

C_{YAw} = Aft lateral wind force coefficient

C_{YFw} = Forward lateral wind force coefficient

The aft perpendicular of a vessel is the axis of the rudder post. The forward perpendicular is a vertical line passing thorough the point of the bow at the loaded water line. Other terms in these equations were defined in subsection 2.3.1.

Fans may be positioned at the forward and aft perpendiculars and on the longitudinal centerline of the vessel as shown in Figure 3-9. Their velocity may then be regulated by a yaw sensor coupled to a computer such that they produce the desired forces on the vessel according to Froude's law scaling. An alternative method which has been proposed is to use only a single fan whose position along the longitudinal centerline, direction, and speed would be controlled by the yaw sensor.

In theory, this method for modeling wind should produce excellent results if the simulated wind forces can be controlled to produce the desired wind effect on the vessel as it moves about on the SPM in response to the total environment. However, this wind modeling technique has not, as of yet, been used for SPM model tests. (Chislett, 1977).

3.6 MODELING OF CURRENT

In SPM model tests, current may be modeled according to Froude's law scaling. The predominent effects of current on the vessel are form drag and, in shallow water, a difference in water level across the vessel due to blockage. The latter phenomenon is a gravity effect and is therefore accurately modeled by Froude's law scaling. Viscous drag effects are minimal except at angles near bow-on to the current. The form or pressure drag is of viscous origin, but appears in general to be independent of Reynolds number in the range of interest in SPM tests with ship-shaped bodies.

It has been estimated that viscous drag contributes only about 5% to the total force on a vessel broadside to the current, but that about 70% of the force is viscous drag when a vessel is bow-on to the current. However, the total force due to current is much less in a bow-on current than in a broad-side current. For example, the force produced on a 250 kdwt tanker with 10% underkeel clearance bow-on to a 1 knot current is 70 kN (15,000 lb), but the force produced when the same tanker is broadside to the current is 1400 kN (300,000 lb). The influence of current on the moored vessel is discussed further in subsection 2.3.4 and 2.3.5.

3.6.1 Modeling Current Forces By Current

The most realistic method of modeling current in SPM model tests is to generate a flow of water in the model basin with a velocity proportional to the prototype current using Froude's law scaling. Such a current flow is usually produced by pumping water through a bank of inlet ducts along the side of the basin, and returning water to the pumps through ducts on the opposite side of the basin. The current velocity generated by this method of modeling is relatively uniform with depth, decaying rapidly near the floor of the basin. Artificial spoilers on the basin floor can be used to make the velocity decay more gradually with depth. However, precise modeling of the velocity profile is not essential in SPM model testing. The current produced in the basin should be calibrated to produce a good match with the prototype velocity at a depth equivalent to about one-half the tanker draft.

Current may be simulated by towing the model of the mooring and the moored tanker in cases where water depth is at least 2 times the tanker draft and the current is nearly in-line with waves. The towing velocity is scaled by Froude's law, and should be determined by the current velocity at a depth equivalent to about one-half the tanker draft. Variations in towing velocity should be avoided.

In the case of low underkeel clearance towing should not be employed because the effect of current flowing under and around the hull cannot be simulated. This effect of underkeel clearance is covered in Section 2. Attempts to simulate the effect of the sea bottom by towing a false bottom with the model will probably not be successful in waves. The false bottom must be very large to avoid edge effects. Unless the false bottom is very rigid, it will deform in the waves, thus modifying both the waves and the current.

3.6.2 Indirect Means of Modeling Current Forces

The several means of indirectly modeling the effect of wind discussed in subsection 3.5.3 might also be considered for modeling current. The use of weights on strings over pulleys is not recommended for modeling current for the same reasons given against its use for modeling wind.

Fans mounted on the vessel could be controlled to produce scaled forces and moments as functions of angle to current. The fans could be programmed to produce the composite forces and moments produced by both wind and current acting simultaneously. However, a number of problems are invisioned in this method of modeling current.

The simulation of current effects by the use of fans is not recommended. The thrust produced by the fans mounted on deck will act far above the vessel's center of gravity, while the real force of current acts below the center of gravity. The thrust of the fans will produce roll in the direction opposite to that produced by current. This effect is especially important in shallow water where roll due to current can be pronounced. In addition the effect of squat, the tendency of the hull of the vessel to be drawn closer to the bottom by water flowing between the hull and the bottom, is not properly modeled by fans. The use of fans to simulate current would be especially unacceptable in model tests intended to measure underkeel clearance requirements.

Current forces on the vessel depend on the absolute velocity of the water with respect to the hull and the velocity of the moving model must be accounted for. On a moving hull the longitudinal current force, given equation 2-6, must be expressed as follows;

$$F_{Xc} = C_{Xc} \frac{k}{2} \rho w \, TL_{BP} (V_c - \dot{X})^2 \qquad (3\text{-}9)$$

where

$\dot{X}$ = Longitudinal velocity of vessel

and other terms are as defined in subsection 2.3.4

Similar equations can be written for the transverse force and the moment on the vessel.

The vessel velocity term is negligible in the case of wind, but for current it must be accounted for. Accounting for the velocity of the vessel hull will complicate indirect modeling of current forces.

Another reason that current should not be simulated by indirect means is there is an interaction between the waves and the current. The presence of a current alters the relation between wave length and wave period as experienced by the vessel.

3.7 MODELING OF SPM AND TANKER

Many different types of SPMs have been proposed and designed. The two most common types now in use are the SALM (Single Anchor Leg Mooring) and the CALM (Catenary Anchor Leg Mooring). The modeling of these two systems will be discussed in detail. Other types will not be discussed because of their very limited use to date. However, much of the following discussion on modeling of the SALM and CALM systems will apply to the modeling of other SPM systems.

3.7.1 The SPM System

The SALM system, shown in Figure 3-10 , consists of a cylindrical buoy which is attached to a mooring base on the sea floor by means of a single pretensioned anchor leg. The principal system parameters that must be modeled are as follows:

- Geometric dimensions of the buoy
- Center of gravity of the buoy
- Center of buoyancy of the buoy
- Anchor-leg length and submerged weight
- Buoy freeboard
- Buoy weight in air
- Buoy gross displacement

The geometric dimensions of the buoy should as a minimum model the length and diameter of the buoy. Fenders and other protuberances which may effect drag and added mass should be included on the model. The center of gravity and center of buoyancy should be located as close as possible to their positions on the prototype design in order to model the dynamic characteristics of the prototype design. The anchor leg length is defined as the distance between the upper and lower universal joints of the system. The net submerged buoyancy of the buoy is of major importance in modeling the elasticity characteristics of the SALM. Net submerged buoyancy is defined as the difference between the weight of the buoy in air and the gross displacement of the submerged buoy.

The CALM system, shown in Figure 3-11 , consists of a cylindrical buoy attached to the ocean floor by a number of anchor legs, usually four, six, or eight. The principal system parameters which should be scaled to accurately model the CALM system are as follows:

- Geometric characteristics of the buoy
- Center of gravity of the buoy
- Center of buoyancy of the buoy
- Anchor chain characteristics
- Buoy freeboard
- Buoy weight in air
- Buoy gross displacement

The depth and diameter of the buoy hull, the diameter and position of the fender ring, and the positions of anchor-chain and bow-hawser attachment points should be modeled accurately. The center of gravity and center of buoyancy should be modeled as closely as possible to the prototype design in order to accurately model the dynamic characteristics of the buoy.

The weight of the CALM buoy in air, the gross displacement of the buoy, and the freeboard of the buoy determine its installed net buoyancy. This net buoyancy along with the anchor chain characteristics determine the elasticity characteristics of the CALM.

The anchor-chain characteristics of the CALM system are very important in determining the elasticity characteristics of the system. The following parameters should be accurately modeled:

- Number of anchor legs
- Length of each anchor leg
- Submerged weight of the anchor chain (and of anchor-chain clumps if used)
- Anchor-leg elongation characteristics (except when chain is very large or short and thus very stiff)
- Anchor-leg pretension
- Position of anchor points on the sea floor
- Depth of water at each anchor point

For any type of SPM, the type of hawser system to be used must be modeled accurately in order to realisticly model the elasticity characteristics of the mooring system. Important hawser modeling parameters are:

- Hawser elongation characteristics
- Hawser-system length

The load-elongation data for used ropes should be used in modeling. This data usually can be obtained from the rope manufacturer. Compared with the load-elongation characteristics of new rope, the used rope load-elongation curves are much less elastic. New and used synthetic rope load-elongation characteristics are discussed in subsection 5.6.3. The actual length of the hawser system to be modeled must be determined by the SPM designer.

As discussed in subsection 2.5.4, the hawser length and elasticity can play very important roles in determining the SPM design loads. Synthetic rope elasticity characteristics are discussed in subsection 5.6.

3.7.2 The Tanker

Generally, a variety of tanker sizes and designs will moor to an SPM. However, only several typical tanker sizes and designs need be selected for model testing. The effect of tanker size and design on mooring loads is discussed in subsection 2.5.

In order to accurately model the selected tankers the principal dimensions and hull lines should be accurately modeled. The distribution of mass must be modeled so that vessel dynamics and stability are properly modeled for each draft condition.

The following are the principal vessel parameters which should be modeled:

- Vessel displacement
- Length between perpendiculars
- Beam
- Molded depth
- Draft and trim
- Forecastle height (bow hawser attachment point)
- Center of buoyancy
- Center of gravity
- Metacentric height
- Natural period of roll
- Natural period of pitch
- Longitudinal distribution of mass

Because the distirbution of mass in prototype tankers is not homogeneous, it is not always possible to model exactly the center of buoyancy, center of gravity, metacentric height, natural period of roll, and natural period of pitch. The natural periods of roll and pitch are usually determined by swinging the model in air while mounted on a special pivoted table. This avoids the influence of damping in water. Weight may be shifted around slightly to obtain the correct periods, and thus the center of gravity may move slightly. This slight variation in center of gravity is preferable to variations in the natural periods.

If wind is to be modeled, vessel freeboard, the deck house, and other major deck structures should be modeled. In tankers of any given size range a variety of deck house designs may be found. The deck house modeled should be typical in design and model typical longitudinal (broad-side) and transverse (head-on) wind areas. Modeling of details such as ladders, stanchions, piping, etc, is not necesary as these have almost no influence on wind loads.

A typical rudder and screw (propeller) should be modeled, especially if current is being modeled. These appendages contribute to the total drag force, especially for small tankers.

3.8 MODEL TEST FACILITY CAPABILITIES

The model test facility which conducts SPM model tests must have the ability to accurately model the important environmental effects identified in Section 2. It must have the capability to model the SPM system and tanker at a large enough scale factor to minimize scale effects and to accurately measure important data. Descriptions of several model test basins which were visited as part of this study are given in Attachment C.

SPM model tests should be conducted in model basins that have the capability of producing irregular waves. Model tests conducted in regular waves are of little value in establishing design mooring loads for SPMs. For most SPM sites waves, wind, and current will come from different directions, and the maximum design mooring loads will probably not be obtained if wind, wave, and current are modeled parallel in direction. Therefore, it may be necessary that the model test facility be able to create the effects of wind and current at various angles to the direction of the waves.

There are many model basins which are capable of producing irregular waves, and a number of these are large enough to permit testing models of VLCCs moored to SPMs at a scale of 1/75 or larger. However, there are only a few which are capable of conducting SPM model tests under the combined influence of wave, wind, and current from varying directions. At the present time the Netherlands Ship Model Basin (NSMB) is the only basin known to have successfully modeled current at a large scale by means of water flowing at angles other than parallel to the direction of wave propagation. Other basins can produce current in line with the wave propagation either by towing the model or by installing false bottoms and circulating pumps. Only a few model basins, including NSMB and Offshore Technology Corporation, have experience in modeling wind by means of a generated uniform wind field. Davidson Labs, and the Danish Ship Research Laboratory have proposed model testing SPM-vessel systems with fans on the vessel to produce active forces which simulate the effect of wind.

The competence and experience of the researchers and technicians conducting the model tests are important considerations. Skill must be exercised in constructing the models and in setting up the model tests. Care must be taken in conducting each model test as careless procedures will seriously jeopardize the results. Instrumentation must be properly calibrated at the start of the test program and must be periodically recalibrated during the test program. Data analysis and reduction are especially critical. Electronic interference can distort measurements, as can improper processing of data. The assistance of skilled and experienced model-test researchers can also be of much value to the SPM designer in determining the scope of the model test program and in interpreting and analyzing model test results.

The model tank that will be used should be large enough to permit the use of large scale models in order to reduce problems associated with scale effects. Model tests in waves should be conducted using Froude's law scaling in order to properly model gravity effects. Viscous drag effects are accurately modeled using Reynolds' law scaling but are distorted using Froude's law scaling. These scaling distortions are minimized by modeling at as large a scale as possible. Conducting model tests at several different scale ratios in order to determine the effect of scale on the results will probably not be meaningful because statistical variations will probably mask any scale effect.

3.9 INSTRUMENTATION AND DATA RECORDING

The quality of the instrumentation and data recording system is very important in a model test program. The care taken in accurately modeling the environment and the system in order to produce scaled forces and motions is lost if the forces and motions are distorted by the instrumentation system.

The instrumentation system must be carefully planned, fabricated, and calibrated to produce accurate measurements. Bulky instrumentation can impose extraneous forces or restraints on the model and thus influence its response. Preprocessing of data before recording should not alter or distort the data.

3.9.1 Wave Height Measurement

Water surface elevation is usually measured by monitoring the change in electrical resistance or capacitance between two vertical closely spaced wires partially emersed in the water. Such instruments are easily calibrated in place by raising or lowering the system with a micrometer-type device. It is reasonable to expect the surface elevation can be measured to an absolute accuracy of ± 0.75 mm (± 0.03 in.), limited primarily by capillary effects, or to a relative accuracy $\pm 3\%$ of maximum range, whichever is less.

Waves should be measured and analyzed to demonstrate matching of the specified wave spectrum. The length of the wave recording to be analyzed should correspond to a minimum of 30 minutes equivalent prototype time. Wave spectra should be checked at the beginning of the test program and, depending on the elapsed length of the program, at the end of the program and several times throughout the program. The generated wave train may vary during the program for reasons such as drift of electronics or wear of mechanical components. It is distressing to discover at the end of a long model test program that the wave spectrum is not the same as it was at the start.

It may be desirable to record waves during each test. The waves recorded with a model vessel in the basin may be different than without a model due to reflection and defraction, and the varying position of the model may also effect the waves. However, a wave record of each test is useful in relating vessel response and mooring loads to the wave height at any given time. For this purpose the wave probe is preferably placed in line with the bow of the vessel and parallel with the waves. Furthermore, such a wave record is very valuable in determining at what point during a test program the wave spectrum changed if it is later discovered the wave spectrum no longer matches the specified spectrum. If the spectrum has changed, without such a record it will probably be necessary to repeat all tests conducted since the last calibration of wave spectrum.

3.9.2 Load Measurement

Axial forces in bow hawsers and anchor legs are usually measured by a strain-gaged proof ring or load cell. The load cell is accurately calibrated in the shop, but it should also be calibrated in place on the model, or at least in the basin near the model, with the instrumentation leads in place.

The accuracy of typical load cells by themselves is usually within ± 0.1 percent of full scale. When they are installed on the model and output is fed through electrical leads, amplified, and then recorded, the measured signal usually has an accuracy of about ± 1.0 percent of full scale. Full scale in this context means the maximum load which the device is designed to measure.

Axial forces, shear forces, and moments in model structures may be measured by strain gages mounted directly on the structure or by special inserts comprised of small high strength members which can be more accurately strain gaged. Sometimes these forces and moments are measured directly or indirectly by load cells which are coupled to the structure by some means.

The accuracy to which structural loads can be measured will depend largely on the cleverness, resourcefulness, and accuracy of the model builder. The instrumentation for structural loads should be calibrated in-place on the model in the basin if at all possible.

3.9.3 Motion Measurement

A variety of motion measurement devices are used at model basins ranging from strings over sheaves to light and laser tracking systems. Any mechanical connection to the model may apply forces to the model or disturb the free motion of the model through friction or inertia. Light and laser tracking systems apply no such forces or restrictions on the model. Some motion measurement systems are very limited in their field of measurement while others may track model motions over the entire basin.

Several basins use pantograph systems to track the motions of the vessel. The pantograph system consists of an articulated parallelogram structure which is pivotally attached at one corner to a fixed point or movable carriage over the basin, and attached at the opposite corner to the model. The angle of the pantograph axis relative to the fixed point and the extension of the pantograph are translated into surge and sway of the model. The angles of the pantograph relative to the fixed point and to the model are translated into yaw. The pantograph may be connected to the model by a vertical rod which is free to rise and fall relative to the end of the pantograph. The elevation of this rod is translated into heave of the model. Angular transducers between this rod and the model measure pitch and roll of the model.

If the pantograph is attached to a fixed point above the basin, the area over which horizontal motions can be measured is limited by the extensibility of the pantograph. At least one model test facility mounts the pantograph on a two-degree-of-freedom motor-driven carriage. The position of the carriage in the horizontal plane is controlled by servo feedback from the position of the pantograph. As the pantograph approaches the limits of its motion the carriage moves to adjust the position of the pantograph. The position of the carriage is combined with the pantograph surge and sway data to give the total surge and sway motion of the model. By this means the position of the model can be tracked over a large horizontal plane area.

Light and laser tracking systems can be set up to measure translational motions over a large area. A set of two tracking systems can be used to measure translational motions at two points on the model to determine yaw of the model. Gyros can be used in conjunction with such a system to measure yaw, roll, and pitch.

Light and laser tracking systems can measure high frequency motions better than pantograph tracking systems. The pantograph system must overcome friction and mass in tracking motions. A small high-intensity light or a reflector is the only apparatus which must be placed on the model to measure translational motions, and no physical contact is required with the model.

The accuracy of static horizontal motions which can be measured with a good quality pantograph or light tracking system is ± 1 mm (0.04 in.) model scale. Angular displacements measured by a rheostat have an accuracy of ± 1 degree. Gyro compasses can typically measure angular displacements with an accuracy of ± 0.5 degree.

Care must be taken that pantographs or other motion-tracking devices attached to the model and instrumentation leads going to the model do not impede the motion of the model or impose extraneous forces on the model. This is especially true in tests conducted in wind. It is generally not practical to measure the motions of very small models, such as the mooring buoy, except by the use of a light-tracking system.

3.9.4 Data Recording

The data measured in the basin must be transmitted to a recording system and amplified and preprocessed prior to recording. Instrumentation leads should not be excessively long to avoid decay of the signal. The amplification should preferably be linear throughout the range of the signal. Amplification factors as well as instrument calibration factors must be carefully noted for future reference during data processing.

Data may be recorded in the basin in either analog or digitized form. The data is usually also recorded on a strip-chart using an oscillograph recorder. This strip-chart data is useful as an immediate check of the performance of the instrumentation system as well as of the model. Peak values can be readily obtained, and a check can be made to assure that amplification factors applied are adequate. Occasionally, a test must be rerun because an instrument has failed or a peak value has exceeded the range of the recording system.

The data recording rate must be fast enough to assure that high frequency signals are adequately recorded. In digitizing wave data, and data which has a moderate to high wave-induced component, the minimum discretization interval should be in the neighborhood of 0.2 second prototype time. This generally requires that the recording be done on magnetic tape. The recorded data may be sampled at a slower rate later during the data analysis process.

3.10 DATA ANALYSIS

The raw data from the model tests is of little use until analyzed. Peaks can be measured from the raw data, and a skilled researcher can scan the data to see trends and to estimates averages, but these cannot be used to predict design loads. The type of analysis performed on the data will depend on the characteristics of the data, the available computational capabilities, and the uses to which the analyzed data will be put. Normally, statistical properties are of more value than simple maximums and minimums, because through statistics the extreme events and their probabilities can be predicted.

3.10.1 Types Of Data

Generally time records of data from SPM model tests will fall into three categories as shown in Figure 3-12.

- Type I The Type I record consists of a fast oscillating motion with the frequency corresponding with the frequency of the waves. This type of record is characteristic of all buoy motions when the buoy is unoccupied by a vessel. It is also characteristic of the heave and, to some extent, the pitch and roll of the buoy when a vessel is moored to the buoy, as well as the roll, pitch, and heave of the vessel.

- Type II The Type II record consists of a fast oscillating motion, with the frequency corresponding in general with the frequency of the waves, superimposed on a slowly varying motion. The amplitudes of the fast oscillations are large relative to the magnitude of the slowly varying motion. This type of record is characteristic of hawser and anchor-chain forces when the buoy is relatively large or the hawser is very stiff.

- Type III The Type III record consists of a slowly varying motion with a superimposed fast oscillating motion where the amplitudes of the fast oscillations are small compared to the slowly oscillating motion. This type of record is characteristic of the yaw, sway, and surge of the vessel and of hawser and anchor-chain forces in most systems. Figure 2-10 is an example of actual model test data which corresponds to Type III.

3.10.2 Spectral Analysis Of The Data

The statistical properties may be calculated directly from the recorded data, or a spectrum analysis may be performed on the data and certain statistical properties then derived from the generated spectrum. The following describes how this is generally done with recorded wave data to determine significant wave height and mean wave period.

The records of irregular long crested waves that are generated by the model basin are of Type I. The disturbance of the water level at a time T is assumed to be composed of an infinite number of components with arbitrary and random phase angles ϵ_n such that

$$H(t) = \sum_{n=1}^{\infty} \bar{\xi}_n \cos(\omega_n t + \epsilon_n) \qquad (3\text{-}10)$$

where

$\bar{\xi}$ = amplitude of n^{th} component of H(t) with a circular frequency of ω_n

The amount of energy contributed by waves with a frequency ω_n and an amplitude $\bar{\xi}_n$ follows from the spectral density, S_H, which is defined by

$$S_H(\omega_n)\, d\omega = \frac{1}{2} \bar{\xi}_n^{\,2} \qquad (3\text{-}11)$$

The moments of the spectrum M_p can be determined as follows:

$$M_p = \int_0^{\infty} \omega^P S_H d\omega \qquad (3\text{-}12)$$

If P = 0, then

$$M_o = \int_0^{\infty} S_H\, d\omega = \text{Area of spectrum} \qquad (3\text{-}13)$$

If P = 1, then

$$M_1 = \int_0^{\infty} \omega S_H\, d\omega = \text{First moment of spectrum} \qquad (3\text{-}14)$$

With the aid of the moments of the spectrum the following quantities can be calculated:

Significant Wave Height

$$H_S = 4\sqrt{M_o} \qquad (3\text{-}15)$$

For narrow spectra the value $4\sqrt{M_o}$ corresponds to the average of the one-third highest waves (double amplitude).

Mean Wave Period

$$\bar{T} = 2\pi \frac{M_o}{M_1} \tag{3-16}$$

For narrow spectra this value corresponds to the mean value of instantaneous period.

3.10.3 Statistical Analysis Of The Data

Type II and Type III data should generally not be analyzed through the spectrum process unless the record is very long and contains many cycles of the long period component of motion. Analysis of such data should be done directly from the digitized record. The statistical properties which are commonly obtained and the manner in which these properties are usually calculated are discussed below.

Data analysis should be preformed on the displacement and force records in order to obtain the following statistical quantities, which are illustrated in Figure 3-13.

Mean value; $\bar{X}$

$$\bar{X} = \frac{1}{M} \sum_{n=1}^{n=M} X_n \qquad M = \text{Number of Samples} \tag{3-17}$$

Root-mean square value; σ

$$\sigma = \frac{1}{M} \sum_{n=1}^{n=M} (X_n - \bar{X})^2 \tag{3-18}$$

Significant peak-to-trough (double amplitude) value: $2\tilde{x}_s$

This is the mean value of the one-third largest double-amplitude values.

Significant peak value: $\tilde{x}_s +$

This is the mean value of the one-third highest peak to zero values.

Significant trough value: $\tilde{x}_s -$

This is the mean value of the one-third largest trough to zero values.

Maximum peak-to-trough (double amplitude) value: $2\,x_{max}$

This is the largest double amplituded value.

Maximum value: x_{max}^{+}

This is the highest peak value.

Minimum value: x_{max}^{-}

This is the lowest trough value.

3.11 DETERMINATION OF DESIGN LOADS

In interpreting the results of SPM model tests for comparison or design purposes, the data must be examined statistically in order to obtain meaningful maximum values. The duration of an individual SPM model test depends on the limitations of the model basin and the type of test being conducted. Typically, it is on the order of 5 minutes model-scale time. In Froude's law scaling, time scales as the square root of the scale factor. Thus, at a model scale of 1:60, the model test duration corresponds to a prototype time of a little more than 30 minutes.

3.11.1 Statistical And Maximum Values

For typical SPM model tests, the statistically derived mean and significant values of the forces in the anchor chains and bow hawsers measured during one test generally vary less than 10 percent from those measured in other tests of the same system in the same environment. Therefore, it may be assumed that each model test has been conducted for a duration of time long enough to justify the assumption that the phenomena modeled in the model test is stationary ergodic. This means that the statistical properties of the phenomena, that is the mean and significant values and the standard deviation, will not change significantly if the test would have been conducted for a longer period of time or if the test had been carried out at another time with the same environment.

The maximum measured values, unlike the statistical properties, can and do vary from test to test. Due to the random nature of the test, the maximum measured values may vary considerably from one test to another carried out under the same conditions. Furthermore, if the test duration is increased, the maximum measured value may also change.

3.11.2 Probability Analysis

Because maximum values will vary from test to test, test data should not be considered individually but should be grouped and analyzed statistically to determine probable maximum values based on storm duration and chance of exceedence. This is commonly done by developing a distribution function of the ratios of the maximum value recorded during each test with its corresponding statistically derived significant value (Haring, Adams, Beazley, and Kripp, 1969 and 1970; Flory and Poranski, 1977).

The value of F_{max}/F_S is called the force ratio, R.

$$R = \frac{F_{max}}{F_s} \qquad (3\text{-}19)$$

where F_{max} = maximum force

F_s = significant force

Force ratios from a number of tests are ranked in ascending order of magnitude. The assumption is made that the values of successive force ratios are statistically independent. Therefore, these occurrences may be treated as Bernoulli trials and each force ratio may be assigned a probability that its value will not be exceeded according to the following formula:

$$P = \frac{n}{M + 1} \qquad (3\text{-}20)$$

where M = Total number of occurrences

n = Rank of occurence being considered

P = Cummulative probability that an event is equal to or less than a particular value

The force ratios and their associated probabilities P are plotted on probability paper. The type of probability theory to be used depends on the data. A typical force ratio probability distribution function using normal Gaussian theory is shown plotted in Figure 3-14. A total of 19 separate force ratios were used to generate the plot. The data is from a typical set of statistically homogeneous tests obtained from an actual SPM model test series in which the duration of each test represented approximately 30 minutes prototype time. The plotted force ratios are for bow-hawser forces.

It is recommended that at least 10 data points and preferably 20 data points should be used in developing the probability distribution. The more data points that are used, the more statistically reliable the results derived from the plot will be.

The data which is plotted must be homogeneous, that is, it must be from tests with similar statistical properties. Figure 3-15 presents a hypothetical probability distribution in which two sets of data have been plotted in one distribution. The data as plotted fits a straight line and, therefore, it might be assumed that the distribution is valid. However, the data is not homogeneous, and in reality the data should be plotted as two distinct distributions as also shown in the figure. Data from tests with ballasted tankers usually exhibits different statistical properties than data from tests with loaded tankers.

Since the duration of each test for which the data has been plotted is 30 minutes, from the distribution of values shown in Figure 3-14, the probability P that the force ratio R will not be exceeded during a period of 30 minutes can be determined. Consequently, the probability that the force ratio will be less than R is equal to (1 - P). The probability P_t that the force ratio will be larger than R during a period of t times 30 minutes the equation:

$$P_t = 1 - (1 - P)^t \qquad (3\text{-}21)$$

where P = Probability of exceedence for test

P_t = Probability of exceedence for design duration

t = Ratio of design duration to test duration

3.11.3 Probability Plotting

When data is plotted on probability paper, the degree to which the plotted points lie on a straight line is an indication of the closeness of fit of the data to the probability theory being used. In Figure 3-14, the data has been plotted on probability paper that assumes a normal or Gaussian distribution. However, the line that has been faired through the data is not a straight line but forms a gentle curve. Since the line is not straight, the data does not exactly fit a normal Gaussian distribution.

Other probability functions may be tried to determine a more applicable probability theory for fitting SPM mooring load data. Gumbel's distribution of extreme event has been found to fit most SPM model test data extremely well. Figure 3-16 presents the same data that was used in developing Figure 3-14 plotted on Gumbel paper. It can be seen that the straight line drawn through the plotted points fits very well.

Gumbel's distribution was developed for the statistical treatment of floods and other climatological extremes. Gumbels distribution provides a method for calculating the line of best fit through the data. This method was used to develop the straight line drawn through the data points plotted in Figure 3-16. Gumbel's distribution for extreme events appears to fit SPM model test data extremely well and it is recommended as an improvement over Gaussian theory for developing the extreme events associated with the SPM mooring loads (Gumbel, 1954; Myers, Holm, and McAllister, 1969).

For convenience, the model test data shown, typically for a 30 minute storm duration, may also be plotted for longer storm durations. Figure 3-17 presents the data of Figure 3-16 plotted on Gumbel paper along with the predicted force ratios for storm durations of 2, 6, and 12 hours. These curves have been developed from the probability theory given in equation 3-21. From these curves the force ratios for various storm durations and probability of exceedences can be obtained without further calculation. Significant hawser forces from the model test data are multiplied by the appropriate force ratios. The following examples demonstrate the procedures that are used.

3.11.4 Example Probability Calculations

Example 1

What will be the chance that the maximum hawser force will be twice the significant force in the hawser during exposure to a constant environment of two hours duration?

From Figure 3-16

P = Chance that the force ratio, R, is equal to or larger than 2.0 during a period of 30 minutes = 0.03

and

$$t = \frac{2 \times 60 \text{ minutes}}{30 \text{ minutes}} = 4$$

From equation 3.21 it can be calculated that $P_t = 0.12$. Alternatively, this probability of 12 percent can be read directly from Figure 3-17. So there is about a 12 percent chance that the maximum force will be larger than twice the significant force during a period of two hours, or four times the duration of the model test results.

Example 2

What will be the force ratio, R, for which the chance of exceedence will have a probability of 50 percent during exposure to a sustained environment of six hours duration?

$$t = \frac{6 \times 60 \text{ minutes}}{30 \text{ minutes}} = 12$$

and, $P_t = 0.50$

From equation 3-21 it can be calculated that $P = 0.056$. From Figure 3-16 it can be seen that this probability corresponds to a force ratio, R, of about 1.9 Alternatively, this value can be determined directly from Figure 3-17.

Example 3

Assume that from model test data it has been determined that for the maximum operational conditions the significant hawser force is 1.7 kN (380,000 lb). What is the maximum hawser force that can be expected in a storm duration of 6 hours with a probability of 70 percent that it will not be exceeded?

A 70 percent probability of non-exceedence corresponds to a 30 percent probability of exceedence. From Figure 3-17 the force ratio for a 6 hour storm duration and a 60 percent chance of the value not being exceeded is 2.2.

$$F_{max} = \left(\frac{F_{max}}{F_S}\right) 6 \text{ hr} \times F_S$$

therefore

$$F_{max} = 2.21 \times 1.7 \text{ kN} = 3.75 \text{ kN}$$

3.12 THE BASIS FOR MOORING LOAD CALCULATIONS

Maximum mooring-load calculations must be based on realistic combinations of tanker size and condition and environments. The duration and chance of exceedence used in the probability calculations must also be realistically selected. If the mooring-load calculations are based on sound data and realistic maximum conditions, then undue conservatism need not be included in factors of safety during the design of the mooring.

3.12.1 Basis For Tanker and Environment

Several combinations of tanker size and loading condition and environment may have to be considered in determining the maximum mooring load. As explained in subsection 2.5.2, the maximum loads may be experienced with either a loaded or a ballasted tanker, and higher loads in some circumstances may be experienced with a tanker smaller than the largest for which the mooring is designed.

Although the highest combination of waves, wind, and current from a given combination of directions will probably produce the highest load, limiting mooring environments may be such that the highest waves, wind and current are colinear, but that waves, wind, and current of lesser magnitude will occur at non-colinear directions. The non-colinear combination of lesser waves, wind, and current may produce the highest mooring loads.

Complete model testing will probably not be necessary for more than two or three cases. However, the possibility of high mooring loads under various conditions should be considered in choosing the cases to be throrougly investigated.

Consider as an example the design of a mooring for a deepwater port off-loading terminal. Mooring loads have been determined for a ballasted 500,000 dwt tanker moored in 5 m (15 ft) significant height waves with 50 knot winds at 45 to the waves and a 2 knot current at 90 to the waves. This is the most severe environment combination for which the mooring is designed. The mooring is to be operated such that no tankers will remain moored in more severe environments. Tankers larger than 500,000 dwt will not be moored. Design studies based on model tests have been conducted which show that higher mooring loads will not be experienced with other smaller tankers moored, or when the 500,000 dwt tanker has a draft greater than the ballasted condition. Then, for this example case, the peak mooring load would be that predicted for the ballasted 500,000 dwt tanker in the cited limiting environment.

3.12.2 Basis For Duration and Chance of Exceedence

The maximum mooring load should be calculated by statistical techniques such as those described in subsection 3.11. The statistics may be developed from model test data. However, to use these data and apply the statistical techniques a realistic duration of exposure to the most severe environment and a reasonable chance of exceedence must be selected.

The duration is usually chosen as that associated with one mooring and cargo transfer operation instead of the cumulative duration of many such operations. Continuing the above example, the 500,000 dwt tanker would be near the ballasted condition only during the last hours of a discharge mooring. Prior to that time, it was more fully loaded, and thus less apt to exert very high mooring loads. Also, because it probably came to the mooring when the environment was mild, the most severe environment will probably only occur during the last hours of a discharge operation. If the total time at the mooring is typically 30 hours, then the duration of the combination of most severe tanker condition and environment may typically be 10 hours or less.

The selection of an appropriate chance of exceedence may take account of the degree of confidence which can be placed in the design data and the consequences of a breakout. Keep in mind that safety factors are applied to the maximum design load in the design of all components of the SPM. If assurances are made that cargo discharge is discontinued and cargo hoses are lowered before the maximum environment is reached and if chances of damage or pollution due to a tanker breakout are small, then conservatism is not necessary and a large chance of exceedence, say 20% or even 50%, may be appropriate. If the consequences of a mooring line failure are severe or if there are unknowns or doubts in the design basis, then a low chance of exceedence, say 5%, may be appropriate.

3.12.3 Probabilities and Factors of Safety

Appropriate factors of safety for the design of various components of the mooring system are recommended in Section 5. Factors of safety are used in engineered designs to account for variations in material properties and construction techniques. Factors of safety are also used to cover overloading due to unforeseen circumstances. If the design-basis setting and mooring-load determination steps described above are followed, then there will be very little probability of overloading.

Consider the probability of the most critical tanker being moored in its most severe condition in the most severe combination of waves, wind, and current. In general, this will be a rare event at a mooring terminal. To this probability must be added the duration of the event and the chosen probability of exceedence under the combination of circumstances. The chance of a mooring load momentarily equaling or exceeding the design mooring load determined in this manner will be very small.

3.13 SUMMARY

An SPM model test program conducted for the purpose of designing a deepwater port must include those combinations of environment, tanker size, and tanker condition which are apt to produce the highest loads, and to acquire sufficient data to permit statistical prediction of maximum loads. It may be necessary to test several tanker sizes and several combinations of waves, wind, and current. The models should be built to as large a scale factor as practical to minimize scale effects and modeling and instrumentation inaccuracies.

The proper wave spectrum as well as wave height and wave period should be modeled. However, reasonable criteria should be placed on the accuracy of the spectrum, and because of wave-frequency-generation limitations, it may be necessary to ignore high-frequency components in defining the spectrum.

Froude's law scaling must be used in model testing to properly scale the effects of gravity waves. Thus, it is preferable to adjust wind veloctiy to induce wind forces which scale by Froude's law. The form-drag and gravity-force effects of current predominate over resistance drag effects and, thus, it is proper to model current by Froude's law. Wind induced forces may be modeled by the reaction forces of fans mounted on the vessel. However, such indirect means of modeling current-induced forces appear to be potentially troublesome.

The important parameters of the mooring system and the tanker must be properly modeled. Of particular importance are parameters which effect the elasticity of the mooring system and the dynamics of the buoy and the tanker.

The model test facility which conducts SPM model tests must have the ability to accurately model the important environmental effects and to model the SPM system and tanker at a sufficiently large scale. Of particular importance may be the ability to model wind and current at angles relative to the waves.

Instrumentation must be carefully prepared and data must be properly recorded and analyzed to assure the accuracy of the measurements. The peak values recorded in a single test are not sufficient to predict maximum values. Generally, statistical functions based on the results of a number of tests must be used to predict maximum values.

TABLE 3-1

SCALE FACTORS FOR FROUDE'S LAW SCALING

Characteristic	Scale Factor	For Length Scale Factor of 50
Length	λ	50
Area	λ^2	2,500
Volume	λ^3	125,000
Time	$\lambda^{1/2}$	7.07
Veloctiy	$\lambda^{1/2}$	7.07
Force*	$S\lambda^3$	128,125
Mass*	$S\lambda^3$	128,125
Acceleration	λ^0	1
Pressure	λ	50
Moment*	$S\lambda^4$	6,406,250
Angular Displacement	λ^0	1
Angular Velocity	$\lambda^{-1/2}$	0.14
Angular Acceleration	λ^{-1}	0.02
Energy*	$S\lambda^4$	6,406,250
Spring Constant	λ^2	2,500
Bending Stiffness*	$S\lambda^5$	320,312,500
Spectral Density	$\lambda^{5/2}$	17,678

* S = Specific gravity of salt water 1.025

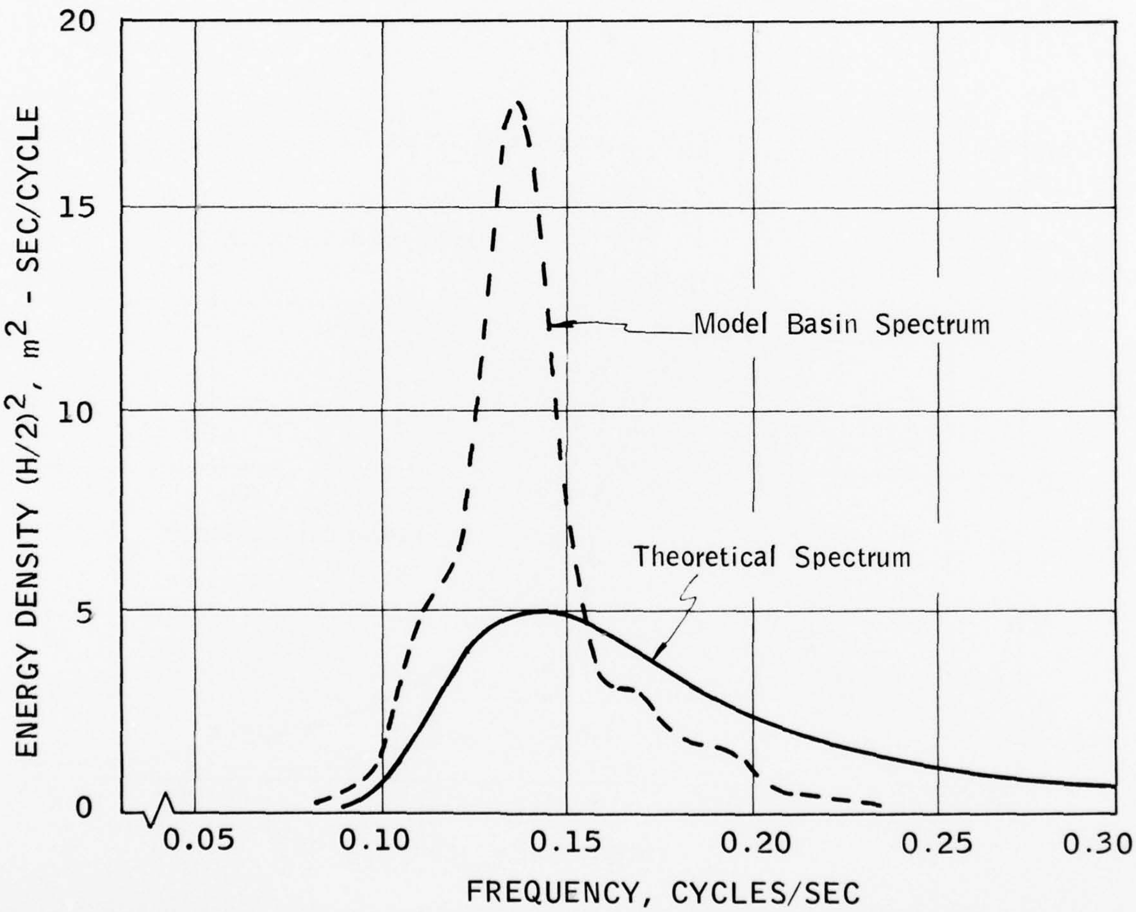


Figure 3-1 - COMPARISON OF THEORETICAL AND MODEL BASIN PRODUCED 2 METER PIERSON-MOSKOWITZ WAVE SPECTRUM

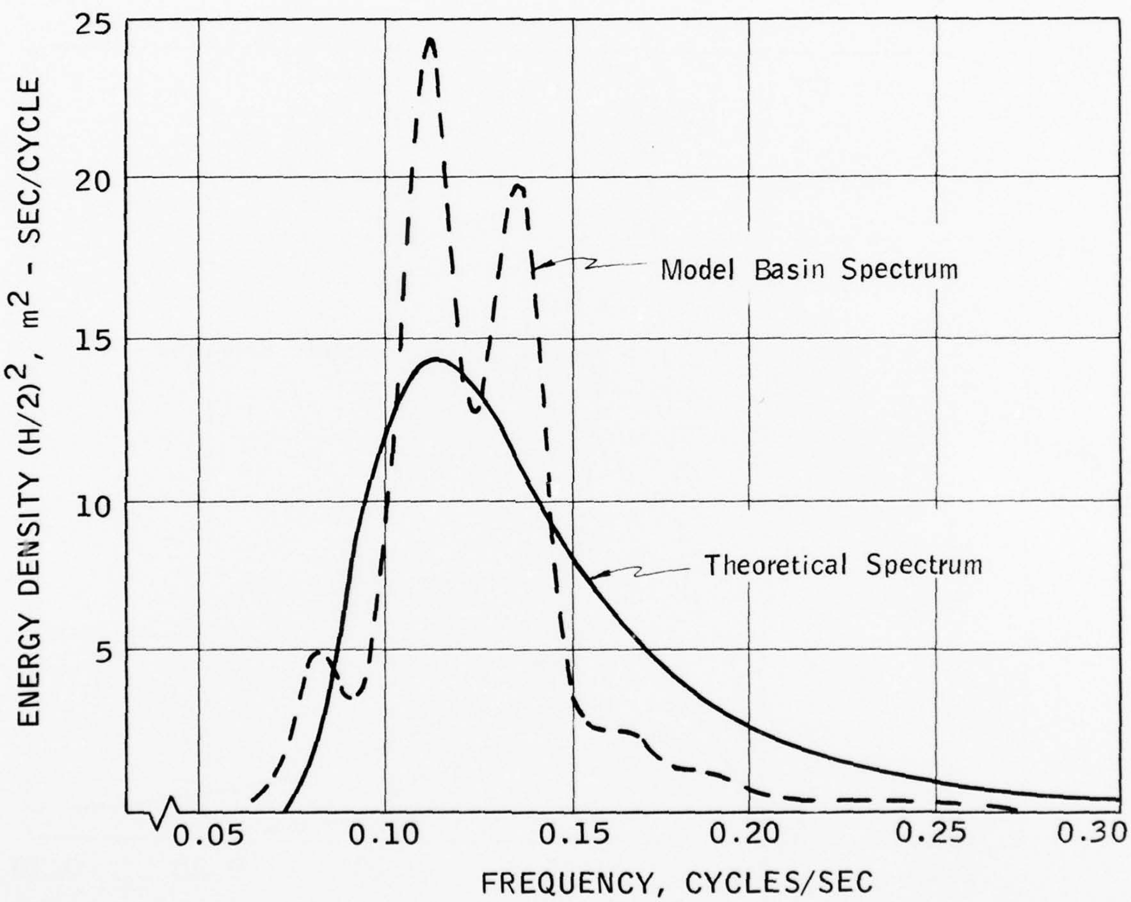


Figure 3-2 - COMPARISON OF THEORETICAL AND MODEL BASIN PRODUCED 3 METER PIERSON-MOSKOWITZ WAVE SPECTRUM

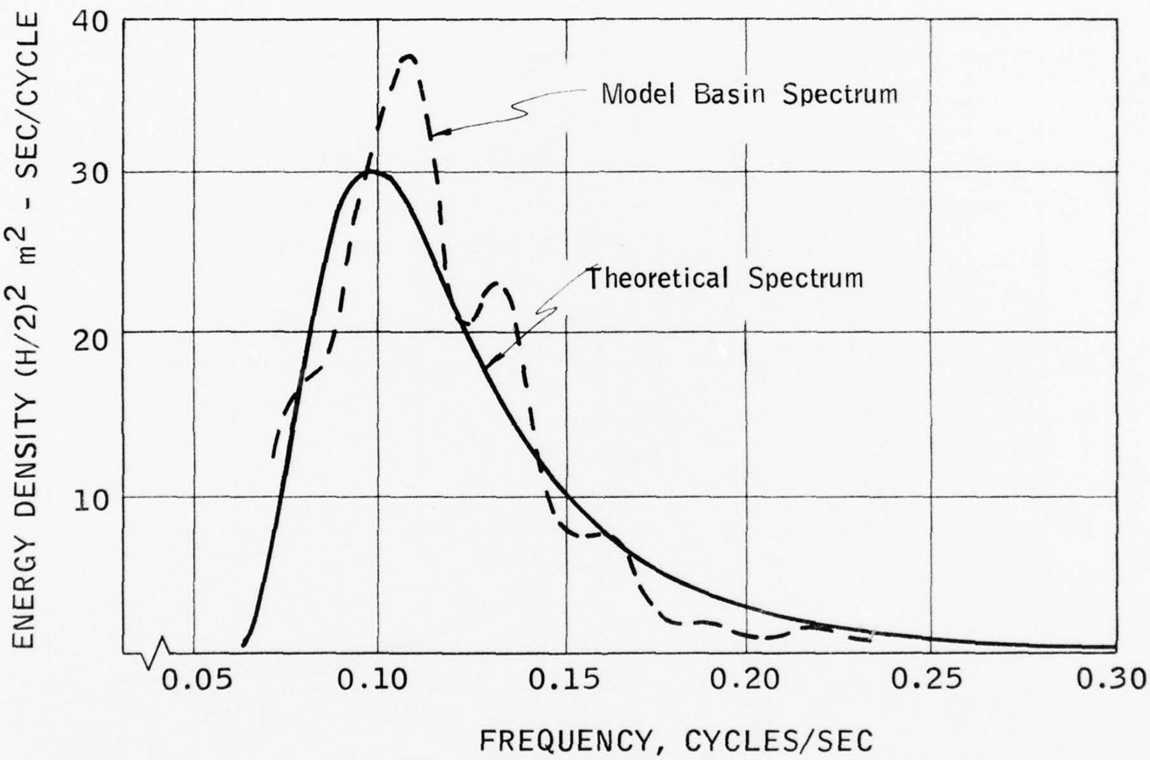


Figure 3-3 - COMPARISON OF THEORETICAL AND MODEL BASIN PRODUCED 4 METER PIERSON-MOSKOWITZ WAVE SPECTRUM

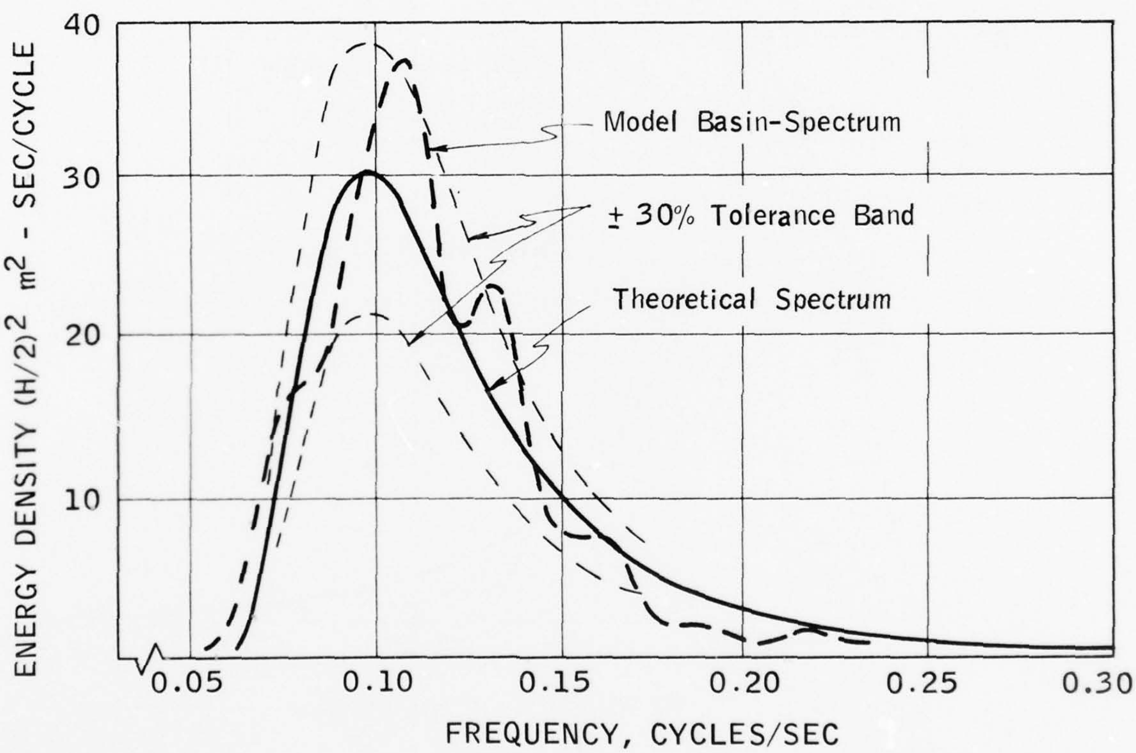


Figure 3-4 - APPLICATION OF ± 30 PERCENT TOLERANCE BAND TO MODEL BASIN SPECTRUM OF FIGURE 3-3

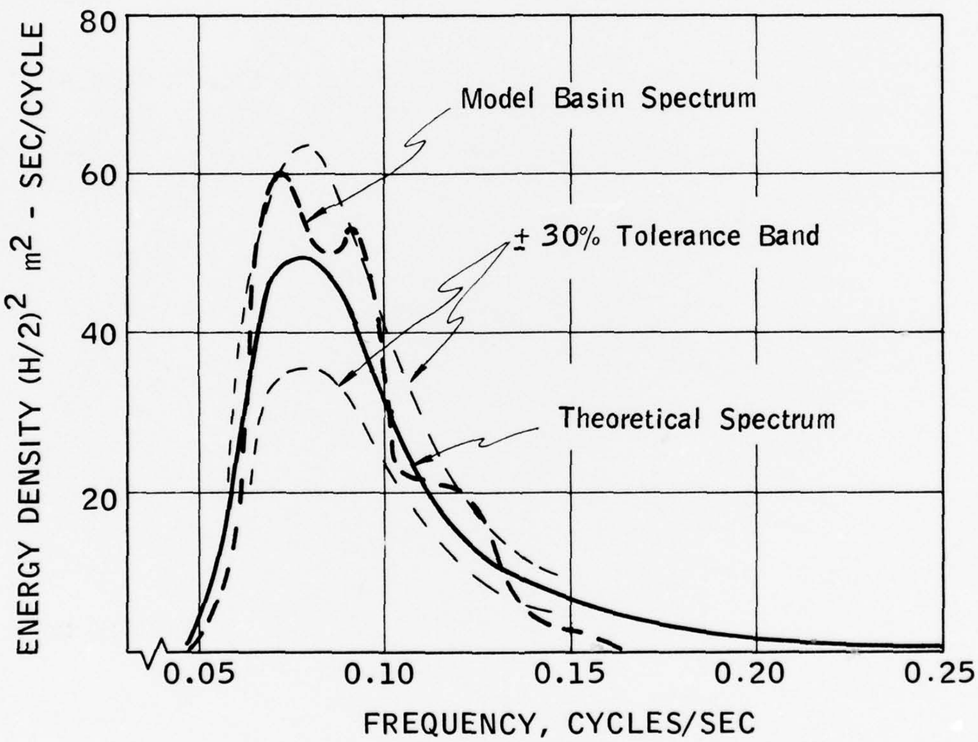


Figure 3-5 - APPLICATION OF ±30 PERCENT TOLERANCE BAND TO 4.6 M WAVE PIERSON-MOSKOWITZ WAVE SPECTRUM FROM ANOTHER BASIN

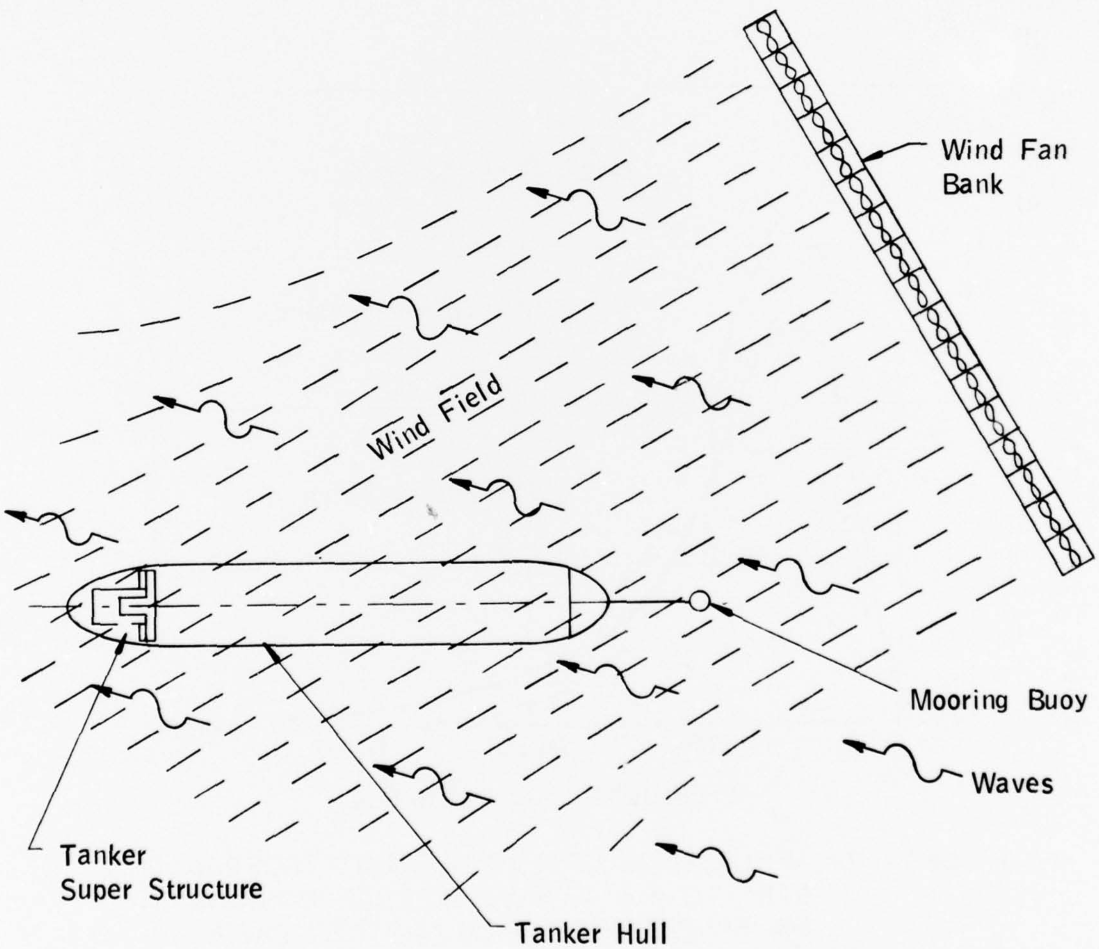


Figure 3-6 - SET-UP FOR MODELING WIND BY FAN GENERATED WIND FIELD

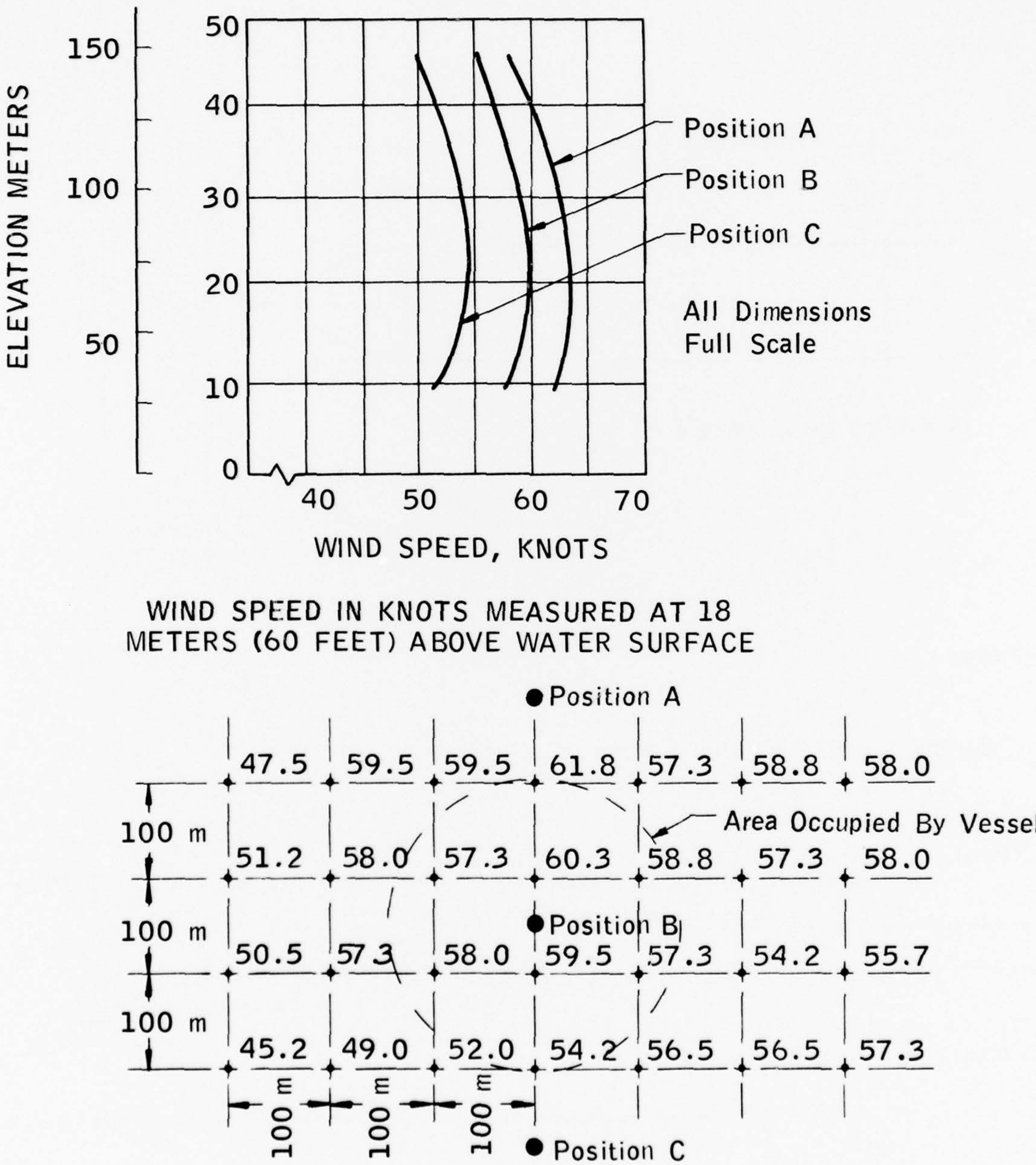


Figure 3-7 - TYPICAL MODEL WIND FIELD CALIBRATION

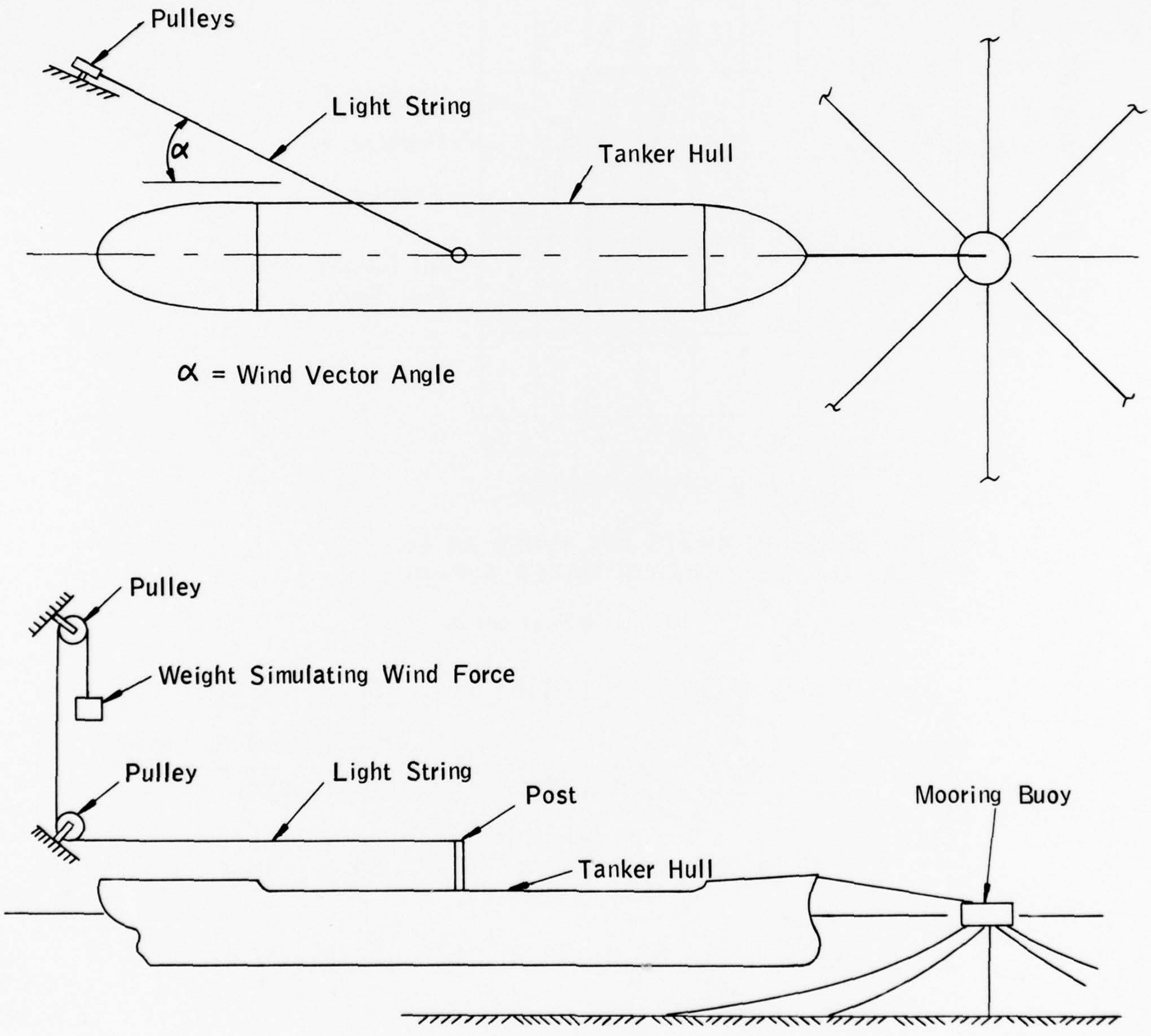


Figure 3-8 - SET-UP FOR MODELING WIND FORCE BY WEIGHT ON STRING OVER PULLEYS

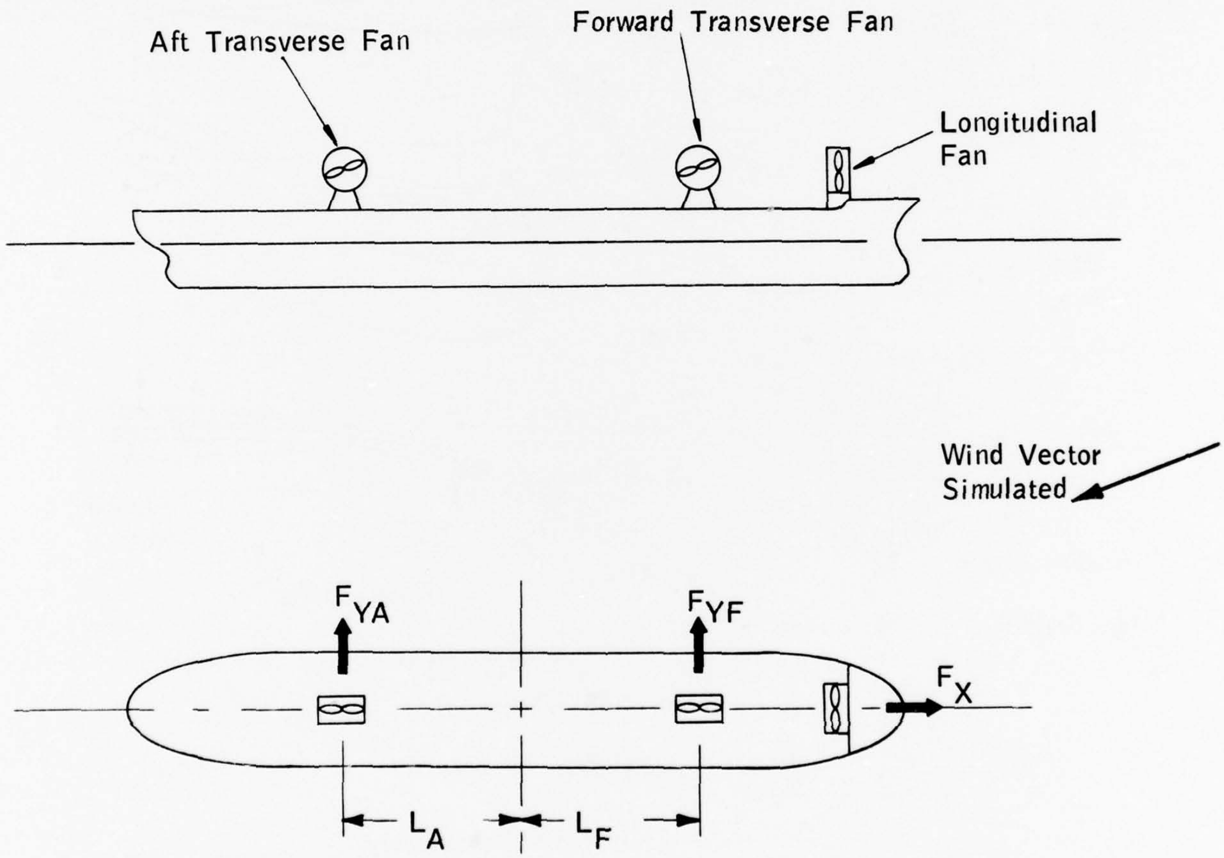


Figure 3-9 - SET-UP FOR MODELING WIND FORCE BY FANS MOUNTED ON VESSEL HULL

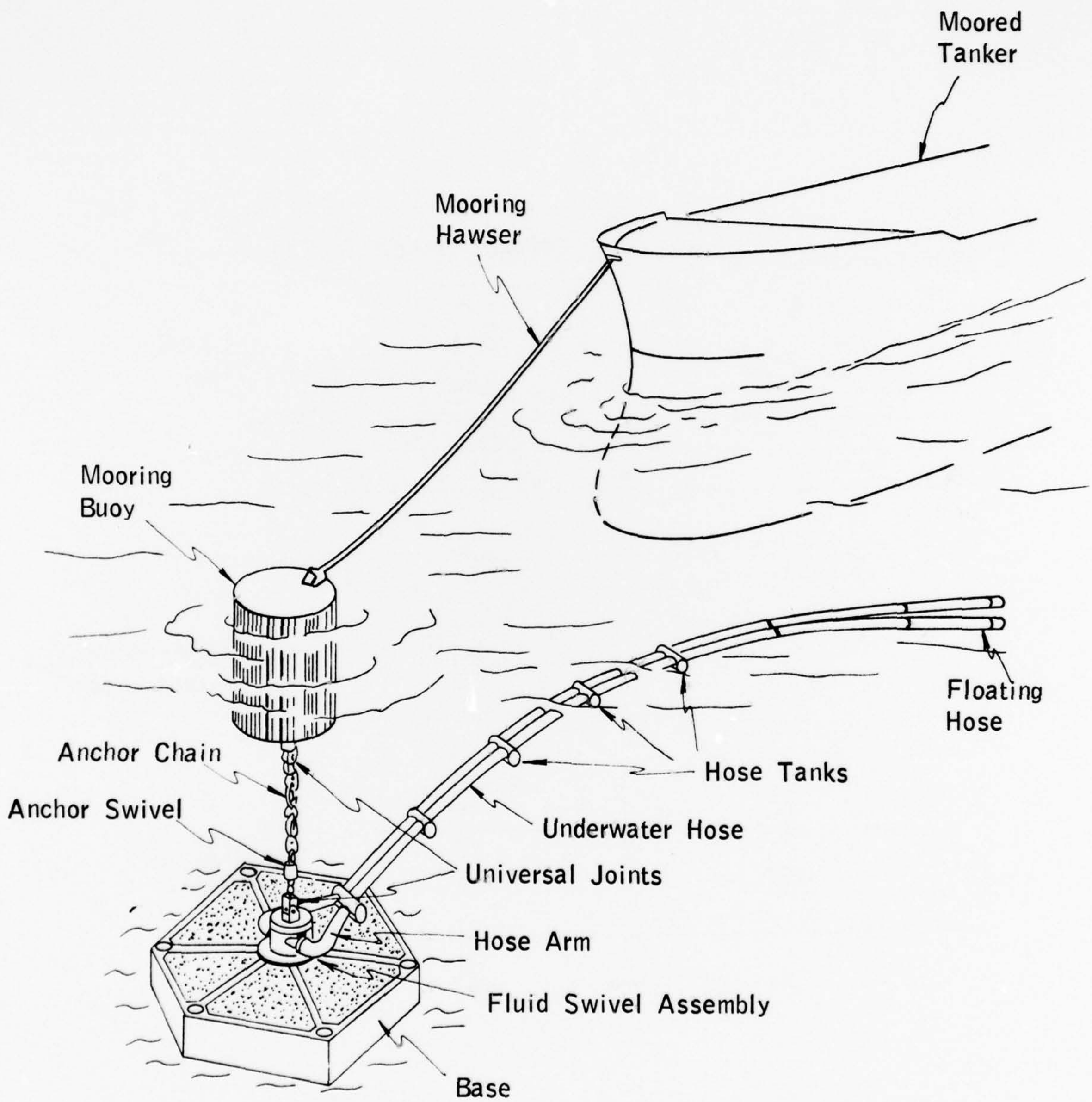


Figure 3-10 - SINGLE ANCHOR LEG MOORING

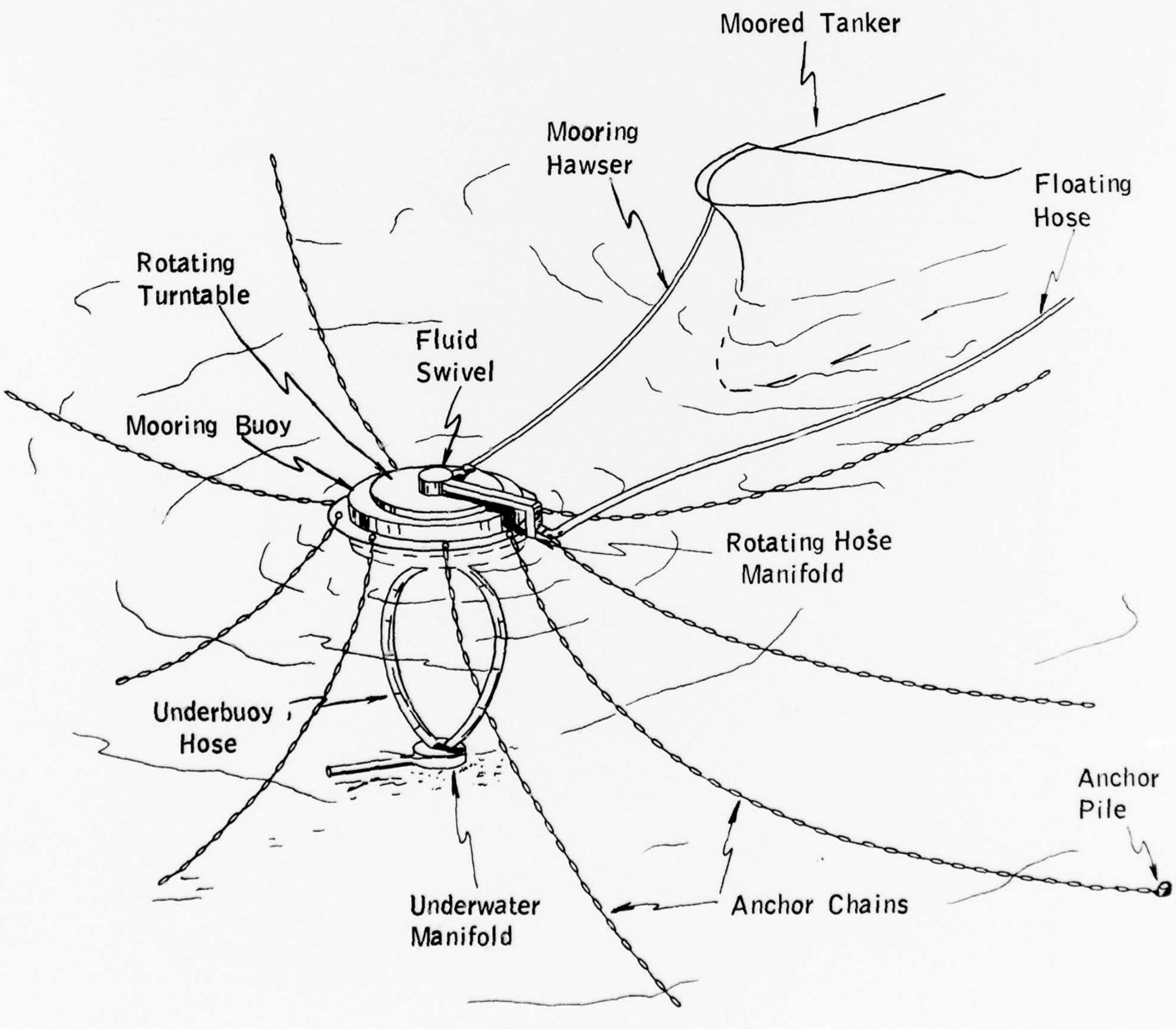


Figure 3-11 - CATENARY ANCHOR LEG MOORING

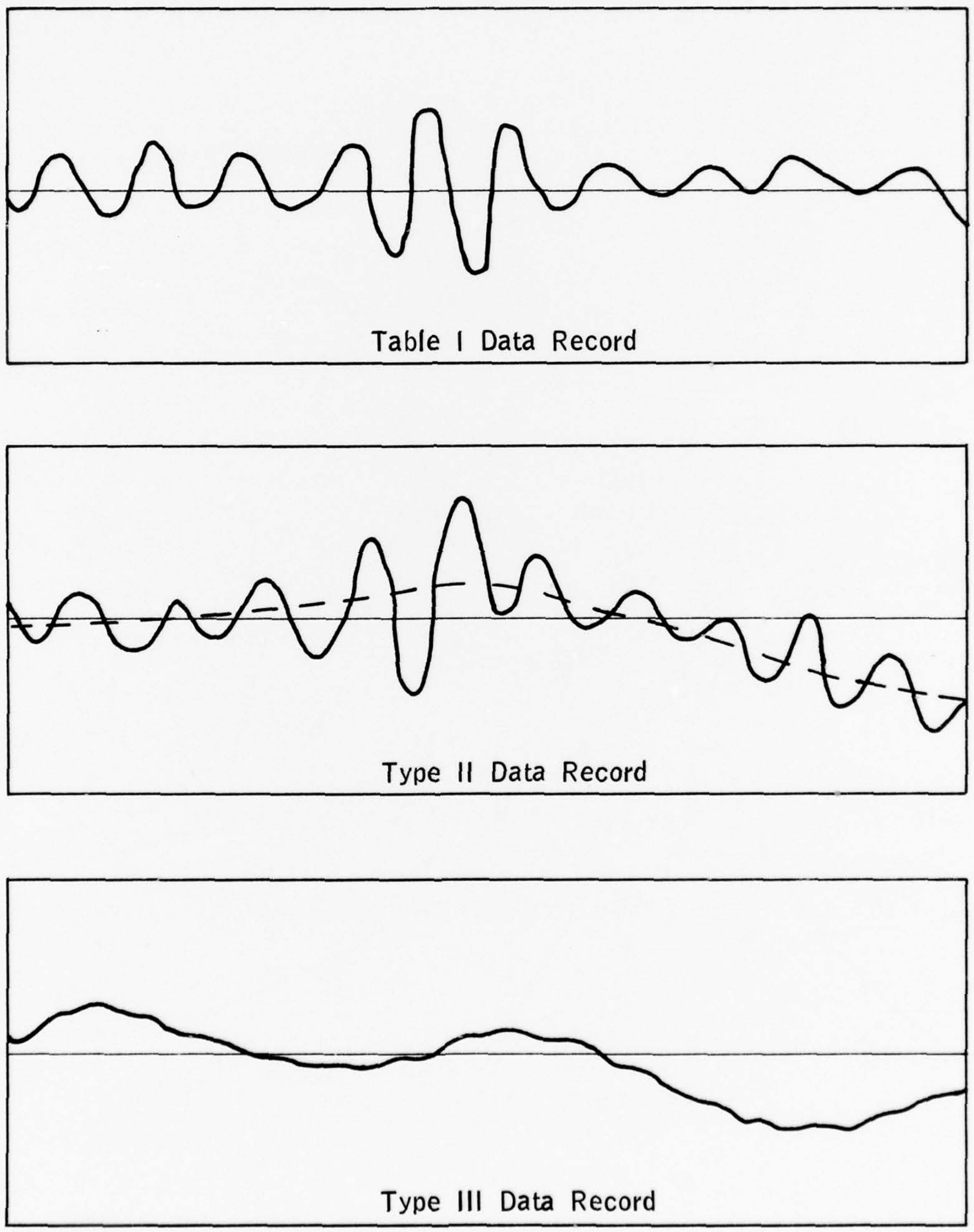
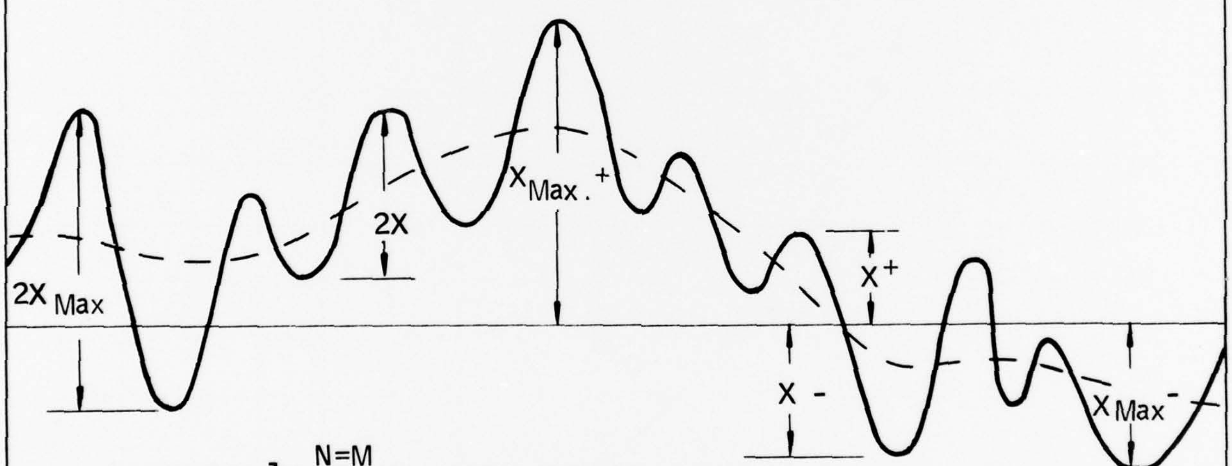


Figure 3-12 - TYPICAL EXAMPLE PLOTS OF THREE TYPES OF DATA RECORD



Mean Value $\bar{X} = \frac{1}{N} \sum_{N-1}^{N=M} X_N$, M = Number Of Samples

Significant Values X_S-, X_S+, $2X_S$, Mean Of Largest One-Third Values

Figure 3-13 - ILLUSTRATION OF TERMS USED IN DATA ANALYSIS

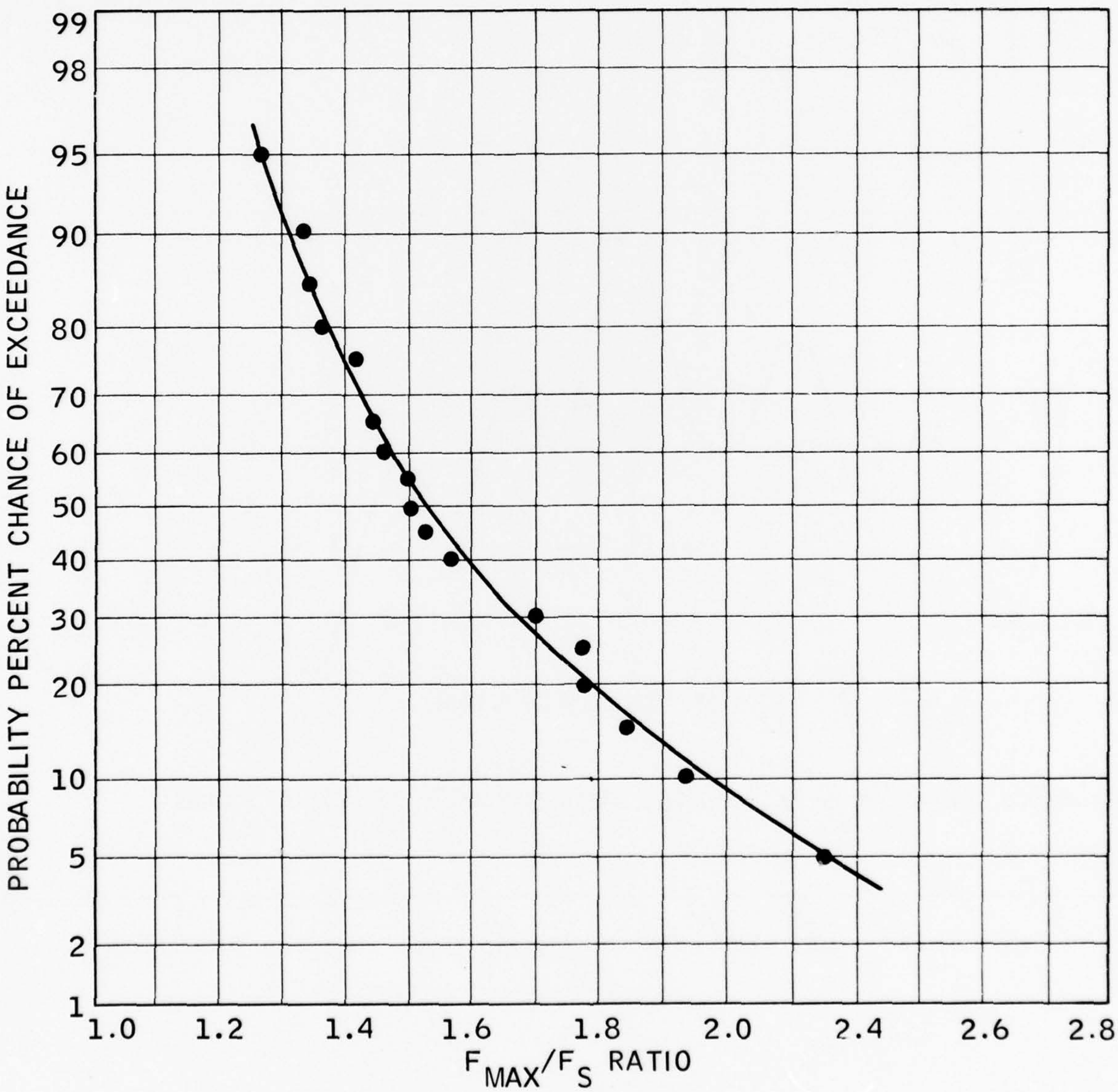


Figure 3-14 - EXAMPLE OF MOORING LOAD DATA PLOTTED BY GAUSIAN PROBABILITY THEORY

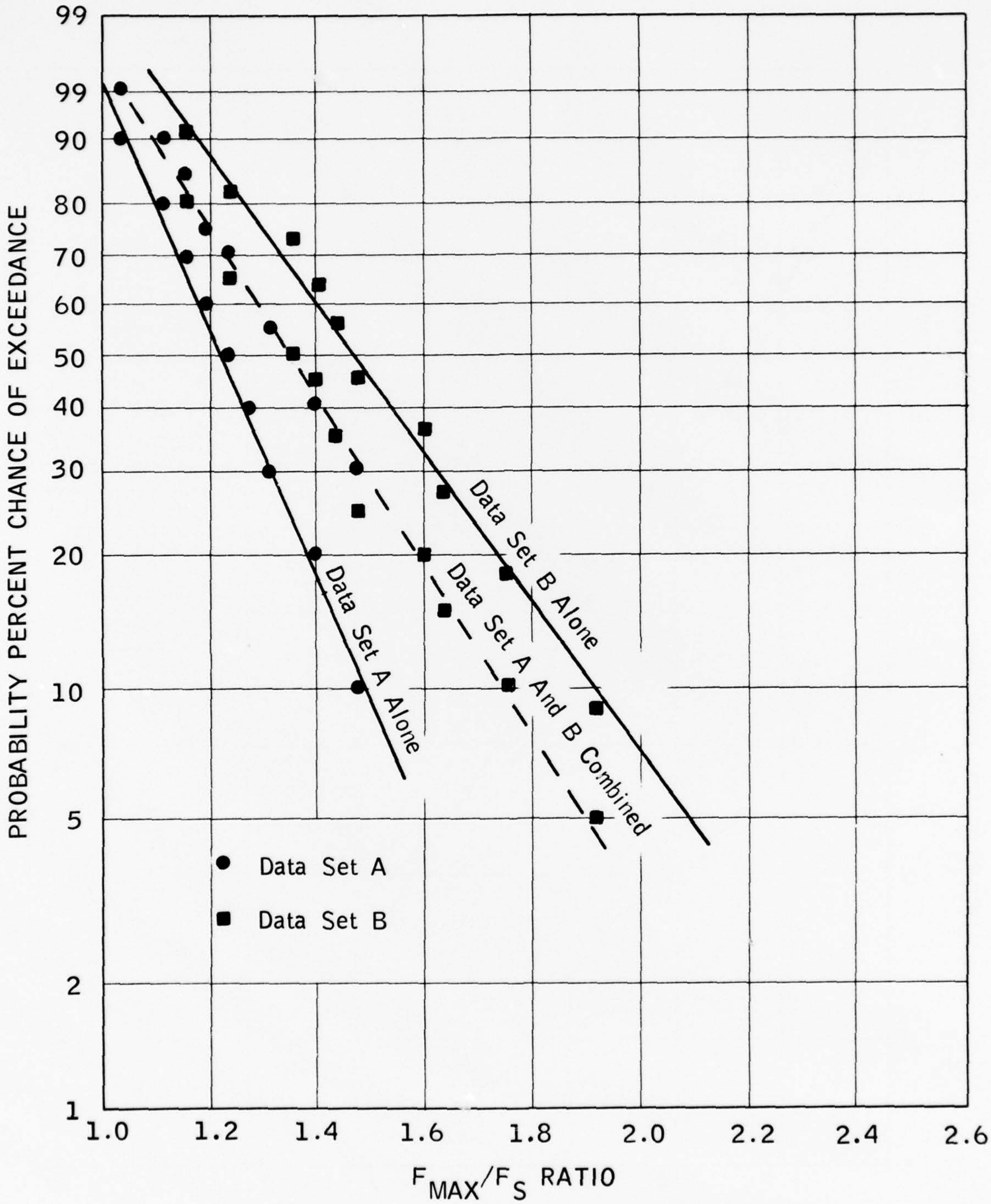


Figure 3-15 - EXAMPLE OF TWO DISTINCT SETS OF DATA PLOTTED ALONE AND COMBINED

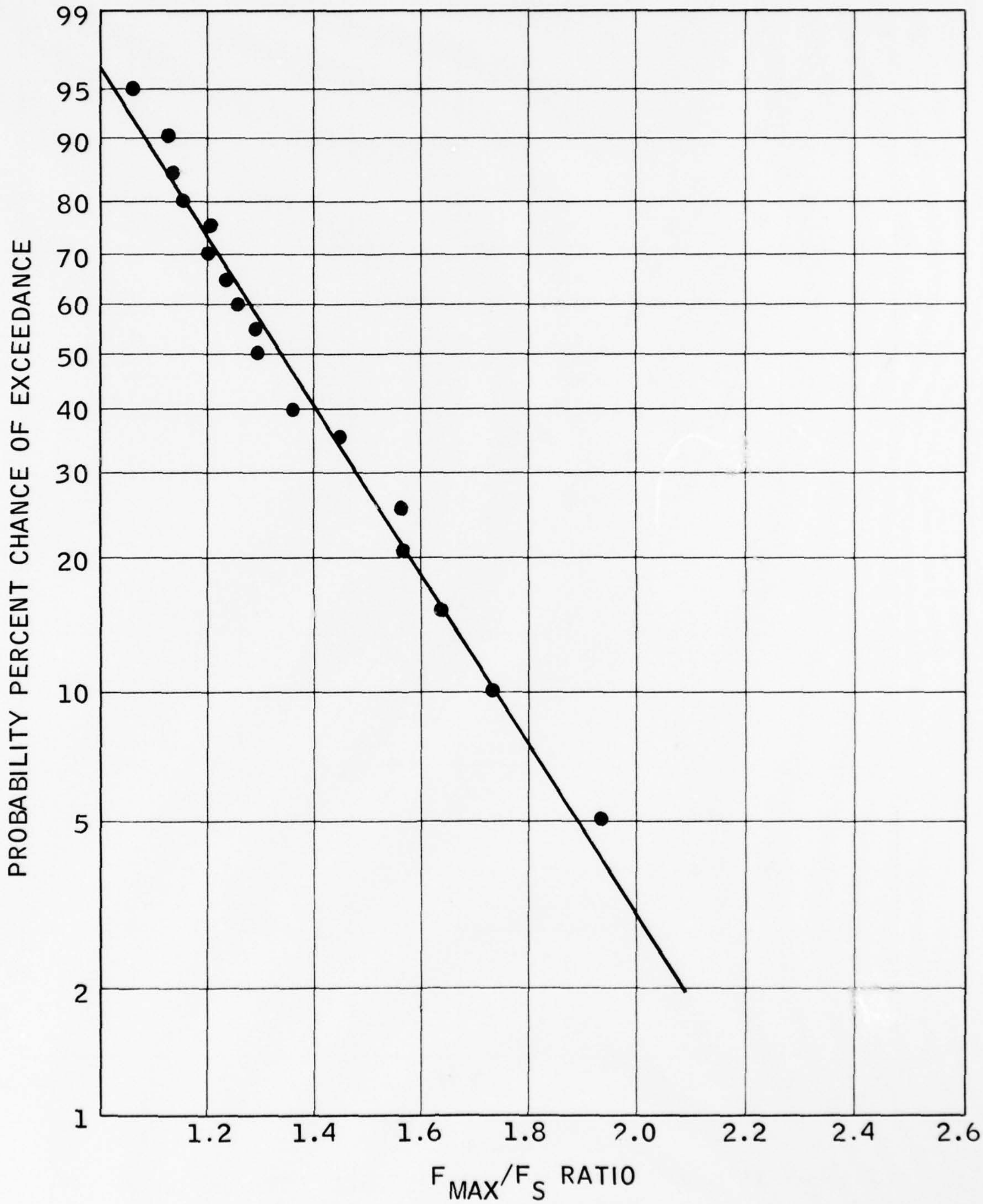


Figure 3-16 EXAMPLE OF MOORING LOAD DATA PLOTTED BY GUMBEL PROBABILITY THEORY

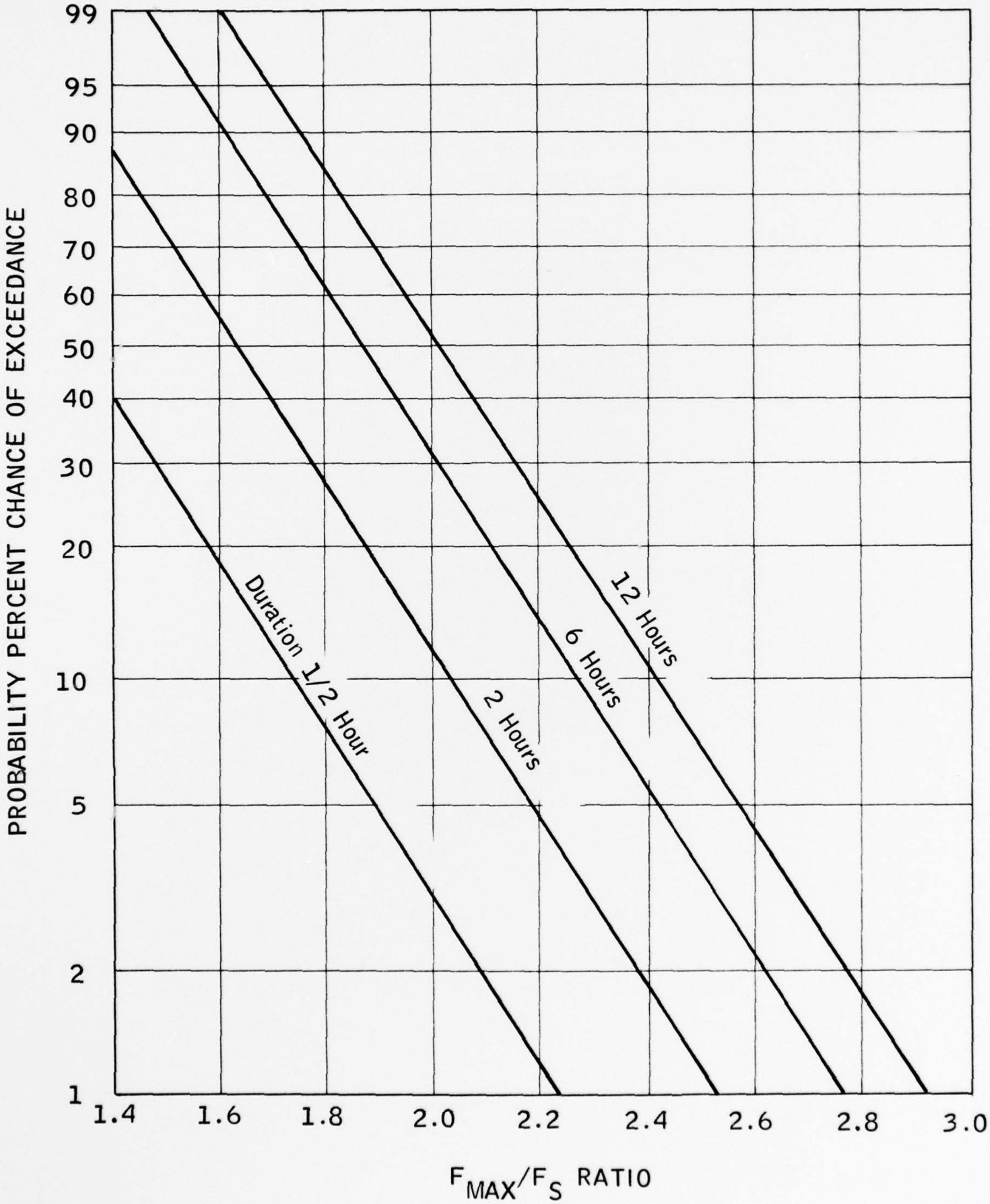


Figure 3-17 - EXAMPLE OF MOORING LOAD PROBABILITY PLOT EXTENDED TO LONGER DURATIONS

SECTION 4

FACTORS WHICH MAY LIMIT MAXIMUM PERMISSIBLE MOORING LOADS

4.1 INTRODUCTION

Designing the SPM system for the mooring loads which can be expected for a range of vessel sizes in various operating environments is not sufficient to insure a safe mooring. Incidents have been reported of tankers breaking out of SPMs due to the failure of shipboard mooring fittings. It is important that deepwater port designers and operators recognize that the type, location, and quality of the mooring fittings aboard the vessels using the port may limit the maximum permissible mooring loads.

Although there has been some effort within the tanker industry toward standardizing shipboard mooring fittings, there are no overall design standards at this time which govern the design, strength, location, and manner of attachment of these fittings. Even if such standards were adopted immediately they would only apply to tankers built in the future, and older tankers might slowly or never be upgraded to meet such standards. Consequently, deepwater ports which will serve a number of tanker fleets can expect to see a wide variety of arrangements of mooring fittings. In some cases, mooring fittings on the tanker will be inadequate to resist the loads imposed on the system in the maximum operating environment for which the SPM is designed. Even when tanker-mounted mooring equipment is capable of carrying the full design load of the SPM, the SPM hawser assembly may not be fully compatible with the shipboard fittings, and alternative means of securing the hawser may be required which are not as strong. This section will describe the various types of fittings found on tankers and establish guidelines for assessing the load capacity of each type fitting.

In addition to factors which limit the maximum permissable mooring loads associated with shipboard mooring equipment, consideration should be given to factors in the environment which affect crew safety and cargo hose handling. Most shipboard mooring and cargo-handling equipment have traditionally been designed with the conventional fixed pier in mind. These pier facilities are generally in sheltered and protected waters. Until recently, little thought has been given by ship designers to the case of ships moored to SPMs where crews must operate mooring and cargo transfer equipment in relatively severe sea conditions. Guidelines are therefore presented for assessing limits related to operational and human factors.

4.2 MOORING PROCEDURES AT SPMs

In order to have a complete understanding of shipboard mooring fittings and how they may limit the maximum permissable mooring loads, it is helpful to first review a typical SPM mooring operation. Generally, two launches are used when mooring at an SPM: a mooring launch to handle the mooring lines, and a hose launch to handle the cargo hoses.

The launches are normally small, approximately 12 to 18 m (40 to 60 ft) in length, and generally cannot carry out their tasks in significant wave heights greater than about 2 m (6 ft). The use of larger launches, perferably with twin screws, thruster units, or other manuvering aids, would increase launch capabilities. Special techniques could also be developed to enhance and extend the crew's ability to perform line-handling functions in higher sea states. In this regard, Gulf Coast workboat operators have stated they believe they can service SPMs in waves up to about 3 m (10 ft) significant height.

The following description is of a typical mooring operation at an SPM. Practices differ at some SPMs. The description is not intended to represent a guide to the best practices, but only as background. Mooring procedures should be developed by tanker captains, pilots, and engineers familiar with the local situation and the design of the specific SPM. A typical mooring operation is shown in Figure 4-1.

4.2.1 Approaching the Mooring

Prior to the approach of the tanker, the launches inspect the hoses and mooring lines and untangle and arrange them if necessary. The mooring launch delivers the pilot and any special tools or fittings required for the mooring to the tanker before it approaches the SPM.

During the mooring approach, the hose launch pulls the hose string to the side, clear of the path which the tanker will take to the buoy. As the hoses are normally attached to the port manifold of the tanker, they are pulled to the port side during the approach. The mooring launch then takes a messenger line from the tanker forecastle and proceeds ahead of the tanker toward the buoy. The approach is usually made along the direction at which the tanker is expected to lie when moored to the SPM. The tanker usually steers a course such that the buoy will pass to port.

The mooring launch attaches the messenger line to the floating pick-up rope connected to the chafing chain at the end of the SPM hawser. The mooring launch then stands off as the tanker continues to approach the buoy. The tanker winches in the messenger line on the gypsy head of its anchor windlass as it closes on the buoy. The tanker should be losing headway as it approaches the buoy and should be nearly dead in the water when the hawser chafing chain is finally winched aboard. The chafing chain is then made fast to fittings on the forecastle of the tanker. In some cases there are two hawsers. Once the tanker has stopped and the first hawser is secured, the launch takes a messenger line from the tanker forecastle and attaches it to the pick-up line of the second hawser. The second hawser is then winched in and made fast.

After the tanker is moored to the buoy, the hose launch brings the end of the hose string(s) to the side of the tanker at the midship manifold. A line from the tanker boom is attached to the end of the hose string or to a tag line on the end of the hose string. The end of the hose is then lifted over the rail and fastened to the tanker's manifold. If two hose strings are provided, the second hose string is then attached to the line from the tanker boom, lifted over the rail, and fastened to the manifold. As pumping begins, one of the launches makes a final inspection of the hose string, the buoy, and the hawser to make certain there are no leaks or other difficulties.

4.2.2 Departing the Mooring

At the completion of cargo transfer operations or in the event of deteriorating weather or other emergencies, the tanker disconnects the cargo hoses and lowers them over the side. The tanker then steams ahead very slowly to remove strain from the hawser system as the hawsers are disconnected from the ship's fittings and lowered to the water. The ship drops back from the mooring to clear the hoses and then proceeds ahead and out of the mooring area.

Although the SPM launches are helpful and generally standby to aid the tanker in disengaging from the mooring, their services are not essential when departing the mooring. Even in sea conditions too severe to allow the launches to operate, the tanker can still safely disengage and leave the mooring.

4.3 TYPES OF SHIPBOARD MOORING FITTINGS

In general, there are three types of fittings used to secure SPM hawsers to the forecastle deck of the tanker. These are mooring brackets, chain stoppers, and mooring bitts. Mooring brackets, often called Smit or towing brackets, and chain stoppers are specially designed to serve as attachment points for the hawser chafing chain and are designed to tranfer the mooring forces to the ship's structure. Mooring bitts are also used for this purpose but are not designed to receive the chafing chain directly. When using mooring bitts, a short line, called a snotter, is secured to the bitts and attached to the end of the chafing chain.

The hawser chafing chain is brought aboard the tanker through chocks, small openings in the ship's bulwark, usually at or near the center of the bow. The ship's winches are used to lift the chafing chains to the forecastle and draw them to the mooring fittings. Fairleads serve as guiding points to lead the messenger lines and pick-up lines from the chocks to the winches. Each of these fittings will be discussed in more detail in the following sub-sections. Table 4-1 summarizes the various types of fittings which can be expected to be found on various size tankers.

4.3.1 Chocks

Chocks are fittings designed to allow lines or hawsers to pass through the ship's railing or bulwark near the deck level. Bow chocks are usually mounted near the bow on the forecastle deck in pairs symetrical about the vessel center-line. Many newer vessels are fitted with a pair of closely spaced bow chocks between 1 to 1.5 m (3 to 5 ft) on either side of the vessel's center-line for use in mooring to SPMs. This is an ideal arrangement for mooring to SPMs. Some older and smaller vessels have bow chocks spaced as much as 15 m (50 ft) on either side of the vessel's center-line. A single large chock, fitted on the vessel's center-line, is sometimes called a Panama chock. It is common practice in the maritime industry to use the name fairlead interchangeable with chock.

The two principal types of chocks found on tankers are closed chocks and roller chocks. Roller chocks are normally provided to reduce wear on lines or hawsers when moored to a conventional fixed pier. A roller chock consists of a steel frame in which are mounted two, and in some cases three, vertical steel rollers on bronze-bushed, lubricated steel pins. Some roller chocks consist of two vertical rollers and two horizontal rollers forming a square opening through which the line passes. Roller chocks are frequently fitted on the forecastle, often close to the bow. However, roller chocks are poorly suited for SPM hawser assemblies as the SPM chafing chain can easily damage the rollers.

Closed chocks, when mounted on the forecastle, are usually called bow chocks. Bow chocks, shown in Figure 4-2, consisting of heavy rings welded into the bulwark, are more suited for SPM hawser assemblies. The minimum bow-chock opening for convenient handling of the chafing chain is about 600 x 450 mm (24 x 18 in.). This size chock will easily pass the largest size of chafing chain which is normally used at SPMs. Chocks which have a minimum opening of 400 mm (16 in.) or smaller may restrict the size of the chafing chain which can pass through the opening. Many older and smaller tankers have smaller chocks and cannot accept chain larger than about 64 mm (2 1/2 in.) diameter.

Bow chocks normally have oblong or oval openings. Two chafing chains will usually fit through a single bow chock. However, the first chain brought aboard tends to center itself in the opening, thus blocking entry of the second chain. For this reason, two closely spaced bow chocks are preferred when dual hawser systems are used.

A special type of bow chock, shown in Figure 4-3 and usually called a bow fairlead, is sometimes fitted on the centerline at the bow of large tankers. It is an ideal chock for SPMs which use single mooring lines because the large radius plates reduce the bending radius and the wear in synthetic hawsers or chafing chains. Also, the large opening allows easier handling of large hawser assemblies which are required to resist the high mooring loads at SPMs. This type of bow fairlead is commonly fitted on tankers which are modified to perform self-service mooring at SPMs in offshore production fields. Two lines can normally be brought through this type fitting.

4.3.2 Mooring Bitts

Mooring bitts, shown in Figure 4-4, are common mooring fittings found on all tankers. They are fitted at convenient locations on open decks to secure mooring lines and rigging lines. They are usually fitted near the edge of decks about 1 m (3 ft) inboard of the ship's side. They are normally located adjacent to and in line with chocks or fairleads on the deck and at convenient locations on the bow and stern. Bitts fitted near the ship's center line on the forecastle deck are used to secure SPM hawsers on smaller and older tankers which are not equipped with mooring brackets or chain stoppers for this purpose. Figure 4-5 shows a typical arrangement of bitts on the forecastle deck of an 80,000 dwt tanker.

Mooring bitts normally consist of two vertical, hollow, steel cylinders mounted on a base plate attached to the deck. They are normally fabricated of welded-steel plate and pipe. Suitable underdeck reinforcement should be provided under the mooring bitts' base plate. A slightly oversized cap is fitted on each cylinder to prevent the line or hawser from riding off the top of the bitts.

A similar fitting, consisting of a single vertical post, is often found on docks and piers for tying off lines from ships. This fitting is called a bollard. It is common practice in the marine industry to refer to the twin-post mooring bitts found on ships as bollards and many references use the two terms interchangeably. In this study, however, only the term mooring bitts will be used.

A line or hawser is normally secured to mooring bitts by "figure-eighting" it around the vertical cylinders until sufficient turns have been taken to hold the line or hawser by friction. The proper procedure to secure the SPM mooring line to mooring bitts using synthetic or wire-rope snotters is discussed in more detail in subsection 4.7.2.

4.3.3 Mooring Brackets

Most newer tankers larger than about 120,000 dwt are fitted with one or two mooring brackets or chain stoppers. Tankers above the range of 200,000 to 275,000 dwt are usually equipped with two of these fittings.

The mooring bracket, shown in Figure 4-6, is a device designed to directly receive and secure the end of the SPM hawser chafing chain. Mooring brackets should be mounted in line with and between 2.75 and 3.75 m (9 to 12 ft) behind the bow chocks.

The mooring bracket consists of two side plates mounted on a deck plate which is welded or bolted to the deck. The end link of the chafing chain is placed between the side walls and a sliding bolt, passing through the side walls, is inserted through the end link of the chafing chain to secure the SPM hawser. The sliding bolt is locked in place by a pin and is driven into the open or closed position by striking a short stud welded to the top of the bolt with a heavy hammer or sledge. This operation allows the mooring bracket to be disengaged under tension in an emergency. Disengaging the mooring bracket with a chain under tension, however, is a hazardous operation as the friction developed between the chain and bolt may require heavy and repeated blows before the chain releases. Injuries have been reported when releasing a mooring bracket under load. SPM operators recommend that the tension be removed from the hawser asembly before releasing the chafing chain from the mooring bracket.

In order to fit into and be secured by a mooring bracket, the end link of the mooring hawser chafing chain must be either a pear-shaped link or an open-link. Subsection 5.9 discusses in more detail the construction and connection of chafing chains. Unfortunately, there are no common design standards for mooring brackets, and brackets having different dimensions are frequently encountered. The space between the side plates and the cross-section of the bolt vary. Some have openings so narrow they will not accept chain large enough to provide the necessary strength required for high SPM mooring loads. Others have pins which are larger than the opening in standard end links and require special arrangements to secure the chain. Some mooring brackets are not properly located on the vessels forecastle to provide the most effective mooring arrangement.

4.3.4 Chain Stoppers

Chain stoppers consist of two side plates mounted on a base plate with a bar or pawl pivoted so it can be lowered to secure the chain. The chain is drawn between the side plates such that every-other link rests horizontally on guide plates and the other links pass vertically through a gap between the guide plates. In the pawl-type chain stopper, shown in Figure 4-7, the pawl is pivoted on a pin between the two side plates and is raised or lowered by a lever. With the pawl-type chain stopper the chain must be threaded between the plates and under the pawl.

The bar in the bar-type chain stopper, shown in Figure 4-8, is pivoted on one side plate and falls into a slot in the other side plate to secure the chain. The bar pivots up and to the side to disengage the chain. The bar is usually counterweighted. With the bar-type chain stopper the messenger line or pick-up rope may be lifted and lowered between the plates so there is no need to thread through the chain stopper.

Most chain stoppers which are intended for securing SPM hawser chafing chains are designed for 76 mm (3 in.) stud-link chain. However, some chain stoppers will not pass the standard enlarged open end link on the end of the chain and this causes difficulties in securing chafing chains designed for mooring brackets.

4.3.5 Fairleads and Winches

Fairleads are fittings which are used to change the direction of a mooring line or hawser on the deck of the vessel. Fairleads are normally of the pedestal or roller type shown in Figure 4-9. A roller fairlead consists of a single roller with bronze bushing set on a vertical pin welded or bolted to the deck.

A fairlead should be placed behind each chain stopper or mooring bracket and slightly off center of a line through the stopper or bracket and the bow chock to allow the messenger line to properly center the chafing chain in the fittings. The offset should be to the side which the rope will lead to the winch so that the messenger line runs in a straight line from the bow chock through the chain stopper or mooring bracket to the fairlead as shown in Figure 4-10.

Tankers are typically equipped with mooring winches on the forecastle for hauling in and holding mooring lines or hawsers when the ship is berthed at a conventional fixed pier. These mooring winches, however, are usually not suitably located for use in hauling in an SPM mooring line. In most cases, the gypsy head on the anchor windlass is used to haul in the SPM mooring line. The gypsy head is a cylinder-like fitting mounted on the end of the anchor windlass shaft. The SPM hawser is hauled in by winding a few turns of the messenger or pick-up rope around the gypsy head, and holding the free end taut manually as the gypsy head turns. A typical arrangement of deck equipment for a 80,000 dwt tanker is shown in Figure 4-5.

4.4 STRENGTH OF SHIPBOARD FITTINGS

Although most components of a deepwater port mooring system may be designed to withstand very high loads, the maximum permissible mooring load will depend on the weakest link in the system. In many cases, this weakest link may be the shipboard mooring fittings. The following sections discuss the strength of individual fittings in more detail.

4.4.1 Strength of Bitts

Well designed, fabricated, and installed mooring bitts on new vessels generally meet the following criteria for safe working loads for various diameter bitts:

Bitt Diameter		Safe Working Loads	
millimeters	inches	k newtons	pounds
500	20	1,520	340,000
550	22	1,864	420,000
600	24	2,452	550,000

Safe working load is defined as one-half the ultimate capacity of the bitts.

Because construction and design defects may not be visible or because internal corrosion may be present, it is often impossible to judge the strength of bitts, especially on older vessels. The strength depends primarily on the method used to attach the bitts to the ship's structure. Properly designed and constructed bitts extend through the baseplate and are welded to both the baseplate and the deck as shown in Figure 4-11. Bitts which are only butted against and welded to the baseplate which is then welded to the ship's deck will have a lower load capacity.

The method of attachment cannot be determined by visual examination. However, the construction may be determined by drilling through the bitt and baseplate to examine the internal structure of the mooring bitt. X-ray examination and ultrasonic inspection are other methods which have been suggested for determining the nature of bitt construction. Examination of the original shipyard drawings may show the method of attachment, however in many cases drawings showing construction details of items such as bitts are not available for older vessels.

4.4.2 Strength of Chain Stoppers and Mooring Brackets

Chain stoppers and mooring brackets are normally designed to the breaking strength of the chain they are sized to accept. Occasionally, they are designed to the proof load (two-thirds of breaking strength) of the chain they are sized to accept. The maximum permissable loads developed in this section are based on the assumption that the fittings are designed to proof loads.

Using grade 3, stud-link chain to establish the required minimum strength of these fittings, a chain stopper or mooring bracket designed to the proof load of 76 mm (3 in.) chain would have a maximum load capacity of about 2880 kN (640,000 lb). A fitting designed to the proof load of 54 mm (2 1/8 in.) chain would have a maximum load capacity of about 1,420 kN (318,000 lb).

In practice, however, chain-stopper and mooring-bracket maximum design loads vary among tanker fleets and mooring equipment manufacturers. Exxon, for example, has developed a design standard for mooring brackets which calls for a maximum design load of 3470 kN (780,000 lb) for a fitting sized to accept 76 mm (3 in.) chain. In almost every case chain stoppers or mooring brackets will be more than adequate to resist the safe working loads of the chains they are designed to accept.

Unlike mooring bitts, chain stoppers and mooring brackets are installed specifically to resist the high loads encountered at SPMs. The designers normally insure that these fittings are structurally adequate. The designers, however, have no control over shipyard practices, and some chain stoppers and mooring brackets, although sufficiently strong themselves, may not be properly connected to the ship.

4.5 ARRANGEMENT OF SHIPBOARD FITTINGS

The location of chocks, bitts, chain stoppers, and mooring brackets can have an effect on the loads experienced and thus should be taken into account when determining the maximum permissible load for a specific vessel.

4.5.1 Spacing of Bow Chocks

When two mooring hawsers are necessary to provide sufficient strength to resist the total mooring load, the spacing of the bow chocks will influence the maximum load experienced in each hawser. If the bow chocks are widely spaced, a high percentage of the total mooring load will be carried by first one hawser and then the other as the vessel yaws from side to side. This type of motion commonly occurs at SPMs and has frequently been observed in both model tests and at actual SPM facilities. The causes of this type of behavior are discussed in subsection 2.4.1.

The degree of sharing of loads in dual hawsers is a function of the spacing of the chocks on the bow of the tanker and the amount of yaw of the tanker. This can be clearly seen in Figure 4-12 where the difference in hawser elongation is shown to be roughly equal to the bow-chock spacing times the sine of the yaw angle for small angles of yaw.

Through model testing, ARAMCO found that dual mooring lines are essentially as effective as a single mooring line provided the chocks are within 2 m (6 ft) of each other (Flory, April, 1977). ER&E has made similar observations through model testing. In model tests with dual mooring lines run to chocks spaced approximately 3 m (10 ft) apart on the bow, ER&E found that the peak mooring loads were shared almost equally, but in the same cross-current environment with the chocks spaced approximately 30 m (100 ft) apart on the bow the higher loaded line took almost 75% of the peak mooring load. Shell has stated that in a two-hawser system the higher loaded line may take two-thirds of the total mooring load (Langeveld, 1975).

To avoid this unequal sharing of loads between dual hawsers on tankers with widely spaced bow chocks, it is preferable to bring both hawsers to one or a pair of fairleads on one side of the center of the bow.

4.5.2 Location of Mooring Fittings on Forecastle Deck

The location of mooring bitts, chain stoppers, and mooring brackets on the forecastle deck is important to insure that they equally share in carrying the total mooring load.

On many vessels, mooring bitts are poorly located for SPM operations, especially when two mooring hawsers are required to resist the total mooring load. Figure 4-13 shows a typical method of using snotters to tie-off the chafing chains of a dual hawser system. Note that because of the location of the mooring bitts, the two snotters required for each chain are of unequal length. Under load both snotters elongate approximately the same amount, but the shorter snotter takes a higher percentage of the load. Also, the angles between the line of the chain and the snotters are unequal. In this example, these two factors result in the nearer bitts carrying a higher share of the load. The unequal length of the snotters is more significant with synthetic snotters because of their non-linear elongation characteristics. If wire snotters were used, the load-elongation characteristic is more linear and thus, this effect is less pronounced.

The location of mooring bitts varies considerably from vessel to vessel and it is difficult to present general guidelines on how to adjust the maximum permissible mooring loads accordingly. SPM operators should be aware of these problems, however, and be able to recognize those cases where one mooring bitt will have to carry a higher share of the load.

Chain stoppers and mooring brackets are specifically mounted for SPM operations and are normally placed an equal distance behind the bow chocks and symetrically about the vessels centerline so that these problems do not occur. SPM operators should be alert, however, for the rare vessel which may have fittings unequally spaced behind the bow chocks. If this occurs, the length of the chafing chains between the hawsers and the fittings should be adjusted, if possible, so that both hawsers share the strain equally.

4.6 MAXIMUM LOADS FOR SHIPBOARD MOORING FITTINGS

It should be evident from the preceeding discussions that the strength of shipboard mooring fittings is highly variable. Not only have various ship designers used different criteria for establishing the strength of the fittings, but shipyards have used different methods of attaching the fittings to the ship's structure. It is prudent, therefore, to establish conservative guidelines without unduely penalizing cargo transfer operations.

Table 4-2, "Maximum Loads for Shipboard Fittings" gives recommended maximum mooring loads for mooring bitts, chain stoppers, and mooring brackets. Loads for two catagories of construction are given depending on the method used to attach the fitting to the ship's structure.

For mooring bitts, "Good Construction" refers to bitts extending through the base plate and welded to both the base plate and the ship's deck with adequate underdeck reinforcement as discussed in Section 4.3.1. The method of attachment cannot be determined by visual examination. Therefore, the loads given in the column "Questionable Construction" should be used to establish the maximum load for mooring bitts unless the ship's personnel can present evidence showing that the bitts are properly designed and attached to the ship's structure.

For chain stoppers or mooring brackets, the loads given under the column "Good Construction" should be used unless inspection reveals that the fitting is poorly constructed or is mounted with inadequate or no underdeck reinforcement. In these cases, the loads listed under "Questionable Construction" should be used.

The loads given in Table 4-2 only apply to well-maintained fittings. For fittings which appear corroded or have split or broken welds, these loads should be reduced. The amount of load reduction is a matter of subjective judgement and would depend on the actual amount of damage to the fittings. If some corrosion or weld damage is visible however, other damage may be present which can't be seen. Therefore, a conservative reduction factor of between 1/3 and 1/2 should be used depending on the degree of corrosion or other damage evident. Fittings which are severely corroded or have evidence of poor welds or severe damage should not be used as it is impossible to estimate the strength of these fittings.

4.7 COMPATIBILITY OF HAWSER ASSEMBLY WITH MOORING FITTINGS

The hawser assembly is defined as the hawser or hawsers, the tanker-end and buoy-end chafing chains, and all associated thimbles, shackles, connecting pieces, floats or buoys, and pick-up ropes. The design of the hawser assembly is covered in detail in Section 5 of this report.

Because of the wide variety of shipboard mooring fittings currently installed on the world's tanker fleet, it is difficult to design the tanker-end chafing chain system of the mooring hawser assembly to be completely compatible with all shipboard mooring fittings which might be encountered. The Oil Companies International Marine Forum (OCIMF) has been addressing this problem for a number of years is currently working on standards for shipboard SPM mooring fittings and SPM hawser chafing chain arrangements.

4.7.1 Attaching to Chain Stoppers and Mooring Brackets

Before attaching the chafing chain to a mooring bracket it is usually necessary to unshackle the pick-up rope from the end link. A ship's wire on a winch may be shackled to the triangle plate of the chafing chain or the chafing chain may be stopped off in some other manner. The tension is removed from the pick-up rope, and the pick-up rope is unshackled from the open link. The open link is then placed between the plates of the mooring bracket and the pin is driven through the open link. Once the chafing chain is secured to the bracket, tension in the ship's wire is relaxed, and the wire is unshackled so the chafing chain can be released quickly in an emergency.

The design of mooring brackets found on tankers vary. Some brackets have large pins which require special large open links on the end of the chafing chains. Other mooring brackets have limited clearance between the side plates, and standard open links cannot be inserted. For these tankers, the chafing chains may have to be attached to mooring bitts as described in subsection 4.7.2.

Recently, several terminals have taken up the practice of placing an open link and shackle in the mooring bracket before the mooring approach is made, and then shackling the end of the chafing chain to this open link or to a short length of chain connected to this open link. Terminals that have adopted this practice find it is more efficient and safer than the normal method of mooring to a mooring bracket.

OCIMF is preparing recommendations which call for all new tankers to be outfitted with chain stoppers. There are several different chain stopper designs. The design may place limitations on the chafing chain design and dictate how the chafing chain is handled as explained in subsection 4.3.4.

Some chain stoppers which are otherwise designed for 76 mm (3 in.) chain cannot pass the enlarged open link usually fitted on the end of the chafing chain. When such chain stoppers are encountered, it may be necessary to place a short length of extra chain in the chain stopper and shackle this to the chafing chain.

When attaching dual hawser assemblies to chain stoppers, care should be taken to insure that the length of chafing chains between the hawsers and the fittings are adjusted so that both hawsers share the strain equally.

4.7.2 Attaching to Mooring Bitts

On tankers which are not fitted with chain stoppers or mooring brackets or if the chain is improperly sized to fit the chain stoppers or mooring brackets, it will be necessary to connect the chafing chain to mooring bitts. This is usually done with synthetic-rope or wire-rope snotters. A snotter is a short length of line with an eye thimble at one end and a large diameter soft-eye at the other end.

The proper procedure to use a snotter is to shackle the eye-thimble end of the snotter to a triangular plate or end link in the chafing chain, pull the snotter tight, and figure-eight it about the mooring bitts. The snotter should always lead from the bottom of the bitts to the chafing chain as this minimizes the bending moment applied to the bitts. At least four or five complete turns should be taken about the bitt, preferably more if there is room. Properly sized and designed snotters allow the moored vessel to utilize the full strength rating of its mooring bitts. SPM operators have generally found that about 25 m (85 ft) is a convenient length of a snotter for most ships.

The ability of the deck crew to conveniently handle wire-rope snotters limits the size wire to about 54 mm (2 1/8 in.) diameter. The average breaking strength of a 54 mm, solid-core, wire rope is about 1950 kN (440,000 lb). An accepted practice is to limit the load in a wire snotter to 55 to 60% of the new breaking strength. This allows for strength reduction due to splices, bending, and deterioration. Since the snotters are easily accessible and can be frequently inspected for wear or damage, this factor is acceptable. By this criteria, each snotter is capable of carrying about 1150 kN (260,000 lb). It is thus usually necessary to use at least two wire-rope snotters per hawser assembly to carry high mooring loads. Different size or strength wire rope may, of course, require more or fewer snotters to maintain the same load.

Many operators dislike using wire rope snotters because of the inconvenience of handling them. Wire ropes are stiff and cumbersome and must be kept well greased to avoid deterioration. After a wire rope has been in service for a short time, it begins to develop short, broken wire strands which can cut or stick into the deck crew's hands causing minor, but painful, injuries. Some operators, therefore, use synthetic snotters to connect the chafing chains to the mooring bitts.

Nylon, because of its high strength, is often used for synthetic rope snotters. Using a factor of safety of 50% of new breaking strength, two nylon snotters of 96 mm diameter (12 in. circumference) would be required to maintain the same load capacity as two 54 mm wire rope snotters. Because of the high-elasticity characteristics of nylon, it is likely that the two snotters would not evenly share the total load. Also, with a high-elasticity snotter the amount of movement, and thus, rate of wear, of the chafing chain in the bow chock as the hawser is alternatively loaded and unloaded will be more than with a relatively stiff wire snotter. In addition, with a nylon snotter, should the hawser or snotter fail under load, the high elasticity will cause a violent snap-back which could injure deck personnel in the vicinity. Nylon is also susceptible to degradation by rust, which is likely to be encountered on poorly-maintained vessels.

Although not as commonly used, polyester is preferable to nylon for synthetic-rope snotters. Polyester has lower elasticity than nylon and is not as susceptible to degradation by rust. The strength of polyester is slightly lower than nylon and, therefore, slightly larger snotters may be required.

Use of a single snotter is not usually practical because the large size which would be required would present handling difficulties for the deck crews. SPM operators generally agree that 112 to 128 mm diameter (14 to 16 in. circumference) synthetic snotters are as large as can be conveniently handled.

4.8 OPERATIONAL AND HUMAN FACTOR LIMITATIONS

In addition to factors limiting the maximum permissible mooring loads associated with shipboard mooring equipment, consideration should be given to crew safety and cargo-hose handling. The following subsections discuss how these factors may limit the maximum environment in which SPMs may be operated.

4.8.1 Maximum Environment In Which Hoses Can Remain Connected

The maximum environment in which a vessel will continue cargo-transfer operation should be slightly lower than the maximum design mooring environment. Although the vessel may be able to remain moored in up to the maximum design environment, the cargo hose should be disconnected and lowered over the side before the mooring system experiences the maximum operational environment. Limitations on the maximum environment in which cargo transfer operations can take place must also take into account considerations for the safety of the crew, problems in handling the hoses, and stresses induced on the hose and the tanker manifold.

Figure 4-14 shows a typical arrangement used in connecting SPM cargo hose to the ships manifold. Under the maximum design environment considerable stresses and motions are induced into the cargo hoses as a result of wave action on that portion of the hoses floating on the water surface. At the same time, the vessel will be experiencing some rolling, pitching, and heaving. The vessel response to the environment depends on the tanker size and will be greater for smaller tankers. Under these conditions, the practical considerations of disconnecting the cargo hoses and lowering them over the side are difficult. The risk of injury to deck personnel is increased as most of this work must be done by hand and the equipment envolved is heavy and cumbersome. It may be impossible to avoid spilling some amount of oil on the deck under these conditions, increasing the hazards associated with the operation.

The maximum environment in which the ship's crew can safely perform the cargo hose disconnection is a matter of subjective judgement and must be decided by the vessel's master who is responsible for his crew. Most SPM operators, however, recommend that the cargo hoses be disconnected when wave heights approach 3.7 m (12 ft) unless the vessel has a specially modified, remotely-controlled, manifold coupling system. To date, only a few vessels have been equipped with such a manifold and these vessels are all dedicated to specific SPM facilities.

In addition to concern for the safety of the deck crew during the cargo hose disconnection, prudent operators recognize that the risk of a mooring system failure is highest when the vessel is moored in the maximum design mooring environment. Should the vessel break out of its moorings with the cargo hoses still connected it is probable that the hoses will be overstressed with possible damage to the hose or other components in the cargo transfer system which may result in cargo spillage and pollution. To avoid this risk, if the hoses have not already been disconnected for the reasons given in the previous paragraph, many operators will order cargo hoses disconnected when the environment approaches about 80% of the maximum mooring environment.

4.8.2 Maximum Environment In Which Hawser Can Remain Attached

Subsections 4.5 and 4.6 have discussed in some detail how the maximum environment in which the vessel may remain moored may be limited by the strength and efficiency of the shipboard mooring fittings. Even when there is no physical danger of a breakout, however, considerations for the safety of the deck crew and the ability of the vessel to disengage safely from the mooring may dictate an early departure if the environmental conditions are deteriorating.

In very severe sea conditions, the vessel may be taking water over the bow. Under these conditions, crew members working to disconnect the hawsers from the shipboard fittings could be injured from a large wave breaking over the forecastle. Communication with the bridge, both visual and radio or telephone, may be poor or impossible due to spray and noise. Efforts to relieve the strain in the hawsers, enabling the crew to safely and efficiently disconnect the chafing chains, may be frustrated by the tendency for the vessel's head to fall off as the ship trys to steam forward.

The vessel's master, as well as the SPM operators, should keep themselves aware of the conditions on the forecastle so that they can disengage before the environment presents a hazard to the deck crew attempting the above operations.

4.9 SUMMARY

In very mild environments where the mooring loads are small, for example less than 1,000 kN (225,000 lb), almost all vessels which may use the mooring will be able to remain moored in the maximum design environment. However, SPM systems designed for more severe environments, especially when large ships are involved, may be designed for mooring loads in excess of 2,000 kN (450,000 lb). In these cases, the strength of the shipboard mooring fittings may require many vessels to leave the mooring before the maximum design environment is reached.

Unfortunately, there is a lack of standardization of shipboard mooring fittings in the world tanker fleet. As a result, a wide variety of types of fittings may be encountered, some of which are underdesigned, poorly fabricated, or incorrectly installed. Visual inspection of these fittings at the time the moor is made cannot always establish the strength of the mooring fittings. This is especially true for mooring bitts, as the strength of the bitt depends on its internal construction. A survey of fittings mounted on tankers and discussions with naval architects has resulted in the list of maximum loads for various fittings given in Table 4-2. Unless the vessel has documented evidence of the strength of the mooring bitts or the method of construction or unless this can be determined by visual examination, the loads classified as questionable construction should be used for mooring bitts.

The use of the loads listed in Table 4-2 will require many vessels to leave the mooring in environments less severe than that which they could remain in if the true capacity of their fittings were known. This is unfortunate as this will result in higher berth outages and will increase costs and congestion at the port. For these reasons, it would be prudent of tanker fleet owners to have their shipboard mooring fittings classified as to actual strength by a recognized classification society such as the American Bureau of Shipping, Lloyds Registrar of Shipping, or Det Norske Veritas. This could be done, at the owner's option, during the normal re-classification survey or by special request when the vessel is at a port where a classification society surveyor is stationed.

TABLE 4-1: TYPICAL SPM MOORING EQUIPMENT FOR VARIOUS TANKER SIZES

Tanker Size Range (DWT)	Typical Maximum Loads	Typical Equipment
Less than 100,000	1180 k newtons (265,000 pounds)	Rarely specially equipped for SPM mooring. Generally do not have bow chocks on center. Bow chocks sometimes not large enough to pass 76 mm (3 in) chain. Bitts almost always must be used to secure mooring lines. Capacity of bitts vary and may not be adequate for loads because of design or condition.
100,000 to 200,000	2670 k newtons (600,000 pounds)	Many equipped with bow chock on center but sometimes chock not large enough to pass 76 mm (3 in) chain. Some equipped with Smits Bracket or chain stopper, however, most must use bitts for mooring. Capacity of bitts vary.
200,000 to 300,000	3920 k newtons (880,000 pounds)	Most have large bow chock on center, some have two. Most equipped with Smits Bracket, many have two Smits Brackets. Many newer vessels have one or two chain stoppers. Usually not necessary to use bitts with single mooring lines but may be required for dual mooring line system. Bitts usually adequate but some deficient because of design or condition.
Greater than 300,000	4895 k newtons (1,100,000 pounds)	Almost all equipped with special SPM fittings for dual mooring line systems. Normally equipped with two bow chocks on center with two Smits Brackets. Many newer vessels have chain stoppers. Bitts usually of sound design but capacity of bitts are generally not adequate for this size load.

TABLE 4-2

MAXIMUM LOADS FOR SHIPBOARD FITTINGS

Mooring Bitts

	Good Construction		Questionable Construction	
Bitt Diameter	k newtons	pounds	k newtons	pounds
450 mm (18 in.)	1160	260,000	800	180,000
500 mm (20 in.)	1520	340,000	980	220,000
550 mm (22 in.)	1864	420,000	1220	275,000
600 mm (24 in.)	2452	550,000	1600	360,000

Chain Stoppers and Mooring Brackets

	k newtons	pounds	k newtons	pounds
For 64 mm (2 1/2 in.) chain	1470	330,000	980	220,000
For 76 mm (3 in.) chain	1950	440,000	1330	300,000

Notes:

1) Loads given are the maximum safe working loads and are defined as approximately one-half the ultimate capacity of the fitting.

2) For mooring bitts, good construction is defined as bitts which are extended through the baseplate to the deck and are welded to both the baseplate and the deck and are supported by adequate underdeck reinforcement.

3) For chain stoppers and mooring brackets, good construction is defined as fittings which are structurally tied to adequate underdeck reinforcement.

4) These loads are for well-maintained fittings. For fittings which appear slightly corroded or have minor split or broken welds, these loads should be reduced by at least 1/3 to 1/2. No values can be assigned to fittings with severe corrosion or evidence of poor welds.

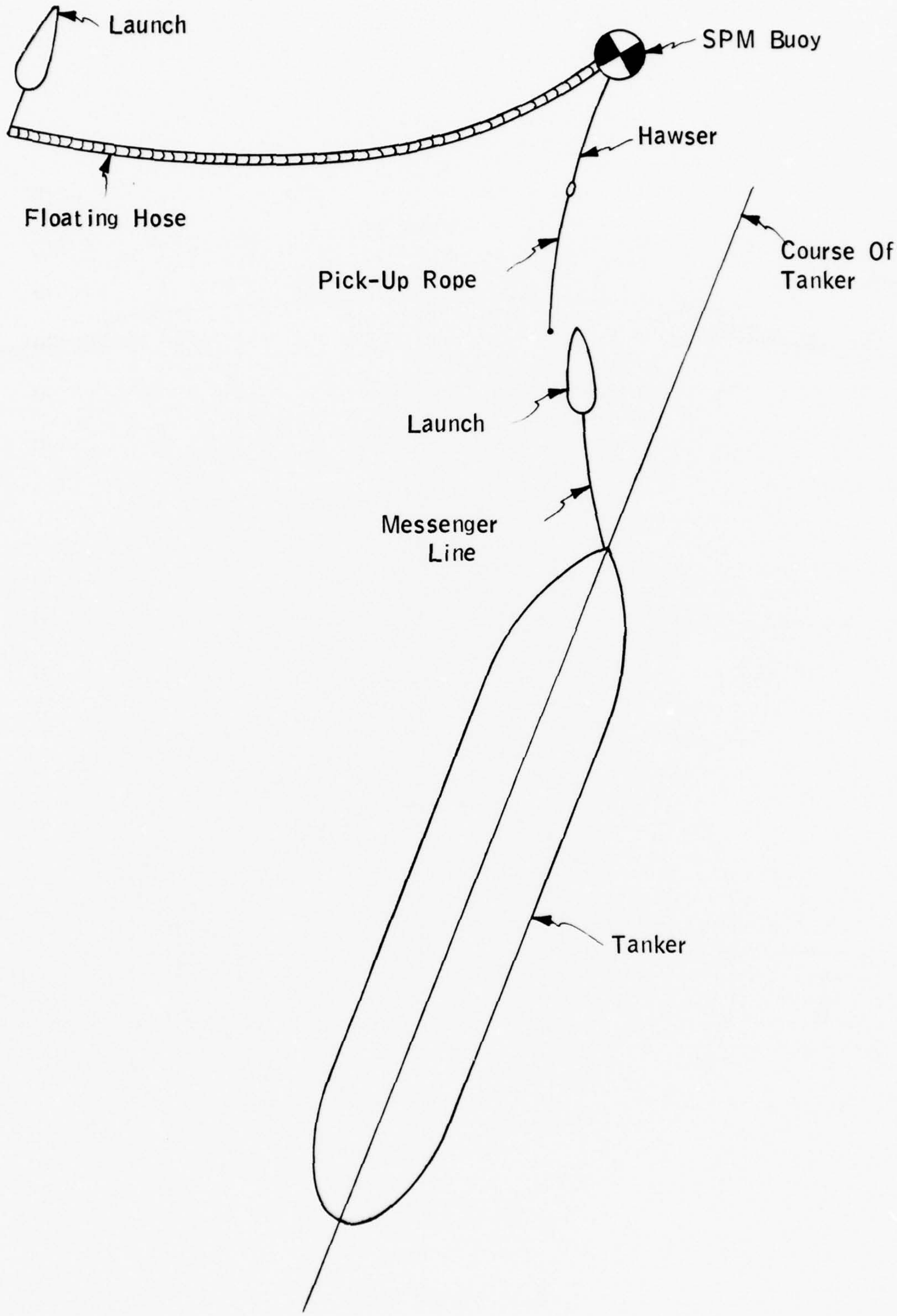


Figure 4-1 - TYPICAL SPM MOORING OPERATION

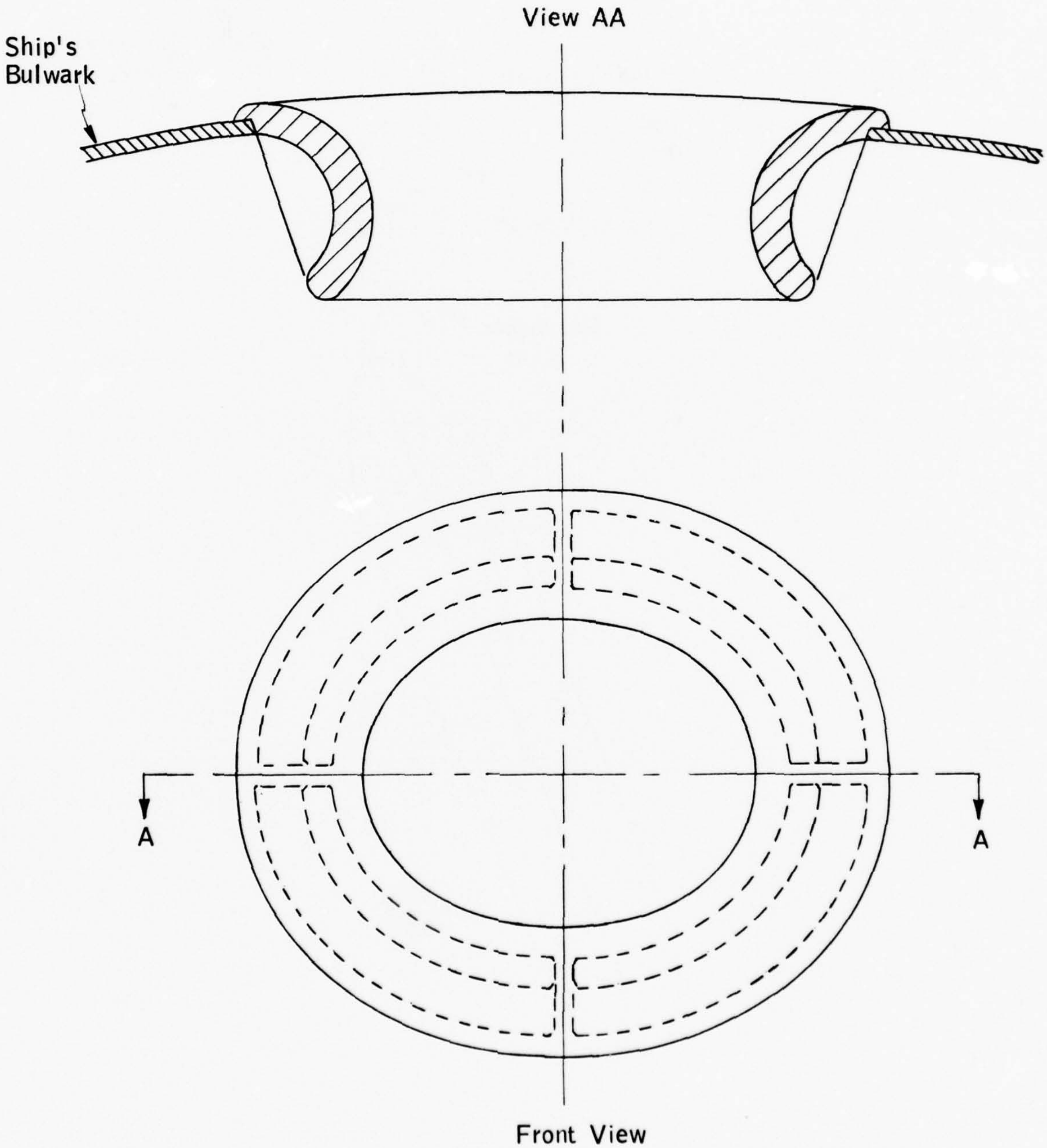


Figure 4-2 - TYPICAL BOW CHOCK

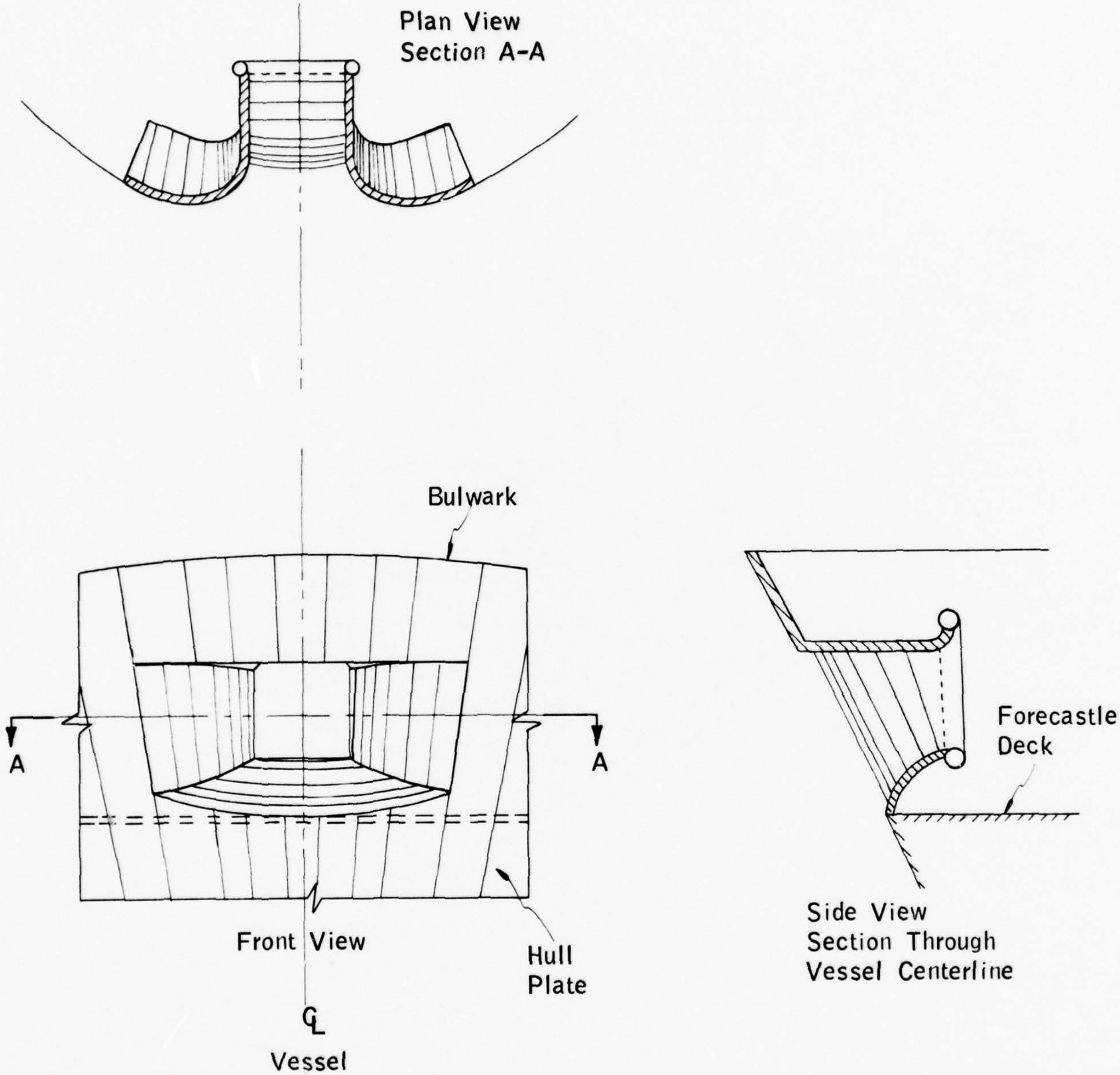


Figure 4-3 - SPECIAL BOW FAIRLEAD

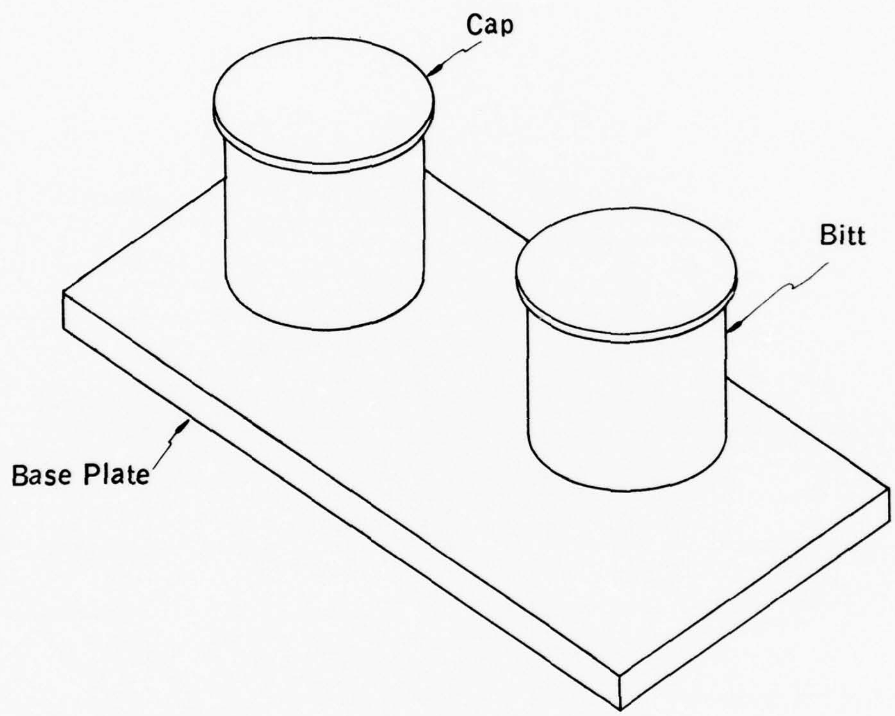


Figure 4-4 - TYPICAL MOORING BITTS

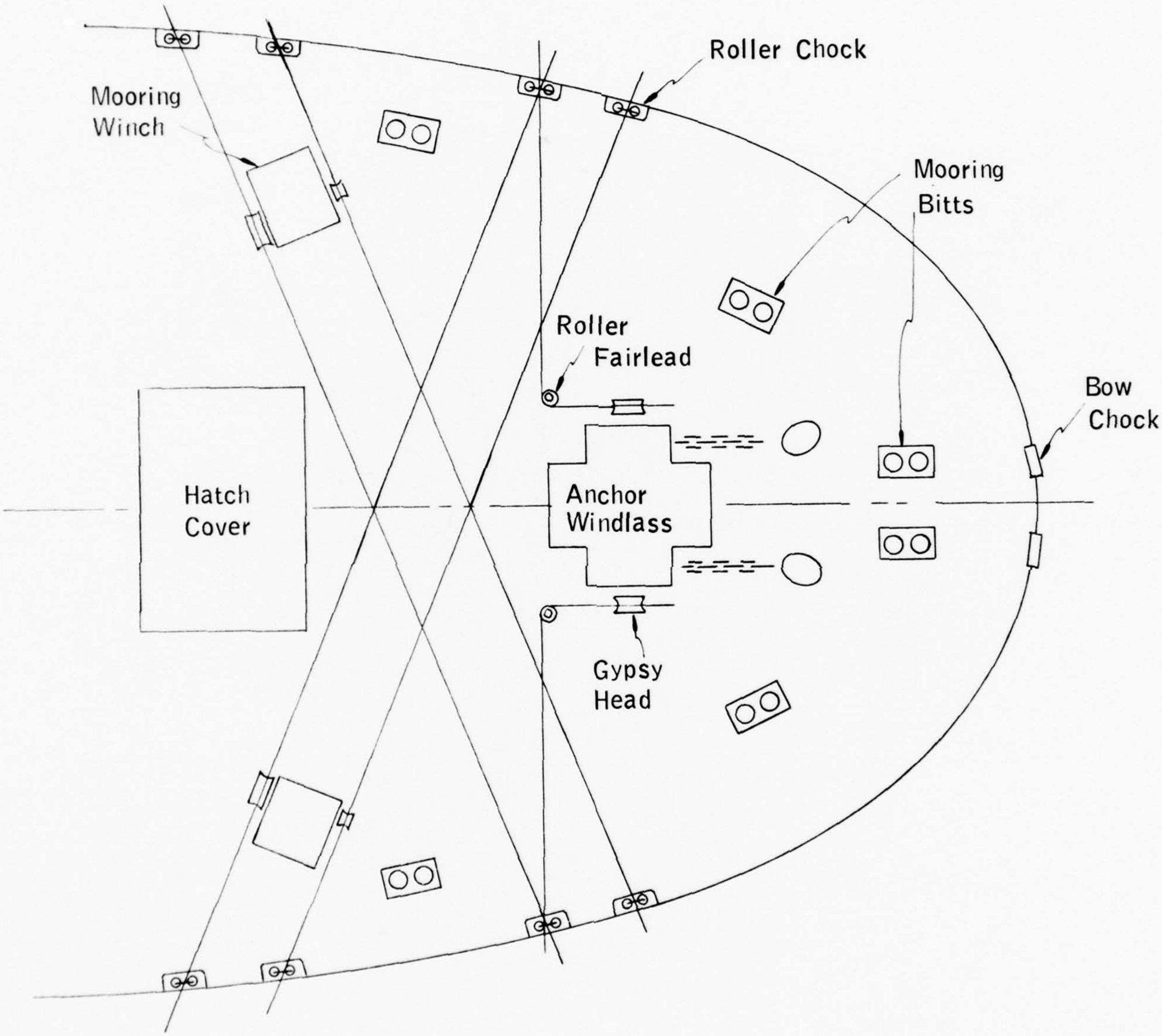


Figure 4-5 - TYPICAL ARRANGEMENT OF DECK EQUIPMENT AND FITTINGS FOR 80,000 DWT TANKER

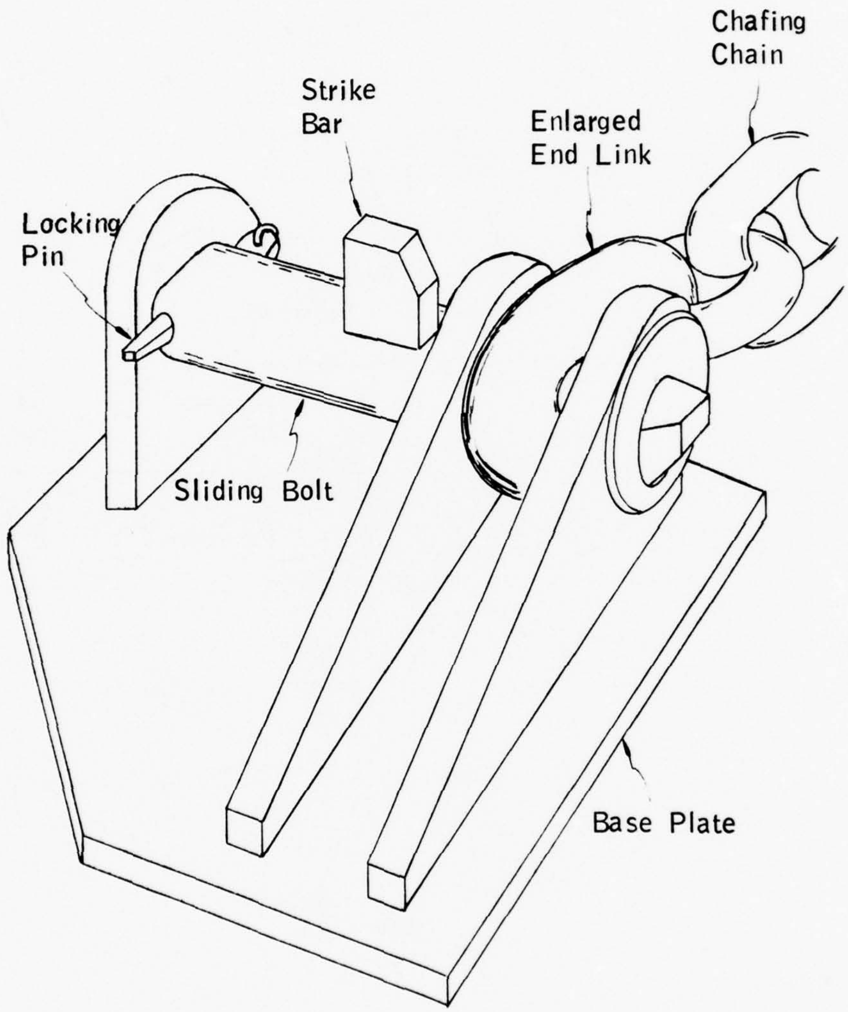


Figure 4-6 - TYPICAL MOORING BRACKET

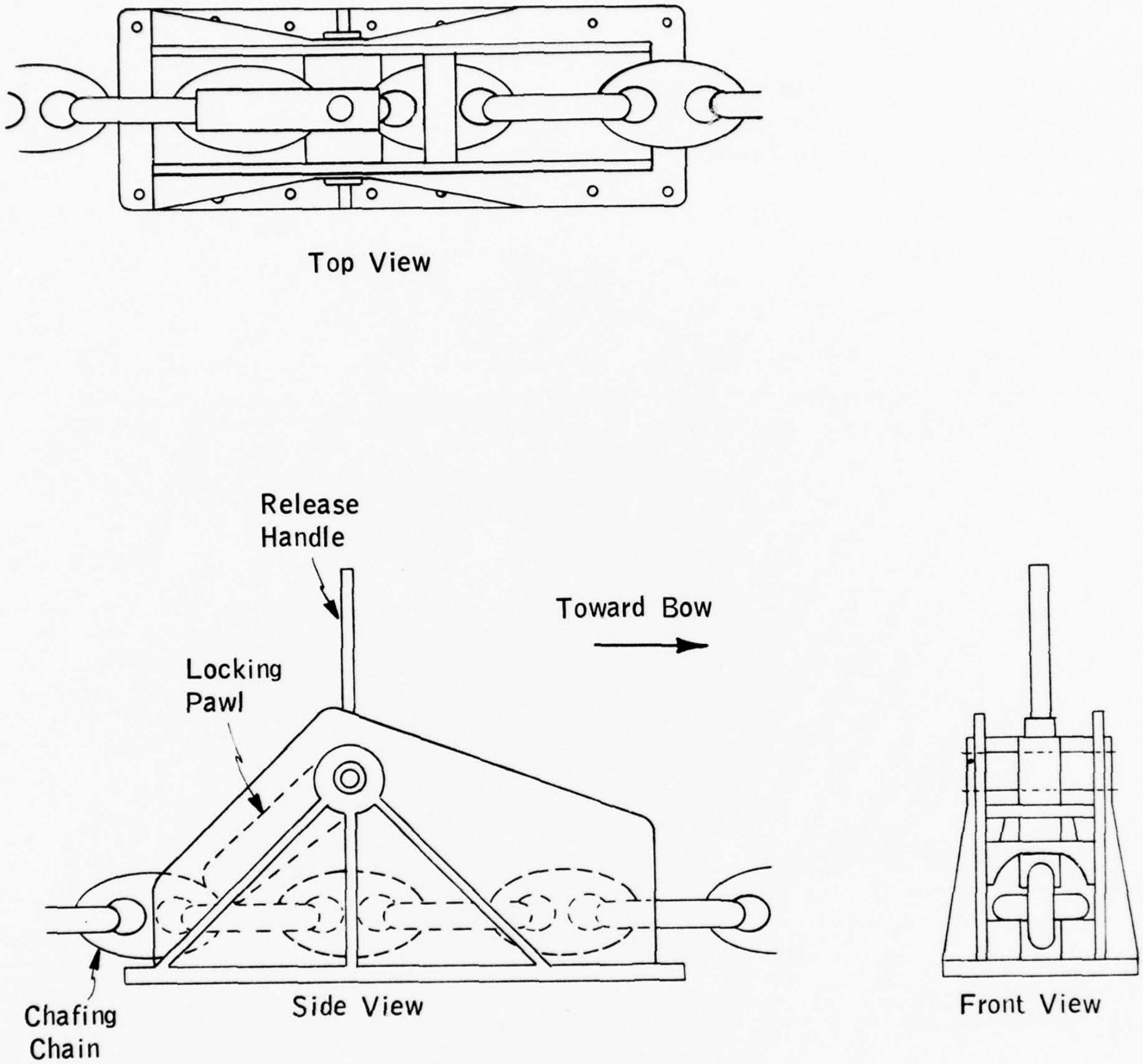


Figure 4-7 - PAWL-TYPE CHAIN STOPPER

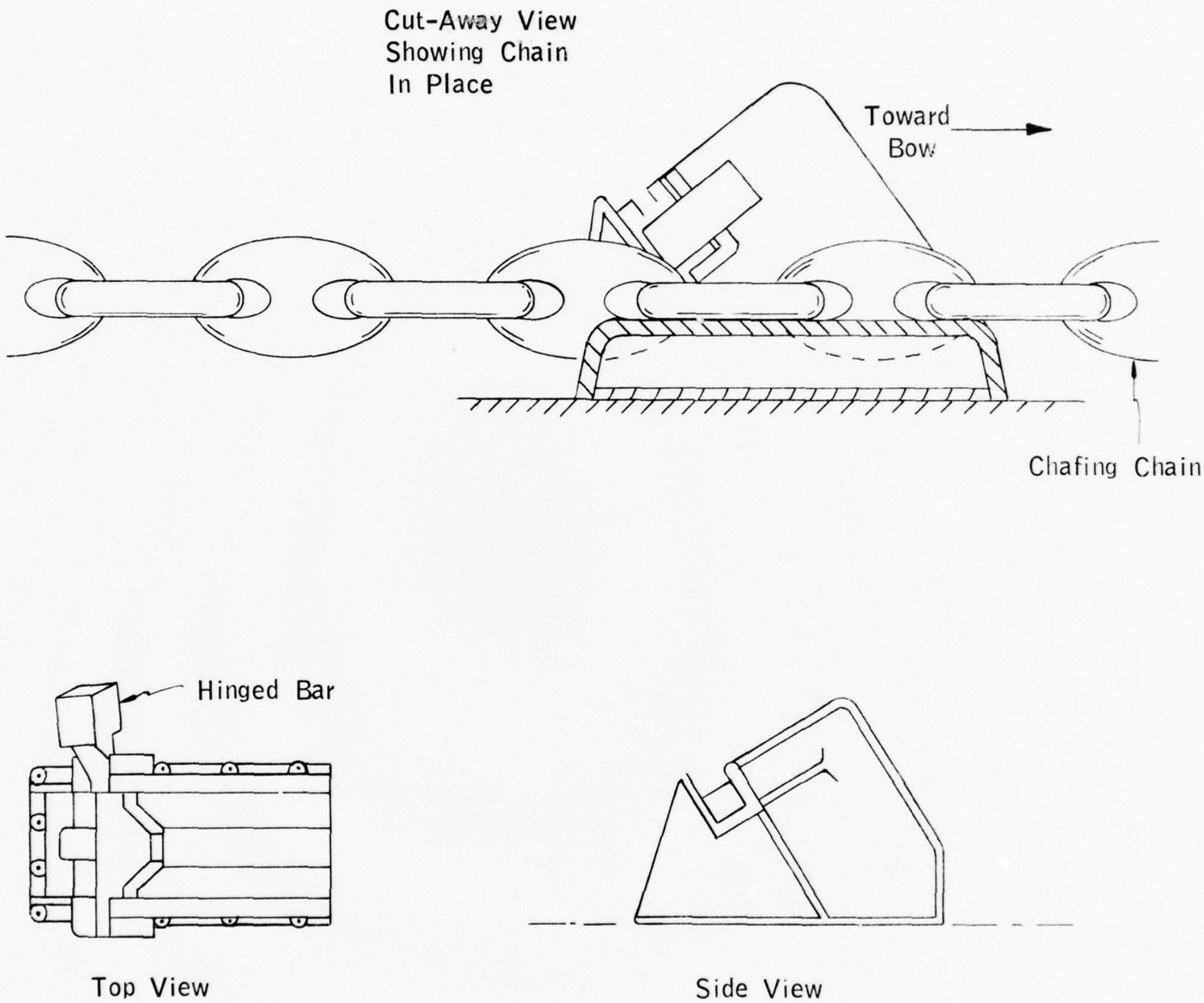
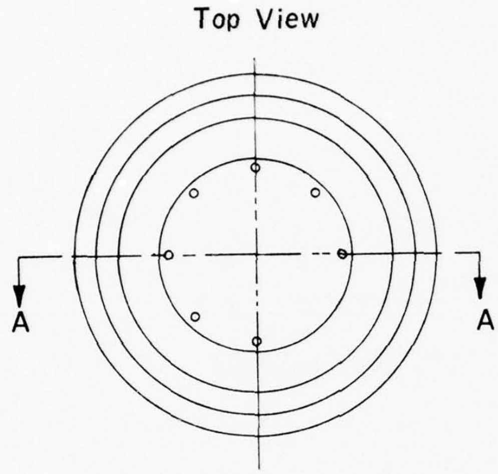


Figure 4-8 - HINGED BAR-TYPE CHAIN STOPPER



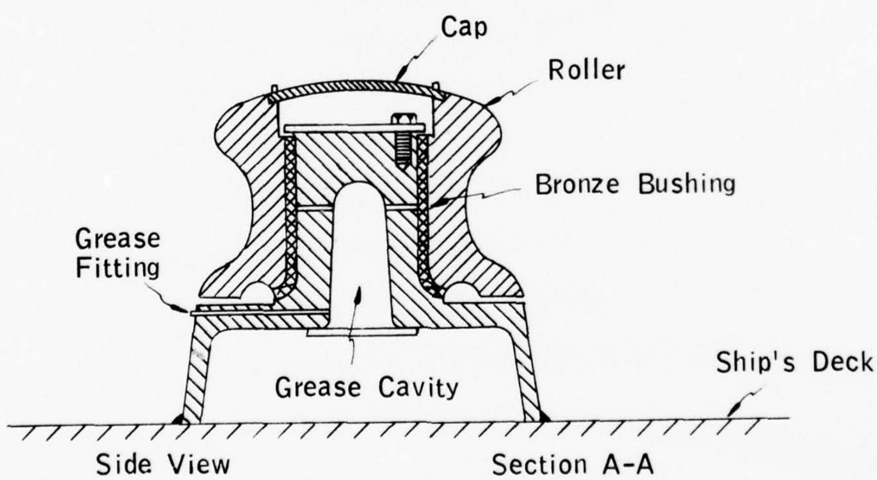


Figure 4-9 - TYPICAL ROLLER FAIRLEAD

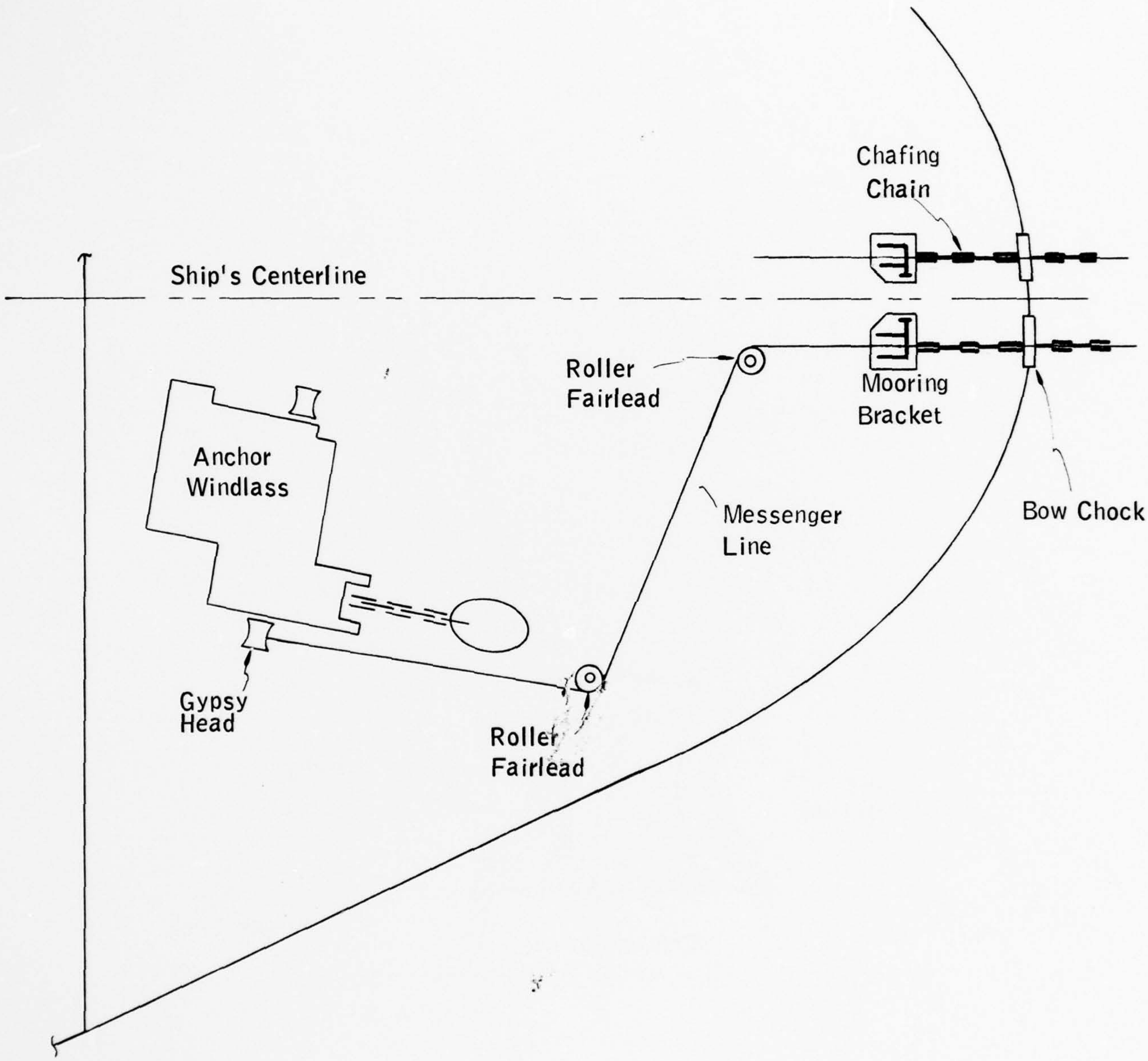
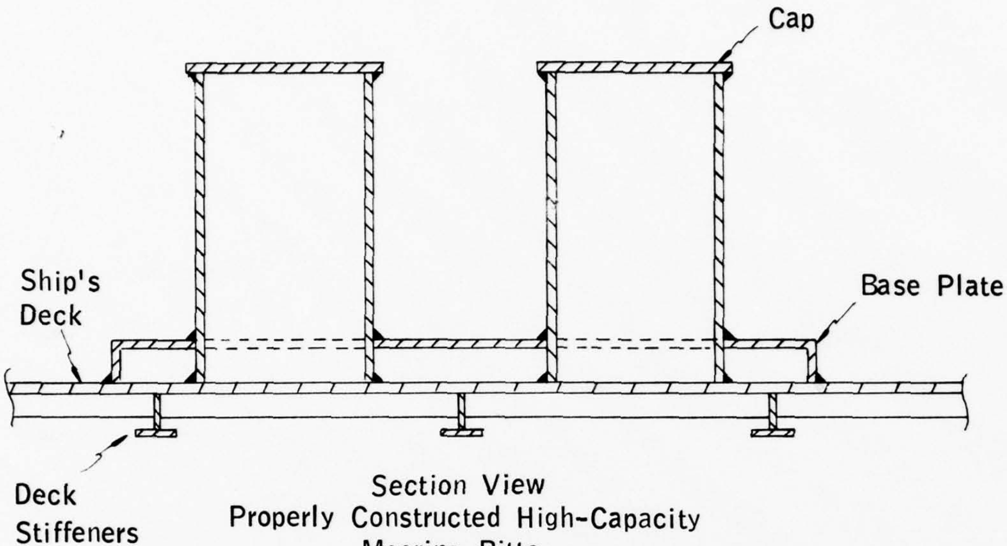
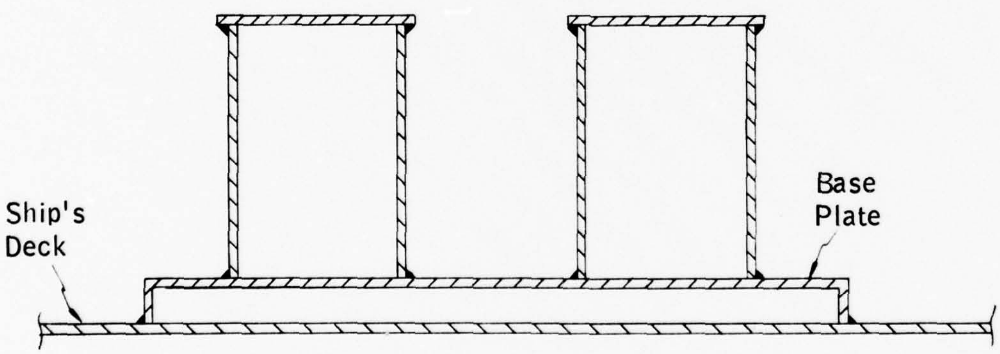


Figure 4-10 - PROPERLY LOCATED ROLLER FAIRLEADS ON FORECASTLE DECK OF 380,000 DWT TANKER

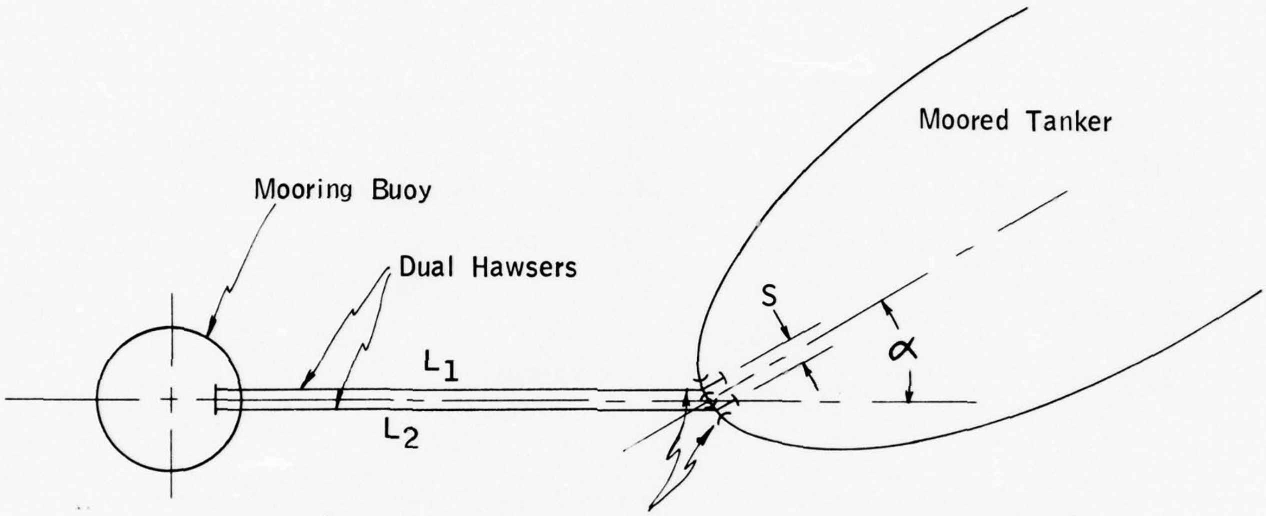


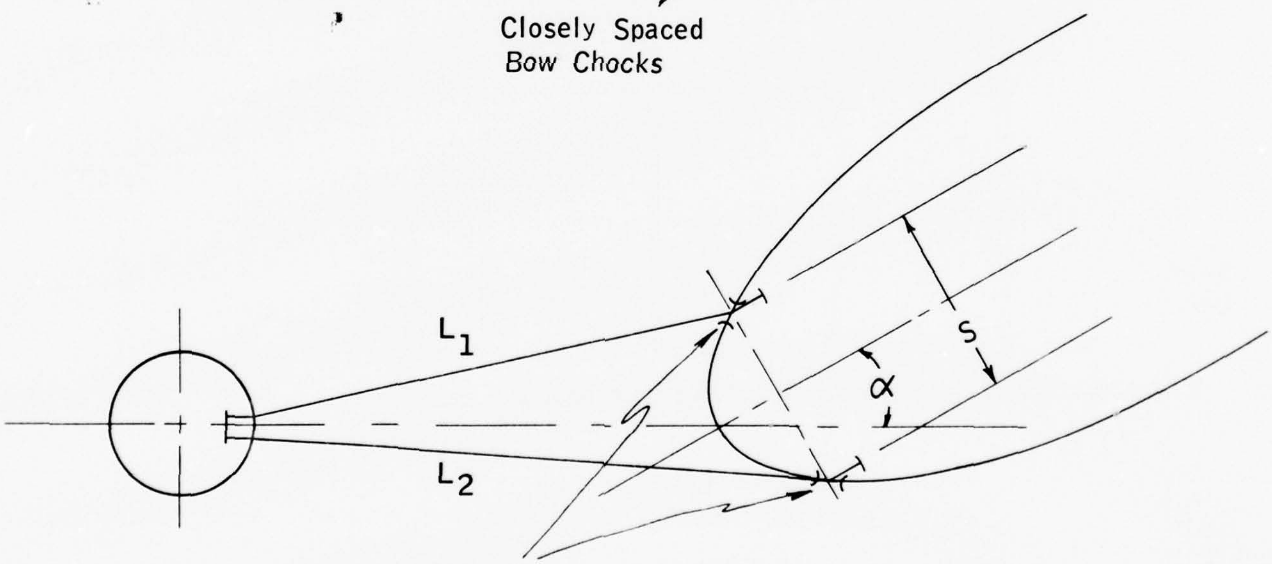
Section View
Properly Constructed High-Capacity
Mooring Bitts



Section View
Poorly Constructed Low Capacity
Mooring Bitts

Figure 4-11 - CONSTRUCTION OF MOORING BITTS





Difference Of Hawser Lengths $= L_2 - L_1 \approx S \sin\alpha$

Where S = Spread Of Bow Chocks
α = Angle Of Tanker Yaw

Figure 4-12 - INFLUENCE OF BOW CHOCK SPREAD ON HAWSER ELONGATION

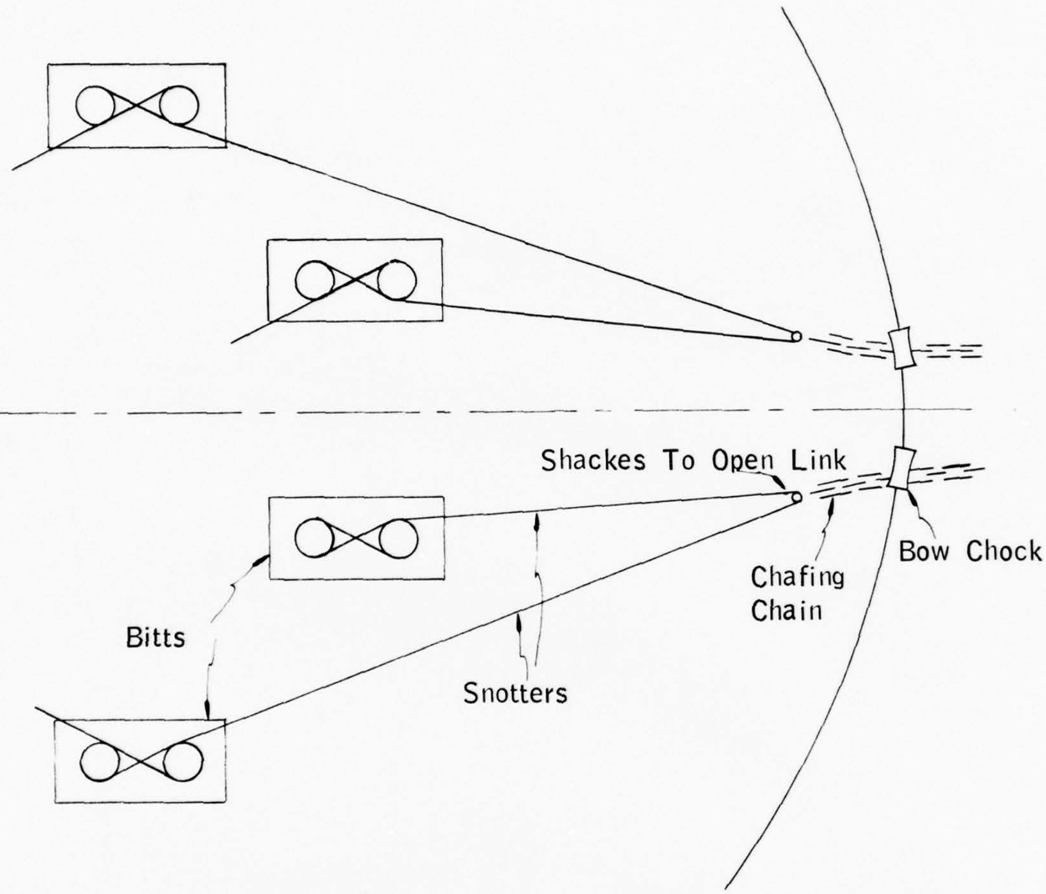


Figure 4-13 - TYPICAL METHOD OF SECURING CHAFING CHAINS USING SNOTTERS

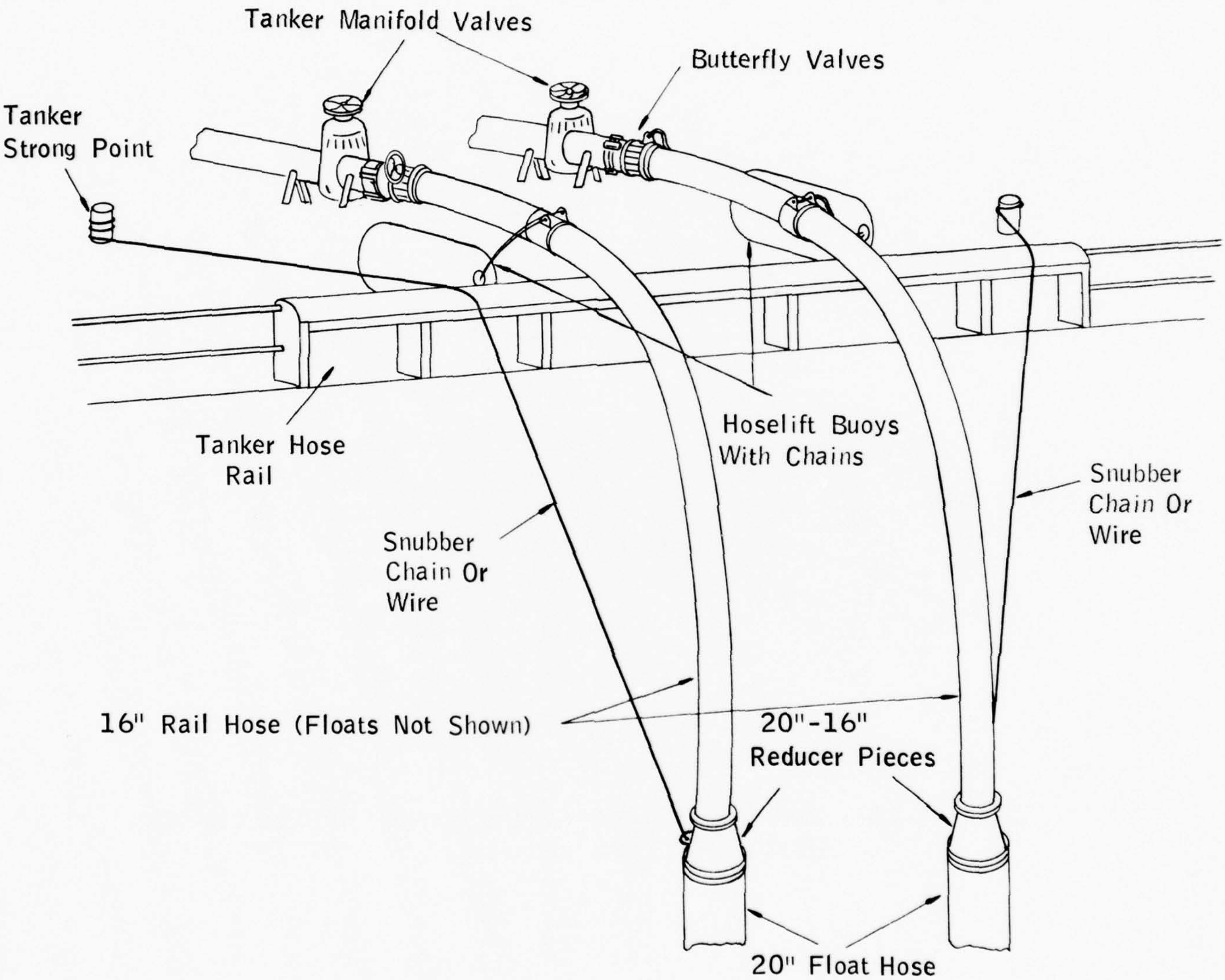


Figure 4-14 - TYPICAL TANKER MANIFOLD HOSE ARRANGEMENT

SECTION 5

GUIDELINES FOR EVALUATING MOORING SYSTEM DESIGN

5.1 INTRODUCTION

The components of the SPM system which carry the mooring load must be designed, fabricated, installed, and maintained in a competent manner to assure the safety of the system. These components are the bow hawser, with associated chafing chains and connectors, the structure of the mooring buoy or tower, the anchor chain or chains, and the mooring base or anchor points.

Data and standards for the design and fabrication of structures, such as the SPM buoy and mooring base, of anchor chain, and of anchor piles are well known. Data and standards for the design and fabrication of SPM hawsers and hawser assemblies are not as well established or readily available. Therefore, most of this section pertains to information on hawser materials, hawser rope construction, the properties of hawsers, and design practices for hawsers and hawser assemblies. Brief discussions of the mooring structure and anchor chains are given at the end of the section.

In the American and English synthetic rope manufacturing industries it has been common practice to refer to the sizes of ropes by their circumference in inches. As these industries convert to metric or SI units the rope sizes are being referred to by their diameter in millimeters. Because rope diameter is generally nominal instead of precise, it is common to multiply the circumference in inches by 8 to determine the nominal diameter in millimeters. Therefore, rope sizes in this report are referred to by their nominal diameter in millimeters with the equivalent circumference in inches given in parenthesis. For example, a 15 inch circumference rope will be referred to as 120 mm diameter (15 in. circumference).

A list of manufacturers of large synthetic ropes is given in Table 5-1. The list includes those manufacturers who have experience in manufacturing large-diameter synthetic ropes in the range of sizes used as SPM mooring hawsers. A list of manufacturers of large chain is given in Table 5-2.

5.2 SYNTHETIC ROPE MATERIALS

SPM hawsers are usually made from one or a combination of the following three synthetic fibers; nylon, polyester, or polypropylene. The synthetic fibers polyethylene and Kevlar* are also used for large rope construction but they have properties which make them undesirable for SPM hawsers. Polyethylene is weaker than other synthetic fibers and it tends to become brittle at normal SPM site temperatures. Kevlar* is very inelastic, which is undesirable in an SPM hawser, and it is more susceptible to damage from bending than other synthetic fibers. The natural fibers manila and hemp

*Kevlar is a DuPont trade name for a high-strength, low-stretch aramid fiber.

were once used for large-rope construction but have been displaced by synthetic fibers. Physical properties of typical nylon, polyester, and polypropylene fibers used in rope manufacturing are presented in Table 5-3.

5.2.1 Fiber Grades and Types

The physical properties of nylon and polyester vary with fiber manufacturers and some manufacturers produce several grades of nylon and polyester. The main difference between the different grades of polypropylene is resistance to sunlight. This property is not important in the large synthetic ropes used at SPMs.

One reason that nylon fiber properties vary is that some rope manufacturers use nylon 6.0 (epsilon amino caproic acid) while others use nylon 6.6 (hexamethylene diamine and adipic acid). A major physical characteristic difference between the two nylons is their melting points. Nylon 6.0 melts at 213°C (415°F) and nylon 6.6 melts at 249°C (480°F). The difference can be significant in abrasion and fatigue resistance of SPM hawsers. Nylon 6.6 is more difficult to manufacture so it is more expensive than nylon 6.0 in some parts of the world. Presently, nylon 6.6 and nylon 6.0 are essentially equivalent in cost in the United States.

Polyester is produced in homo-polymer and co-polymer forms. Co-polymer polyester is the weaker of the two forms and is not generally used in rope construction.

The basic nylon and polyester fibers are coated to enhance their qualities. The coatings put on the fibers differentiate their grades. Each coating is formulated to enhance fiber qualities important for a specific fiber use. The nylon and polyester fibers used for synthetic ropes are the same as those used as tire cord in tires. The coatings for these fibers are formulated for the highest strength and abrasion resistance possible.

Each manufacturer designates the different grades of fibers which he produces by his own nomenclature. There is no industry-wide convention for classifying the different fiber grades, and the fiber manufacturers hold this data as proprietary. Therefore, it is difficult to compare the qualities of fibers produced by different manufacturers. Rope users must rely on characteristics of high quality fibers and experience with and reputation of the different fibers to judge quality. At this time, DuPont nylon 6.6 type 707 and Allied polyester type 1W81 are representative of the best nylon and polyester fibers available. Other fibers may be of equal quality.

Nylon fibers absorb water up to 10 to 12% of their weight. As a result, the filaments swell, shrink in length by about 10%, and lose about 10% of their strength when wet. Recognizing this strength reduction, the strength of nylon ropes for some applications should be reduced about 10% below the catalog values. However, the shrinkage of nylon filaments increases their energy absorbtion capabilities and, thus, SPM mooring loads may be lower when the fibers are wet. Some manufacturers believe that the increase in energy-absorbtion capability compensates for the reduction in strength of wet nylon fiber for SPM hawsers. Recommendations on the strength reduction for wet nylon hawsers are given in subsection 5.7.

The third fiber used for SPM hawser construction, polypropylene, is relatively easy to manufacture. Many rope manufacturers purchase polypropylene in granular form and then extrude their own polypropylene fibers. Because polypropylene fiber manufacturing operations are usually small and because of the great number of polypropylene fiber manufacturers, it is difficult for the rope user to determine polypropylene fiber quality and to judge the quality control exercised by the different manufacturers.

Rope users must specify high quality fiber characteristics from rope-manufacturers with good reputation and experience to obtain the best quality. If specific nylon- or polyester-fiber manufacturer's products are not called for in rope specifications, characteristics of high quality fibers should be specified. Characteristics for specifying high-quality nylon, polyester, and polypropylene fibers are given in Table 5-4.

Synthetic fibers are usually manufactured in either monofilament or multifilament form. Denier per filament is sometimes used to differentiate monofilament fibers from multifilament fibers. Denier is a measure of filament size. It is the weight in grams of 9000 meters of filament. Wall Ropes stated that in the U.S. cordage industry filaments below approximately 40 denier per filament are classfied multifilament and those over this value are classified monofilament. Wall Ropes further classified monofilament fibers as those large enough to be used singularly in the rope manufacturing process whereas multifilament fibers are those which must be twisted together to form a workable filament size. The division point between monofilament and multifilament fibers is not universally defined, and it varies between countries and between industries. Canadian Standard 40-GP-13 defines the division between monofilament fiber and multifilament fiber to be 25 denier per filament. Filaments used in rope manufacturing are usually significantly below or above the range of 25 to 40 denier per filament. Samson Ropes have stated that multifilament fibers are generally stronger than monofilament fibers.

Monofilament polypropylene fiber is sometimes cut in approximately 1 m (3 ft) lengths to form staple fiber. This fiber form was originally constructed to resemble natural fibers such as Manila. The cut filaments are grouped and twisted together in a random overlapping fashion for rope construction. Friction between the filaments prevents the rope from slipping apart. Ropes manufactured with staple fiber have a hairy appearance like natural material fibers. This can make a rope easier to handle and give the rope abrasion protection. Ropes constructed with continuous monofilament or multifilament fibers form the same type of fuzzy surface once abrasion begins. Similar abrasion protection is then obtained. The ability to handle a rope is not usually an important criteria for SPM hawsers.

Staple fiber is not believed to be superior to monofilament or multifilament fibers so it is not ususally specified for SPM hawsers. However, staple-fiber ropes are as strong as continuous monofilament or multifilament ropes and compare favorably in other rope properties, so use of staple-fiber ropes should not be ruled out at SPMs.

Film fibers are also manufactured for some rope applications. Film filaments are thin tape-like strips resembling movie film. Polypropylene is usually the only material manufactured in this form. Film fibers as presently manufactured are generally weaker than monofilament or multifilament fibers. Therefore, film fibers are not usually used for high strength applications such as SPM hawsers. The advantage of film fibers for some applicatons is low cost because it is relatively easy to manufacture.

5.2.2 Fiber Elasticity

The hawser between the SPM buoy and the moored tanker must provide a certain amount of elasticity. At many SPMs, especially those in relatively shallow water, a hawser with a large amount of elasticity is needed to keep mooring loads within acceptable limits. The hawser elasticity also serves as a shock absorber between the buoy and the tanker. As the buoy moves in response to waves it can impose high loads in a stiff hawser. Buoy-induced loads are minimized in an elastic hawser. The importance of mooring system elasticity is discussed further in subsection 2.6.

Hawser elasticity is determined by the hawser fiber, hawser construction, and hawser length. The effect of different hawser constructions is discussed in subsection 5.6 and limitations of hawser length are discussed in subsection 2.6.

Comparative fiber load-elongation curves for the various materials are not available. Comparative load-elongation curves for nylon, polyester, and polypropylene ropes of the same construction are given in Figure 5-1. Nylon is more elastic than either polypropylene or polyester. The high elasticity of nylon gives it high energy-absorbtion capabilities. Therefore, nylon is usually used in SPM hawsers, especially at shallow-water SPMs.

Polypropylene fiber is slightly more elastic than polyester fiber. However, polyester fibers which have been heat set as part of the rope-making process, may be slightly more elastic than polypropylene fibers, which are normally not heat set. This effect causes apparent discrepancies in comparison of elasticities for ropes of different materials. Other possible reasons for discrepancies are discussed in subsection 5.6.1.

5.2.3 Fiber Abrasion Resistance

Abrasion resistance of the different fibers is a function of fiber melting point and the fiber overlay coating or fiber surface finish. As stated in the discussion on grades of fibers, the coatings put on nylon and polyester are formulated to increase strength and abrasion resistance. Polypropylene is not usually coated. Unfortunately, because the fiber manufacturing industry treats data on fiber coatings as proprietary, little quantitative data could be obtained on fiber coatings. As a result, the physics behind the coating effects on fibers and ropes cannot be discussed.

Fiber and rope manufacturers run their own abrasion tests to judge the quality of various fibers. Test data indicate that polypropylene is inferior in abrasion resistance to nylon and polyester. Its inferiority is due at least in part to its lower melting point.

Results of abrasion tests on nylon and polyester fibers vary a great deal. Part of the variation is due to the wide variety of testing techniques used. However, the largest variations in test results apparently occur because different types and grades of fibers have been tested and compared. The specific type and grade of fiber used is not usually presented in test results. Because of its lower melting point, nylon 6.0 has poorer abrasion resistance than nylon 6.6, and abrasion test results on the two nylons will differ. Fiber coating plays a large part in abrasion resistance, so tests performed on fibers of various grades and from different fiber manufacturers will give different results. Water also has an effect on some fibers and coatings, so abrasion tests on wet fibers can give different results than tests on dry fibers.

Data available at this time does not conclusively prove either nylon or polyester to be significantly superior in abrasion resistance. Top grade polyesters such as Allied polyester type IW81 may be slightly more abrasion resistant than top grade nylons such as DuPont nylon 6.6 type 707. Further abrasion testing on new improved grades of polyester may show polyester to be significantly more abrasion resistant than nylon.

5.2.4 Fiber Strength

Hawsers at SPMs must withstand very high mooring loads and, therefore, must be made from very strong fibers. Fiber strength is measured by tenacity. Tenacity is defined as the breaking strength of the filament. In the SI system tenacity is expressed in Newtons per tex; Traditionally it is expressed in grams divided by denier. Tenacitites of typical nylon, polyester, and polypropylene fibers are given in Table 5-3.

There are stronger synthetic fibers than nylon, polyester, and polypropylene but certain physical properties make these stronger fibers undesirable for use in SPM hawsers. Of the three fibers used in SPM hawsers, nylon is the strongest, polyester ranks second, and polypropylene is weakest. The strength difference between nylon and polyester is small and is decreasing as improved grades of polyester are being developed. Strength values for some of the newest polyester fibers are not yet published but DuPont has stated their new Dacron* type 608 is very nearly as strong as their nylon 6.6 type 707. The polypropylene fiber representated in Table 5-3 is only about 70% as strong as the typical nylon represented in that table.

5.2.5 Fiber Specific Gravity

As shown in Table 5.3, polypropylene is the only fiber of the three with a specific gravity less than that of water. Polypropylene hawsers float without the need of additional floatation. Nylon and polyester hawsers need some form of additional floatation because their specific gravities are greater than that of water.

Polypropylene's ability to float can be an advantage at SPMs although not necessarily a strong advantage. Without the need for additional floatation, an entire polypropylene hawser can remain visible and be easily inspected for external abrasion and cuts. In addition, because there is no floatation to remove, polypropylene hawsers can be quickly winched by traction winches at SPMs designed for self-service mooring. Because of polypropylene's ability to float it is commonly used for pick-up ropes which are used to winch the mooring hawsers to the tanker forecastle.

As explained in subsection 5.8.2, the cover of floatation material serves to protect the hawser from abrasion and cuts. Although hawsers with additional flotation are harder to inspect, they are less likely to be cut or abraided because of the protection provided by the covering floatation. The need for additional floatation on mooring lines which must be handled on traction winches is a disadvantage. However, floatation devices are available which can be quickly removed so the hawser may be winched. Traction winches and self-service mooring would not normally be used at general SPM terminals.

5.2.6 Fiber Ultraviolet and Chemical Resistance

Ultraviolet and chemical fiber degradation resistance of SPM hawsers are not usually important factors. Because of their large sizes, SPM hawsers do not experience any significant strength loss from ultraviolet exposure due to sunlight. Most SPMs are located in relatively clean water precluding hawser strength loss from chemical attack. However, a general understanding of these two forms of degradation is still important.

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*Dacron is a DuPont trade name for polyester.

Of the three fibers, polyester is most resistant to ultraviolet degradation, nylon is also very resistant, but polypropylene readily degrades when exposed to ultraviolet rays. Many ropes of polypropylene and some of nylon have ultraviolet inhibitors such as dyes added to slow ultraviolet degradation.

Ultraviolet rays attack only the outer fibers of a rope. Rope strength loss due to ultraviolet degradation will vary inversely to the ratio of a rope's volume to its surface area. Rope volume is proportional to the square of rope diameter while rope surface area is proportional to rope diameter. Therefore, the ratio of rope volume to rope surface area increases in proportion to rope diameter. Relating this to ultraviolet degradation resistance, the larger the rope is, the more resistant it is to ultraviolet rays. Rope manufacturers have stated that polypropylene ropes over 24 mm diameter (3 in. circumference) will not be significantly affected by ultraviolet rays. Nylon and polyester ropes are even more resistant. As SPM hawsers are usually at least 120 mm diameter (15 in. circumference), strength loss due to ultraviolet degradation is negligible.

Chemical resistance of the three fibers is generally good. In general terms, strong acids degrade nylons and strong bases degrade polyesters. Polypropylene seems to be very resistant to almost all chemicals. Test data shows a saturated solution of chlorine at 21°C (70°F) will degrade polypropylene (DuPont Technical Report X-225). However, exposure to the chlorine in seawater should not cause significant effects.

Chemical resistance of a typical nylon and a typical polyester to many chemicals are given in Table 5-5 (DuPont Technical Report X-226). The chemicals that degraded the nylon and polyester fibers were either concentrated or at high temperature. The presence of chemicals at either high temperature or in high concentration is very unlikely at SPMs.

Rust can cause chemical degradation of polyester and nylon ropes. Besides producing flakes of rusted metal which can cut synthetic fibers, rust produces an acid which has been shown to readily degrade nylon, and to degrade polyester to a lesser degree. A general evaluation for determining remaining strength of rust-stained ropes is given in DuPont Technical Report X-226. If half of the cross-sectional area is rust-stained, rope strength is reduced by one-half. This evaluation may not be accurate, especially for polyester ropes, but it demonstrates the importance of keeping nylon and polyester SPM hawsers away from rusting metal. Polypropylene SPM hawsers are not chemically affected by rust but should still be kept from rusting metal to prevent the fibers from being cut by rusted metal flakes.

Very little data has been found on the effects of petroleum products on synthetic fibers. Data from cyclic load tests conducted by United Ropes discussed in subsection 5.5.2 indicates lubricating oil or grease reduces the strength of nylon rope. However, Samson Ropes have stated some U.S. West Coast fishermen soak nylon ropes in diesel fuel before use because it allegedly improves the strength of the rope.

Unless very unusual conditions exist at a SPM, such as proximity to a chemical outfall, chemical degradation of SPM hawsers in service is not an important factor (except perhaps chemical degradation caused by rust). Precautions must be taken against chemical degradation during storage or handling. Synthetic ropes should not be stored near sources of certain chemical fumes, such as in paint lockers, or near chemical process operations. To prevent degradation by rust, SPM hawsers must be kept from rusting metals. Proper storage of hawsers onboard tankers is discussed in Section 4.

5.2.7 Material Recommendations

No one material, nylon, polyester, or polypropylene, can be recommended for every SPM hawser application. Design criteria for each SPM will indicate the best material to use. At most SPMs nylon is used for the hawsers and polypropylene is used for the pick-up ropes.

Nylon's high energy-absorbtion capability along with high strength and abrasion resistance makes it preferable for many SPM hawser applications. The preference for nylon is based on the use of nylon 6.6; nylon 6.0 is inferior to both nylon 6.6 and polyester in abrasion resistance. New grades of polyester now being developed may approach the strength of nylon and may be significantly more abrasion resistant than nylon 6.6. If the new grades of polyester prove to be superior to nylon they may be preferable at SPMs in deep water where nylon's energy absorbtion capability is not as important.

Different materials should generally not be mixed in the same rope. Materials are sometimes mixed to obtain the best qualities of each material, such as the strength and abrasion resistance of nylon or polyester and the buoyancy of polypropylene. However, because of the different extensibilities of the materials and polypropylene's susceptibility to creep, one of the component fibers may take a larger percentage of the load. The strength of mixed-material ropes may degrade more rapidly than single-material ropes. For this reason, special care must be taken in selecting and specifying mixed-material ropes. Special mixed-material ropes which are properly engineered to take advantage of the different material properties may be suitable for some applications.

The best quality nylon and polyester fibers are overlay finished. The nylon-and polyester-fiber overlay finish processes are proprietary and, therefore, cannot be rigidly specified. Polypropylene fiber quality is hard to specify because many rope manufacturers produce their own polypropylene fibers. To insure quality fibers are used in rope manufacturing, high quality fiber characteristics should be specified from manufacturers with good reputations and experience.

5.3 SYNTHETIC ROPE CONSTRUCTIONS

Rope making is a very old art and there are many different forms of rope construction in existence today. Some of the more common constructions are parallel-lay, three-strand (also known as regular lay), eight-strand (also known as squarebraid or plaited),nine-strand (also known as cable lay), and double-braid. Eight-strand and double-braid ropes are the only rope constructions used extensively at SPMs. The main reason that cable lay and three-strand ropes are not used is that they are subject to hockling. Hockling is very detrimental to the rope. Parallel-lay ropes must be held together by a cover which makes them hard to bend. They are also hard to splice and they lack the elasticity which is needed at SPMs. Three-strand, eight-strand, and double-braid rope constructions are discussed in the following subsections.

5.3.1 Strands and Yarns

The preparatory treatment of the fibers in making the various common rope constructions is the same. The process of twisting filaments into yarns and then into strands is shown in Figure 5-2. Monofilament or multifilament synthetic fibers are brought together and twisted to form yarns. Yarns are brought together and twisted to form strands. The size of the rope and the final construction technique dictates the number of filaments used to construct yarns and the number of yarns used to construct strands. The process by which the strands are formed into ropes varies with the type of rope construction.

The direction and tightness of fiber twist in the yarns, of yarn twist in the strands, and of strand twist in the rope are important in determining rope characteristics. Among the characteristics affected by the twist of the different components are a rope's tendency to twist, its breaking strength, and its elongation characteristics.

To prevent the final rope from having an inclination to twist, rope strands are twisted in the opposite direction of the yarns. Eight-strand and double-braid ropes are usually made of equal numbers of strands with opposite twists. This gives them excellent resistance to twisting and hockling. The uneven number of strands in three-strand ropes makes their construction unbalanced and prone to hockling.

The twist of the fibers in the yarns and the twist of the yarns in the strands are measured by their cyclic length. The greater the cyclic length, the less elastic and stronger the component will be.

The twist of the strands in the rope is measured by the helix angle or the number of picks per unit length. The helix angle is the inverse tangent of the strand cycle length divided by the rope circumference. The picks per unit length is a count of the number of strands on the surface of a rope in a distance of one unit length parallel to the rope axis. Although in different forms, both measures give the same information. As above with fibers and yarns, the more parallel the strands are to the rope axis, the less elastic and stronger the rope will be.

There is a lack of uniformity in the definition of strand and in the distinction between strand and yarn. Wall Ropes uses the term strand to refer not only to the major twisted subcomponents of three-strand and eight strand rope, but also to the next subdivision of twisted components.

British Ropes calls each major twisted component of double-braid rope a strand. Samson Ropes calls the adjacent parallel group of twisted components which are wound on the same bobbin and are braided together through the rope a strand. The bobbin or spool is called a carrier and, thus the group of components which are wound on a single bobbin is also sometimes referred to as a carrier. Each individual major twisted component is called an end. Thus, British Ropes calls each end a strand and Samson Ropes calls the components on each bobbin a strand.

In this report, the term strand will refer to an end, that is, a single major twisted component.

5.3.2 Three-Strand Ropes

Three-strand rope is the traditional form of rope construction. However, because it is susceptible to hockling, it is generally unacceptable for use in SPM hawsers.

Three-strand rope construction is shown in Figure 5-3. Filaments are twisted into yarns which are then twisted into strands. Three strands are twisted together to form three-strand rope.

Because three-strand rope is the simpliest of the three rope constructions, it is the least expensive. Cost is the only advantage of three-strand rope. Three-strand ropes are not well suited for SPM hawsers because of their tendancy to hockle. Hockles result from twisting of layed rope in a direction opposite to the direction of lay. Hockles resemble knots on a rope, as shown in Figure 5-4. Hockling greatly reduces the strength of a rope. Hockling of three-strand rope can be caused by a number of reasons, including pulling out kinks and rapid release of load. Both events are common in SPM service and thus hockles are likely to occur. In some services, the hockled portion of a rope can be cut out and the rope can be respliced. However, because SPM hawsers must be of a certain length, and field splices are of questionable strength, this practice is not acceptable at SPMs. Because of the tendency to hockle, three-strand ropes are being replaced in many applications by braided ropes.

5.3.3 Eight-Strand Rope

Eight-strand rope construction is shown in Figure 5-5. Filaments are twisted into yarns which are twisted into strands. Two pairs of right-turn strands and two pairs of left-turn strands are braided together to form the rope. Right turn and left turn refer to the direction of yarn twist in the strands. The major advantage of eight-strand rope over most other rope constructions is that it will not hockle.

Eight-strand rope construction forms a tight rope, so it holds its shape well when under no load. Double-braid rope does not hold its shape as well as eight-strand rope and thus is more susceptible to grapnel damage when used as a pick-up rope. Eight-strand rope may also be easier to man-handle by launch and tanker personnel. Because of the tight rope construction, eight-strand rope is harder to bend than double-braid rope. This is not viewed as either an advantage or a disadvantage at SPMs.

New eight-strand ropes have much greater elasticity than double-braid ropes of the same material and breaking strength. However, when both ropes are broken-in the difference in elasticities is slight. Elasticities of eight-strand and double-braid ropes are discussed in subsection 5.6.

Eight-strand ropes have a reputation of being easier to field splice than double-braid ropes. Ability of field splicing is not a major advantage in SPM hawsers. SPM hawsers should not be field spliced because inferior splices are likely to result. Pick-up ropes may be field spliced to remove worn sections and eight-strand ropes are sometimes preferred for pick-up ropes for this reason.

If severely cut, eight-strand rope can quickly degrade in strength The surface of eight-strand rope has a knuckled appearance. A deep cut in one strand of an eight-strand rope will essentially destroy its ability to take loads. Damage to one strand in an eight-strand rope will cause uneven loading in the rope, and will cause some of the remaining strands to take a much higher load. This will lead to break-down of the entire rope under sustained or repeated loading.

Eight-strand rope is the traditional form of large ropes. As such, there are a number of eight-strand rope manufacturers. Presently, Hawkins and Tipson Ropemakers can manufacture the largest eight-strand rope. Their braiding machine is capable of making up to 192 mm diameter (24 in. circumference) eight-strand rope.

5.3.4 Double-Braid Rope

Double-braid rope construction is shown in Figure 5-6. The number of strands used in construction depends on the size of the final rope and on the manufacturer. Both British Ropes and Samson Ropes double-braid ropes in the size range of 120 to 168 mm diameter (15 inch to 21 inch circumference) have 64 strands in the cover. British Ropes double-braid construction has 32 strands in the core and Samson Ropes has either 24 or 36 strands in the core. Note that the strands in the core are larger than those in the cover. Refer to subsection 5.3.1 for the definition of strands as used in this report.

When double-braid rope is spliced, the core and cover are joined together so they share the loads. The percentage of the load shared between core and cover is dependent on the number of fibers and the twist of each member. British Ropes have stated that the loads are shared 50-50 in their double-braid ropes. Samson Ropes have stated that in their double-braid ropes the core and cover share the loads 50-50 at near the rope breaking strength, but in the normal working load range the loads are shared 60 percent in the core and 40 percent in the cover. The redistribution of load sharing occurs because the filaments of the core and cover become more closely aligned as the load is increased.

Double-braid rope will not hockle. As stated previously, this is a very important rope characteristic for marine hawser use.

Double-braid rope is a looser construction than eight-strand rope. Therefore, double-braid rope is easier to bend than eight-strand rope. However, because of the loose construction, double-braid rope is more susceptible to grapnel damage. Also, because of its looser construction, double-braid ropes may be harder to manhandle by launch and tanker operators than eight-strand rope.

Double-braid rope is more likely to retain its strength when damaged by cuts than eight-strand rope. Because the core is protected by the cover, there is very little chance of the core being cut. Also, the strength of the core is not affected by a cut in the cover. Therefore, the core is likely to retain its entire strength when the rope is damaged. Because all strands of eight-strand rope are exposed, eight-strand rope does not have this safety protection. Another reason double-braid rope is more likely to retain its strength is that the cover is made up of many more strands than eight-strand rope. The strength loss from one or two cut strands of the 64 outer strands of double-braid rope will be significantly less than the strength loss from one or two of the eight strands of eight-strand rope being cut.

Comparing ropes of the same circumference, double-braid rope is stronger than eight-strand rope. There are two mechanisms responsible for this. Double-braid rope is made up of many small strands; thus the construction is denser, and there are more fibers in double-braid rope per unit area than in eight-strand rope. For two ropes of the same circumference, double-braid rope will have more fibers to carry the load than eight-strand rope. The second reason relates to the twist of the filaments in a rope. The more parallel the filaments are to the axis of a rope, the stronger the rope will be. Because of construction differences, double-braid rope filaments are more parallel to the rope axis than eight-strand rope filaments, thus resulting in a stronger rope size-for-size. New double-braid rope is less elastic than eight-strand rope for this same reason.

A new type of braided rope, 12-strand single-braid rope, has recently come on the market. The construction of 12 strand rope is shown in Figure 5-7. Single-braid rope is now being used as the pick-up rope at several SPMs. Its major advantage over double-braid rope is lower cost. Like double-braid rope, single-braid rope is a loose construction so it tends to be more susceptible to grapnel damage and harder to handle than eight-strand ropes.

5.3.5 Rope Construction Guidelines

The two rope components of SPM hawser systems, the hawser and the pick-up rope, have different functions favoring different rope constructions. Double-braid construction is recommended for SPM hawsers and eight-strand rope is recommended for pick-up ropes. Three-strand rope is not recommended for SPM hawsers or pick-up ropes because of its detrimental tendency to hockle.

SPM hawsers must have maximum strength and maximum strength retention. SPM hawsers must also be immune from hockling. Eight-strand and double-braid ropes will not hockle. Of these two constructions, double-braid ropes are stronger size-for-size because more filaments are contained per unit area and the filaments are more parallel to the rope axis.

Double-braid ropes are more likely to retain their strength with use for a number of reasons. Strand damage will cause less strength loss in double-braid ropes because loads are distributed through sixty-four cover strands rather than eight strands. The core strands of double-braid rope are well protected from abrasion and cuts because of the cover braided over the core. Eight-strand rope does not have this reserve protection because all eight strands are exposed. Double-braid ropes are also more abrasion resistant, because the outer surface of double-braid rope is much smoother than the outer surface of eight-strand rope. For the reasons of maximum strength and strength retention, double-braid rope is recommended for SPM hawsers.

The most important criteria for pick-up ropes are ease of handling and resistance to damage from grapneling or other methods used to pick up the rope. As with SPM hawsers, pick-up ropes must be immune to hockling so three-strand rope is not recommended.

Eight-strand, double-braid, and single-braid ropes are presently used successfully as pick-up ropes at SPMs. All three rope constructions are adequate for pick-up ropes, but presently eight-strand construction is preferred. Because construction of eight-strand rope is tight, it is easier for launch and tanker crews to handle. It is much harder to pull strands out of eight-strand rope so eight-strand ropes are more resistant to grapnel damage or other damage which might occur during rope pickup. For the above reasons, eight-strand rope is preferred for pick-up ropes.

5.4 SYNTHETIC ROPE BREAKING STRENGTHS

SPM hawsers are sized according to rope breaking strength. At SPMs, tankers are attached to the mooring by only one or two hawsers, so the hawsers must have high breaking strengths to take the mooring loads. As SPMs are being designed to moor larger tankers in more severe environments, rope manufacturers have increased their rope size capabilities in recent years.

5.4.1 Rating Large-Rope Strengths

The rated breaking strengths given by manufacturers of large synthetic ropes are in general extrapolated from smaller-rope breaking strengths or breaking strengths of rope components. Typical rated breaking strengths for eight-strand and double-braid ropes are given in Table 5-6.

Extrapolation has been necessary because the size and breaking strengths of large diameter ropes have increased at a faster rate than the capabilities of rope break-testing aparatus. Recently, a few large ropes in the size range used at SPMs have been break tested. New testing machines with increased break testing capabilities are being installed, and more testing of large ropes will soon be done. However, presently there is no test program underway to systematically and accurately determine the breaking strengths of large ropes such as those used at SPMs. Rope testing is discussed further in Section 6.

In general, European and United Kingdom rope manufactuarers publish minimum breaking strengths and United States rope manufacturers publish average breaking strengths. Minimum breaking strength is defined as the minimum rope strength guaranteed by the rope manufactuarer. There is general agreement that, for ropes less than or equal to 72 mm diameter (9 in. circumference), minimum breaking strengths are about 10% below average breaking strengths.

Above about 72 mm diameter (9 in circumference) most synthetic-rope breaking strengths are determined by extrapolation or by using realization factors. British Ropes and Hawkins and Tipson continue to publish minimum breaking strengths in this range. British Ropes have break tested a few ropes up to 144 mm diameter (18 in. circumference) and the results they have released exceed their published minimum breaking strengths. It should be noted that enough break tests to form a statistical distribution would be needed to confirm projected minimum breaking strengths. Wall Ropes have stated they have break tested all of their rope sizes and publish average breaking strengths for all ropes they manufacture. Wall Ropes stated their minimum breaking strengths are 10% below the published catalog strengths. All published Samson rope strengths are listed as approximate averages.

5.4.2 Determination Of Breaking Strengths By Extrapolation

Samson Ropes extrapolates the breaking strengths of large ropes from breaking strengths measured in tests of smaller ropes. This approach is used because Samson has not been able to conduct extensive break testing of ropes larger than about 80 mm diameter (10 in. circumference). A trend curve is developed based on the small-rope average breaking strengths. Expected breaking strengths for large ropes are extrapolated from the curve. (Benham, Trip Memorandum, Dec., 1976).

The breaking strengths listed in Samson's catalog for large ropes were determined by this manner. For ropes 80 mm diameter (10 in. circumference) and below, the size range for which they have extensive break-test data, Samson claims the published values are conservative average breaking strengths, but does not state what the minimum breaking strengths are. The extrapolated breaking strengths which Samson publishes for large ropes are claimed to be even more conservative. Samson's published breaking strengths include the reduction for splices.

Extrapolation of breaking strengths is potentially inaccurate and should be used with caution. Rope breaking strength is affected by a number of factors which may vary with rope size. Such factors are the twist of the yarns and strands, the helix angle of the strands, and the tensions of the braiding machines used in rope manufacture.

5.4.3 Determination of Breaking Strengths By Realization Factors

British Ropes, Hawkins and Tipson, and United Ropes relate the total strength of large ropes to the strength of rope components. The relationship between the strength of the component and the strength of the complete rope, known as the realization factor, is developed from tests on small ropes. The total rope breaking strength is determined by the following formula.

$$TS = CS \times NC \times R \qquad (5\text{-}1)$$

where TS = Total rope breaking strength

CS = Component breaking strength

NC = Number of components

R = Realization factor

The rope components most frequently used in establishing realization factors are strands or yarns. A number of samples (usually 15) of the rope component are removed and break tested. The average component breaking strength is determined. Samples of the complete rope are broken and, depending on whether minimum or average rope breaking strengths are desired, a minimum or average sample-rope breaking strength is determined. The realization factor is determined by dividing the sample-rope breaking strength by the component breaking strength times the number of components.

The tests are repeated for a series of rope sizes and realization factors are determined for each rope size. From these realization factors for smaller ropes, realization factors for larger untested rope sizes are extrapolated. To determine breaking strengths of large ropes, sample components of the large rope are break tested and the component breaking strength is determined. Total rope strength is then determined using the appropriate realization factor in equation 5-1.

The realization-factor method of determining rope strength is used in Europe for three-strand, eight-strand, and double-braid rope construction. Realization factors differ for each rope construction and may also vary for different rope materials and different rope sizes. The British Standards Organization and the International Standards Organization have published realization factors for three-strand and eight-strand ropes in sizes up to 96 mm diameter (12 in. circumference) (British Standards Specifications 3758, 3977, and 4928; ISO Standard 2307). Realization factors for larger three-strand and eight-strand ropes are being studied by the British rope manufacturers organization and will be published soon by British Standards.

British Ropes have determined their own realization factors for double-braid rope. To check the accuracy of their realization factors, British Ropes are conducting a program of break testing large ropes. To date, they have break tested ropes in sizes up to 144 mm diameter (18 in. circumference) (Woehleke, Trip Memorandum, Dec. 10, 1976). The tests on the few ropes which British Ropes have broken indicate their realization factors are sufficient. Repeated break testing of large ropes may show statistical variations which necessitate adjusting the realization factors.

The nature of the rope component on which the realization factor is based may be an important factor in the accuracy of the method. The breaking strength of individual synthetic filaments vary randomly, and a very large statistical sample would be required to determine a reliable average (Phoenix, 1976). Therefore, the strengths of individual rope filaments should not be used to determine total rope breaking strength. The component on which the realization factor is based must contain enough filaments to avoid the influence of statistical variations in filament breaking strength. The number of filaments needed cannot be defined, but the more filaments that are in the component the more accurate the breaking-strength determination will be. The realization factors published by British Standards and International Standards Organization are based on rope yarns. To obtain representative component breaking strengths, British Standards and International Standards Organization state the average breaking strength from fifteen yarns of the sample rope should be used with the realization factors.

Like extrapolation of rope strength, use of realization factors to determine total rope strength is potentially inaccurate and should be used with caution. Industry-standardized realization factors which relate composite rope strength to the strength of a rope component may not be accurate for all rope manufacturers because of variations in rope construction. As stated before, some of the factors which may vary are the twist of the yarns and strands, the helix angle of the strands, and the tensions of the braiding machines used in rope manufacture. Therefore, if realization factors are to be used to determine rope breaking strength, tests should be run to develop realization factors or to confirm the adequacy of standard realization factors for the ropes of a particular manufacturer.

Further caution must be exercised when using realization factors to determine double-braid rope strength. Samson Ropes have stated they believe separate realization factors should be determined for the core and the cover of double-braid rope (Benham, Trip Memo, Dec. 17, 1976). The separate realization factors are needed because geometric factors which affect rope breaking strength vary between the core and cover of double-braid rope. British Ropes take fifteen strands from the core and fifteen strands from the cover to determine a realization factor for each size of double-braid rope. (Woehleke, Trip Memo, Dec. 10, 1976).

5.4.4 Effect of Splices On Breaking Strengths

Splices are generally the weakest point in SPM hawser systems. Increased stresses in the hawser at a splice cause a reduction in hawser strength. Results of a 1975 Oil Companies International Marine Forum survey for SPMs showed that most of the broken hawsers reported in the study parted in the area of a splice. This report is discussed further in section 6. Both double-braid ropes and eight-strand ropes are reduced in strength approximately 10% when spliced, although the methods of splicing differ.

Eight-strand rope is relatively easy to splice because the strand ends are simply braided back into the rope. Double-braid rope splicing is more complicated. A simplified representation of the double-braid splicing procedure is shown in Figure 5-8. A length of the core and cover are separated at the hawser's end. Then part of the cover is threaded into the core and part of the core is threaded into the cover. The exiting and entering points of the core and cover in the splice are buried within the cover of the hawser. These general splicing techniques for eight-strand and double-braid ropes apply to both eye splices and end-for-end splices. An eye splice and an end-for-end splice in double-braid rope are shown in Figure 5-9. Complete splicing instructions can be obtained from the rope manufacturers.

When a double-braid rope is tensioned, the splices tighten up, thus preventing separation. However, when a double-braid rope is cycled under very low loads, double-braid rope splices may loosen and partially pull apart. To prevent this, all splices on double-braid ropes should be siezed and sewn with small line to keep them intact under low load.

The cross-sectional area of a splice is larger than the rest of the hawser. The change in load carrying area at the end of splices causes stress concentrations in the hawser in this transition zone. The stress concentrations decrease the rope strength, increase interstrand abrasion, and increase fatigue.

The catalog rated breaking strengths of double-braid ropes generally account for the strength reduction due to factory splices, but those for eight-strand ropes generally do not include the reduction in strength due to splices. Because they cannot be effectively gripped or otherwise terminated, double-braid ropes have normally been break tested with splices. Samson and British Ropes have stated their rated breaking strengths of double-braid rope include the effect of splices.

Eight-strand ropes are usually break tested with special end terminations. Therefore, the breaking strengths measured do not include the effect of splices. British Ropes, Hawkins and Tipson, and United Ropes, do not account for splice reduction in their published eight-strand rope rated breaking strengths. Wall Ropes claim they do include the reduction for splices in their published eight-strand rope rated breaking strengths.

Unless it is specifically stated in their catalog, the rope manufacturer should be consulted to determine if the catalog rated breaking strengths include the reduction for splices. It is preferable that break testing of large synthetic ropes be conducted with typical splices and that the effect of these splices be included in the rated breaking strengths.

5.4.5 Effect of Bends on Breaking Strengths

Any bend in a rope will decrease its strength. The amount of strength reduction is dependent on the radius of the bend. A gentle curve decreases rope strength negligibly but a sharp bend can cause a significant decrease in strength.

Three mechanisms cause strength reduction in bent ropes. The strength of the individual synthetic fiber is reduced when bent. However, the amount of fiber strength reduction due to bending is probably negligible in large ropes. The other two mechanisms are more important and will require a more detailed explanation.

When a rope is sharply bent, the outer fibers are under tension even when the rest of the hawser is slack. When a rope is loaded, the outer fibers of the bent portion are put under higher tension than the rest of the rope. Therefore, the outer fibers fatigue faster and reach their breaking strengths quicker than the rest of the rope filaments. The increase in tension in the outer fibers of the bent portion of a rope depends on the sharpness of the bend and on the rope diameter. As the bend in a rope becomes sharper the elongation of the outer fibers is greater so the tension difference also becomes greater. For equal bending radii, the elongation of outer fibers in larger ropes is greater than that in small ropes so the tension difference is greater.

Rope strength is also reduced at bends due to compression of the inner fibers. As a rope is bent over a radiused surface, such as a bitt or a thimble, the inner fibers are compressed against the surface. When a rope is bent over a very small radius or an uneven surface, these compression stresses can be very high, causing deformation and cutting of the filaments.

The degree of strength reduction from the three mechanisms depends upon the sharpness of the bend. To keep strength reduction to a minimum, rope manufacturers recommend minimum ratios of bending diameter to hawser diameter. A general rule of thumb is the bending diameter should be at least three times the rope diameter. If the hawser is to be flexed around the bend, as a rope run around a pulley, the ratio should be 8 to 1. (Samson Rope Manual).

Most thimbles supplied by rope manufacturers have a thimble diameter to rope diameter ratio of between 2 and 2.5 to 1. This is less than the recommended ratio of 3 as given above. However, the rope around the thimble in an eye splice is better protected against wear and is not as highly loaded as an ordinary bend in a rope. The effect of thimbles on eye splices and strops is discussed in subsection 5.8.1.

SPM hawser assemblies usually have loops, referred to as eyes, at each end for connection purposes. The hawser effective strength is decreased in the eyes because the hawser is bent. Part of the strength reduction is caused by a mechanism different from that discussed above.

The strength reduction mechanism is shown in Figure 5-10. An eye is formed by an eye splice or by lashing together the two legs of a strop. At the crouch of an eye the rope is bent. Ropes can carry loads only along their axis. Therefore, if each leg of a strop is under tension T, the tension of the rope in the eye is $T/\cos \alpha$. α is the angle between the axis of the hawser and the rope leading from the crouch to the thimble. The tension in eyes of single-leg hawsers which have eye splices, is $T/(2 \cos \alpha)$. The tension is lower because the single rope under tension T splits into two legs which share the load.

This increased tension in the eyes of strops is one mechanism which decreases strop strength. Strops and single-leg hawsers are discussed further in subsection 5.8.3.

5.4.6 Rope Breaking Strength Guidelines

As presented above, large-rope breaking strengths are presently determined by extrapolation, because the requirements for testing large ropes generally exceed the capacities of presently available break testing facilities. Until the true breaking strengths of the large ropes are determined through testing of a number of samples, it is recommended that extrapolated breaking strengths be treated as provisional breaking strengths and, thus, be used with appropriate caution.

It is recommended that a test program be initiated to determine new-rope breaking strengths for the large size ropes which must be used at SPMs. A sufficient number of tests should be run for each major material and construction of large rope to permit statistical variations to be determined. Such a test program would require a high load-stroke facility. Coordinated Equipment Company of Long Beach, Calif. is presently installing a very-high-load break-test machine which should be capable of break testing the largest ropes presently available (Flory, Trip Memorandum, Jan. 27, 1977). Testing of large synthetic ropes is discussed further in Section 6.

5.5 STRENGTH REDUCTION OF USED ROPES

SPM hawser strength may degrade in service due to fatigue, cuts, and abrasion. Synthetic ropes can also be degraded by exposure to chemicals or ultraviolet radiation, but, as explained in subsection 5.2.6, SPM hawsers are not usually affected by these strength reduction mechanisms because of their size and the nature of their service.

5.5.1 Fatigue Of Synthetic Ropes

Experiments have shown that synthetic-rope strength may degrade due to cyclic loading. The strength degradation of SPM hawsers due to cyclic loading is caused by a number of mechanisms. Synthetic-rope strength degradation is similar to metal fatigue in that the strength decreases gradually under the influence of cyclic loading. Hawser-strength degradation due to cyclic loading is referred to, therefore, as fatigue. However, the mechanisms causing loss of strength are not necessarily the same as those which cause metal fatigue.

The mechanism of synthetic rope fatigue is not fully understood and cannot be accurately predicted. Part of the strength degradation may be caused by fatiguing of the hawser fibers but it is believed interstrand abrasion causes the major strength degradation.

When a rope elongates, the fibers rub against each other causing interstrand abrasion. As explained in subsection 5.3, the fibers are twisted and intertwined in constructing the rope. Each fiber elongates when a rope is loaded, but the elongation is also caused by the straightening of the twisted and intertwined fibers. As the fibers are elongated and straightened, they move relative to each other. The ultimate result is interstrand abrasion and hawser strength degradation.

One factor which may affect the rate at which the fibers abraid due to elongation is their position in the hawser. When a rope is loaded, the twist of the fibers relative to the axis of the hawser creates a compressive load. The compressive load is greatest at the center of the hawser and zero at the outer surface. This compressive load on the inner fibers increases the interstrand abrasion caused by the fibers rubbing against each other. The increasing interstrand abrasion caused by the compressive load is dependent on the loads and the size of the hawser. It is theorized that larger ropes may fatigue more rapidly than small ropes. No data has been found relating effect of synthetic rope size on the rate of fatigue. However, because rope size may be a factor in interstrand abrasion, use of fatigue tests on small ropes to determine the fatigue of large ropes may not be accurate.

Fiber temperature is another important factor in the rate of interstrand abrasion. Under cyclic loading heat builds up in the rope. The higher the temperature, the more rapid the interstrand abrasion. As fiber temperature approaches the material melting point, interstrand abrasion can be very rapid.

Rate of loading can affect rope strength in several ways. Rapid cycling of a rope will cause heat build-up which results in interstrand abrasion and loss of strength. However, rope manufacturers report a rapid rate of loading during break testing can result in the rope breaking at a higher strength than would have been expected at a moderate rate of loading. Very rapid loading of ropes, amounting to shock loading, can cause ropes to lose strength or break even though the shock load did not approach the breaking strength of the rope. Shock load conditions are generally not experienced in SPM hawsers, but may be the cause of some hawser failures.

Most of the energy stored in an SPM hawser as it elongates is released as the hawser contracts, however, some energy is dissipated as heat when the hawser contracts. The amount of heat produced depends on the magnitude and frequency of the loads. Heat build-up is not usually a problem in SPM hawsers as they are usually wet when a tanker is moored and get periodically immersed while a tanker is moored. Generally, SPM hawser temperatures would probably not increase sufficiently to increase interstrand abrasion and accelerate fatigue.

Tight lashing or seizing on the SPM hawser may affect the rope fatigue rate. The two legs of a strop are usually lashed together. Single-leg hawsers are usually seized at splices and sometimes lashed with small rope to hold buoys in place. The lashing rope puts a compressive load on the load carrying fibers. Nylon rope swells and shrinks in length when wet. Thus, nylon lashing applied to a nylon rope will tighten when wet. The quantitative effect of lashing-rope-induced compressive loads on SPM hawsers is not known. However, Aramco has attributed a failed SPM hawser to lashing which was applied too tight (Flory, Trip Memorandum, April 18, 1977). To prevent detrimental compressive loads, it is recommended that the lashing rope tension be kept to a minimum and that polyester, or some other material be used instead of nylon for the lashing rope.

Another factor which may affect rope fatigue is the load range over which the rope is cycled. Available data indicates a rope which is cyclically loaded over a given load range with zero load as the lower limit will fatigue quicker than if it were cycled over the same load range with a load above zero as the lower limit. Because the load-elongation curve for synthetic ropes is non-linear, as shown in Figure 5-1, the rope is cycled over a longer stroke when cycled to zero load. For example, on the curve for nylon the rope elongates 24% when loaded from 0 to 20% of breaking strength, but elongates only 12% when loaded from 10 to 30% of breaking strength. Interstrand abrasion is higher, and thus loss of strength is greater when the rope is loaded over a 20% of breaking strength range with 0 as the lower limit than when it is cycled over the same range with 10% of breaking strength as the lower limit.

5.5.2 Results of Cyclic Load Tests

As stated above, hawser fatigue is not fully understood. A number of cyclic-loading test programs have been conducted on synthetic ropes. These tests have usually been conducted to compare the fatigue characteristics of different rope materials or fiber coatings for the same type of rope construction. Important parameters and test procedures have varied between the test programs, so it is difficult to compare test results. Some of the more important factors which vary between the various test programs are; fiber manufacturer, fiber type, fiber grade, length and arrangement of test specimen, rate of loading, and the load ranges over which the test ropes were cycled. Brief discussions of the results of several cyclic loading test programs follow.

Hawkins and Tipson have performed cyclic loading tests on small diameter three-strand nylon 6.0, polyester, polypropylene, and supermix ropes (Woehleke, Trip Memorandum, Dec. 10, 1976). Supermix is a rope produced by Hawkins and Tipson using a mixture of polypropylene and polyester fibers. Nylon, polyester, and supermix ropes were cycled to 75% of their breaking strengths. Each loading cycle lasted approximately one minute. Two nylon 6.0 ropes were tested, one broke after 49 loadings and the other broke after 55 loadings. The cycling was stopped after 200 loadings on the one polyester rope and one supermix rope that were tested. The two ropes were then loaded to breaking. The polyester rope broke at its original strength and the supermix rope broke at 91.5% of its original strength.

Hawkins and Tipson repeated the tests, loading four ropes to 50% of their breaking strengths. The rope materials were nylon 6.0, polyester, polypropylene staple, and polypropylene film. All ropes survived 200 loadings and were then broken. Their final breaking strengths follow.

Nylon 6.0	79.5% of original strength
Polyester	100.0% of original strength
Polypropylene staple	93.4% of original strength
Polypropylene film	102.0% of original strength

Shell Oil has run a more complete fatigue testing program. 56 mm diameter (7 in. circumference) ropes of various materials and constructions were tested. Complete details of the test program and its results have not been made known. The results published by Langveld (Oct. 15, 1973) are shown in Figure 5-11. The figure shows a classic cyclic-fatigue-failure curve with a horizontal asymtote at about 30% of original breaking strength indicating failure will probably not occur if cyclic loads do not exceed this value. The results were used by Shell to help develop factors of safety for SPM hawsers (see subsection 5.7).

United Ropes ran cyclic loading tests to compare nylon 6.0 and nylon 6.6 fibers (Flory, Trip Memorandum, Oct. 27, 1976). Several types of nylon 6.6 fibers with different coatings were tested. 56 mm diameter (7 in. circumference) 9-strand ropes were cycled from zero to 30% of breaking strength at one load cycle per minute. United first ran comparison fatigue tests on nylon 6.0 and nylon 6.6 ropes in a dry condition. No significant difference in performance was noted in these dry tests up to 5000 cycles.

The same tests were then conducted on nylon 6.0 and nylon 6.6 ropes wetted with fresh water. After as few as 1500 cycles, the nylon 6.0 showed severe abrasion and melting between the adjacent strands. Nylon 6.6 did not exhibit this effect after 5000 cycles.

United Ropes then conducted fatigue tests on nylon 6.6 ropes with several surface treatments. Ropes with and without surface treatments were tested in the wet condition. Ropes impregnated with oil or grease for lubrication were also tested wet. Table 5-7 gives the results of these comparison tests. The results show that the coated ropes lost little strength after cycling but that th noncoated ropes lost about 10% of their rated breaking strength after 5000 cycles. The results also show that the oil lubricated ropes lost about 15% of their rated breaking strength. The oil tended to extrude out of the fibers and strands and was not effective as a lubricant. Its presence, however, was apparently detrimental to the fibers of the rope.

Cyclic loading tests were run by Wall Ropes a number of years ago to assess different nylon 6.6 coatings and to compare polyester and nylon (Woehleke, Meeting Memorandum, Feb. 2, 1977). The tests were performed on wet 48 mm diameter (6 in. circumference) three-strand ropes. Each test sample was cycled six times in the morning and six times in the evening to 60% of the new rope break strength. The rate of loading was about 30 cm/minute (12 in./minute). The rest period in testing allowed the ropes to regain some of their stretch and it made the testing apparatus available for other work. A few nylon ropes with experimental coatings failed after approximately 50 loadings. The average for the nylon ropes was about 600 loadings. The best was 900 loadings. The polyester rope tested was cycled 2,000 times before the test was terminated. The cycled polyester rope was then loaded until it broke. The rope had retained nearly its original strength.

British Ropes cyclically loaded 14 mm diameter (1.75 in. circumference nylon, polyester, and polypropylene ropes (British Ropes, 1966). The ropes were loaded 120 times between 0 and 75% of their breaking strengths. All sample ropes surived the cyclic loadings. The reactions of the ropes to the cyclic loading were studied, and it was concluded that nylon and polyester ropes have better cyclic loading characteristics than polypropylene ropes.

A fatigue-testing program on small, wet nylon and polyester ropes is presently being conducted by DuPont. Three-strand ropes are cyclicaly loaded from 0 to 75% of their breaking strengths. Not enough tests have been run to draw any conclusions yet.

Exxon Research and Engineering Company cyclically loaded 64 mm diameter (8 in. circumference) nylon three-strand ropes and nylon yarns at the National Engineering Laboratory in Scotland (N.E.L. Report, March 16, 1976). In a preliminary test, a test rope was cycled between 20 and 40% of its breaking strength. No apparent strength degradation was produced even when cycled more than 30,000 times.

Different loading ranges were then tested. Test ropes were cycled between 24% and 44% of their breaking strength in some tests and 24% and 75% of their breaking strength in other tests. The loading period varied between two and three cycles per minute. The results of the Exxon tests are given in Table 5-8.

5.5.3 Conclusions Based on Available Cyclic-Loading Test Data

On the basis of these limited test results, conclusions which were and can be drawn with regards to SPM hawsers are:

1) For top-grade nylon 6.6 rope, the cyclic loading pattern must include at least transient loads exceeding 60% of breaking strength for rope strength to degrade quickly.

2) Low-grade nylons may be subject to rapid degradation due to fatigue, especially when wet.

3) Polyester apparently has excellent resistance to fatigue.

4) Because of the many factors which may effect rate of fatigue of synthetic ropes and the random cyclic-loading nature of SPM mooring loads, it would be difficult to devise a cyclic loading rope test program which could accurately model the life of an SPM hawser.

5.5.4 Effect of Cuts and Abrasion

SPM hawser strength can also degrade from cuts and abrasion. SPM hawser systems should be designed to minimize the chance of cuts and abrasion. However, because of the type of service SPM hawsers experience, cuts and abrasion are likely to occur.

If abrasion is observed, action should be taken to prevent any additional damage. If abrasion is allowed to continue hawser strength may decrease rapidly. When a cut or abraided area is observed, the hawser should be examined closely and evaluated to determine if the hawser should be retired. Cuts and abrasion of ropes are discussed further in Section 6.

5.5.5 Strength of Used Ropes Guidelines

The strength of an SPM hawser may degrade gradually with time due to fatigue. However, a moderate number of cycles not exceeding 60% of breaking strength will not cause serious loss of strength, and top-quality nylon can withstand very many cycles not exceeding 30% of breaking strength. Cuts and abrasion may also decrease hawser strength. To check for cuts and abrasion, SPM hawsers should be frequently inspected. Due to fatigue, SPM hawsers should be periodically replaced irregardless of appearance. SPM hawser inspection and replacement criteria are discussed in detail in Section 6.

5.6 ROPE ELASTICITY CHARACTERISTICS

The SPM hawser elasticity is an element in the elasticity of the mooring system, and the mooring system elasticity has a significant influence on mooring loads. This relationship is discussed fully in subsection 2.6. In many SPMs a high degree of hawser elasticity is desired.

5.6.1 Apparent Discrepancies in Available Data

Although many elasticity curves for synthetic ropes of various constructions and materials in new and broken-in condition have been published, comparison of various curves is difficult and at times confusing. Sufficient information on the exact type of material or construction and on the manner in which the tests were run and the results analyzed and presented are usually not included. For tests on broken-in or used ropes, data on the nature of the break-in or use are usually lacking.

Differences in the manner of testing and the presentation of data appear to be the causes of most of the discrepancies in elasticity data. Normally a rope is tensioned to a slight nominal load before recording of load-elongation characteristics. However, some of the published data is from tests in which the rope was initially completely relaxed. Such data characteristically shows a much larger elongation.

Rate of loading will affect load-elongation characteristics as well as breaking strength. Elongation is normally plotted against percent of breaking strength, but the breaking strength is seldom defined. Some data may be plotted against rated minimum or average breaking strength, and other data may be plotted against the break strength obtained in the particular tests.

Comparison of elongation data for broken-in or used ropes is made even more difficult by other factors. After cycling, the rope is permanently set at a longer length and the elasticity is less. Elasticity remaining after the rope has been cyclic loaded will depend on the magnitude and rate of loading, if tension was maintained for some time, and on the length of time and treatment of the rope since loading. Much of the elongation data for broken-in rope is plotted against percentage of new breaking strength. The elongation as a percent of the length after permanent set is of more importance to SPM designers. The percent of permanent set over original length should be given.

It is hoped that in the future investigators and manufacturers will be more consistent in their test procedures and more complete in their presentations. It is recommended that further basic work be done on new and used synthetic-rope elasticity characteristics. The following discussion presents some data from which general comparisons can be made.

5.6.2 Rope Material and Elasticity

Elasticities of new polypropylene, polyester, and nylon three-strand ropes are compared in Figure 5-1. The differences in elasticities of the various fibers are discussed in subsection 5.2.2.

There is some disagreement between U.S. and English data on the relative elasticities of polypropylene rope and polyester rope. English rope manufacturers usually heat set three-strand and eight-strand polyester and nylon rope. Heat setting causes the fibers to better conform to the twisted construction, but it also lowers the strength of the rope slightly. U.S. military specifications prohibit heat setting, and thus, U.S. manufacturers usually do not heat set their ropes. Heat setting of nylon and polyester rope increases the percentage of elongation at a given load, at least when new.

Comparison of elasticity curves for non-heat-set nylon, polypropylene, and polyester ropes, as shown in Figure 5-1, shows nylon is most elastic, polypropylene is less elastic, and polyester is least elastic. However, comparison of the elasticity curve for heat-set polyester rope with that of non-heat-set polypropylene rope, as published by English rope manufacturers, will show polyester is slightly more elastic than polypropylene.

5.6.3 Rope Construction and Elasticity

The elasticities of new three-strand, eight-strand, and double-braid nylon ropes are shown in Figure 5-13. This data is for wet rope. The samples were soaked in water for 24 hours prior to testing. No data for dry ropes of these constructions tested in a comparable manner could be found.

This data shows eight-strand rope to be the most elastic and double-braid rope to be the least elastic. Three-strand rope is only slightly less elastic than eight-strand rope, and data from some sources indicates the two constructions have essentially identical elasticity characteristics.

Elasticities of all three constructions will vary with the details of construction, such as the twist of fibers, the length of lay, and the tightness of construction. Thus, data for ropes of the same construction but from different manufacturers may show variations. The strength of the rope will generally vary inversely with elasticity for ropes of the same construction.

5.6.4 Elasticity of Broken-In Ropes

Figure 5-12 shows elasticity curves for the polypropylene, polyester, and nylon three-strand ropes presented in Figure 5-1 after they have been broken-in. The elasticity of each rope in the broken-in condition is approximately two-thirds of that when new. Apparently, most of this loss in elasticity is due to tightening of the rope structure. No data has been found to indicate that the elasticity properties of any of the three materials substantially changes as the ropes are broken-in.

Comparison of elasticity curves of new three-strand, eight-strand and double-braid ropes show that double-braid rope is substantially less elastic than either three-strand or eight-strand rope. Comparison of elasticity curves for broken-in rope based on percent of original length indicates three-strand rope and eight-strand rope remain more elastic than double-braid rope. Figure 5-14 shows the broken-in elasticities of the three samples of Figure 5-13 after cycling 10 times to 50% of the breaking strength and then maintaining that load for 24 hours. The elongation is plotted in percent of new-rope length.

Elasticity of broken-in ropes is more meaningfully compared on the basis of elongation in percent of the broken-in rope length when the energy-absorption capacity of the rope is a consideration. Comparing the data of Figure 5-14 with that of Figure 5-13 shows the broken-in double-braid rope had a permanent elongation of about 8% while the eight-strand rope had a permanent elongation of about 18%.

Data provided by British Ropes for broken-in three-strand, eight-strand, and double-braid nylon rope is shown in Figure 5-15. These ropes were cycled in the dry condition to approximately 40 percent of the new breaking strength. The data shows that, when plotted on the basis of broken-in length, there is very little difference in the load-elongation characteristics of the three types of rope construction. Comparison of this data with other data is complicated by the facts that elongation was measured from a relaxed condition, and that breaking strength is plotted on the basis of percent of broken-in strength. Data provided by Samson Ropes (1969) for eight-strand and double-braid dry nylon rope cycled 200 times also shows that the load-elongation curve for broken-in eight-strand rope is almost the same as that for broken-in double-braid rope.

The differences in elongation characteristics for new ropes of various constructions but of the same material are due to construction differences, primarily the length and tightness of the lay of the fibers, yarns, and strands. After the ropes are broken-in, the tightness of construction is increased and the lays may become more equivalent. The primary mechanism of elongation is then the elasticity of the material. This may explain why three-strand, eight-strand, and double-braid ropes have different load-elongation characteristics when new but similar load-elongation characteristics when broken-in.

Data on the elongation characteristics of used or broken-in rope from various sources does not show good agreement. Figure 5-16 compares data for nylon double-braid rope from various sources. In their catalog, Samson states that nylon double-braid rope looses 6 percent of its elasticity when broken-in. The test data from Wall is after 10 cycles as shown in the prevous figure. The test data from Samson is apparently taken after many more cycles. It is not clear if the various data shown in the curve were plotted on the basis of new or broken-in rope length.

5.6.5 Rope Elongation Guidelines

For the purpose of design and analysis of SPMs, the load-elongation characteristics of broken-in rope should be used. As explained in sub-section 2.6, the elasticity characteristics of the mooring system are important in determining mooring loads, and, especially in relatively shallow water, the elasticity of the hawser is an important contributor to mooring system elasticity. After only a few loadings, the hawser will have a permanent elongation and the load-elongation characteristic will be essentially that of a broken-in rope.

When compared on the basis of broken-in length, there is little difference in the elasticity curves of three-strand, eight-strand, and double-braid rope. New three-strand and eight-strand ropes are significantly more elastic than double-braid rope, and therefore, might be preferred. However, on the basis of broken-in elasticity there is no reason to favor one construction over another.

Data on the elasticity of used or broken-in ropes does not show good agreement. The manners of cycling the ropes and the methods of plotting the data have varied. There is a need for a coordinated program of obtaining load-elongation characteristics for ropes of various constructions and materials on a consistent basis.

5.7 SPM HAWSER FACTORS OF SAFETY

The factors of safety for single and dual-hawser systems which are now called for by the American Bureau of Shipping (ABS) Rules for SPMs (1974) are 1.67 and 2.5 respectively. In this study, the bases by which SPM hawser factors of safety have been set in the past are re-assessed. The recommended practices in which mooring loads are defined and determined, the certainty with which hawser breaking strength can be determined in both the new and used condition, and the manners in which tankers are moored to SPMs are considered. Based on this review, slightly higher factors of safety than those specified by ABS are recommended for SPMs at U.S. deepwater ports because of their critical sensitive nature.

5.7.1 Factors Which Limit Hawser Factors of Safety

The view might be taken that the cost of very large hawsers is not high, and therefore, very high factors of safety should be set for the hawser portion of the mooring system. It is true that economics should not be a criteria in sizing hawsers. However, other more practical and technical considerations require that hawser size and, therefore, hawser factor of safety can not be excessive.

The influences of over-all mooring system elasticity and of hawser elasticity on mooring loads have been discussed in subsection 2.6. Hawser stiffness increases proportional to hawser breaking strength. An increase in hawser stiffness will cause an increase in mooring loads, not only in the hawser, but also in other portions of the mooring system, such as the anchor chain, the buoy structure, and the shipboard mooring fittings. The increase in mooring load is not proportional to the increase in hawser strength; an increase in hawser strength will lead to an increase in hawser factor of safety, but the increase in factor of safety will be less than the increase in strength.

The increase in factor of safety achieved by increasing the breaking strength of the hawser may actually decrease factors of safety in other portions of the mooring system. This fact may be overlooked when increasing hawser size on existing SPMs. In a new design, the strength of the buoy and anchoring system can be increased, although at some cost. However, the strength of shipboard mooring fittings cannot generally be increased. Limitations of shipboard mooring fittings are discussed in Section 4.

Although no part of the mooring system should be designed to fail, the consequences of a failed hawser are usually less severe than the consequences of a buoy, foundation, or anchor-chain failure which in turn could cause failure of cargo piping or hose, or the consequences of a shipboard mooring-fitting failure which could cause personal injury. The SPM hawser may preferably be the weakest part of the mooring system, provided that cargo loading is discontinued, and hoses are disconnected in environments approaching the design environment.

A practical limitation on hawser size is the difficulty of handling very large hawsers and of lifting them and the associated thimbles and hardware to the forecastle of the tanker. The winching capacity of many tankers is limited. ARAMCO personnel have stated some tankers can only lift about 2,300 kg (5,000 lbs) of hawser and hardware (Flory, ARAMCO Trip Memorandum April 18, 1977). The thimble alone for a 192 mm diameter (24 in. circumference) hawser weighs approximately 750 kg (1,700 lbs), and a hawser of this size when wet would weigh approximately 300 kg per meter (200 lb per foot) of length. Thus, a hawser of this size would probably be impractical unless special provisions were provided onboard the tanker.

Unreasonably high hawser factors of safety which would require the use of very large, very stiff hawsers would cause difficulties in design; may increase the risk of failure of other parts of the mooring system; and may be impractical because of shipboard mooring fitting and winch limitations. Realistic factors of safety should be prescribed and met through rational design practices and adequate hawser inspection and replacement criteria.

5.7.2 Definition Of Mooring Load And Breaking Strength

The factor of safety of the SPM hawser is defined as the ratio of the new-hawser breaking strength to the predicted maximum mooring load. A realistic basis for determining hawser loads and a consistent definition of maximum mooring loads are required in setting factors of safety. The maximum mooring load at an SPM is a function not only of the environment, the tanker, and the mooring system, but also of the statistics of the randomly varying load in the hawser. The influences of the environment, tanker, and mooring system are discussed in Section 2. The manners in which mooring loads are determined and maximum mooring loads calculated by statistical analysis are discussed in Section 3.

There will only be a very small chance that the maximum mooring load will be exceeded when the design basis is realistically set, that is: if the mooring loads are accurately determined; the most critical tanker, tanker condition, and environment are used to determine the maximum mooring loads; and the proper statistics are used in the analysis. Furthermore, because of the nature of the statistics of the mooring loads and the combination of factors which produce the loads, there will only be a small chance that the maximum mooring load will be approached. Therefore, if the maximum mooring load is realistically determined and defined, the factor of safety which should be applied to that load does not need to account for loads in excess of that load or for frequent applications of that load.

The strength on which the factor of safety is based should be defined as the minimum breaking strength of the synthetic rope as rated by the manufacturer. Most rope manufacturers will state the minimum breaking strength which they will guarantee. Rope manufacturers indicate the minimum breaking strengths are typically approximately 10% less than the average breaking strength.

The breaking strength on which the factor of safety is based must take into account the effect of splices. Splices are frequently a point of weakness of hawsers in service. Generally, double-braid rope is tested with typical splices, and thus the rated breaking strength of this rope accounts for the reduction in strength due to splices. Eight-strand and three-strand ropes are generally tested without splices, and thus a reduction, generally 10%, must be made in the rated breaking strength of these ropes for splices.

Although the new breaking strength of strop-type hawser arrangements may be as little as about 1.7 times the strength of the individual legs due to the effect of bend in the eyes, as discussed in subsection 5.8.3, this does not appear to be a point of weakness of strops in service. A more important factor is that the bulk of each leg in a strop is not as great as that of a single-leg hawser of the same strength; thus the strop strength may degrade faster in service due to damage and wear. Therefore, it is recommended the breaking strength of strop-type hawsers should be taken as 90% of the total strength of the two legs. This topic is covered further in subsection 5.8.3.

The strength of nylon rope is known to be reduced by approximately 10% when wet. The hawser at an SPM floats in the water between moorings and dips to the water occasionally when connected to the moored tanker. Thus, the SPM hawser is usually wet when loaded. Nylon is the most common material for SPM hawsers and has given good service in the past without taking into account its reduced strength when wet in setting safety factors. Manufacturers have stated that, because nylon has been used successfully for years, the properties of nylon are well known, and the reduction in strength when wet is at least partially offset by an increase in elasticity, there is no need to set a higher criteria for nylon than for other synthetic rope materials. This reasoning appears sound; no reduction in strength need be taken for wet nylon in setting SPM hawser factors of safety.

5.7.3 Single-Hawser System

The number and arrangement of hawsers between the SPM and the moored vessel must be considered in setting safety factors. When only one hawser is used to moor the tanker, total reliance must be placed on this single hawser as there is no backup in case of failure. A strop-type assembly or an assembly having two or more hawsers joined to a single chafing chain at the tanker end are considered to be single hawsers for the purpose of this discussion. Breaking load is defined as the effective total breaking strength of such assemblies.

As a result of extensive SPM operating experience and rope testing, Shell set criteria that the working load in a single-hawser system should be no more than 40% of the breaking strength and that the maximum load should not exceed 60% of the breaking strength (Langeveld, 1973). Shell gave no definition of working load to be used in conjunction with the criteria. ABS incorporated the criteria that the maximum mooring load should not exceed 60% of the breaking strength of a single-hawser system in its Rules for SPMs. Stated another way, the ABS criteria is the factor of safety of a single-hawser system should be 1.67 based on breaking load to maximum mooring load (ABS, 1974).

Re-examination of this criteria shows that it may be unconservative for critical applications. The principle consideration is there is no redundancy in a single-hawser system and the tanker is unrestrained in the event of an SPM hawser failure. If hoses are still connected and cargo transfer is still underway, an oil spill will probably result. If the mooring is in restricted waters the tanker may be involved in a collison or grounding before it can recover control and get underway. Other considerations are that the SPM hawser may fail at a load below its rated breaking strength due to either latent quality defects or to undetected wear and damage, and that mooring loads higher than the predicted maximum mooring load may occur due to extreme event statistics or operational mishaps.

A factor of safety of 2.0 on breaking load to maximum mooring load is proposed as being more prudent for single hawser systems in sensitive locations. This criteria limits peak mooring loads to half the rated breaking strength. This criteria is slightly higher than that now generally used. Few SPM hawser failures were experienced with systems designed to the previous criteria. However, the consequences of the failure of an SPM hawser at a U.S. deepwater port warrant this revised criteria.

5.7.4 Dual-Hawser System

An arrangement of two or more independent SPM hawsers in parallel between the SPM and the tanker provides redundancy in the event one line fails. However, in the event of failure of one line, the remaining hawser or hawsers must be strong enough to withstand the entire mooring load. Also, because of the arrangements of mooring fittings on board tankers, the mooring load may not be shared equally between the hawsers. Systems with three or or more hawsers are generally impractical because of the arrangement of mooring fittings on tankers. Therefore, consideration is given here to dual-hawser arrangements only.

For dual-hawser systems, the ABS Rules for SPMs (1974) require a factor of safety of 2.5 on the maximum mooring load to the combined strength of the two lines. This criteria was essentially in agreement with criteria which had been independently developed and successfully employed by Exxon, Shell, IMODCO, and SBM Inc.

With dual-hawser systems the total mooring load may not be shared equally between the two hawsers. As discussed in subsection 4.5.1 and shown in Figure 4-12, if the two hawsers are not brought through the same or adjacent bow chocks, one hawser will take more of the mooring load as the tanker yaws. Shell has stated that in a dual mooring hawser system the higher loaded line may take two thirds of the total mooring load. Based on this philosophy and the factor of safety criteria of 1.67 cited set for single hawsers, Shell set a factor of safety of 2.5 for dual-hawser systems (Langeveld, 1975).

Factors in the sharing of loads between dual hawsers are the degree of elasticity of the hawsers, the matching of length and elasticity of the two hawsers, and the matching of positions at which the hawsers are attached on the tanker forecastle. As the tanker yaws, the percentage of sharing of loads between two hawsers is a function of the difference in elongation between the two hawsers and the elasticity of the hawsers. With very stiff SPM hawsers, only a small difference in elongation can cause a large difference in loads between the two lines. Also, if the two hawsers are of unequal length or of different elasticities, the loads will not be equally shared even if the chocks are closely spaced and the tanker does not yaw. If one hawser is attached to a fitting very near the chock and the other hawser is attached to a fitting far from the chock the effect is the same as with hawsers of unequal length.

In this study calculations were made of the sharing of hawser load experienced in a dual-hawser system due to tanker yaw. A maximum tanker-yaw angle of 45° occurring simultaneous with the design mooring load was assumed. Two broken-in 50 m (165 ft) long double-braid nylon hawsers were used in the analysis. The relationship of differential hawser length to tanker yaw and spread of bow chocks were those shown in Figure 4-12 and discussed in subsection 4.5.1.

With a factor of safety of 2.5 and a chock spacing of 2 m (6.5 ft), the higher loaded line was loaded to approximately 50% of its breaking strength, an acceptable load level. However, with a chock spacing of 10 m (33 ft), one hawser will take all the load and be loaded to 80% of its breaking strength. With a factor of safety of 3.0 on a dual-hawser system under the same circumstances of one hawser taking all of the load, that hawser is loaded to only 67% of its breaking strength.

A dual-hawser system provides a redundancy or safety backup in that if one hawser breaks the tanker can be held until corrective action is taken. This is only effective, however, if each hawser is strong enough to withstand the maximum mooring load by itself. With a dual-hawser factor of safety of 2.5, one hawser would be loaded to 80% of its breaking strength in the absence of the other hawser. A load of this magnitude could fail a used hawser. Although it would appear to be unlikely that two very high mooring loads would occur in sequence, on several occasions the second line of a dual-hawser system has failed shortly after the first line failed. With a dual-hawser system factor of safety of 3.0, one line alone would be loaded only to 67% of its breaking strength.

A factor of safety of 3.0 is more prudent for dual hawser systems at SPMs at U.S. deepwater ports than the 2.5 factor of safety now generally used. The 2.5 factor of safety is adequate provided that both hawsers share the mooring load within a ratio of 2:1. However, for terminals which must serve a wide variety of tankers which may not be equipped with proper mooring fittings and chocks, a high inbalance of load may occur. Also, if one hawser parts, the second hawser may be loaded to near its breaking strength if a factor of safety of 2.5 is used.

5.7.5 Alternative Factor of Safety On Significant Load

In some mooring applications, repeated cycling to loads slightly lower than the maximum mooring load could cause fatigue of the ropes, thus reducing the breaking strength and causing failure. For example, at a site where long period swell was the predominent factor, the nature of variation in mooring loads may be almost regular and the ratio of maximum load to significant load may be relatively small.

Based on the analysis of cyclic-load fatigue data given in subsection 5.5.2, a factor of safety of 4.0 on significant mooring load appears to be prudent. Applying a factor of safety of 4.0 on the significant mooring load would essentially restrict cyclic loads to less than 25% of breaking strength. Cyclic-load data for top-quality nylon ropes show they will not be fatigued by even a large number of cyclic loads up to 30% of breaking strength.

The safety factor criteria of 4.0 based on the significant mooring load corresponding to the maximum mooring load is proposed to preclude degradation of hawser strength due to cyclic loading. The significant mooring-load safety-factor criteria will govern when the maximum mooring load is less than twice the significant load for single-hawser systems. It is unlikely that situations will exist which will cause this criteria to govern in the case of dual-hawser systems.

5.7.6 Recommended Factors of Safety

The above re-examination of the bases for factors of safety concludes that factors of safety slightly higher than those now recommended by ABS and commonly used by the industry should be used for hawsers at SPMs for U.S. deepwater ports. A factor of safety of 3.0 for dual-hawser systems is recommended. A factor of safety of 2.0 is recommended for single hawsers. These factors of safety are defined as the ratio of the rated minimum breaking strength of the synthetic rope making up the hawser system, with splices and other factors accounted for, to the predicted maximum mooring load. Another criteria of a factor of safety of 4.0 on significant mooring load is recommended to limit fatigue effects.

These factors of safety are based on the premise that the maximum SPM hawser load has been realistically and accurately determined and the new breaking strength of the SPM hawsers have been properly assessed. They are based on the practices that the SPM hawsers are frequently inspected and periodically replaced to assure their breaking strengths have not seriously degraded.

5.8 SPM HAWSER ASSEMBLIES

The SPM hawser consists of a synthetic rope spliced to form an endless strop or with eye splices at each end. The ends of the strop or the eye splices are enclosed by thimbles and are attached to short lengths of chain to prevent chafing on the buoy and the tanker. Chafing chains and related hardware are discussed in subsection 5.9. Unless the synthetic rope is lighter than water, floatation must be placed on the hawser assembly.

5.8.1 Hawser Thimbles

Hawser eyes should generally be protected by thimbles. The term hawser eye refers to the loops formed at the end of the hawser. Thimbles protect the hawser from abrasion, keep the hawser at the proper bending radius, and distribute contact stresses over a sufficient length of hawser. Eyes without thimbles may be acceptable where the eyes are placed over large bollards on the SPM. A typical thimble is shown in Figure 5-17.

Factors which are important in thimble design are thimble size and shape, abrasion protection placed between the hawser and thimble, and thimble material. Thimbles are sized according to rope size to prevent significant strength decrease due to hawser bending and to provide adequate surface contact area between the hawser and thimble. As explained in subsection 5.4.5, a sharp bend in a hawser will significantly reduce its strength. Hawser abrasion caused by rubbing between the thimble and the hawser must also be protected against.

The mouth or entrance opening in the thimble should allow for hawser elongation. When loaded, the hawser eye elongates resulting in the crotch of the eye pulling away from the thimble. If the thimble housing tightly encloses the legs of the eye near the crotch, the outside of the legs will rub against the thimble, as shown in Figure 5-18, resulting in hawser abrasion. Therefore, the opening in the thimble should be designed to allow the eye to elongate without contacting the lip of the opening.

The shape of the thimble which contacts the rope should be smooth and without sharp corners. Any sharp corners will cause stress concentrations in the hawser which will reduce hawser strength. Half-circular shaped thimbles as shown in Figure 5-18, present abrupt corners which cause stress concentrations. The shape of the hawser should be continuous and be smoothly faired. The thimble shape of Figure 5-17 is well designed.

Rope manufacturers are now producing special detachable thimbles. Detachable thimbles have the advantage the thimbles can be reused by removing them from used hawsers and placing them on replacement hawsers in the field. Ordinary thimbles must be shipped back to the rope manufacturer for resplicing into a new hawser. Most SPM operators find this too costly and bothersome to do.

It is important that detachable thimbles are properly designed. Care must be taken so that the pin holding the thimble to the hawser cannot accidentally come out. A typical detachable thimble is shown in Figure 5-19.

Different design criteria may apply to the buoy-end and tanker-end thimbles. Minimum weight is usually an important criteria for tanker-end thimbles because they must be supported by a buoy and must be lifted by tankers to moor. A heavy thimble with the appropriate support buoy may exceed winch capabilities of many tankers. The buoy-end thimble is not lifted by the tanker; it is supported by the SPM buoy so thimble weight is not as important a criteria. Minimum abrasion is the most important criteria for buoy-end thimbles. The thimble-to-buoy connection design varies between SPMs and the specific design will dictate the best thimble to use. A thimble specially designed for use on the buoy end of an SPM hawser is shown in Figure 5-20.

A coating or covering is usually placed between the hawser and thimble to prevent hawser abrasion. Leather typically is placed around the eye up to, and sometimes past, the crotch. The leather acts as a skin over the hawser, taking the abrasion which would otherwise take place on the hawser. Leather's effectiveness depends on its thickness, on how well it is placed on the eye, and on the thimble design.

Polyurethane has recently been introduced as a means of protecting the eye from chafing. In standard closed thimbles the polyurethane is cast in place by pouring it into the cavity between the rope and the thimble wall. Polyurethane encapsulation holds the hawser securely in the thimble so no abrasion between hawser and thimble can occur. As polyurethane is applied in the liquid state, cloth or plastic film should be wraped around the protion of rope to be covered to prevent polyurethane from adhering to or entering the hawser fibers. If polyurethane does adhere to or penetrate the hawser fibers, the rope may be stiffened, and rope strength may degrade more rapidly due to wear. Because polyurethane is a semiflexible material, it may be applied along the rope beyond the thimble to provide further protection against abrasion.

On unthimbled eyes and on eyes which will be placed in detachable thimbles, polyurethane may be sprayed on the eye to provide chafe protection. The polyurethane coating remains flexible. As with cast polyurethane, the rope should be wraped with cloth or plastic to prevent the sprayed polyurethane from adhering to or penetrating the fibers.

The third factor affecting performance of the rope in the thimble is thimble material. It is important that rust does not form on the thimble in the vicinity of the rope. This is particularly true of nylon rope. Rust forms an acid which readily attacks nylon causing strength degradation of the rope. Polyester is not as severely attacked by the acid formed rust and polypropylene fibers are immune to attack by the acid formed by rust. In addition, flakes of rusted metal from the thimble may get into the rope fibers, cutting fibers and causing rope strength degradation.

Thimbles are usually made of stainless steel or galvanized steel. If properly manufactured, galvanized steel thimbles will not rust during the service life of an SPM hawser. However, galvanized thimbles should not be reused unless they are thoroughly inspected for rust or are regalvanized.

Stainless-steel thimbles are preferable to galvanized thimbles for SPM hawsers. Stainless-steel thimbles are resistant to rust so they may be reused if properly maintained. The surface of stainless steel is smoother than the surface of galvanized steel, so the risk of fiber abrasion on the hawser surface is less. For the above reasons stainless-steel thimbles are recommended for SPM hawsers.

5.8.2 Hawser Floatation

The specific gravities of nylon and polyester are greater than the specific gravity of water, so nylon and polyester SPM hawsers need some form of floatation material to stay on the surface. The specific gravity of polypropylene is less than that of water so polypropylene ropes float without additional floatation.

Those types of floatation which cover the hawser also provide external abrasion protection. The floatation cover is usually placed along the entire hawser length to provide complete abrasion protection even though complete coverage by floatation material is not needed to keep the hawser afloat. Three types of permanent floatation are commonly used: bead floats, tube floats (also known as floatation sleeves or hose floatation), and collar floats (also known as lace-on floats). The three types of floatation are shown in Figure 5-21.

Bead floats are doughnut-shaped rings consisting of closed-cell plastic foam or a hollow hard plastic shell filled with foam. They are threaded onto a hawser before both ends are eye spliced. To permit the hawser to flex and contract, bead floats should not be packed tightly onto the hawser. It is not practical to attach each bead to the hawser itself. Therefore, bead floats are free to move along a hawser and tend to bunch at one end. To restrict movement of bead floats on a hawser, small rope is wrapped around the hawser or special collars are tied to the hawser at intervals. Experience has shown bead floats to be undesirable. Bead floats have been known to bang against each other sometimes shattering or breaking off, especially in cold weather.

Tube floats are long tubes made of closed-cell foam covered with a flexible jacket. One form of tube float consists of a thin-walled rubber hose covered by closed-cell foam. Like bead floats, tube floats are threaded onto the hawser before both ends are eye spliced. If tube floats are not tied to the hawser they tend to bunch up at the lower end. Some manufacturers wrap small line around the hawser to restrict tube float movement.

Collar floats are split tubes or sheets of closed cell foam covered with soft plastic or plastic-impregnated cloth with an axial split seam. Collar floats are typically about 1 meter (3 ft) long and about 25 to 50 mm (1 to 2 in.) thick. The edges of the split tude or sheet are provided with eyelets. The collar may be placed over the hawser and then laced together with small rope.

Collar floats and tube floats are usually sewn or tied onto a hawser. If not properly sewn on to compensate for hawser elongation, collar floats will break off when a hawser is elongated.

Collar floats have several advantages over bead and tube floats. As they are attached directly to the hawser, collar floats will not bang together or bunch up at one end of the hawser. Collar floats need not be put on at the rope factory so the hawser and floats can be shipped separately. This makes shipping easier as the hawser can be shipped on a reel or packed in a relatively small crate. As collar floats can be put on the hawser in the field by the SPM operating personnel, they can also be removed and placed on new hawsers in the field. Collar-float life is dependent upon environment conditions and handling care. If the SPM is in a mild location and if the hawser is carefully handled, collar floats may be reused a number of times.

5.8.3 Comparison of Strops and Eye-Spliced Hawsers

A rope can be spliced to construct either an eye-spliced hawser or an endless loop called a strop (also known as a grommet). Both constructions are used at SPMs. A strop and an eye-sliced hawser are shown in Figure 5-22.

A strop is constructed by end-for-end splicing the four ends of two ropes to form a single loop. It is important that loads are equally shared by both legs in the strop. The two sides of the loop, refered to as legs, are usually lashed together with small line to hold them together and prevent them from tangling.

It is important that one splice be made in each leg to insure equal loading, and the splices should be adjacent but preferably stagered so they do not chafe against each other. Splices affect hawser elasticity. If the elasticities of the two legs are uneven, the stiffer leg will elongate less and experience higher loads, resulting in more rapid wear and a reduction in strength of the strop. Strops are sometimes constructed by end-for-end splicing the two ends of a single rope. This type of strop has one splice. To equalize the elongations of the legs, the unspliced leg is sometimes pre-stretched. This method is not adequate in equalizing the elongations and loadings of the two legs.

The breaking strength of a strop may not be as high as the combined breaking strength of the two legs, because of the manner in which the eye is loaded. The arrangements of the eyes in eye-spliced and strop hawsers are shown in Figure 5-23.

In an eye-spliced hawser the area of rope around the thimble is equal to the area of rope in the midsection. Under tension T, the total tension is present in the midsection, but the tension in each leg of the splice around the thimble is T/2 (neglecting the effect of the angle of the crotch covered in subsection 5.4.5). Although the strength of the rope around the radius of the thimble is reduced due to curvature, the rope is under less tension around the thimble, and does not tend to break in this area.

In the strop hawser the area of rope around the radius of the thimble is the same as the area of each individual leg, but is only half the total area of rope in the midsection. Therefore, the rope around the thimble is under the same tension as each leg in the midsection. Because the strength of the rope around the thimble is reduced due to curvature, the rope tends to break at this point at a load less than the combined breaking strength of the two legs which make up the midsection.

The different manufacturers do not agree on the strength reduction in the eyes of strops. Samson Ropes and Wall Ropes believe the predominate strength reduction comes from the hawser being bent. A test conducted by Samson Ropes indicates the strength of a strop may be 1.7 times the minimum breaking strength of the rope which makes up each leg. (Samson, TR 12-74, 1974). This reduction in strength is attributed to the bend in the hawser at the eyes. British Ropes and Hawkins and Tipson believe the reduction in strength in the eye of a strop is negligible.

Based on experience in service, the reduction in new breaking strength in the eyes of strops does not appear to be significant with respect to the strength of used hawsers. The section of rope in the bend of the eye, which may be the weakest point in a new strop, is well protected by the thimble against cuts, abrasion, and other external damage. Although strops have failed in service due to wear or other factors along their unprotected length, no incident of a used rope failing in the eye section is known.

A more important factor in the strength of used strops may be the bulk of rope in each leg of a strop as compared with the bulk of rope in an eye-spliced hawser. External damage of a given magnitude, for example a cut of a given depth or abrasion over a given area, will cause a greater strength reduction in a strop hawser than in an eye-spliced hawser of the same new breaking strength. For example, a 25 mm (1 inch) deep cut in a 120 mm diameter (15 in. circumference) leg of a strop hawser would cause a larger reduction in strength than the same depth of cut in a 168 mm diameter (21 in. circumference) eye-spliced hawser. Thus, the strength of a used strop hawser may not be as great as that of a used eye-spliced hawser of the same original strength.

It is recommended the strength of a strop hawser be rated as 1.8 times the strength of each individual leg; that is, the total breaking strength of a strop hawser should be reduced by 10%. This reduction in rated breaking strength is in addition to that which should be taken into account in determining the minimum breaking strength of the rope as discussed in subsection 5.4. This reduction in rated breaking strength for strop hawsers is to account for the possibilities that the new breaking strength may be reduced by the effect of bends in the eyes and the breaking strength may be reduced more rapidly in service than that of an eye spliced hawser.

Eye spliced hawsers should be preferred over strops for SPMs. Strops were favored for a number of years because at the time ropes of sufficient strength to singly take the high loads at SPMs were not available. Splicing the rope in a strop was the only way to produce a hawser assembly capable of withstanding the high loads. Since the development of strops, hawser size has increased so that eye-spliced hawsers can now take the high loads.

5.8.4 Pairing of Dual Hawsers

Where dual hawsers are used at an SPM, care must be taken that the pair of hawsers are matched in length and elasticity. If one hawser is shorter than the other, the shorter hawser will take more than half the mooring load. Likewise, if one hawser is stiffer than the other, the stiffer line will take more than half the mooring load.

Rope manufacturers can splice hawser assemblies to a tolerance of ±1% of a specified length. Thus, it can be specified that the two hawsers of a matched pair are spliced so their lengths are within ±1% of each other. No significant unequal sharing of the mooring load would occur between hawsers in a pair spliced to this length tolerance.

The two hawsers in a pair should be matched as to construction and material. Obviously, an eight-strand hawser should not be used with a double-braid hawser. Also, a nylon hawser should not be used with a polyester hawser. Slight variations in elasticity can exist in hawsers made on different machines in which twisting and braiding tensions are different, and can also exist in hawsers made from different batches or grades of fibers of the same material. Therefore, paired hawsers should preferably be made on the same machine and should be made from the same grade of material.

Because the length of the hawser will increase with use, hawsers which are manufactured as pairs should continue to be used as pairs. If one hawser is retired from service due to damage, then the paired hawser should also be retired. An exception could be made if the used undamaged hawser can be paired to another undamaged used hawser which was essentially identical to it when new, which has had a similar service history, and is of essentially the same length in the used condition. A used hawser should never be paired with a new hawser.

5.9 CHAFING CHAINS

Chafing chains are usually placed between the buoy and the hawser and at the tanker end of the hawser. They serve to protect hawsers from rapid strength degradation caused by chafing, abrasion, and cuts at the buoy and tanker ends of the hawser.

The arrangement and design of chafing-chain assemblies varies and it is difficult to furnish a chafing chain assembly which is compatible with the SPM mooring fittings on all tankers. A comittee of OCIMF is now studying chafing-chain assembly design and intends to issue standards for compatible designs of tanker fittings and SPM chafing chains in the near future. A typical tanker-end chafing-chain assembly is shown in Figure 5-24. Typical dimension for chain links and shackles are given in Figure 5-25. Typical chain breaking strengths are given in Table 5-9 .

5.9.1 Chafing Chain Componenets

SPM chafing chains are made up of stock stud-link chain and chain fittings which are available from numerous manufacturers. The technology for manufacturing chain in the SPM chafing-chain size range is well established. Classification societies have standards for manufacturing and testing such chain and chain fittings. The American Bureau of Shipping (ABS) rules for manufacturing and testing chains are given in Rules for Building and Classing Steel Vessels.

SPM chafing chains are made from short lengths of chain so it is not usually practical to perform all of the testing procedures specified by the classification societies for each length of SPM chafing chain. It is, therefore, important that SPM chafing chains come from batches of chain which have been manufactured and tested to classification society specifications. When special alloy chain, not covered by any classification society, is used for SPM chafing chains, classification society testing procedures should still be followed to insure high quality and strength chains. Chain manufacturers insure their chain is high quality by proof loading all chain as part of the manufacturing process. Proof loading also improves elasticity properties of chain. Test and inspection certificates should be provided by the chain manufacturer or supplier.

The chafing chain of Figure 5-24 is made of 76 mm (3 in.) diameter chain. Many SPMs are now fitted with chafing chains of this size and the SPM mooring fittings on most large tankers are sized for this chain. This size is in accordance with the proposed OCIMF recommendation that chain stoppers on new tankers be sized for 76 mm (3 in.) chain. If an SPM is to be designed for small (in the size range up to 100 kdwt) generally older tankers, smaller chain and fittings may be required. The bow chocks and mooring fittings of many small tankers have chafing chain size restrictions. Also, the winches on some older tankers cannot lift chafing chain assemblies weighing more than about 2,300 kg (5,000 lb). Chafing chains for such vessels are usually made of 54 mm (2-1/8 in.) chain and the fittings used in assembling the chafing chain are sized appropriately.

Three methods are used to manufacture stud-link chains. The most common manufacturing method for chain in the SPM-chafing-chain size range is to flash weld chain out of bar stock. A simplified representation of the flash-welding procedure is shown in Figure 5-26. A length of bar stock is heated and bent to form a link interlocking with the end of the preceeding chain. After the link is formed, a stud is inserted in the link and the link is squeezed together. The compressive load put on the link when the stud is inserted holds the stud in place. The studs increase the strength of chain by up to 20%, so it is important that they stay in place. To insure studs remain in place, chain manufacturers will weld them in place for a small additional fee. The studs used by several chain manufacturers have protruding lugs at each end which penetrate the chain stock to hold the lugs in place. As chafing chains are located in positions where they can be frequently inspected to detect missing studs, it is not necessary that special provisions be made to retain the studs.

Other methods of stud-link chain manufacturing are to cast interlocking chain links or to assemble forged or cast chain-link halves. Baldt Corporation produces Di-Lok* chain by pressing two mating halves together. Di-Lok chain is claimed to be slightly more abrasion resistant than flash welded chain, but it is also more expensive. For SPM chafing chains the extra cost of Di-Lok chain is not generally warranted over that of flash-welded chain.

The shackles, end links, and other fittings used to make up chafing chain assemblies are generally stock items suppied by chain manufacturers. The assembly fittings should be manufactured to the same standards as the chain. Because tanker-end chafing chains are usually sized at 76 mm (3 in.) diameter to fit common ship-board mooring fittings, they may be stronger than necessary to withstand the mooring load. Fittings used in the assembly need not have the same strength as the chafing chain, but should be strong enough to withstand the mooring load with the appropriate factor of safety. Proof loading of fittings should be performed where possible. Certificates should be furnished for the various fittings which make up the load-carrying portion of the chafing-chain assembly.

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*Di-Lok is a registered trademark of Baldt Corporation.

5.9.2 Chain Support Buoy

Tanker-end chafing chains must be supported by buoys when not connected to a tanker so the chafing chain does not sink to the bottom, dragging the hawser and pick-up rope with it. Two typical chain support buoys are shown in Figure 5-27. The pendent-type of support buoy is connected to the chafing chain by a short length of small chain shackled to the chafing chain at a point near the hawser thimble. The chafing chain of Figure 5-24 is provided with a special link for this purpose. It is important that the support buoy is connected at the hawser-end of the tanker-end chafing chain so the support buoy will hang outboard of the tanker.

The hawse-type of chain-support buoy of Figure 5-27 consists of a closed-cell-foam buoy or steel-shell buoy built around a pipe with an inside diameter slightly larger than the maximum chain dimension. The hollow buoy is placed over the chafing chain and secured directly to it by a pin passing through the chain link. A disadvantage of the hollow type of chain support buoy is that the chain which it covers cannot be drawn through the tanker chock. Thus, more chain is required and the weight of the assembly is greater. The hawse-type buoy eliminates the problem of the buoy chafing against the hawser, which sometimes occurs with pendant type buoys..

There is no definitive criteria for sizing the necessary buoyancy of chain support buoys. Generally, chain support buoys are sized so there is at least a 20% and preferably a 50% reserve buoyancy with the chafing chain and buoy thimble attached. Other criteria important in selecting chain support buoys are that the support buoy should not interfere with SPM mooring operations and it should not abraid the hawser.

5.9.3 Chafing Chain Arrangement

Mooring fittings on tankers which visit SPMs vary widely in construction and strength. It would be difficult to design a chafing chain compatible with all tankers. Each SPM situation should be examined to determine the best chafing chain design compatible with most tankers which will moor. For most SPMs, it is important for tanker-end chafing chains to be compatible to as wide a variety of mooring fittings as possible.

Depending on the tanker mooring fittings, the chafing chain is connected to the tanker in different ways. The three types of mooring fittings which are common on tanker bows are mooring brackets, chain stoppers, and bitts. These mooring fittings and the methods in which chafing chains are attached to these fittings are discussed in Section 4.

If a tanker does not have either a mooring bracket or a chain stopper, snotters are normally used to attach the SPM chafing chain to bitts on the tanker forecastle. The snotter is shackled either to the open link at the end of the chafing chain or to the triangle plate in the chafing chain assembly. The manner in which snotters should be attached to bitts is discussed in subsection 4.7.2.

The design of mooring brackets found on tankers vary. On some tankers, the mooring bracket may not be compatible with the chafing chain and the chafing chain may have to be attached to bitts.

OCIMF is preparing recommendations which call for all new tankers to be outfitted with chain stoppers. There are several different chain-stopper designs. The design may place limitations on the chafing-chain design and dictate how the chafing chain is handled. Chain stoppers fitted on tankers for the purpose of mooring to SPMs are normally sized for 76 mm (3 in.) chain. However, some designs of chain stopper do not permit passage of the enlarged open end link on this chain. On such vessels, a short length of chain may have to be shackled to the open link to pass through the chain stopper.

Buoy-end chafing chains are simpler in design than tanker-end chafing chains. Buoy-end chafing chains are usually permanently connected to the buoy so their versatility is not important. The SPM buoy supports the buoy-end chafing chain so chain support buoys are not needed. Buoy-end chafing chains are usually made of the same chain size and grade as the tanker-end chafing chain. Buoy-end chafing-chain length depends on the SPM buoy design. They must be long enough to prevent the hawser from abraiding against the buoy. Buoy-end chafing chains often are as short as 3 m (10 ft). Open links are usually placed on both ends of buoy-end chafing chains so they may be shackled to the hawser thimble link and the SPM buoy mooring bracket.

5.9.4 Chafing-Chain Factor of Safety

Because the strength of chafing-chain assemblies are more adequately assured, chafing chains generally need not be as strong as the synthetic rope which makes up the hawser. The strength of synthetic rope degrades in service in an unpredictable manner due to cuts, wear, abrasion, and fatigue. Also, the strength of new rope cannot be accurately determined through testing. Stud-link chain and other components of the chafing-chain assembly can be proof tested to assure their strength. Under normal circumstances, the strength of the chain will degrade slowly due to abrasion and corrosion. Thus, the chafing chain is not likely to fail in an unpredictable manner.

The stud-link chain used as chafing chains for SPMs which serve VLCCs is usually 76 mm (3 in.) chain. Several chain grades having different strengths are available. Grade 3 chain is usually used, because it has high strength and generally is no more expensive than Grade 2 chain. Grade 3 76 mm (3 in.) chain has a breaking strength of 4.315 kN (970,000 lb). Thus, chafing chains may be significantly stronger than the SPM's hawsers. This extra strength is not viewed as a detriment because the extra strength gives a larger margin for abrasion fatigue and corrosion. Therefore, the chafing chain can remain in service longer than the synthetic rope.

For convenience in handling and mooring, some components of the chafing chain, such as shackles and open links, may be designed for strengths less than that of the stud link chain. To size these components a minimum factor of safety should be defined.

As discussed in subsection 5.7, the minimum factor of safety of individual lines in dual mooring line systems should be 1.5. Commensorate with the philosophy that each line in a dual mooring line system should be capable of taking the entire mooring load, the chafing chain assembly in each leg of a dual mooring line system should have a minimum factor of safety of 1.25.

The minimum factor of safety of 2 which is recommended on a single-hawser system is more than is justified for the chafing chain assembly in such a system. A factor of safety of 1.75 based on breaking strength of the chain and other components of the chafing chain assembly will provide a safety factor of approximately 1.2 based on proof load of these components. This factor of safety of 1.75 on breaking strength is recommended for the chafing chain assembly of single-hawser systems.

5.9.5 Chafing Chain Inspection and Replacement

SPM chafing chains should be inspected along with SPM hawsers during every tanker mooring. They should also be periodically removed from the water for a close inspection. SPM chafing chains can remain in service longer than SPM hawsers, but should be replaced when wear or damage reduces their strength below an acceptable level.

It is impractical to inspect chafing chains before a tanker moors. Tanker-end chafing chains are submerged when a tanker is not moored. Thus, it is more convenient to visually inspect them after the tanker moors. Shortly after the chafing-chain assembly is secured to the tanker, it should be visually inspected for missing studs, deformation, and excessive abrasion or corrosion. If excessive abrasion or corrosion is observed, measurements should be made to determine the strength of the chain.

During periodic close inspections of the mooring lines, the chafing chains should be brought on the deck of a launch or work-boat and closely checked for deformation, fatigue cracks, excessive corrosion, and abrasion. Remaining chain strength can be determined by measuring the size of links during close inspections. Shackles should be examined to assure pins are properly secured.

Fatigue cracks indicate the chafing chain is approaching catastrophic failure. SPM chafing chains are unlikely to develop fatigue cracks except after very long use. If fatigue cracks are observed, the chafing chain and fittings that have been in use as long as the chafing chain should be retired.

If chain links are deformed, the entire chafing chain should be removed form service because it has probably been overloaded. Loose studs may be welded in place provided proper attention is paid to welding procedures to avoid stress concentrations and metalurgical problems. Individual links with missing studs or excessive abrasion may be replaced by Kenter shackles of strength equivalent to the chafing chain.

SPM chafing chains are more likely to degrade from abrasion and corrosion than from fatigue or deformation. It is desirable to minimize abrasion and corrosion, but, due to the nature of their use, SPM chafing chains will abraid and corrode with time. Chain strength will degrade gradually due to these causes and at some point the chain should be retired. The remaining strength of chafing chain should be assessed during periodic close inspections to determine if the chafing chain should be retired.

Precise means of determining the remaining strength of used chain are not known. The approximate strength of used chain can be determined by measuring the remaining area of material at points of wear or corrosion. The minimum cross-sectional area corresponding to the minimum desired strength may be determined by equating the minimum area to the cross-sectional area of a smaller size chain of the same grade and construction having the minimum desired strength. The chafing chain should be retired when the cross-sectional area at any point in a link corresponds to this minimum cross-sectional area.

At a point of abrasion, the cross-section of the chain stock will not be round and it will be difficult to determine the exact cross-sectional area. Several methods may be used to estimate the cross-sectional area at the point of maximum abrasion. One method is to average the maximum and minimum chain stock diameters at the point of abrasion. The average diameter is squared and multiplied by $\pi/4$ to obtain the cross-sectional area. The minimum diameter at the point of abrasion may be used to conservatively estimate the cross-sectional area. This method is not as accurate and is probably over-conservative. The cross-sectional area can be accurately determined by using micrometers or other instruments to measure the exact shape of the cross-section but this method is believed to be unpractical for field use.

It is recommended that the average-diameter method be used to estimate the cross-sectional area for the purpose of replacing chains. Chafing chains should generally be replaced when their remaining strength, as determined by this method, is 90% of the strength required to meet the factors of safety given in subsection 5.9.4.

5.10 SPM STRUCTURAL DESIGN

The techniques for structural design of floating structures, such as SPM buoys, and for fixed structures, such as SPM towers or mooring bases, are well established. In the past, the design of SPM buoys has generally been guided by the rules of several classification societies for ship and barge hulls. Practices for offshore drilling structures and for onshore structures have been applied to the design of SPM towers and mooring bases as well as to SPM buoys. Recently, several classification societies have issued specific rules for SPMs.

5.10.1 ABS Rules

The American Bureau of Shipping Rules for Building and Classing Single Point Moorings (1974) contains an extensive section on structural design and another section on welding. These structural-design rules cover various types of loading and combinations of loading conditions. In general, the rules limit tensile stress in structural components to less than 80% of yield strength under combined conditions of gravity, waves, wind, current, and mooring loads. The criteria and methods of analysis given in the rules follow the ABS rules for ship hulls. Although generally intended to apply to buoy hulls, the rules are broad enough to apply to mooring bases and SPM towers.

The ABS rules call for the structure to be stress analyzed and suggests the use of finite-element techniques. An analysis of transmission of the hawser load from the hawser attachment point to the anchor-leg attachment points or to the foundation is called for.

The ABS Rules also call for an analysis of loads imposed under survival conditions without a tanker at the mooring. In the case of a buoy-type SPM, the survival loads on the buoy and anchoring system will probably not be as high as the maximum loads with a tanker moored. However, for a mooring tower the peak loads imposed by waves, wind, and current in the survival conditions may control the design of the structure.

The ABS Rules call for a factor of safety of 2 against destructive yielding for bearings which carry the mooring load.

5.10.2 Other Applicable Structural Rules

The American Petroleum Institute (API) document API RP2A Recommended Practice for Planning, Design and Construction of Fixed Offshore Platforms (1977) contains much applicable material for the design of SPM towers and mooring bases. Although primarily intended for fixed structures, the API Practice includes material appropriate for buoy design.

The Norwegian classification society Det Norske Veritas has just formulated tentative Rules For The Design, Construction, and Inspection of Offshore Loading Systems (1977). These rules are not generally applicable to SPMs for terminals in relatively shallow water, as at U.S. deepwater ports. The Norske Veritas Rules have been developed primarily for SPMs at production fields in relatively deep water. However, they may contain items applicable to SPM terminals in relatively shallow water.

5.11 SPM ANCHORING SYSTEM DESIGN

The SPM buoy or tower must be securely anchored to the sea floor in order to resist the mooring load. In a SALM (single anchor leg mooring) the anchoring system consists of a single very large chain attached to a base on the ocean floor. In a CALM (catenary anchor leg mooring) the anchoring system consists of a number of anchor chains extending radially from the buoy in catenaries to piles or anchors at some distance from the buoy. The SALM and CALM are shown in Figures 1-3 and 1-2 respectively. A tower type SPM, such as that shown in Figure 1.4, would be anchored directly to the sea floor much in the same manner as a SALM base. This subsection discusses important aspects in the design of the anchoring system of SPMs.

5.11.1 Anchor Chains

The chains used in anchoring SPM buoys are large stud-link anchor chains. The construction of stud-link chain is discussed in subsection 5.9 Flash-welded or cast stud-link anchor chain as large as 178 mm (7 in.) are available from several manufacturers. Chain size refers to the diameter of the cross-section of the chain. Table 5-9 gives representative strengths of large stud-link anchor chain.

Several grades of material are used in stud-link anchor chain. Normal strength, high strength, and extra high strength chain are classified as grade 1, grade 2, and grade 3 chain respectively by classification societies. American Bureau of Shipping refers to these categories simply as "Grades". Lloyds Register prefixes the grade numbers with "U". Both Norske Veritas and Germanischer Lloyd prefixes them with "K". Bureau Veritas (French) prefixes them with "Q". The test loads for chains of these grades are the same for each of the above mentioned classification societies.

The designation Oil Rig Quality, ORQ, has no official significance, however, most chain manufacturers list chain by this designation, and the proof and break loads claimed by these manufacturers are equivalent, apparently by mutual agreement. The American Petroleum Institute (API) specification 2F, Mooring Chain, (1977) was developed to serve as an inspection criteria for chain for drilling vessels and is now commonly used by oil companies in procuring ORQ chain. The API mooring chain specification covers metallurgy and inspection procedures and calls for studs to be welded in place.

The minimum proof and break loads called for in the API specification do not correspond to the proof and break loads for ORQ chain listed by the manufacturers. The API specified minimum proof and break loads follow the following formula.

$$\text{Proof Load (kN)} = .014d^2\,(44-.08d) \qquad (5.2)$$

$$\text{Break Load (kN)} = .0211d^2\,(44-.08d) \qquad (5.3$$

where d = nominal diameter in mm

$$\text{Proof Load (lb)} = 2030.5d^2\,(44-2.032d) \qquad (5.2a)$$

$$\text{Break Load (lb)} = 3060.3d^2\,(44-2.032d) \qquad (5.3a)$$

where d = nominal diameter in inches

Although ORQ chain is rated by the chain manufacturers as being higher in strength than grade 3 chain, it is not necessarily stronger. The minimum proof and breaking strengths given by the manufacturers for grade 3 chain are prescribed by classification society rules while the values listed by the manufacturers for ORQ chain are not so governed. The minimum tensile strength for steel used in grade 3 chain is 70 kp/mm^2 (99,600 lb/in^2) as per classification society rules (ABS Rules for Steel Vessels). ORQ chain made in accordance with the API specification must be of steel with a minimum tensile strength of 65 kp/mm^2 (93,000 lb/in^2).

Special high-strength chains are also available from the various chain manufacturers. Such chain is referred to by various names such as super-alloy, special alloy, or extra strength chain. The test loads listed by the manufacturers for these special grades of chain vary. Although the testing loads for these chains are not listed by the classification societies, the chains can be tested and classed by the testing societies to the rated loads.

Table 5.9 gives break test loads for various grades of chain in typical sizes in the range from 54 mm (2 1/8 in.) to 178 mm (7 in.). Table 5.2 lists manufacturers of chain larger than 5 inch. Not all manufacturers manufacture all grades of chain in the larger sizes. There is generally no cost difference between grade 2 and grade 3 chain and, therefore, grade 2 chain is not normally used to anchor SPMs.

5.11.2 Anchor Leg Design

The size and strength of chain used to anchor the SPM buoy depends on the type of mooring and on the mooring loads. A short length of chain as large as 152 to 178 mm (6 to 7 inch) would normally be used to anchor an SALM buoy. As the length of anchor chain on a SALM is very short, large, very-high-strength chain can be used with little increase in cost.

Four, six, or eight very long anchor chains, generally in the range of 102 to 127 mm (4 to 5 inch) are normally used to anchor a CALM buoy. The unit weight of anchor chain is an important factor in determining the elasticity of the CALM system. Heavy chain may be desirable to provide the proper catenary elasticity characteristics. Therefore, the chain may be sized to provide the proper weight as much as to provide strength. Therefore, chains stronger than grade 3 are generally not used at CALMs.

The length of anchor chain at a CALM is another important parameter in determining the system elasticity and also the loads on the anchoring points. Generally, the chain should be long enough so that no vertical load is applied to the anchor point. If all of the chain in an anchor leg is lifted from the sea floor then an upward load as well as a horizontal load is applied to the anchor point. This is only permissible if the anchor point is designed to withstand an upward load. The catenary elasticity characteristic changes when all the chain is lifted, and this characteristic should be reflected in the design analysis.

Instead of employing heavy chain throughout the anchor leg at a CALM, heavier chain may be used in only a portion of the anchor leg to achieve the desirable elasticity. Alternatively, weights, refered to as sinkers, may be attached to the anchor leg to alter the catenary elasticity characteristics.

5.11.3 Anchor Chain Loads and Safety Factors

The maximum mooring load for the anchor chain of an SPM should be determined based on the same design conditions and same prediction techniques as employed for hawser loads. The anchor-chain loads should preferably be calculated directly from the results of model tests instead of being related to hawser loads. The load in the anchor leg due to a steady load on the hawser in calm water can be calculated by statics. However, the dynamic loads in the anchor chain may be influenced by the dynamic response of the buoy and by wave forces on the chain. Therefore, it is preferable that the anchor-chain load statistics be separately analyzed.

The ABS Rules for Classing SPMs (1974) call for anchor legs of buoy-type SPMs to be designed with a factor of safety of 3 based on the breaking load of the chain. This safety factor is consistent with design practices for anchor chains on multiple buoy moorings used by the U.S. Navy and others for many years.

The factor of safety for anchor chains must account for corrosion and chain wear. Wear between adjacent chain links will reduce the effective area of the chain and reduce its strength. Corrosion will also reduce chain area and strength. In catenary anchor chains the portion of the chain which lies on the sea floor and is picked-up and laid down, known as the dip section, will be abraided if the sea floor is sand, coral, or rock. For such moorings where abrasion is severe, the factor of safety of 3 has proven to be satisfactory. On some moorings, the dip section of the chain is oversized to allow for abrasion.

At either a CALM or a SALM, an anchor chain failure may result in damage to the cargo system. Failure of an anchor chain at an SALM will permit the buoy and the moored tanker to drift free. If prompt action is not taken the cargo hoses connecting the underwater fluid swivel with the tanker manifold may be overstressed or broken. However, if cargo operations were suspended and the hoses disconnected before failure of the anchor chain, then there would be no risk of pollution.

Failure of any one leg of the anchoring system of a CALM could result in overstressing or breaking of the underwater hose connecting the buoy with the pipeline manifold on the sea floor. The buoy is held in place over the manifold by the anchor legs, and if one leg is broken the buoy will shift position. Under a high load, the buoy will probably shift beyond the limits of the underwater cargo hose. This would occur even if the floating cargo hoses were disconnected.

Failure of an individual anchor leg on a CALM will not cause the buoy to drift free. However, the mooring load is not shared equally among the several anchor legs, and substained or repeated high loads from the same direction might lead to failure of additional anchor legs.

5.11.4 Anchor Chain Inspection and Replacement

The anchor chains of an SPM should be periodically inspected for wear and corrosion. The short single anchor leg of a SALM can be easily inspected by divers. The underwater fluid swivels of a SALM should be removed and brought to the surface for detailed inspection and overhaul every several years. Depending on the design and on other factors this may be necessary at intervals of approximately 5 years. The SALM anchor chain may be brought to the surface and inspected in detail at the same time.

The long anchor chains of a CALM should also be periodically inspected by divers. It is more difficult to inspect that portion of the chain which lays on the sea floor. At long intervals, for example whenever the buoy is brought ashore for overhaul, the anchor chains of a CALM should be brought to the surface for detailed inspection.

Fatigue of anchor chains has not been known to be a problem at SPMs. The pretension applied to the anchor chains and the dynamic loads imposed on the anchor chains in normal service are much lower than the breaking strength of the chain. The pretension applied to the anchor chain of a SALM is about 15 to 20% and the maximum load is not more than 33% of the breaking strength of the chain. The pretension applied to each anchor chain in a CALM is about 2 to 5%, and the maximum load is not more than 33% of the breaking strength. Maximum loads in the anchor chains are approached only rarely. Although the percent of breaking load applied as pretension is higher in the SALM, the load range, in terms percent of breaking load, which is dynamically applied to the chain is larger in the CALM.

During inspection of anchor chains attention should be given to evidence of fatigue cracking. If fatigue cracking is noted in any portion of a chain the entire chain should be carefully checked, and unless the fatigue cracking is found by detailed inspection to be confined to only one or several links which can be replaced, the entire chain should be replaced.

The condition of studs in the chain and of shackle pins and other connection devices should be checked during inspections. Missing studs reduce the strength of stud-link chains. Loose studs may fall out of the chain. Studs should preferably be welded in place during manufacturing, especially in the short anchor chain of a SALM where the additional cost is relatively small.

Pins of shackles should be secured by bolts or welding during assembly of the anchor chain to assure they will not work loose in service. The integrity of the means of securing these pins should be checked during inspection. Signs of deformation of shackles and pins should also be checked, as deformation is a sign of overloading of the anchor chain system.

5.11.5 Strength of Used Chain

The rate of chain wear between links should be determined during inspections. The approximate strength of the worn chain can be determined by measuring the remaining cross-sectional area of the chain as described in subsection 5.8.4.

A definitive replacement criteria for SPM anchor chains has not been determined by the industry. As a tentative criteria it is recommended that anchor chains be replaced when the strength is reduced by wear such that the remaining strength is 33% below that required to satisfy the factor of safety of 3 criteria for new anchor chain.

5.11.6 Design of Piles and Anchors

The ABS Rules for Classing SPMs (1974) specify the minimum factor of safety against pullout of an anchor point should be 2. For anchor points employing conventional anchors, a pull test at the maximum design load should be applied to each anchor. Piled anchor points are much preferred to conventional anchors because their holding security can better be determined through design and analysis. Where conventional anchors are used on catenary anchor legs, their position and security should be frequently checked, as movement of an anchor can destroy the balance of loads in the anchoring system and may shift the position of the buoy so far that the underwater cargo hoses are overstressed. Resetting of anchors and repretensioning of the anchor chains may be necessary in service.

The ABS Rules recommend that anchor piles and pile foundations be designed to comply with API RP2A Recommended Practice for Planning, Designing and Construction, Fixed Offshore Platforms (1977). The API Practice is a fairly comprehensive guide to underwater pile design and installation. It covers such subjects as soil investigation, pile capacity, pile penetration, material, wall thickness, and grouting. Investigation of soil conditions is important to assure adequate pile and foundation designs. Pile driving records and grouting records are important to assure the installed piles are adequate for the design load.

5.12 SUMMARY

The components of the SPM system which carry the mooring load are the hawser assembly, the buoy or tower structure, the anchor chains, and the mooring base structure or anchor points. The design and fabrication of structures, anchor chain, and anchor points should be done in accordance with published rules and standards. Data and standards for the design and fabrication of large synthetic-rope SPM hawsers are not as well established or readily available.

The SPM hawser normally consists of a large-diameter synthetic rope spliced either with eyes at each end or in a continuous loop, known as a strop. The eyes or ends of the strop are normally protected by thimbles to limit the radius of bending and to protect against wear.

Nylon is preferred as a material for SPM hawsers because of its elastic characteristics, however, polyester and polypropylene are sometimes used. Nylon is highest in strength of the three common SPM hawser materials. The properties of nylon, polyester, and polypropylene vary with the grade of material and the proprietary process of the fiber manufacturers. The results of various tests indicate nylon 6.6 is better than nylon 6.0 for SPM hawsers.

The newest grades of polyester are almost as strong as nylon. Polypropylene is significantly weaker. Nylon looses approximately 10% of its strength when wet, but this is not a serious deficiency and experience with nylon hawsers has been good. Polyester and polypropylene are significantly stiffer than nylon and, therefore, are not as desirable for SPM hawsers. Chemical and ultraviolet degradation would not normally be a problem in large diameter SPM hawsers.

Double-braid rope is preferable to eight-strand or three-strand rope as an SPM hawser because it is stronger when compared on an equivalent-size basis and is less likely to be damaged in service. Three-strand rope is undesirable because of its tendency to hockle. Although three-strand and eight-strand ropes are more elastic than double-braid rope when new, once the ropes are broken-in their elasticities are similar. Eight-strand rope may be preferable for pick-up ropes.

The rated breaking strengths of very large synthetic-ropes are generally based on extrapolations from the strengths of the rope components or from the strengths of smaller ropes. Only a few break tests have been conducted on very large ropes to date because of the limitations of testing machines, but the results of these tests confirm the rated breaking strengths.

The minimum breaking strength of the rope, as guaranteed by the rope manufacturer, should be used for design purposes. This breaking strength should account for the effect of splices. The strength of a strop should be rated as 90% of the total strength of the two legs to account for the possible influence of bends and the possibility of more degradation of strength in service due to wear as compared with an eye-spliced hawser of equal strength.

Cuts, abrasion, and other wear are the principle causes of rope strength degradation in service. Careful inspection practices should be established and followed to detect such damage. Significant strength degradation due to cyclic loading or occasional high loads will probably not occur with top quality nylon or polyester hawsers. Results of cyclic-loading tests on top-quality nylon ropes indicate strength will not decrease rapidly unless the loads exceed 60% of the breaking strength, and that cyclic loading below 30% of breaking strength will not cause failure of the rope.

A re-examination of the bases for hawser factors of safety has been made. For dual-hawser systems, a factor of safety of 3.0 is recommended. A factor of safety of 2.0 is recommended for single-hawser systems. These factors of safety are defined as the ratio of the rated minimum breaking strength of the synthetic rope, with splices and other factors accounted for, to the predicted maximum mooring load. These factors of safety are intended for hawsers at SPMs for U.S. deepwater ports and are slightly more conservative than those specified in the ABS Rules for SPMs.

Thimbles, floatation devices, chafing chains, and other hardware are important components of the SPM hawser asembly. Thimbles should be designed to avoid points of stress in the hawser and to protect the hawser from wear. Protection in the form of leather or plastic covering is normally provided on the rope within the thimble. Floatation is necessary to support nylon or polyester hawsers, and the covering of floatation material serves to protect the hawser. The chafing chain at the tanker end of the hawser should be designed, as much as possible, to be compatible with the various shipboard mooring-fitting arrangements which are likely to be encountered.

The ABS Rules for Building and Classing Single Point Moorings (1974) cover most aspects of the design of buoy and tower-type SPMs. Other standards and rules also contain applicable material and may be used to supplement the ABS rules. Classification society rules cover the testing of anchor chains.

TABLE 5-1

LARGE-ROPE MANUFACTURERS

Manufacturer	Address	Rope Manufacturing Capabilities
British Ropes Limited	Anchor and Hope Lane Charlton, London, England SE7 7SB	Double-braid rope up to 240 mm diameter (30 in. circumference)[1] Eight-strand rope up to 150 mm diameter (18 in. circumference) Three-strand rope up to 96 mm diameter (12 in. circumference)
Hawkins and Tipson Ropemakers Limited	Marlow House, Hailey Road Erith, Kent, England DA18 4AL	Eight-strand rope up to 192 mm diameter (24 in. circumference) Three-strand rope up to 160 mm diameter (20 in. circumference)
Samson Ocean Systems, Inc.	99 High Street Boston, Mass. U.S.A. 02110	Double-braid rope up to 240 mm diameter (30 in. circumference)[1] 12-strand single-braid rope up to 96 mm diameter (12 in. circumference)
Tokyo Rope Mfg. Co., Ltd.	Yu-Man Building, No. 5 3-Chome, Minamihoumachi Higashi-Ku, Osaka, Japan	Double-braid rope up to 112 mm diameter (14 in. circumference)[2] Eight-strand rope up to 120 mm diameter (15 in. circumference) Three-strand rope up to 100 mm diameter (12 in. circumference)
United Rope Works	Rotterdam, Holland	Eight-strand rope up to 144 mm diameter (18 in. circumference) Three-strand rope up to 112 mm diameter (14 in. circumference) Nine-strand rope up to 192 mm diameter (24 in. circumference)
Wall Rope Works	Beverly, New Jersey U.S.A. 08010	Eight-strand rope up to 140 mm diameter (17 in. circumference)

1. Only test specimens of 240 mm diameter (30 in. circumference) double-braid ropes have been produced by the manufacturers. British Ropes have stated they believe they can manufacture double-braid rope up to 320 mm diameter (40 in. circumference).

2. Tokyo Rope's double-braid rope is not presently marketed in the United States.

TABLE 5-2

LARGE-CHAIN MANUFACTURERS

Manufacturer	Address	Chain Manufacturing Capabilities
Hamanaka Chain Mfg. Co. Ltd.	Ko-770 Shirahama-cho Himeji, Japan	Grade 3 chain up to 162 mm (6-3/8 in) ORQ chain and "Extra Strength" chain up to 102 mm (4 inch) (have indicated capability to manufacture 178 mm (7 in) chain)
Ljusne (L-W) Products	Bergvik Och Ala AB Ljusne (LW) Products S-820 20 Ljusne Sweden	Grade 3 chain up to 162 mm (6-3/8 in) ORQ chain up to 152 mm (6 in) "Ljusne Super" chain up to 130 mm (5 in) (have indicated capability to manufacture 180 mm (7-1/8) chain)
Osaka Chain and Machinery, Ltd.	Matsumura Bldg. 15-1, 2-chome Kyobashi, Higashi-ku Osaka, Japan	Grade 3 chain up to 162 mm (6-3/8 in) ORQ chain up to 102 mm (4 in) (have indicated capacity to manufacture 178 mm (7 in) chain)
Rammas Bruks AB	Bulten-Kanthal AB Rammas Bruk Division S-730 60 Ramnas Sweden	Grade 3 chain up to 152 mm (6 inch) ORQ and "Super" chain up to 152 mm (6 inch)
Sirot Metallurgie S.A.	Boite Postale 21 1 rue de Petite Repas 59230 Saint-Amand-les-Eaux France	Grade 3 chain up to 152 mm (6 in) ORQ chain up to 100 mm (3-15/16 in)
August Thiele	August Thiele 5841 Kalthof/Kreis Iserlohn West-Germany	Grade 2 and Grade 3 chain up to 152 mm (6 in)
Vicinay S.A.	P.O. Box 956 Bilbao, Spain	Grade 3 chain up to 152 mm (6 in) "Special Alloy" chain up to 120 mm (4 3/4 in)
Vicinay Chain Co. Inc.*	6610 Harwin, Suite 110 Houston, Texas 77036	Grade 3 chain up to 140 mm (5-1/2 in)

All manufacturers also supply Grade 1 and Grade 2 chain

Maximum sizes listed are from most recent catalog unless otherwise indicated

*Vicinay Chain Co. Inc. is U.S. affiliate of Vicinay S.A. with a chain manufacturing plant in Corpus Christi Texas

TABLE 5-3

TYPICAL PROPERTIES OF NYLON, POLYESTER, AND POLYPROPYLENE FIBERS

	Nylon 6.6	Polyester	Polypropylene
Tenacity of dry fiber (grams/denier)	8.0	7.0	5.5
Retained strength when wet	85-90%	100%	100%
Specific gravity	1.14	1.38	.91
Elongation at breaking point[1]	35%	22%	24%
Susceptibility to creep	Negligible	Negligible	High
Resistance to rot, mildew, and deterioration from marine organisms	Excellent	Excellent	Excellent
Chemical resistance	Some acids can degrade	Some alkalis can degrade	Very resistant to chemical degradation
Sunlight resistance	Good	Excellent	Low
Abrasion resistance	Excellent	Excellent	Good
Critical temperature[2]	177°C (350°F)	177°C (350°F)	149°C (300°F)
Melting temperature	249°C (480°F)	249°C (480°F)	166°C (330°F)

Note: It would be difficult to quantify some of the above fiber properties such as abrasion resistance. The qualitative fiber properties given above are generally agreed upon throughout the cordage industry.

1. These elongations are for broken-in eight-strand ropes.

2. The critical temperature is the point at which fiber degradation is caused by temperature alone.

TABLE 5-4

CHARACTERISTICS OF HIGH QUALITY NYLON, POLYESTER, AND POLYPROPYLENE FIBERS

Nylon	- Bright White Virgin Overlay Finished Continuous Filament Light Resistant Minimum Tenacity of 8.0 Grams per Denier
Polyester	- Bright White Virgin Overlay Finished Continuous Filament Minimum Tenacity of 7.0 Grams per Denier
Polypropylene	- White Virgin Continuous Filament Light Resistant Minimum Tenacity of 5.5 Grams per Denier

COMPARATIVE CHEMICAL RESISTANCE OF "DACRON" AND DUPONT NYLON FIBERS*

		Exposure Conditions	Per Cent of Original Strength Retained after Exposure: DACRON* polyester fiber	Du Pont nylon fiber
STRONG MINERAL ACIDS				
1% Hydrochloric		70°F. (21°C.) for 1,000 hrs.	97	81
1% "		160°F. (71°C.) for 10 hrs.	103	90
10% "		70°F. (21°C.) for 10 hrs.	100	82
10% "		70°F. (21°C.) for 1,000 hrs.	87	—
37% "		70°F. (21°C.) for 100 hrs.	86	—
10% Hydrofluoric		70°F. (21°C.) for 10 hrs.	84	97
10% Nitric		70°F. (21°C.) for 10 hrs.	105	91
10% Phosphoric		70°F. (21°C.) for 10 hrs.	97	95
1% Sulfuric		70°F. (21°C.) for 1,000 hrs.	98	80
1% "		250°F. (121°C.) for 10 hrs.	92	11
10% "		70°F. (21°C.) for 10 hrs.	103	57
10% "		70°F. (21°C.) for 1,000 hrs.	103	—
70% "		70°F. (21°C.) for 100 hrs.	98	—
ORGANIC ACIDS				
40% Acetic		70°F. (21°C.) for 10 hrs.	89	88
3% Benzoic		210°F. (99°C.) for 10 hrs.	90	19
40% Formic		70°F. (21°C.) for 10 hrs.	103	99
90% "		70°F. (21°C.) for 10 hrs.	100	A
5% Oxalic		70°F. (21°C.) for 10 hrs.	90	82
3% Salicylic		210°F. (99°C.) for 10 hrs.	91	25
STRONG ALKALI				
1% Sodium hydroxide		210°F. (99°C.) for 10 hrs.	84	90
10% " "		70°F. (21°C.) for 10 hrs.	103	101
10% " "		210°F. (99°C.) for 10 hrs.	A	94
40% " "		210°F. (99°C.) for 10 hrs.	—	84
BLEACHING AGENTS				
3% Hydrogen peroxide	(pH- 6)	70°F. (21°C.) for 10 hrs.	92	97
3% " "	(pH-10)	160°F. (71°C.) for 10 hrs.	105	26
2% Peracetic acid	(pH- 4)	70°F. (21°C.) for 10 hrs.	103	94
2% " "	(pH- 4)	210°F. (99°C.) for 10 hrs.	95	70
0.7% Sodium chlorite	(pH- 4)	70°F. (21°C.) for 10 hrs.	106	93
0.7% " "	(pH- 4)	210°F. (99°C.) for 10 hrs.	101	20
0.4% Sodium hypochlorite	(pH-11)	70°F. (21°C.) for 10 hrs.	97	94
0.4% " "	(pH-11)	160°F. (71°C.) for 10 hrs.	105	7
1% Sodium perborate	(pH-10)	70°F. (21°C.) for 10 hrs.	104	97
1% " "	(pH-10)	210°F. (99°C.) for 10 hrs.	118	95
REDUCING AGENTS				
1% Sodium bisulfite	(pH- 4)	160°F. (71°C.) for 10 hrs.	102	97
1% Sodium hydrosulfite		160°F. (71°C.) for 10 hrs.	99	98

	Exposure Conditions	Per Cent of Original Strength Retained after Exposure: DACRON* polyester fiber	Du Pont nylon fiber
SCOURING COMPOUNDS			
1% Soap	210°F. (99°C.) for 10 hrs.	100	99
1% Sodium carbonate	210°F. (99°C.) for 10 hrs.	102	96
1% " "	250°F. (121°C.) for 10 hrs.	93	99
SALT SOLUTIONS			
3% Copper sulfate	70°F. (21°C.) for 1,000 hrs.	94	96
3% " "	210°F. (99°C.) for 10 hrs.	104	88
3% Ferric chloride	210°F. (99°C.) for 10 hrs.	112	56
3% Sodium chloride	70°F. (21°C.) for 1,000 hrs.	94	98
3% " "	210°F. (99°C.) for 10 hrs.	99	88
3% Zinc chloride	210°F. (99°C.) for 10 hrs.	99	103
ORGANIC SOLVENTS			
100% Acetone	70°F. (21°C.) for 1,000 hrs.	94	90
100% Amyl alcohol (normal solution)	70°F. (21°C.) for 1,000 hrs.	93	90
100% Benzene	70°F. (21°C.) for 1,000 hrs.	91	90
100% Carbon disulfide	70°F. (21°C.) for 1,000 hrs.	91	91
100% Carbon tetrachloride	70°F. (21°C.) for 1,000 hrs.	90	90
100% Chloroform	70°F. (21°C.) for 1,000 hrs.	90	95
100% Ether	70°F. (21°C.) for 1,000 hrs.	92	90
100% Ethyl acetate	70°F. (21°C.) for 1,000 hrs.	93	90
100% Ethyl alcohol	70°F. (21°C.) for 1,000 hrs.	95	90
100% Methyl alcohol	70°F. (21°C.) for 1,000 hrs.	90	90
100% Perchloroethylene	210°F. (99°C.) for 10 hrs.	101	99
100% Stoddard solvent	160°F. (71°C.) for 10 hrs.	96	101
100% Tetrachloroethane	70°F. (21°C.) for 1,000 hrs.	90	90
MISCELLANEOUS CHEMICALS			
10% Acetaldehyde (in water)	70°F. (21°C.) for 1,000 hrs.	84	96
28% Ammonia (in water)	70°F. (21°C.) for 1,000 hrs.	B	94
100% Benzaldehyde	70°F. (21°C.) for 1,000 hrs.	92	90
100% Cottonseed oil	70°F. (21°C.) for 1,000 hrs.	92	90
10% Formaldehyde (in water)	70°F. (21°C.) for 1,000 hrs.	91	89
100% Glycerine	210°F. (99°C.) for 10 hrs.	104	99
100% Glycol	210°F. (99°C.) for 10 hrs.	111	94
3.5% Iodine (in ethyl alcohol)	70°F. (21°C.) for 10 hrs.	103	95
100% Lard	70°F. (21°C.) for 1,000 hrs.	93	70
100% Linseed oil	70°F. (21°C.) for 1,000 hrs.	88	90
100% Mineral oil	210°F. (99°C.) for 10 hrs.	97	96
100% Nitrobenzene	70°F. (21°C.) for 10 hrs.	103	65
5% Phenol (in water)	70°F. (21°C.) for 10 hrs.	107	43

*Du Pont registered trademark A dissolved B degraded (brittle) — not tested

*From DuPont Technical Report X-226

TABLE 5-6

TYPICAL SYNTHETIC ROPE STRENGTHS

		Eight-Strand Rope						Double-Braid Rope					
		Nylon		Polyester		Polypropylene		Nylon		Polyester		Polypropylene	
Diam. mm	Circumf. inch	kN	kips*	kN	kips	kN	kips	kN	kips	kN	kips	kN	kips
72	9	881	198	707	159	574	129	1268	285	1112	250	712	160
80	10	1076	242	867	195	707	159	1432	322	1246	280	863	194
88	11	1286	289	1036	233	845	190	1708	384	1495	336	1023	230
96	12	1503	338	1228	276	1005	226	2006	451	1761	396	1201	270
104	13	1784	401	1423	320	1157	260	2326	523	2051	461		
112	14	2064	464	1646	370	1334	300	2664	599	2362	531	1557	350
120	15	2353	529	1882	423	1535	345	3025	680	2696	606	1779	400
128	16	2651	596	2122	477	1713	385	3407	766	3047	685		
144	18	3350	753	2682	603	2171	488	4226	950	3812	857	2491	560
168	21							5605	1260	5115	1150		
192	24							6339	1425	5560	1250	3292	740
208	26							7326	1647	6405	1440		
224	28							8380	1884	7295	1640		
240	30							9488	2133	8229	1850		

*kip = 1000 pounds

TABLE 5-7

UNITED ROPE WORKS WET FATIGUED STRENGTH TESTS
56 mm DIAMETER (7 in. CIRCUMFERENCE) NYLON 6.6 ROPE

Sample	Construction	Treatment	Breaking Strength* Kilograms Force
A	3 x 3 strand**	Coated	54,000
B	3 x 3 strand**	Not coated	50,000
C	3 x 3 strand**	Oil lubricated	47,650
D	4 x 3 strand	Coated	53,325
E	4 x 3 strand	Noncoated	50,500
F	4 x 3 strand	Oil lubricated	45,325

All samples were wetted with fresh water.
All samples were cycled 5000 times to 30% of breaking strength at one load cycle per minute and then loaded to breaking point.

*Catalog breaking strength is 56,000 kgf and this is claimed to be 10% conservative.

**3 x 3 strand is also referred to as 9-strand or cable-lay rope.

TABLE 5-8

EXXON CYCLIC LOADING TESTS
64 mm DIAMETER (8 in. CIRCUMFERENCE)

Load Range	Cycles/min	No. Of Cycles	Breaking Load	Percentage Retained Strength
167-304 kN	3	10,000	667 kN	97
167-304	3	20,000	706	100
167-304	3	10,000	559	81
167-520	2	20	520	-*
167-520	2	750	520	-*
167-520	2	10,000	471	68

*Rope failed during cyclic loading at indicated number of cycles

TABLE 5-9

LARGE CHAIN BREAKING LOADS

Size		Grade 3		Oil Rig Quality		Special Alloy	
mm	in.	kN	kips*	kN	kips	kN	kips
54	2 1/8	2,269	510	2,438	548		
76	3	4,315	970	4,648	1,045	6,036	1,357
102	4	7,259	1,632	8,881	1,996.5	10,160	2,284
127	5	10,707	2,407	12,900	2,900	14,973	3,366
152	6	14,488	3,257	15,871	3,568		
178	7	18,627	4,188	20,594	4,630		

*1 kip = 1000 lbs

Note:

Proof loads are approximately 70% of breaking loads.

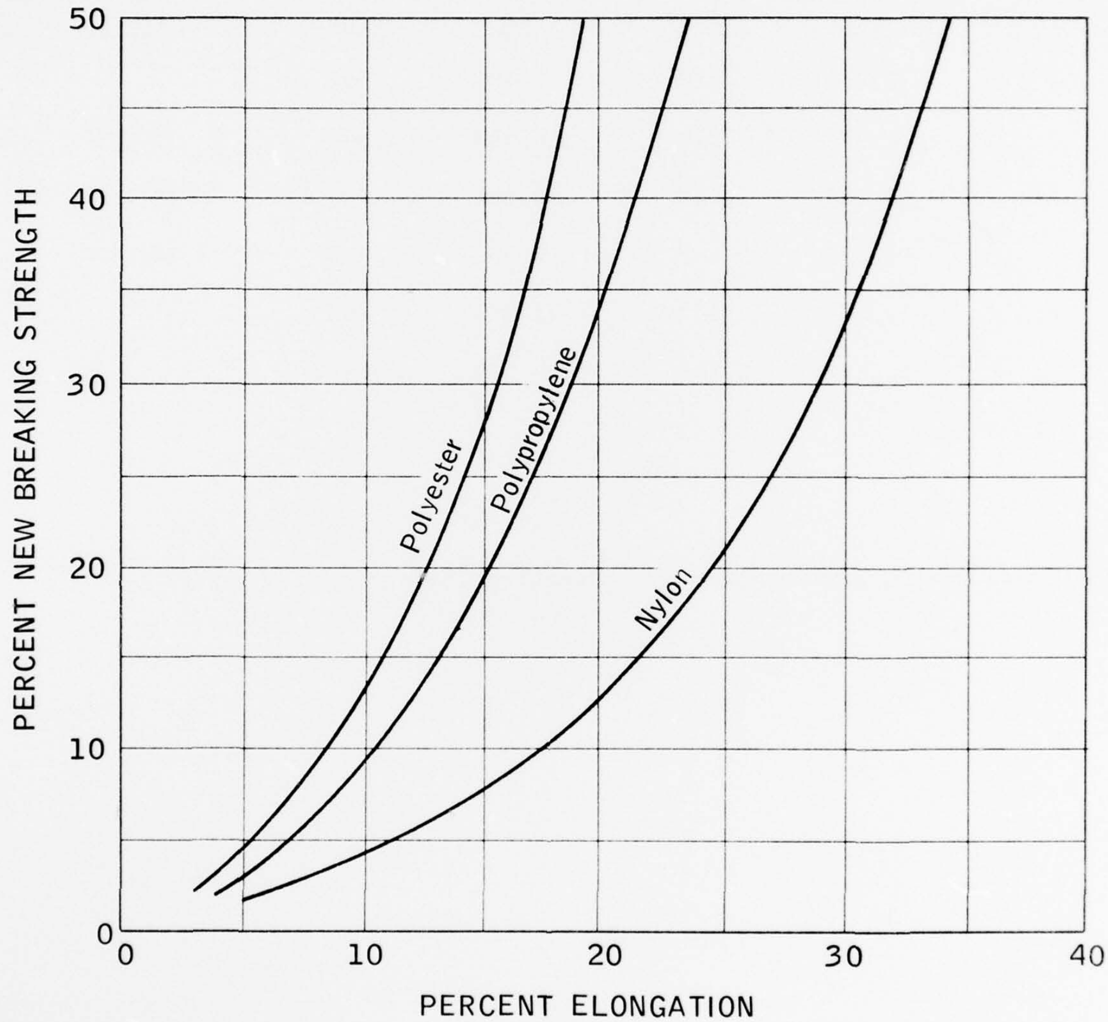


Figure 5-1 - NEW THREE-STRAND ROPE LOAD-ELONGATION CURVES FOR THREE MATERIALS (Wall Ropes 1963-64)

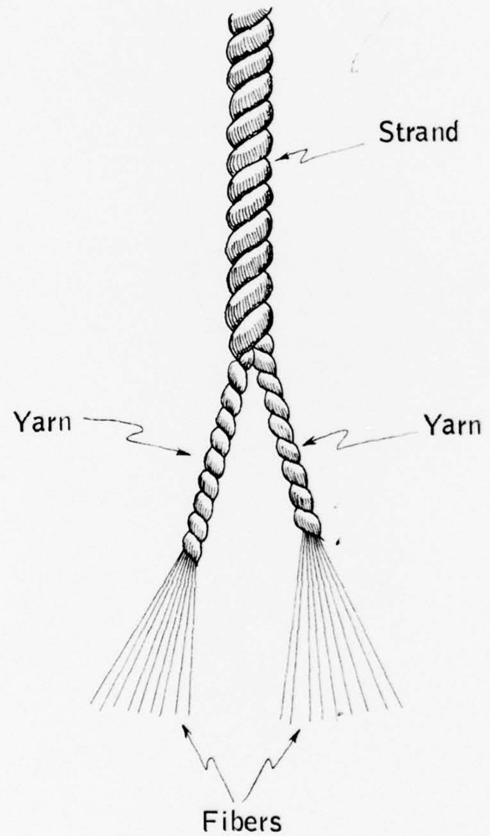
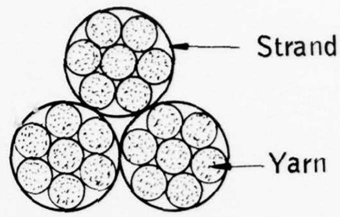


Figure 5-2 - PROCESS OF TWISTING FILAMENTS INTO YARNS AND STRAND



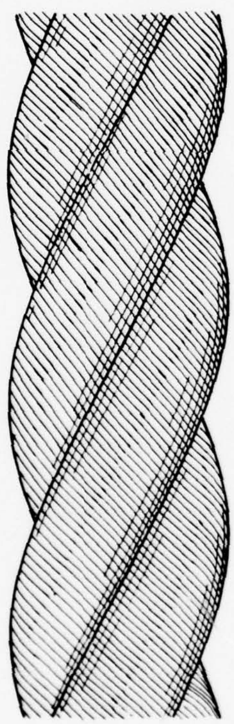


Figure 5-3 - EXAMPLE OF THREE-STRAND ROPE

Figure 5-4 - EXAMPLE OF A HOCKLE IN THREE-STRAND ROPE

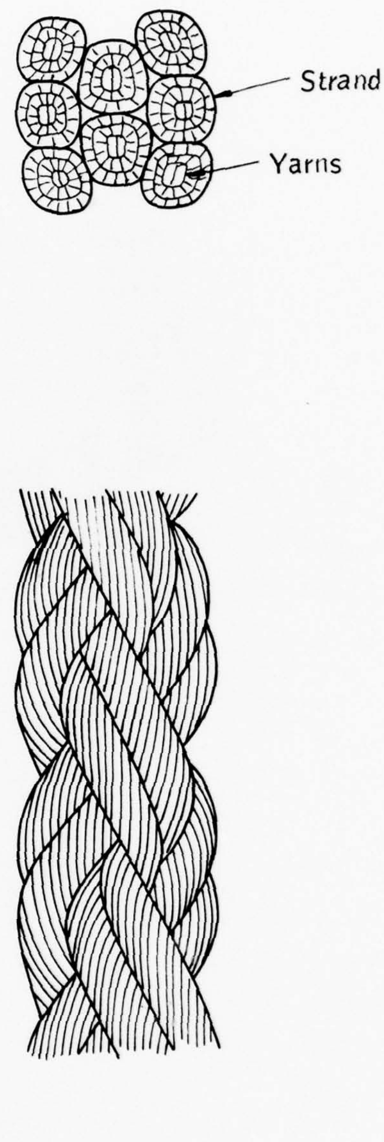


Figure 5-5 - EXAMPLE OF EIGHT-STRAND ROPE

Figure 5-6 - EXAMPLE OF DOUBLE-BRAID ROPE

Figure 5-7 - 12 STRAND SINGLE-BRAID ROPE

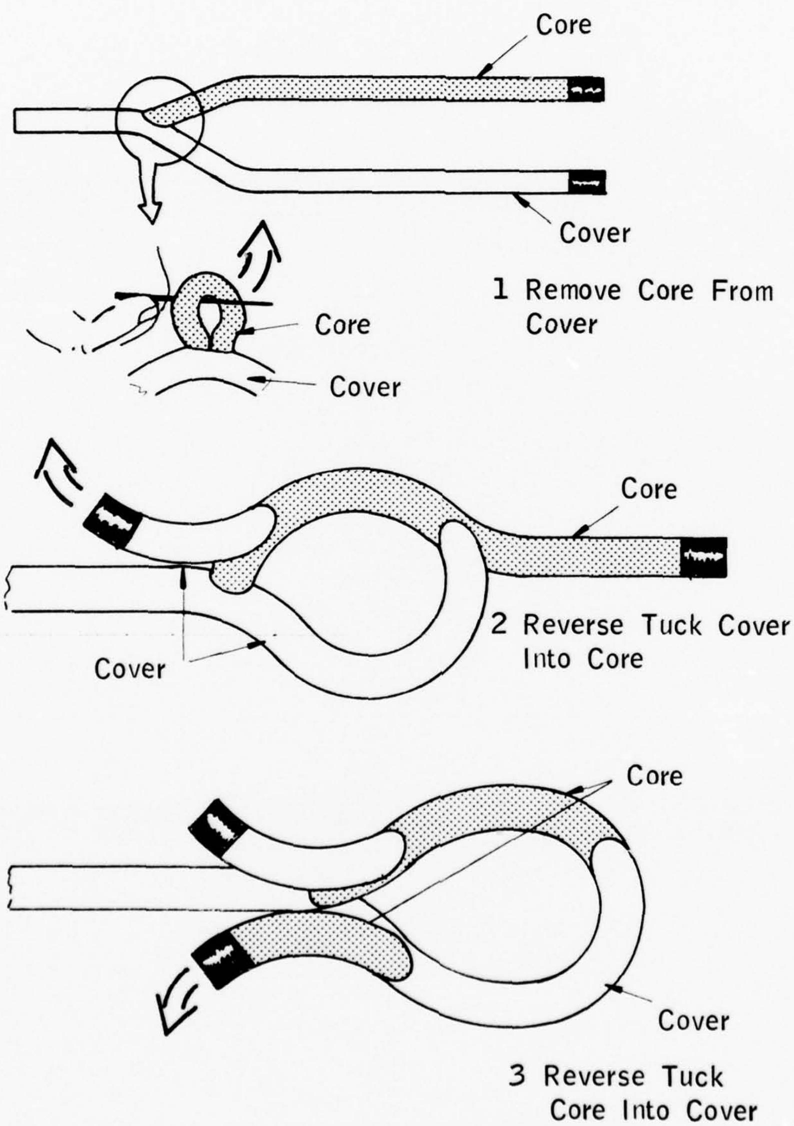
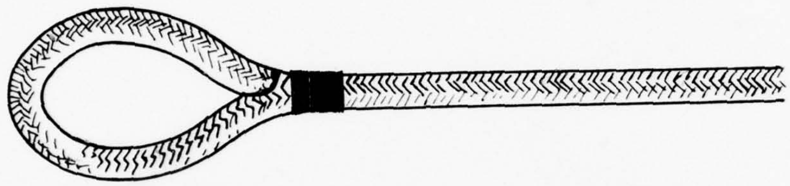


Figure 5-8 - SIMPLIFIED DOUBLE-BRAID ROPE SPLICING PROCEDURE



Eye Spliced Rope

End-For-End Spliced Rope

Figure 5-9 - EXAMPLES OF EYE SPLICE AND END-FOR-END SPLICE IN DOUBLE-BRAID ROPE

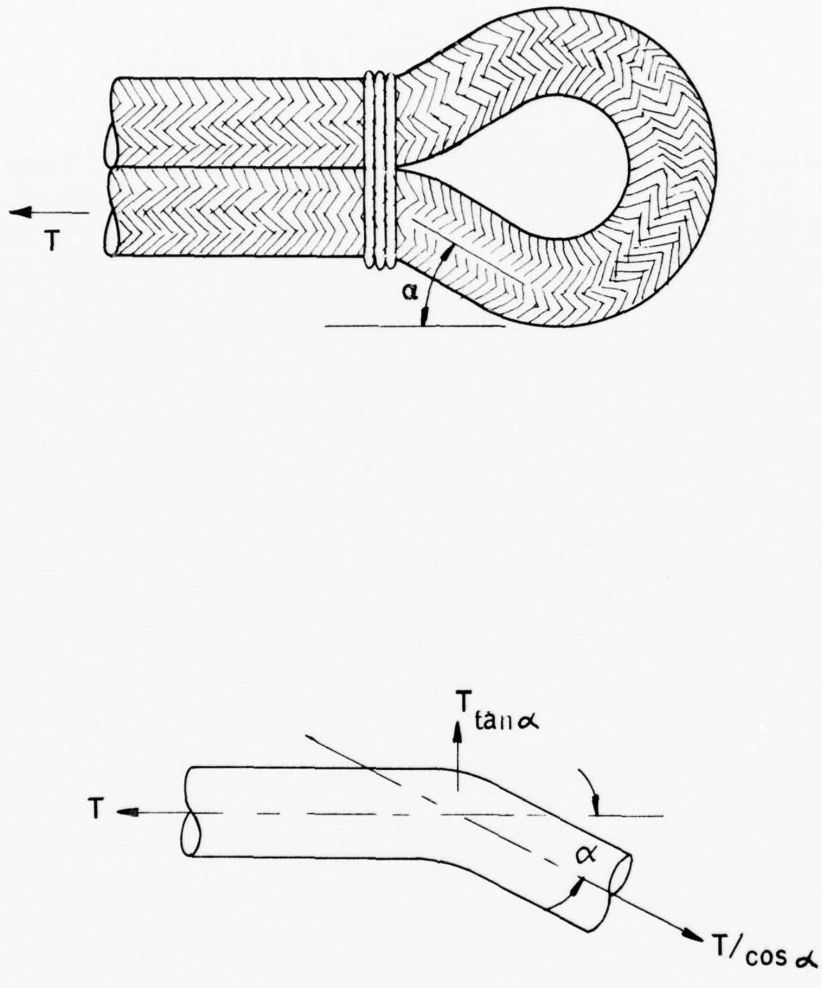


Figure 5-10 - STRENGTH REDUCTION MECHANISM DUE TO ANGLE OF ROPE IN EYES

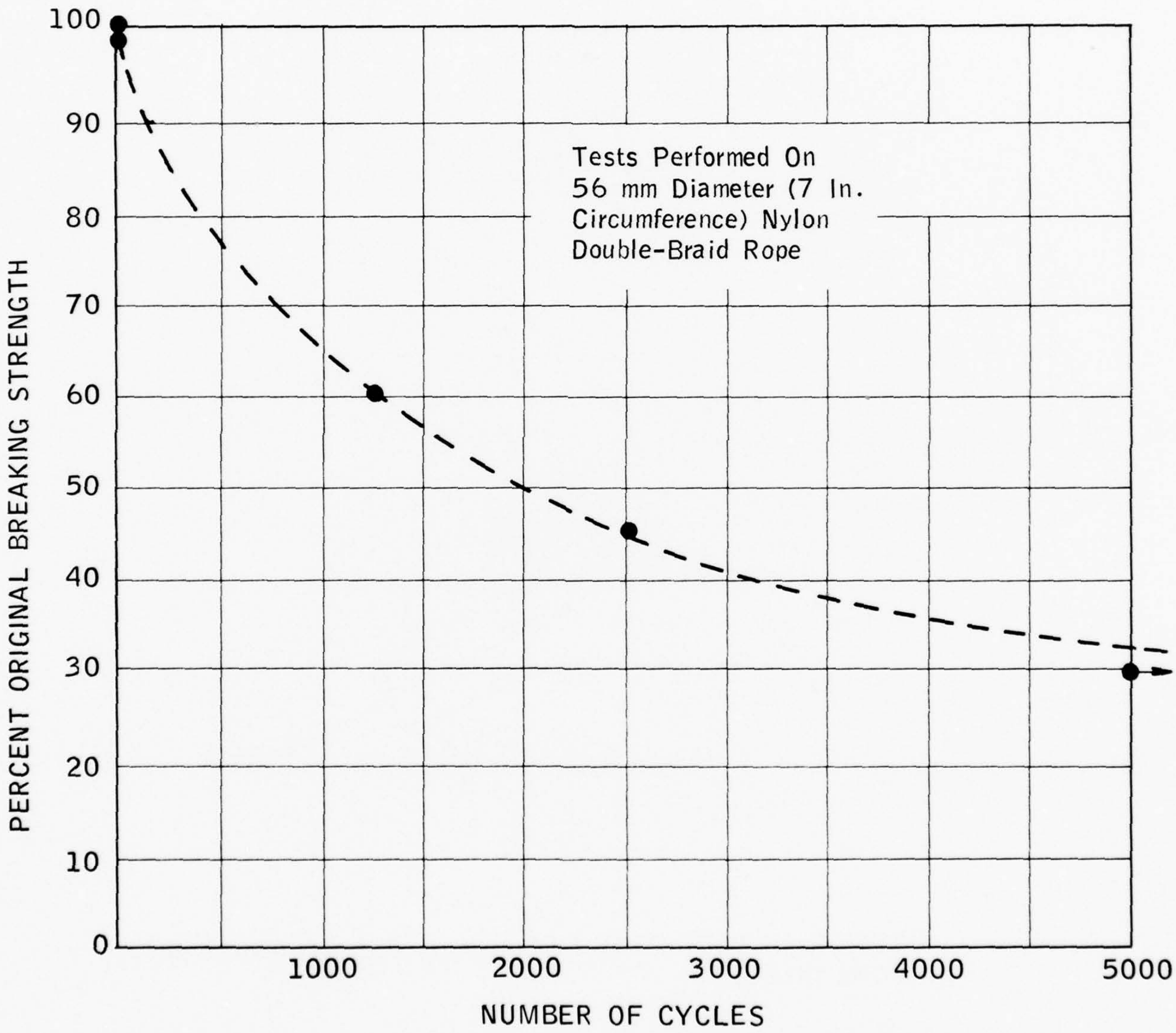


Figure 5-11 - SHELL ROPE FATIGUE TESTS (Langeveld, 1974)

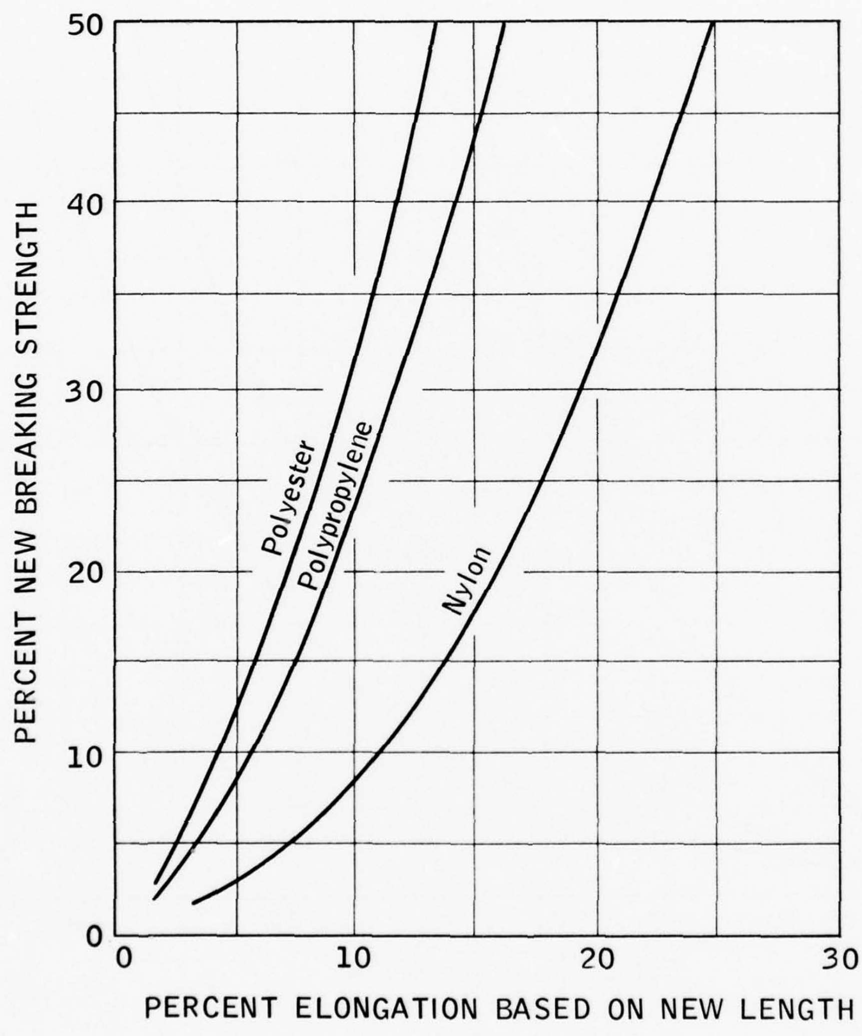


Figure 5-12 - BROKEN-IN THREE-STRAND ROPE LOAD-ELONGATION CURVES FOR THREE MATERIALS (Wall Ropes 1963-64)

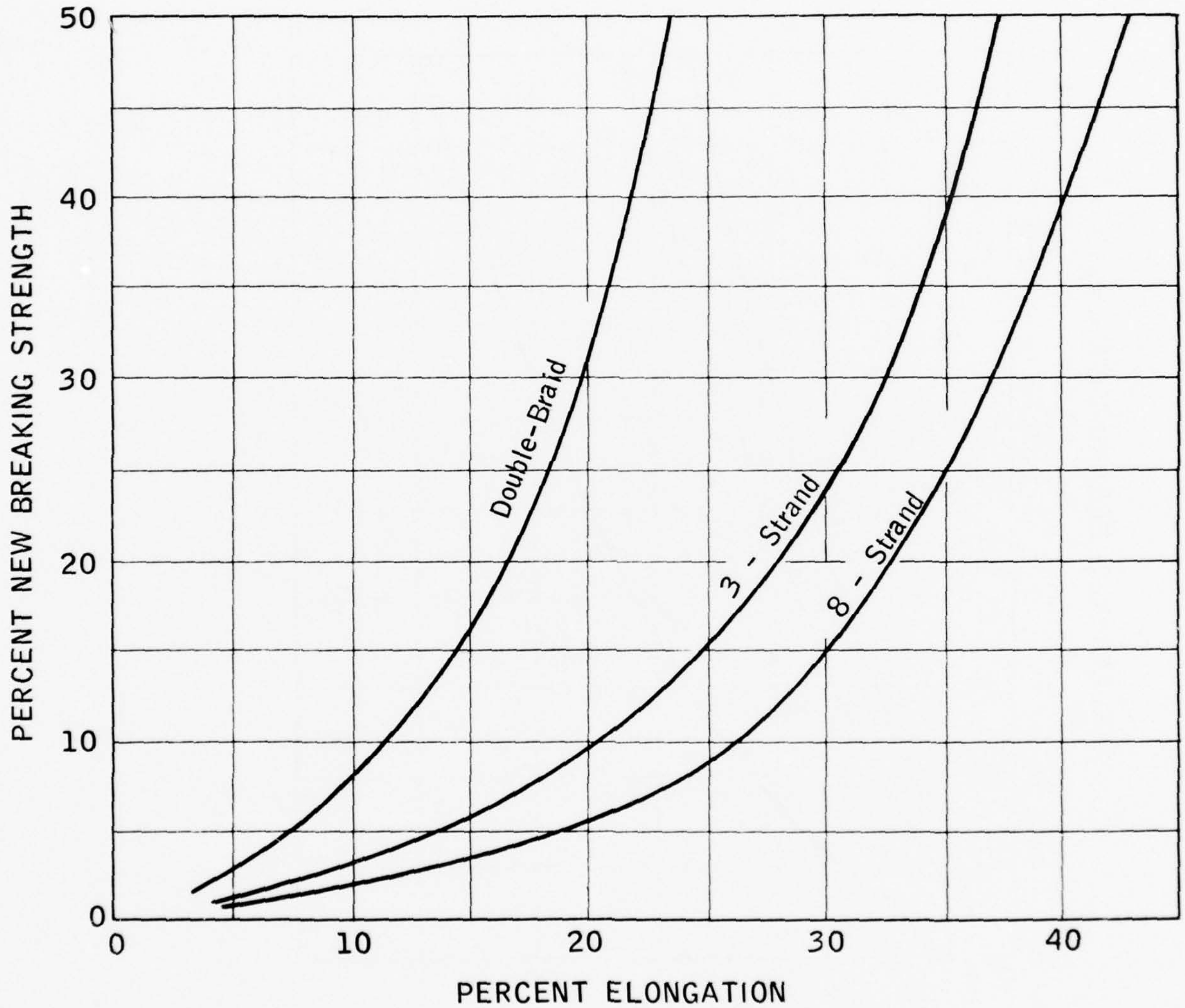


Figure 5-13 - NEW (WET) NYLON ROPE LOAD-ELONGATION CURVES FOR THREE CONSTRUCTIONS (Wall Ropes 1977)

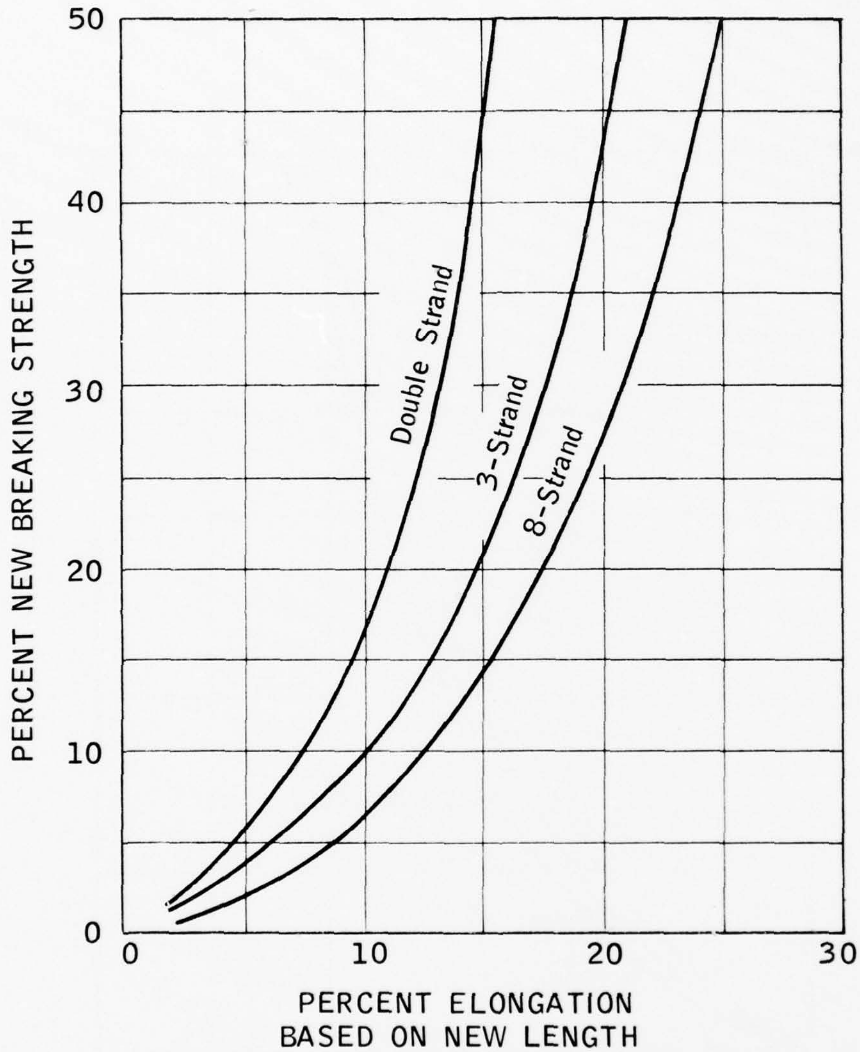


Figure 5-14 - BROKEN-IN NYLON ROPE LOAD-ELONGATION CURVES FOR THREE CONSTRUCTIONS (Wall Ropes 1977)

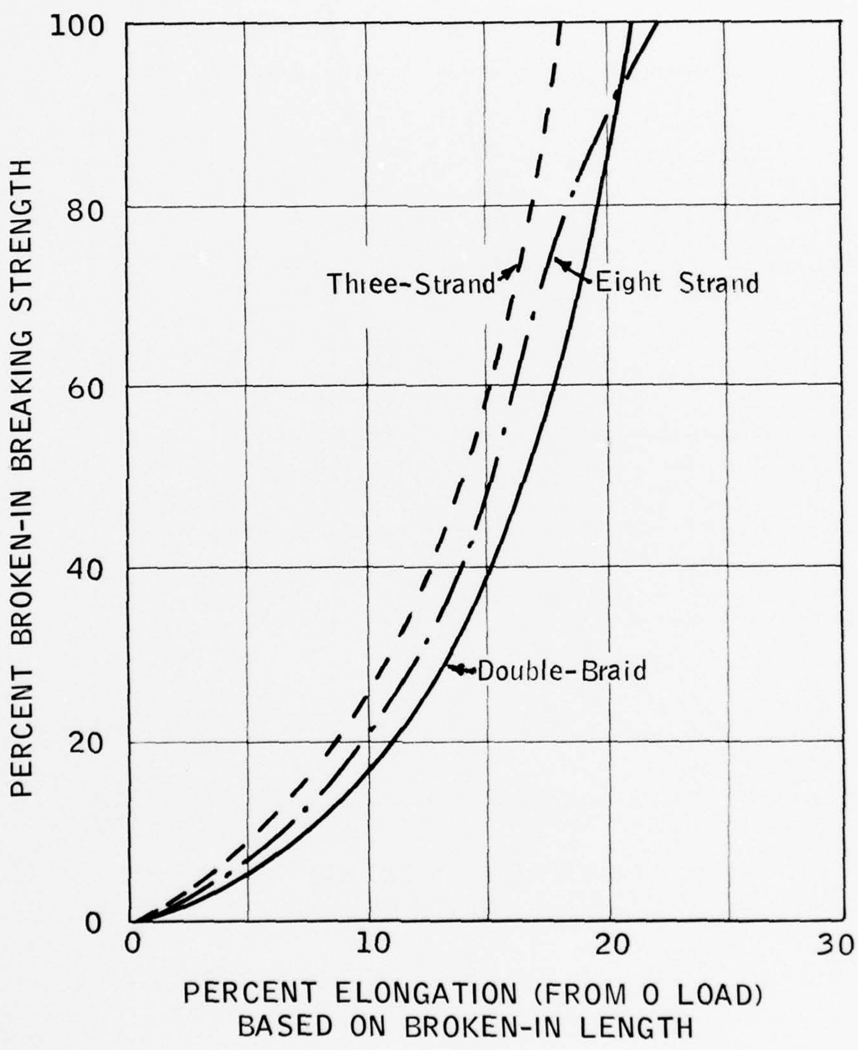


Figure 5-15 - BROKEN-IN NYLON LOAD ELONGATION CURVES BASED ON BROKEN-IN LENGTH (From British Ropes Graph 1200.69)

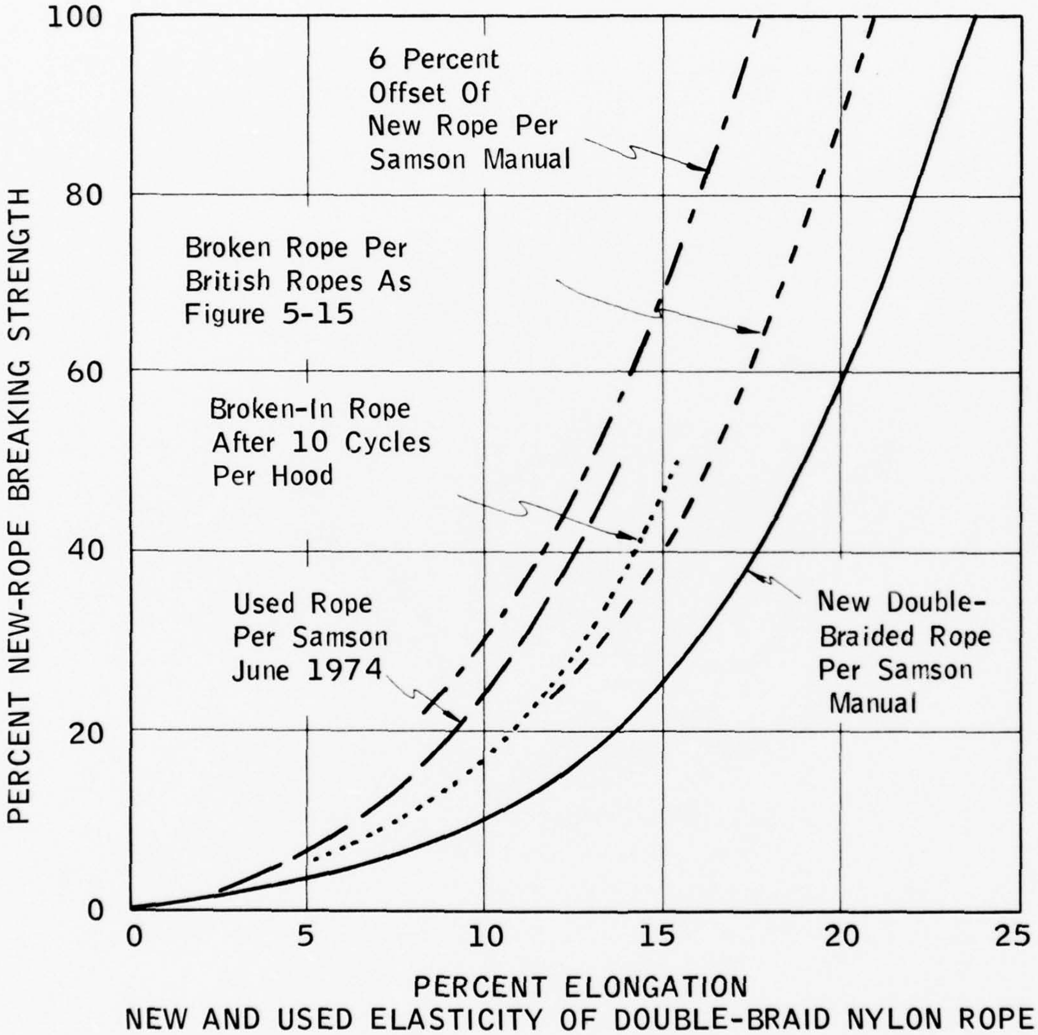


Figure 5-16 - BROKEN-IN AND USED NYLON ROPE LOAD-ELONGATION CURVES FROM VARIOUS SOURCES

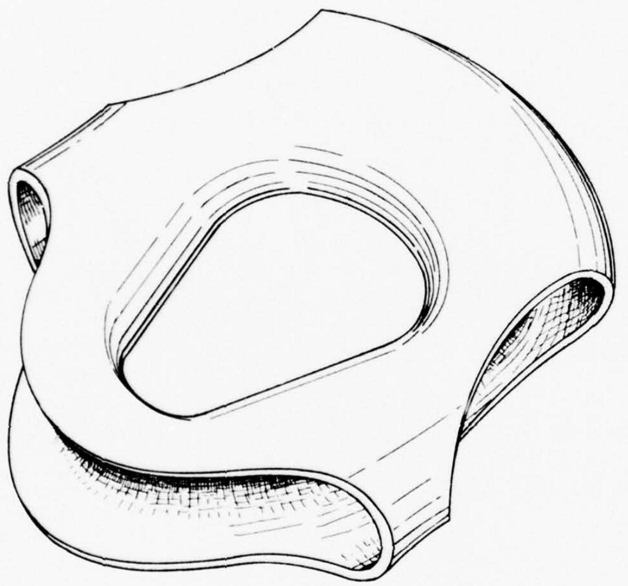
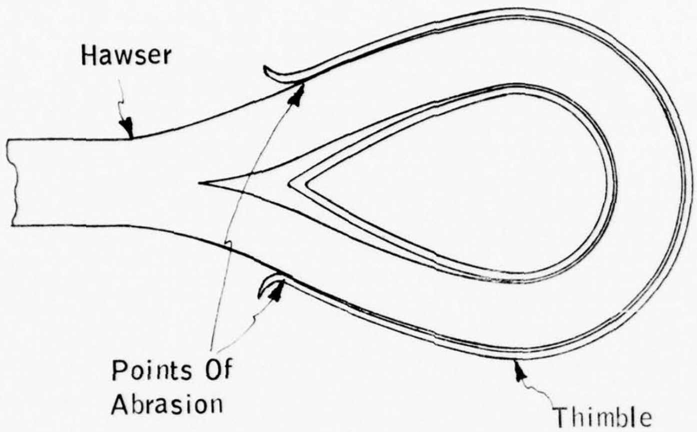
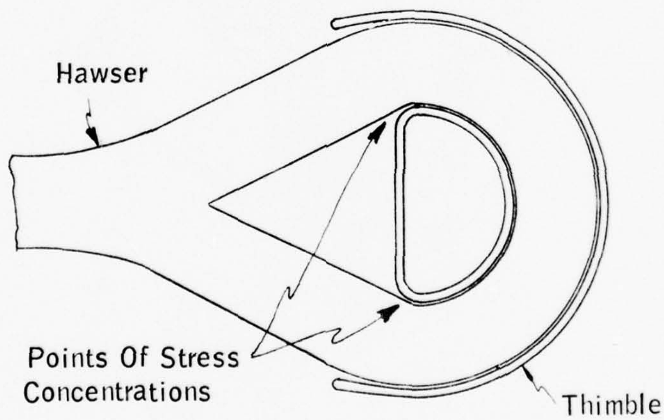


Figure 5-17 - EXAMPLE OF A TYPICAL HAWSER THIMBLE



A NARROW-MOUTH THIMBLE



A HALF-CIRCULAR SHAPED THIMBLE

Figure 5-18 - EXAMPLES OF POORLY DESIGNED THIMBLES

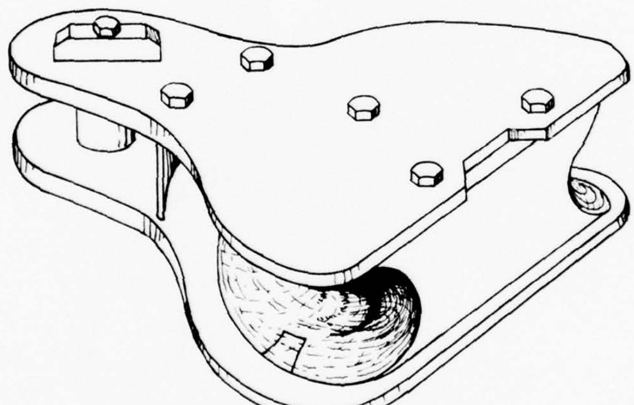


Figure 5-19 - EXAMPLE OF A DETACHABLE HAWSER THIMBLE

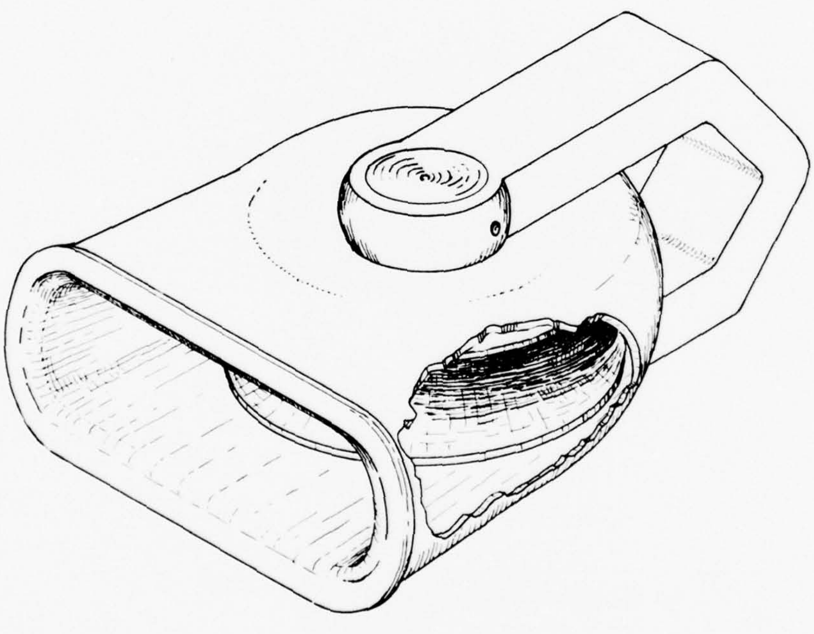


Figure 5-20 - EXAMPLE OF A BUOY-END HAWSER THIMBLE

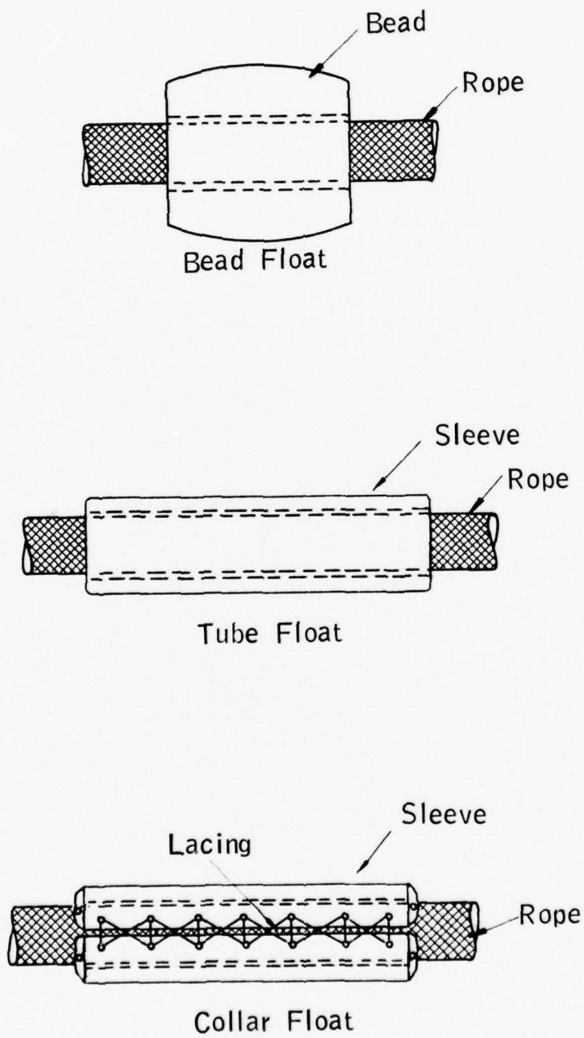


Figure 5-21 - EXAMPLES OF DIFFERENT TYPES OF HAWSER FLOTATION

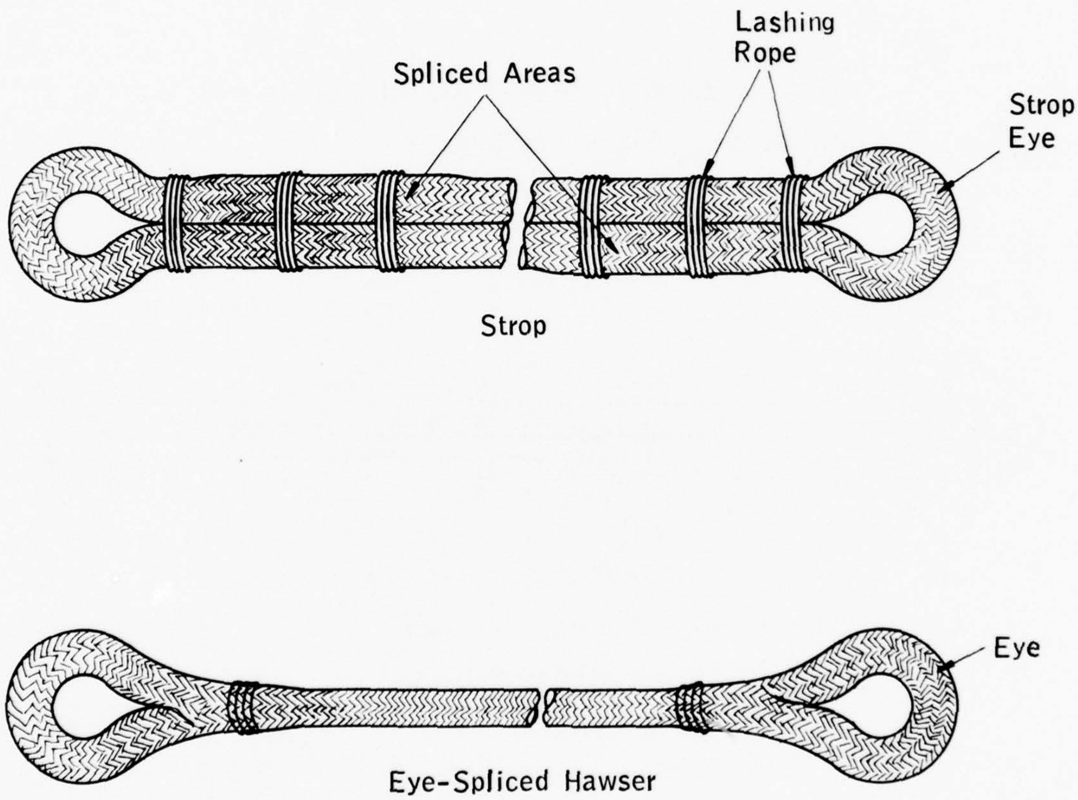
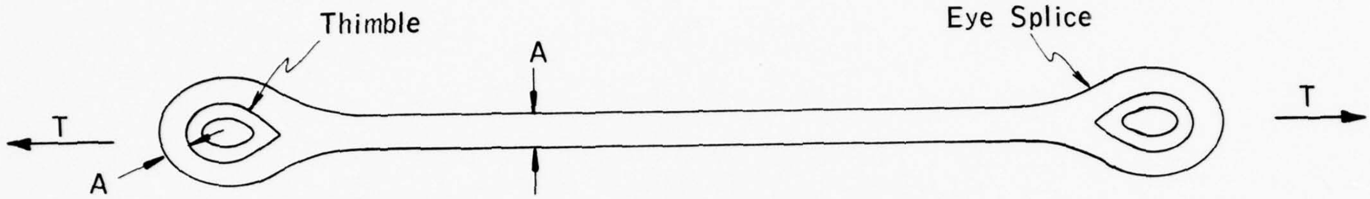
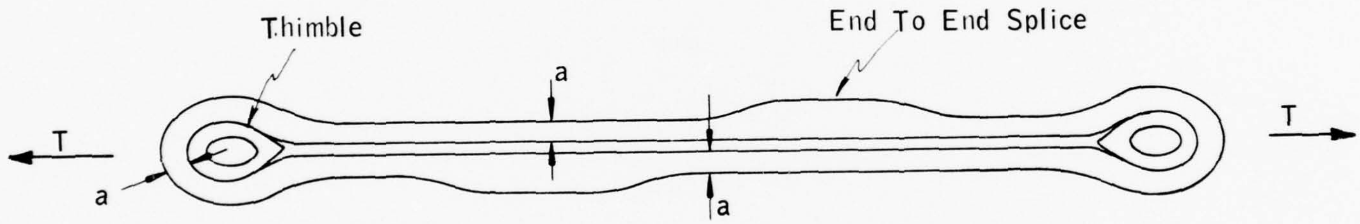


Figure 5-22 - EXAMPLES OF STROP AND EYE-SPLICED HAWSERS



Area In Midsection = Area Around Thimble = A

Single Leg, Eye Splice Ends



Total Area In Midsection = 2a = A'

Area Around Thimble = a = A'/2

Double Leg, Midsection Splices

Figure 5-23 - SINGLE-LEG AND DOUBLE-LEG (GROMMET) HAWSERS

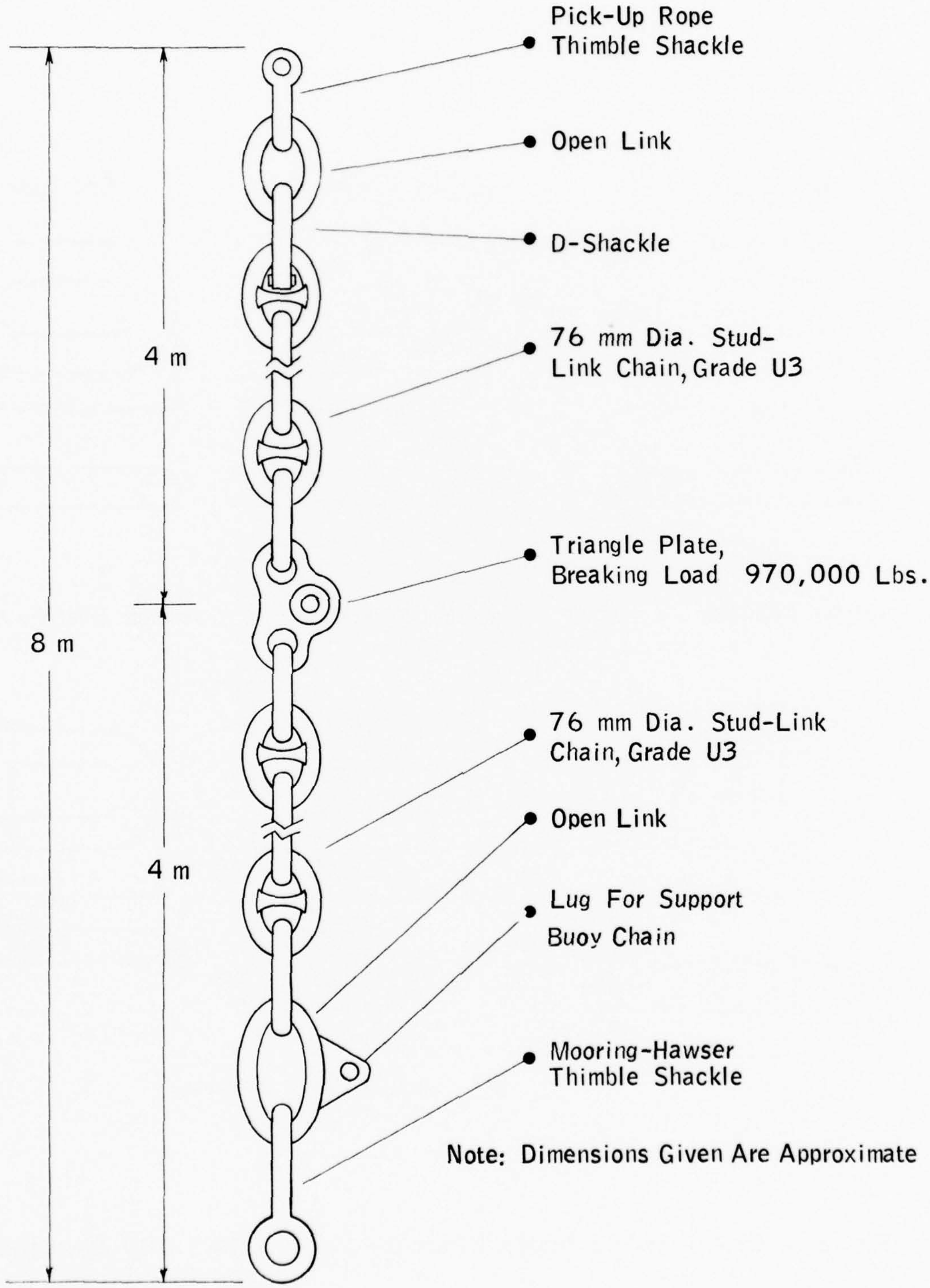


Figure 5-24 - EXAMPLE OF TYPICAL TANKER-END SPM CHAFING CHAIN

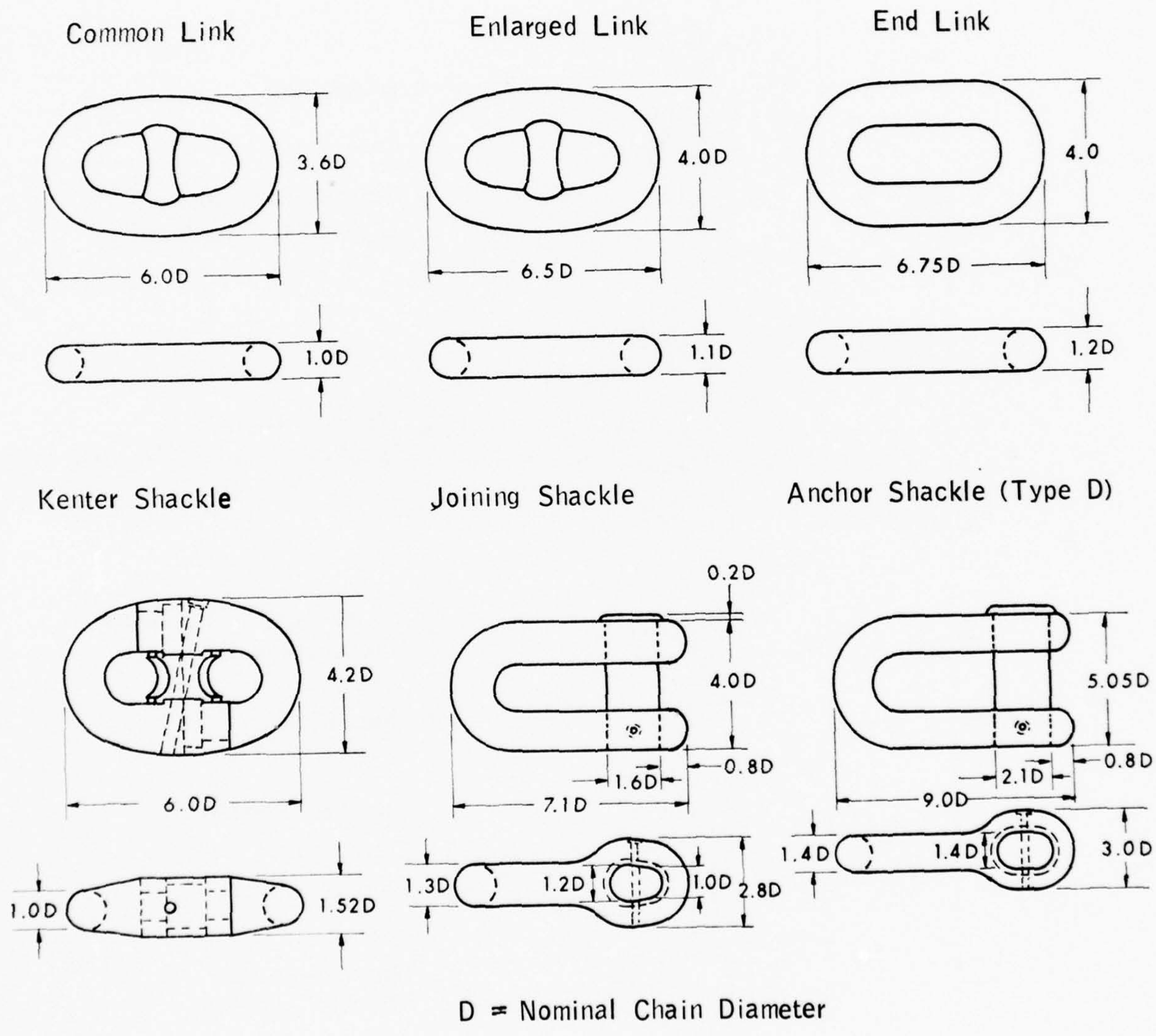


Figure 5-25 - TYPICAL DIMENSIONS OF CHAIN LINKS AND SHACKLES

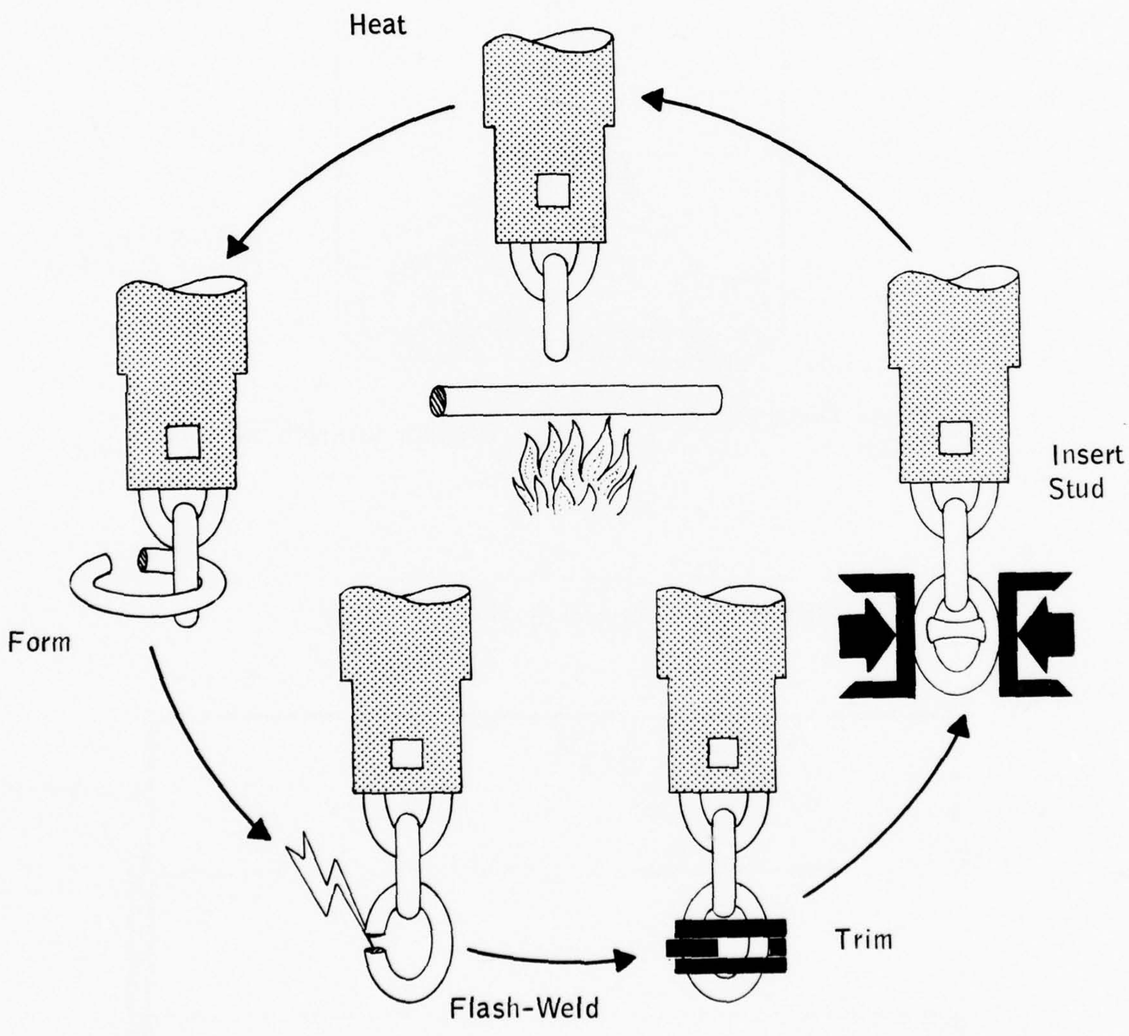
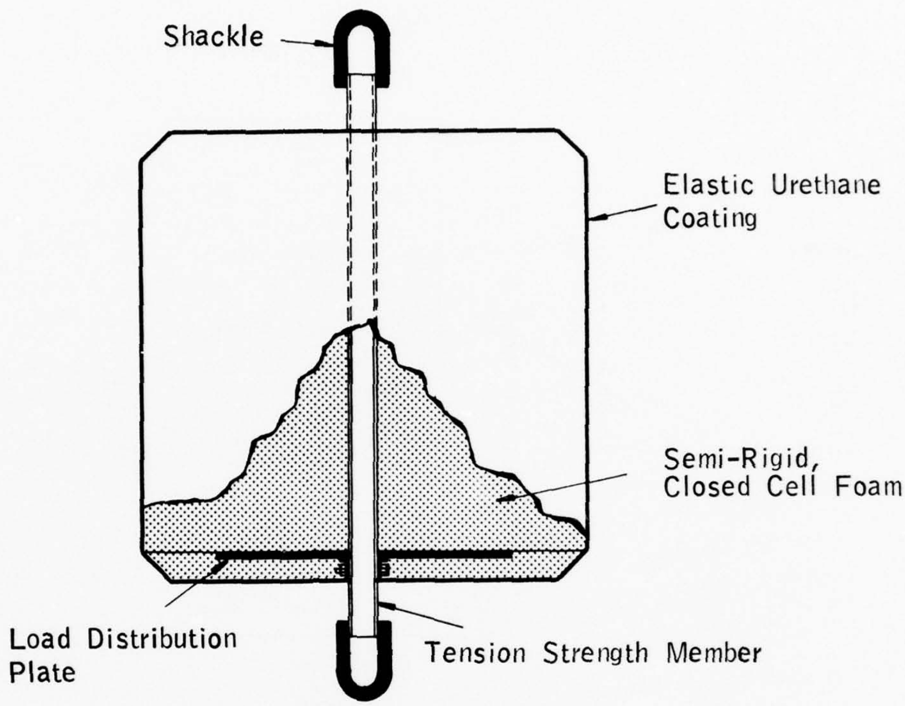


Figure 5-26 - PROCEDURE FOR MANUFACTURING FLASH-WELDED CHAIN

PENANT-TYPE CHAIN-SUPPORT BUOY



HAWSER-TYPE CHAIN SUPPORT BUOY

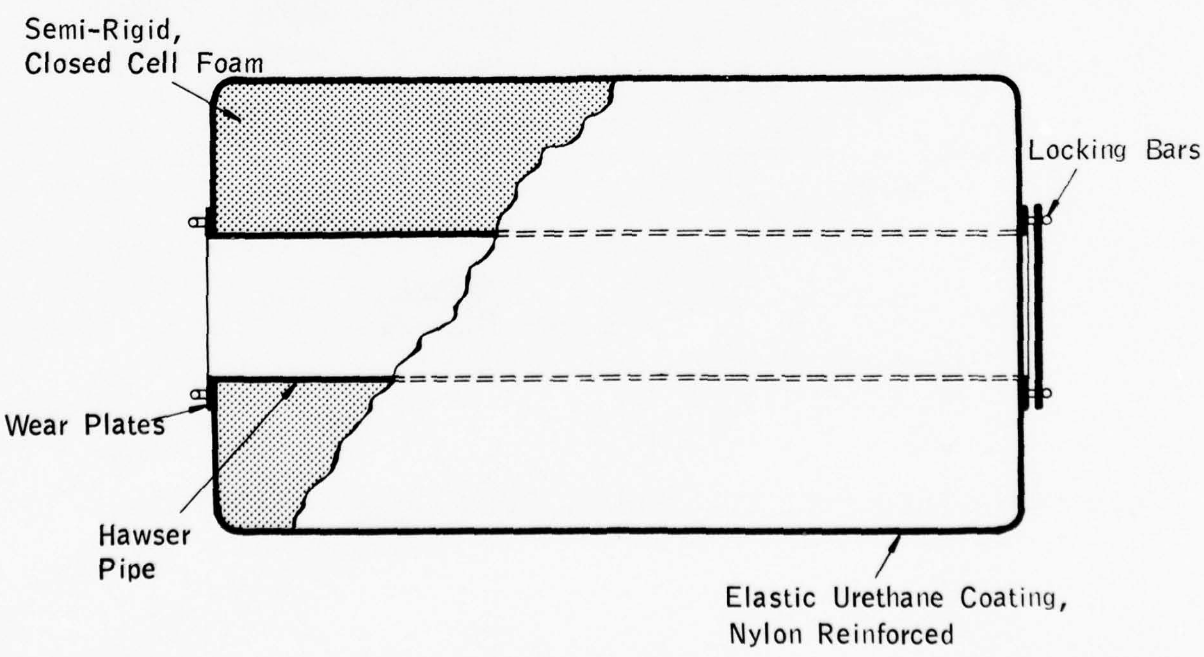


Figure 5-27 - EXAMPLES OF CHAIN SUPPORT BUOYS

SECTION 6

TESTING, INSPECTION, AND REPLACEMENT CRITERIA FOR SPM HAWSERS

6.1 INTRODUCTION

Although synthetic ropes have been used in the marine industry for many years, relatively little information exists on rope testing, inspection, and replacement criteria of the large-size synthetic ropes used as SPM hawsers. Most of the synthetic-rope testing that has been carried out to date has been with smaller-size ropes (<72 mm diameter, 9 in. circumference) due to a greater demand for this size of rope and due to the limitations of existing testing equipment.

This section will document available information on rope testing, inspection, and replacement criteria. This information is based on the results of a literature search, discussions with several of the major European and U.S. rope manufacturers, discussions with rope testing organizations, and a review of recent SPM user experience. The major emphasis will be on information related to larger-diameter (>72 mm) ropes. However, reference will also be made to certain efforts on smaller ropes when this information is believed to be of significance for use with the larger ropes.

6.2 ROPE TESTING

6.2.1 Present Status of Testing Large Ropes

Extensive testing of synthetic ropes in sizes larger than about 72 mm diameter (9 inch circumference) has not been carried out. There are several reasons for this. First the demand for large-diameter ropes is generally limited and hence will not support major research and development programs by the rope manufacturers. The major use of these large-diameter synthetic ropes is as hawsers for SPMs. Secondly, testing equipment capable of developing the high tension loads and long strokes needed to break large-diameter rope specimens is not readily available at present.

Generally, most rope manufacturers have based the strength and performance criteria of the larger-sized marine ropes on extrapolations of test information for smaller-sized synthetic ropes made of the same material and of similar or identical construction. Several rope manufacturers have carried out limited testing on the larger ropes in order to confirm their extrapolation methods and procedures. However, only a few such tests have been conducted, and in some instances special testing setups and methods have been required to enable the use of existing test equipment. Therefore, the rated breaking strengths listed in the rope manufacturers' catalogs for the larger sized ropes have generally not been verified by testing.

Very few tests on used SPM hawsers have been performed to date. In addition to the problems cited above, splicing of used ropes is difficult and,therefore, preparing a short test specimen from a long used hawser is not usually possible. ARAMCO is presently carrying out a program of testing used hawsers. This program will be discussed in more detail in subsection 6.5.4.

6.2.2 Large-Rope Testing Equipment

The manufacturers of large diameter synthetic ropes do not have equipment capable of loading these ropes to break. Several commercial testing firms have performed limited breaking tests on large-diameter synthetic ropes. These organizations, along with a listing of their testing equipment and large rope testing experience, are given in Appendix D.

The National Engineering Laboratory (NEL) in Scotland has developed tentative plans for the addition of a new testing machine specifically for testing large-diameter ropes. It would appear, at this time, that government or industry funding will be necessary to help support NEL in development of this equipment (Benham, November, 1976).

New testing equipment presently under construction at Coordinated Equipment Company (CEC) in California appears to meet the requirements to perform static breaking tests on new and used SPM ropes (Flory, January, 1977). The test bed is long enough to accommodate large sections of rope and, perhaps, entire lengths of SPM hawsers. In addition, the hydraulic cylinders have the capacity to achieve the catalog breaking strengths of the largest hawsers now made. However, installation of this test apparatus is not yet completed so its capabilities are yet to be proven. Several rope manufacturers are studying the possibility of break testing ropes at CEC.

6.2.3 Rope Testing Specifications

A number of specifications are available which cover the requirements for testing of synthetic ropes. Table 6-1 is a listing of the more common specifications and the issuing agencies. Unfortunately, most of these specifications were developed for testing of smaller-sized ropes (<96 mm diameter) and have limited applicability for testing very large ropes. In addition, most of the specifications were developed for ropes of stranded and plaited construction.

Several of the issuing agencies are currently in the process of updating the specifications to include larger ropes and double-braided rope construction. Until these specifications are updated and expanded, use of the existing documents for testing of larger ropes should be done with care. Items in the existing documents, such as minimum specimen length and rate of specimen loading, may be misleading or not applicable to large ropes.

The SPM Rope Committee of the Oil Companies International Marine Forum have been advocating the need for developing a comprehensive testing specification covering large diameter SPM hawsers. The need for such a specification is more urgent now that testing equipment capable of performing tensile tests on large ropes will soon be available. Industry use of a specification covering large rope testing will assure that test data can be compared and correlated on the same test basis.

6.2.4 Types of Rope Tests

Rope testing for SPM hawsers can be divided into two main categories. The first category is tests carried out on new ropes to demonstrate or confirm that the product in the new conditon meets the claims of the manufacturer and the specification of the user. These tests would include:

- Tensile tests to demonstrate new rope breaking strength, load-elongation characteristics, and splice reliability.

- Cyclic loading tests to access the effect of fatigue loading on rope fibers and on ropes of various materials and types of construction.

- External abrasion tests to access how various rope constructions and materials will perform in an abrasive field application.

- Exposure tests to assess how various rope constructions and materials sustain the effects of ultraviolet light and chemical attack.

The second main category of rope testing are tests carried out on used SPM ropes to determine the condition and retained strength after various periods and types of service. These tests would normally consist of tensile breaking strength and load-elongation tests performed on rope specimens taken from SPM hawsers retired from service.

At present, there is no simple and reliable non-destructive field test that can be performed on a hawser to determine when it should be replaced. Comprehensive test programs could be carried out on hawsers after various lengths of service to determine which measureable properties accurately indicate rope condition and residual strength. These types of programs will be discussed in subsection 6.5.4.

The following subsections discuss several of the above-mentioned types of tests in more detail and outline certain key items for consideration when carrying out such tests.

6.2.5 Static Tensile Tests

Based on discussions with rope manufacturers and personnel experienced in rope testing, the following parameters are considered to be the most important when performing static tensile tests on ropes:

- Specimen length
- End-connection details
- Type and quality of rope splices
- Rate of specimen loading

Rope test specimens should be of sufficient length to minimize effects in the test that would not be present in the actual rope being used in service. Normally the rope should be tested with an eye at each end through which the pin or bollard of the test machine would be passed. Each eye should be formed with a typical eye splice. The specimens should have a minimum length of undisturbed rope equal to about 8 to 12 times the rope circumference between the ends of each eye splice (See Figure 6-1). The sum of the lengths of the two eye splices and the undisturbed rope will produce a test specimen with an overall minimum length of about 20 to 25 times the circumference of the rope being tested.

The testing equipment must have the capability to stretch the specimen to the breaking point. This could require a machine stroke of up to 50 percent of the unstretched specimen length depending on rope material, type of construction, and whether the rope is new or used. Thus the overall length of the test bed may have to be as much as about 40 times the circumference of the rope.

For testing large-diameter ropes, an eye-spliced test specimen between two large pins is the best mounting arrangement. Pin diameters should be approximately two times the rope diameter to simulate the condition of a thimble and to avoid overstressing of the eye. Such an arrangement not only tests the strength of the pure rope, but also tests the strength of eye-splices similar to those used on SPM hawsers. Testing of smaller three-strand and eight-strand rope specimens has in the past been done with the ends secured by wedge grips. The effects of the wedge grips on the breaking strength of the specimen is questionable, and this arrangement does not simulate typical hawser conditions. Personnel experienced in rope testing claim that for larger-diameter ropes the effect of the wedge grips would probably be even more pronounced than for smaller ropes. Hence, they recommend not using wedge grips for testing large-sized ropes.

Generally rope tensile tests carried out on an eye-spliced specimen result in failure at or near the splice, as this is usually the weakest point in the rope. Hence, the test results should accurately reflect the effects of the splice. Splices used in test specimens should be carried out by personnel trained in rope splicing, and the splices should be made in the same manner as that to be used on the final SPM hawser.

When tensile testing used SPM ropes, the ideal arrangement would be to test the entire rope as a single test specimen. However, this will probably not be possible due to test-equipment bed-length and stroke limitations. A representative test specimen containing one of the original eye splices should be taken from the used rope. Splicing a new eye on the other end of a used rope specimen will most likely be difficult or impossible, as the rope becomes very tight and the strands become hard with use. It may, therefore, become necessary to use some type of bitt or cleat arrangement to tie-off the free end of the rope specimen. Selection of the proper diameter of end fitting is important to avoid failure at the bitt or cleat due to excessive rope bending.

In situations where it is not possible to test used rope specimens, testing of yarns from the used rope may reveal qualitative information on the strength degradation of the used rope.

Another item that can cause variations in the results of tensile tests is the rate of load application to the test specimen. Some testing specifications recommend the rate of loading be constant regardless of specimen size or length. Other specifications recommend that the strain rate be constant, that is the rate of loading be proportional to the length of the test specimen. Because of their energy absorbing characteristics, synthetic ropes will break at different loads depending on the rate at which they are loaded. In general, the faster a synthetic rope is loaded, the higher the breaking load will be (this does not apply when the rate of loading approaches a shock loading situation). This question of loading rate appears to be an area requiring further research, especially in regard to testing of large ropes of different materials and constructions.

6.2.6 Cyclic Loading Tests

When performing cyclic loading tests on rope specimens the following items, in addition to those mentioned in subsection 6.2.5, are considered as the more important parameters:

- Frequency of loading
- Pattern of loading

Cyclic loading tests on smaller sized ropes, 64 and 72 mm diameter, (8 inch and 9 inch circumference) have shown that frequency of loading will significantly affect rope performance. For a cyclic loading pattern of constant amplitude, as the frequency of loading increases the rope strength generally deteriorates more rapidly, that is the strength will decrease by a certain percentage in fewer cycles.

Discussions with rope manufacturers and personnel from rope testing organizations have confirmed the importance of the loading pattern in cyclic load tests. A rope cyclically loaded from zero load to 40 percent of its new rope breaking strength may experience more strength deterioration than the same rope cycled the same number of times from 10 to 50 percent of its new-rope breaking strength. This is discussed further in subsection 5.5.1.

Because of the importance of so many parameters in cyclic tests, comparison of available test data is extremely difficult. Standards for cyclic load tests need to be developed for large-diameter synthetic ropes. The standards for cyclic loading should be developed keeping in mind the use of the rope as an SPM hawser and the type of loading history the rope is most likely to see in service.

6.2.7 Other Tests

There are no universally accepted standards for abrasion testing of ropes. Various types of abrasion tests have been developed by rope fiber manufacturers and rope manufacturers. Most of these tests are carried out on rope fibers or very small (about 12 mm diameter) sample ropes. These tests are conducted to determine the qualities and relative performance of different rope fibers, fiber finishes, and rope constructions. Because of the lack of standardization in this area, comparisons of test data are extremely difficult. Information on the relative abrasion properties of the various synthetic rope fibers is covered in subsection 5.2.3.

Tests have been carried out to determine the effects of chemical attack and ultraviolet light on rope fibers and small diameter ropes. Results of some of these efforts are covered in subsection 5.2.6.

6.3 IN-PLANT INSPECTION OF NEW ROPE

Inspection of ropes can be divided into two categories; in-plant inspection and field inspection. In-plant inspection should be carried out on new rope at the rope manufacturer's plant to assure that the rope has been made in accordance with the purchase specification and has the properties claimed by the manufacturer.

There is no established industry specification covering in-plant inspection of large synthetic ropes for SPM service. The most applicable specifications in this area presently in the public domain are the U.S. Military Specifications listed in Table 6-2. Unfortunately, these specifications at present cover ropes only through 96 mm diameter (12 in. circumference). However, a number of the quality assurance provisions in these specifications are applicable for larger-sized ropes.

6.3.1 In-Plant Inspection Items

The general types of inspection that should be carried out as part of an in-plant inspection of new SPM ropes are as follows:

- Check that rope size, materials, and type of construction are in accordance with the purchase specification.

- "New-rope length" should be measured and witnessed or certified by the manufacturer. New-rope length is usually defined as the distance from center to center of the two end thimbles when the rope is under a tension $T = 1.54\ d^2$*, where T is the rope tension in Newtons and d is the nominal rope diameter in mm. The new-rope length should agree with the purchase documents. A check of new-rope length is especially important on dual-hawser SPM systems to insure that the lengths of the two ropes are such that the system will share the loads as designed.

- The general condition of the rope should be checked along its entire length. Defects such as pulled or missing strands or yarns, fusion, chafe, abrasion, cuts, or discoloration should be noted. Such defects should be reviewed carefully with the manufacturer to assess the cause and extent and to determine if the rope is still fit for field service.

- All splices should be inspected for compliance with standard manufacturer's splicing instructions for the type of rope construction. Double-braided ropes should be examined to insure that the splices have been lock-stitched to prevent them from working loose under low or no-load conditions.

- Floatation should be checked to insure that the proper type and size floatation devices have been supplied and that the installation has been carried out properly.

- Thimbles should be examined prior to any encapsulation to insure that no sharp edges are present where the rope could chafe or abrade. Chafe protection in the thimble area should be examined to insure that it has been properly applied.

- All hawser hardware, including chains, plates, connecting links, and shackles should be checked for compliance with the purchase specification. Material, testing, and certification documentation should be presented where applicable.

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*In the English System of units $T = 200\ d^2$, where T is the rope tension in pounds and d is the nominal rope diameter in inches.

6.3.2 Determination Of Strength Of New Rope

Because the large diameter rope used in SPM hawsers is very expensive to make and very difficult and expensive to test, it is not practical to test a sample of each lot of rope to break. The U.S. Military Specifications call for such testing of smaller ropes which are bought in large quantities. Means of determining the breaking strength of a new manufactured rope from tests on samples of components of the rope may be practical.

A method of testing specimens of rope components and applying a realization factor to determine the strength of a new rope has been developed and is being applied by some rope manufacturers in England. This realization factor method is described in subsection 5.4.3. By this method, an inspector can select sample yarns or strands for break testing from the spools used in making the rope. Based on the breaking loads achieved by these yarn or strand specimens and on the realization factor developed for the rope construction, the breaking test of the manufactured rope can be determined. U.S. rope manufacturers question the reliability of the realization factor method of determining breaking loads. Thus far, realization factors have not been established for the products of U.S. manufacturers.

The realization factor method of determining rope breaking strength should be investigated further. If it is found to be reliable, it should be adopted as a means of inspecting and certifying large-diameter synthetic ropes for hawsers at deepwater ports.

6.4 FIELD INSPECTION OF USED ROPE

SPM hawsers should be examined frequently while in service to detect deterioration such as cuts, abrasion, fusing, snagging, or normal progressive wear of strands. Through such field inspections the rope user may be able to estimate the approximate residual strength in the used rope and determine if the rope should remain in service or be discarded. Unfortunately, there are no known inspection techniques or test procedures which accurately determine the residual strength of used ropes.

6.4.1 Preparation For Field Inspection

The principal causes of reduction in SPM hawser strength are external wear and damage. An experienced inspector will be able to detect signs of wear and damage and may be able to assess their effects on rope strength. The qualifications and experience of the field inspector are important. The inspector should be a technically oriented person knowledgeable in the basic principles of rope design. The inspector should be familiar with the construction and appearance of SPM ropes and splices when new. In addition, he should be familiar with the nature and appearance of the types of damage which can degrade rope strength, such as abrasion, cut or pulled strands, friction burns, damaged splices, and effects of chemical degradation.

Discussions with the manufacturer of the rope will help in determining the manner in which defects should be assessed and the methods of estimating the amount of strength loss due to defects. Formulas may be developed relating the number of damaged strands or yarns to the loss of hawser strength. The field inspector should also be aware of the manufacturer's rated strength of the rope being inspected and either the design factor of safety for the particular rope application or the residual strength at which the rope should be retired.

6.4.2 Field Inspection Items

The following general rules of thumb for the strength of used ropes have been suggested by rope manufacturers.

The outer yarns on three-strand, eight-strand, and nine-strand ropes constitute about 10% of the structure of each strand, and the failure of any one strand would amount to failure of the rope. Therefore, if a substantial proportion of the outer yarns on any one strand are damaged, for example by fusion or abrasion, then the rope should be rated at only about 90% of its original strength.

Typically, a large double-braid SPM hawser consists of an outer cover made up of 64 strands and a core of 24 to 36 strands. The structure of double-braid rope is discussed in subsection 5.3.4. If it is assumed that the rope cover is designed to carry about 50% of the load in the rope, then the effect of cover strand wear on the strength of the rope can be estimated by comparing the remaining intact bulk of the damaged cover with the bulk of an undamaged section of the rope. As a general guideline, rope manufacturers recommend that if the bulk of the surface strands of the cover of a double-braided rope has been reduced by 50% or more for a distance along the axis of the rope of four or more rope diameters then the rope should be replaced.

The following guidelines for the field inspection of <u>double-braided ropes</u> are presented based on discussions with personnel experienced with the construction of this type of rope:

<u>Cut or Pulled Strands</u> - Based on the previous discussion concerning number of strands in the cover and portion of load carried by the cover, each cut or pulled strand in the cover would, in principle, result in about a 2% rope strength loss. If one cut strand reduces the rope breaking strength by 2%, then for example, 3 cut strands would theoretically reduce the rope strength by 6%. This would be true if the cut strands in the rope were at random. However, because abrasion surfaces are normally localized, cut strands are usually adjacent to each other. In this case strength loss is progressive and other guidelines are needed. As a general rule, for double-braided ropes 40 mm diameter (5 inch circumference) and larger, eight or more adjacent cut strands indicate severe damage and the rope should be discarded.

Pulled strands represent a potential hazard for snagging some foreign object during rope operation. A pulled strand may be re-incorporated back into the rope by hand working the loop back through the intersections of the braided surface. In the event that it is not possible to do this, the loop formed by the pulled strand should be cut and the strand ends buried. Seizing the rope at this point would provide added protection. If there should be eight or more adjacent pulled strands that cannot be re-incorporated back into the rope, then the rope should be replaced.

Core Abrasion - The core contributes approximately 50 percent of the strength of a new double-braid rope and retains most of the strength in service even if the cover is severly worn or damaged. Normally the core is protected against wear and damage by the cover. However, the core should be spot-checked periodically by parting the cover strands. If the core shows visible damage then the rope should be replaced. With ropes having a nylon cover, heavy use and shrinkage may make the cover strands impossible to separate in order to inspect the core. This condition would indicate a rope that has probably been subjected to considerable loading and it would be wise to consider replacing the hawser.

Internal wear of the core caused by repeated flexing of the rope, particularly when wet, and by particles of grit worked into the core may be indicated by the presence of powdered fibers. Such conditions indicate the rope should be replaced.

Core Failure - Core failure of a double-braided rope can be detected by visual examination of the rope under moderate load. Any section of the rope that necks-down (diameter is reduced) under this load condition should be suspected of a core failure. A more detailed inspection of the core should be carried out for any suspected areas and the rope replaced if a core failure is detected. This type of inspection is quite easy for polypropylene ropes which do not require any external floatation devices. However, for nylon and polyester ropes, an in-service visual inspection for core failure will be difficult to perform due to the floatation devices covering all or a portion of the rope.

The following inspection guidelines would be applicable to all types of rope construction:

Friction Burns - Friction of synthetic ropes under high tension or rendering over bits and capstans can generate heat sufficient to melt or fuse together the outer fibers. This type of fiber melting can lead to serious degradation of the rope. Normally the melted or fused section will be axial to the rope and could affect the strength of the entire cover. If burns or melting are visible for a length of four or more rope diameters, the rope should be discarded.

Visual inspection can readily detect melting or fusion. However, caution must be taken because there is no fool-proof method of detecting a fiber or strand that may have been weakened from high temperatures. These weakened fibers or strands will have the same appearance as any other fibers or strands. This type of damage represents an extremely difficult item to inspect. If there is any question that the rope has undergone excessive heating, then it should be replaced.

Eyes and Splices - Each eye and splice should be checked during field inspection to assure that excessive abrasion in and beyond the thimbles has not taken place. The condition of the chafe protection provided in the thimble and eye area of the rope will be a good indicator of the amount of abrasion being experienced in that part of the hawser. In addition, on double-braid rope the splice should be checked to insure that the sewing or seizing has not come loose and that the splice is not pulling out.

Mooring Hardware - The field inspection should also include the chains and other hardware items covered in subsections 5.8 and 5.9. The condition of the floats is generally a good indicator of areas where rope abrasion and damage may have taken place. If the floats have been badly chafed or damaged then the rope in those areas should be carefully inspected for similar type damage. If the floats are in good condition, then there is a high probability that the rope in those areas has not been subjected to severe abrasion or cuts.

6.4.3 Frequency and Method Of Field Inspection

The interval between detailed SPM hawser inspections should depend on the frequency of tanker moorings, the severity of service, and the manner in which the ropes are handled between moorings. A terminal such as a U.S. deepwater port where tankers would be moored very frequently, for example several times a week, would experience a high utilization of the mooring hawsers. At ARAMCO, where the hawsers are in almost continuous service, the following inspection frequency and procedures are being successfully implemented:

- The hawsers are patroled and a visual inspection is made with special attention given to critical locations around the splices and thimble areas prior to each mooring and again while the tanker is moored.

- Once each month the hawsers are inspected in more detail by bringing the over the deck of the SPM service vessel. Here a detailed visual inspection of the rope, floatation equipment, and associated hawser fittings can be carried out. Any signs of hawser chafe, abrasion, cuts, or other damages are noted and if severe enough, the rope is retired from service. Generally, the hawser inspection can be carried out without removing the flotation material. Floats are usually slid around to expose that portion of rope where inspection is desired.

At other locations where the terminal is used infrequently, for example once a month, it may be appropriate to carry out a detailed inspection prior to each mooring at the SPM. This would be especially true if the mooring hawsers were left on the buoy between vessel moorings and the environment between moorings is relatively severe. The action of waves and current and collision with drifting objects or passing boats may cause chafing, abrasion or cuts of the hawsers while left unattended at the SPM. At some locations where frequency of vessel moorings is low or excessive rope wear between moorings has been experienced, the hawsers are removed from the SPM and towed to a protected location or brought on-shore between moorings. This practice minimizes the possibility of hawser damage between tanker moorings and also facilitates easier inspection of the ropes between moorings.

6.5 ROPE REPLACEMENT CRITERIA

At present there are no universal rules or guidelines used by SPM operators to determine when to replace bow hawsers. Generally, hawser replacement criteria are established at each site based on operating experience and the results of field inspections of used ropes.

As discussed in subsection 5.8.6, both ropes of a dual-hawser SPM system should be changed out at the same time. This should be done even if one rope appears to be in reasonably good condition. Should only one of the ropes be replaced, the two hawsers (one new and one used) will have significantly different load/elongation characteristics. This will result in a disproportionate share of the load being taken by the older or stiffer rope and could lead to a premature hawser failure.

The principal methods now used for determining when SPM hawsers need replacement are: fixed replacement interval, rope elongation criteria, and replacement based on number of tankers moored.

6.5.1 Fixed Replacement Interval

At most SPMs the hawsers are inspected on a regular basis in a manner similar to that described in subsection 6.4. If the inspection uncovers potential weak points in the ropes, the hawsers are retired and replaced with new ones. The interval between inspections depends on site and operating conditions.

Some terminals keep hawsers in service until damage or deterioration is detected during a periodic inspection. Hawsers have been known to remain in service for several years without severe damage or deterioration.

Most terminals, however, retire SPM hawsers after a set period of time in service regardless of their apparent condition. This fixed replacement interval varies from four months to eighteen months or more depending on the frequency of hawser use and severity of environmental conditons at the site. ARAMCO is currently carrying out a test program on used hawsers to better define the optimum fixed replacement interval at their facilities. This test program is discussed further in subsection 6.5.4.

6.5.2 Rope Elongation Criteria

Some SPM users have suggested using an elongation criteria as a basis for hawser retirement. By such a criteria the length of the hawser would periodically be measured, and when a defined limit of permanent elongation was reached, for example 15% permanent elongation, the hawser would be retired.

Unfortunately, this proposal has several drawbacks. Certain synthetic rope materials, such as nylon, undergo shrinkage when wet and remain shortened until loaded again. In addition, synthetic ropes retain some elongation after loading but with time return to a shorter length. These factors will mask the true permanent elongation of the used rope. Another problem will be that of obtaining an accurate measurement of rope length in the field. The rope must be detached from the SPM, brought to shore and laid out on a test bed. A slight pretension or reference load, explained in subsection 6.3.1, must be applied to the rope to obtain a length measurement comparable to the new-rope length. Humidity and temperature may also have to be controlled, to obtain a comparable measurement.

6.5.3 Replacement Based on Number of Tankers Moored

Another method sometimes used as a criteria for hawser replacement is to record the number of tankers which have moored, or the number of hours that a tanker has been moored to the buoy. Hawser service life is limited to a certain number of vessel moorings or a certain total duration of mooring hours. The obvious problem with these types of hawser replacement schemes is they fail to account for tanker size differences, frequency of calls, and environmental conditions during the mooring. All of these items play an important part in the loading history of the mooring line and the resultant service life of the rope.

At some locations, hawser replacement criteria may be a combination of the above mentioned methods. At other locations, no established rules are followed and hawser replacement is based completely on periodic field inspections.

6.5.4 Operating Experience

During 1975, the Rope Sub-Committee of the Oil Companies International Marine Forum undertook a survey of SPM users to determine frequency of rope failures. Twenty-four rope failure reports were received as a result of this survey. Over 85% of the failures reported occurred during the first 12 months of rope use. Over 50% of the reported failures occurred in the rope splice or thimble areas. Although sufficient detail was not given, these failures were most probably a result of abrasion or chafe of the ropes in the thimble area. Discussions with SPM operators and rope manufacturers have substantiated that chafing or abrasion at thimble areas is the major cause of rope deterioration and subsequent failures.

Of the SPM users surveyed, 8% of the failure reports were for ropes retired from service without a true rope "failure". The reports indicated that excessive abrasion was the reason for replacement.

ARAMCO is presently replacing hawsers at their SPMs after six months of service if they have not been retired earlier because of damage. During the first year of operation approximately 25% of the mooring lines were retired before six months of service. Most of these were retired because of chafing damage. Chafing on the thimble and the ship's anchor flukes were the most common cause of this damage. Approximately 5% of the lines were damaged by the work boat, principally by the propeller. Improved design and operational practices have now almost eliminated these causes of damage. ARAMCO retired only one mooring line due to damage before 6 months of service during 1976.

ARAMCO is currently carrying out a test program at the National Engineering Laboratory in Scotland to access residual strengths of bow hawsers after various in-service periods. (Flory, April, 1977). Tests are planned on several hawser specimens that have seen service of 2, 4, 5 and 6 months respectively. In this manner, it is hoped that a relationship between service life and residual hawser strength can be developed. Based on these tests, and perhaps tests on ropes which have been in service for longer periods of time, ARAMCO will be able to better evaluate their present hawser replacement criteria and be able to establish better guidelines for field inspection of SPM ropes.

6.6 SUMMARY

The areas of testing, inspection, and replacement criteria for large-diameter SPM hawsers require additional research and standardization efforts. Extrapolating test results and operating experience on smaller ropes to predict performance characteristics of large ropes will always leave questions unanswered. The only sure way to develop guidelines and standards for the use and replacement of large synthetic ropes is to carry out appropriate testing and research on typical sizes and types of ropes.

Testing equipment capable of performing proper tests on new and used SPM hawsers is just now being developed. Test programs, such as ARAMCO's, on hawsers with different service-lives will help to develop replacement criteria for SPM hawsers.

Until these test programs have been completed, continued close field inspection and conservative rope design, operating, and retirement criteria should be employed.

TABLE 6-1

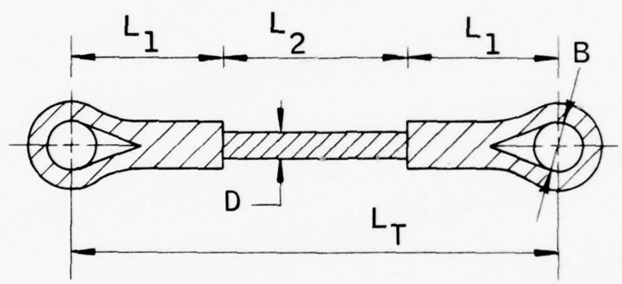
ROPE TESTING SPECIFICATIONS

Title	Number	Issuing Agency
Textile Test Method	FED-STD-191	U.S. Army Research and Development Command
Standard Test Methods For Stranded and Plaited Ropes	April 4, 1974 issue	Cordage Institute 2300 Calvert St. Washington, D.C.
Methods of Sampling and Testing Cordage	40-GP-1	Canadian Government Specifications Board Department of Supply & Services Ottawa, Canada
Methods of Test For Cordage and Allied Articles	BS-5053	British Standards Institute 2 Park Place London, England
Ropes-Determination of Certain Physical and Mechanical Properties	ISO-2307	International Organization for Standardization

TABLE 6-2

U.S. MILITARY PURCHASE SPECIFICATIONS FOR ROPE

Specification Name	Identification Number
• Rope Polypropylene	MIL-R-24049A
• Rope, Fibrous, Polyester/Polypropylene Dual Fiber	MIL-R-43942
• Rope, Nylon	MIL-R-17343D
• Rope, Nylon (Spun Yarn)	MIL-R-43161B
• Rope, Nylon, Double-braided	MIL-R-24050B
• Rope, Nylon, Plaited	MIL-R-24337
• Rope, Polyester	MIL-R-30500B
• Rope, Polyester, Fiber	MIL-R-24335



B = Pin diameter

D = Rope diameter

L_1 = Length of eye plus eye splice

L_2 = Length of pure rope

L_T = Length of total rope specimen

Figure 6-1 - EYE-SPLICED TEST PIECE BETWEEN TWO PINS

SECTION 7

TOPICS FOR FURTHER INVESTIGATION

7.1 INTRODUCTION

This study has been the first comprehensive investigation of the manner in which mooring loads for SPMs are determined and the manner in which the SPM is designed to withstand those loads. The time and resources available for the study did not permit an in-depth investigation of all pertinent topics. As topics were studied and data were gathered, it became evident that some additional investigation would be required.

The principal topics which require further investigation pertain to the engineering properties of large-diameter synthetic ropes of the type used on SPM hawsers, and the rate and manner in which the strength of these hawsers degrades in service. The data obtained on these topics during this study was sufficient to establish general guidelines for engineering properties and to recommend factors of safety which are believed to be conservative. Nevertheless, further data on these topics is needed to backup, or, if necessary, to modify the guidelines and recommendations.

Four topics for further research are recommended as follows:

- Establish Test Procedures for Large-Diameter Synthetic Ropes
- Determine Large-Diameter Synthetic Rope Properties
- Develop Non-destructive Means and Practices for Determining New-Rope Strength
- Develop Means and Practices for Determining Used-Rope Strength

The Coast Guard should consider initiating studies in these areas. More detailed discussions of each of these topics are given in the following subsections.

7.2 ESTABLISH TEST PROCEDURES FOR LARGE-DIAMETER SYNTHETIC ROPES

As pointed out in subsection 6.2.3, there are no testing specifications now available which specifically and adequately apply to large-size (generally larger than 96 mm diameter, 12 in. circumference) synthetic ropes. Several agencies are now in the process of updating their specifications to include these very large ropes. However, it cannot be foreseen how soon these specifications will be available or how adequate and applicable these specifications will be to SPM hawsers.

Specifications are needed to cover the following general categories of large synthetic SPM hawser testing.

- Tensile testing of new ropes
- Cyclic loading of new ropes
- Abrasion testing of new ropes
- Tensile testing of used ropes

For tensile testing and cyclic loading, the specifications should, as a minimum, cover the following parameters:

- Specimen length
- End-connection details
- Type and quality of rope splices
- Rate of specimen loading

Specifications for cyclic loading should include the manner in which data is presented. The means of selecting and preparing specimens of used rope should be specified. A method of abrasion testing should be specified which adequately simulates the types of abrasion which are experienced by SPM hawsers in service.

The Coast Guard should acquaint itself with the efforts of the several agencies which are now preparing specifications for testing large ropes. If necessary and feasible, the Coast Guard should become involved in and support the efforts of one or several of these agencies to assure the specifications adequately cover the needs of testing SPM hawsers. If it appears participation with these agencies will not lead to timely or adeqaute SPM-hawser testing specifications, then the Coast Guard should sponsor the development of such specifications.

7.3 DETERMINE LARGE-DIAMETER ROPE PROPERTIES

One objective of the present study was to gather and assess the data available in publications and from rope manufacturers on strengths and other characteristics of large-diameter synthetic ropes of the types which are used in SPM hawsers. Of particular interest are data on the statistical variation of breaking strength, the load-elongation characteristics of new and used rope, and the effects of cyclic loading, abrasion, cuts, and other types of damage on breaking strength. We had hoped that through a careful analysis of such data more complete recommendations and guidelines for field inspection and hawser replacement criteria could be developed. Only a limited amount of such data was obtained. Direct comparison of data from different sources, or even from the same source, was difficult because of known or apparent discrepancies in the materials or rope constructions, the methods by which the ropes were tested, or the manners in which the data were analyzed and reported.

There is a need, therefore, for a well-planned and coordinated program of testing large-diameter synthetic ropes of the types commonly used on SPM hawsers. The following are the principal types of testing which should be conducted:

- Comparison of break tests for various nylon grades and types,
- Determination of statistical variations of breaking loads,
- Comparison of cyclic-loading strength degradation for various nylon grades and types,
- Determination of break-load strength retention as a function of cyclic-loading history, and
- Determination of new and used load-elongation properties of ropes of various materials and constructions.

Much of the testing could be done with relatively small rope, for example approximately 40 mm diameter (5 in. circumference). This testing could be conducted relatively inexpensively on most existing test machines. Some of the testing, particularly the cyclic loading testing, should be partially conducted or repeated with large ropes to verify the influence of scale. Such testing would be relatively expensive.

It is recommended the Coast Guard contract for a comprehensive test program to quantitatively determine the engineering properties of very-large synthetic ropes. In preparation for such a test program, the methods of testing must be defined, as detailed in the preceeding subsection. A more accurate knowledge of the engineering properties of large-diameter synthetic ropes may permit decreasing or possibly require increasing the factors of safety recommended in this report.

7.4 DEVELOP NON-DESTRUCTIVE MEANS OF DETERMINING NEW-ROPE STRENGTH

A reliable means of determining the breaking strength of new large-diameter synthetic rope by non-destructive testing as part of in-plant inspection is needed. The method must not reduce the strength of the rope or require breaking a sample length of the rope.

The large ropes used in SPM hawsers are very expensive, costing roughly $175 per meter ($50 per foot) for a typical 168 mm diameter (21 in. circumference) nylon double-braid rope. Only approximately 100 m (330 ft) of rope may typically be ordered at a time. A specimen of such rope for break testing prepared with eye splices would cost approximately $4,000. Machines are generally not available for break testing such large ropes, and the cost of testing would be high. Thus, it is impractical to require a destructive break test of such a large-size rope for each SPM hawser order.

The realization factor method of determining the strength of a synthetic rope from break tests conducted on sample components of the rope, such as strands or yarns, is described in subsection 5.4.3. This method is now employed in England for certifying the strength of large-size synthetic ropes. However, U.S. rope manufacturers question the reliability of the realization factor method of determining breaking strengths.

The Coast Guard should sponsor an investigation of methods of non-destructively determining the strengths of new large-diameter synthetic ropes. The study should include an investigation of the realization factor method, and also seek out and investigate alternative methods of determining new rope strength. As a result of such a study, a method of certifying the new breaking strengths of hawsers for deepwater ports would be developed.

7.5 DEVELOP MEANS AND PRACTICES FOR DETERMINING USED-ROPE STRENGTH

There is also a need for a method of quickly, easily, and reliably determining the strength of used rope through inspection and non-destructive testing in the field. Routine determination of the remaining strength of a used rope by break testing to determine if it is fit for further service is impractical. Not only are test machines with sufficient capacity, bed-length, and stroke not commonly available, but broken rope would be unfit for further service.

Much data which could lead to a field inspection and testing method would be gained from a program of testing hawsers which have been retired from service. Facilities such as ARAMCO's Ju'aymah deepwater port would probably be willing to furnish used hawsers at no cost. The new Coordinated Equipment high-capacity test machine will probably be capable of performing such tests.

To obtain meaningful data, careful preparations must be made for testing. The service history of the hawser should be documented. Preferably the hawser should come from an SPM equipped with load-monitoring and recording equipment, and the complete loading record should be available for analysis. Prior to testing, a detailed inspection of the hawser should be conducted. Any indication of significant damage or defects should be noted and measured as to its extent and its position on the hawser.

Photographs of such damage and defects should be taken. A load-elongation record should be made as the hawser is loaded. After breaking the hawser, the location and nature of the break should be noted and correlated, if possible, with the damages and defects noted during the inspection.

Through analysis of the results of a series of such tests, hopefully some pattern of correlation between types and extents of damage and defects with the remaining strength of the rope can be made. Possibly an album of pictures of typical defects and damage and the resulting strength reductions can be prepared for use in field inspections. Possibly some observation or discovery can be made which will lead to the development of a non-destructive field-testing method. Also, some relationship may be established between the service history or, if available, the loading record and the strength of the used hawser.

The Coast Guard should investigate the feasibility of conducting such a series of tests. Two important requirements are the availability of a sufficient number of suitable used hawsers and the availability of a long-stroke high-capacity test machine. A number of facilities throughout the world would probably be willing to furnish specimens. The machine now being installed at Coordinated Equipment Co. will probably be suitable for conducting the testing.

APPENDIX A

WAVE DESCRIPTION CONCEPTS

This Appendix is intended as a primer on ocean waves and the methods by which they are described. Ocean waves are very complex. Winds acting on the ocean surface produce a system of wave trains of different heights and lengths propagating in various directions. The result of this simultaneous superposition of waves of many shapes and energy contents is a very irregular appearance of the sea surface, which is difficult to describe mathematically. Therefore, various efforts to simplify the description of ocean waves as they occur in nature have been undertaken.

The first concept to be introduced in modern times is that of the significant wave height. Following this, it was recognized that the only way to deal with the complex sea surface, as it occurs in nature, is by statistics or statistical summaries. A number of studies were undertaken in the decade folowing the introduction of the significant wave height for the purpose of characterizing the sea surface by statistical correlations. The next major development in describing the irregular sea surface was the introduction of various spectral concepts. At the present time, attention is being focused on certain aspects of long-period components of the wave spectrum.

The following discussion is a summary of these concepts, statistical distributions, and matters of current interest.

A.1 Significant Wave Concept

The concept of the significant wave was introduced by Sverdrup and Munk (1947)*, but according to Kinsman (1965) the work was actually completed in 1943, being initially classified. The forecasting aspects associated with the significant wave concept have been revised many times, most notably by C. L. Bretschneider. Readers should consult the latest references for information on this subject. Our concern here is a brief review of the significant wave height as a means of describing the irregular sea surface.

What is the "significant wave height"? The significant wave height is not an identifiable wave form which propagates like a physical wave as predicted by classical wave theories. The significant wave height and its associated parameter the significant period are statistical properties defined as follows:

H_s = The average height of the highest one-third of the waves on a wave record (significant wave height).

T_s = The average period of the highest one-third of the waves on a wave record (significant period).

The symbols H_s and T_s will be used throughout this report. The symbols $H_{1/3}$ and $T_{1/3}$ are frequently used in other papers on ocean waves.

* From personal communication with R. L. Wiegel, March 17, 1977, M. P. O'Brien should also be given credit for introducing this concept.

The significant wave height is a particularly important value for a number of reasons. An observer tends to neglect the small waves and notice only the larger waves when visually estimating wave heights. The significant height has generally been found to correspond to wave height estimates based on visual observations. The practice of reporting sea conditions on a significant height basis is widespread, and most observed and recorded wave data is presented in the form of significant wave heights. Also, the statistical distribution of wave heights and most energy-spectrum techniques have been related to the significant height. In fact, the significant height carries the major portion of energy of the spectrum. Finally, the effects that irregular seas have on many types of fixed and floating objects and on various shore processes have been related to significant wave heights, at least with sufficient accuracy for many engineering applications.

In summary, the irregular sea can be described by two terms, the significant wave height and the significant period. However, designers sometimes have need to know other statistical features of the sea, such as the maximum wave or maximum water surface elevation. This need led to a number of studies which attempted to establish the statistical correlations for real ocean waves. A brief summary of these correlations is presented in the next subsection.

A.2 Statistical Distributions

The examination of wave and swell records by numerous investigators shows that the height of waves follows a certain statistical distribution which conforms closely to the random-walk distribution studied by Raleigh (1880). This is a probability distribution of the positive skew type, shown in Figure A-1. The theoretical curve has been found by Putz (1950) and Barber (1950) to conform closely to the scatter of real ocean wave data. The area under the curve represents the total number of waves present in any record of suitable length.

The abscissa through the centroid of the total area defines the average wave height, H_o. The ordinate through the centroid defines the probability of occurrence of the average wave, is less than the maximum ordinate which determines the probability of occurrence of the most frequent or most probable wave. Stated another way, the wave height of the most probable wave is less than the average wave height.

Figure A-1 illustrates these concepts. On the basis of this distribtuion, the average wave height (H_o), the significant wave height (H_s), and the 1/10 highest wave height ($H_{1/10}$) may be approximated as follows:

$$H_o = 0.89 \sqrt{\overline{H^2}} \quad \text{(A-1)}$$

$$H_s = 1.41 \sqrt{\overline{H^2}} \quad \text{(A-2)}$$

$$H_{1/10} = 1.80 \sqrt{\overline{H^2}} \quad \text{(A-3)}$$

where $\overline{H^2}$ is the mean squared wave height or the average of all the squared values of wave heights in a wave record.

From these equations it follows that:

$$H_o = 0.625\ H_s \qquad \text{(A-4)}$$

$$H_{1/10} = 1.27\ H_s \qquad \text{(A-5)}$$

As indicated in the preceeding section, the maximum wave height is a criteria of importance, but because of its infrequent occurrence, it is difficult to recognize and to measure. Its determination is more satisfactorily accomplished on the statistical basis of its relationship to the height of the average wave, H_o, or the height of the significant wave, H_s.

A few selected results of wave height statistical correlations are given in Table A-1.

From these data, it is seen that the most probable maximum wave height, H_{max}, can range between 1.5 and 1.9 times the significant wave height, H_s. It is also known that the maximum wave height will depend on the duration of the storm or length of the wave record. The following relationship may be used to approximate this maximum wave height.

$$H_{max} = 0.707\ H_s\ \text{Ln}\ N \qquad \text{(A-6)}$$

where N is the number of waves in the record. When N is not known, a practical assumption for this maximum wave height is

$$H_{max} = 1.77\ H_s \qquad \text{(A-7)}$$

Equations A-6 and A-7 should be used with caution for very severe storm waves or for waves in very shallow water near the breaking point.

Various investigators do not agree as well on wave-period distribution as they do on wave-height distribution. Generally, however, studies have shown that the significant period, T_s, is about equal to the average wave period, $\overline{T}$, for typical irregular sea conditions where the average period ranges from four to ten seconds. For longer period waves, the average period can be as little as 75 percent of the significant period. The International Ship Structures Congress generally concludes that the average period may be taken as 90 percent of the significant period. Mean apparent period, T, is a term occassionally used and it is considered generally to be equal to the average period.

A.3 Wave Spectrum Concepts

A major advance in the description of irregular ocean surface waves was proposed by Pierson (1952), who merged certain concepts from classical mechanics and the theory of stochastic processes with the energy or power spectrum in order to predict the behavior of ocean waves. For a complete description of the development and use of the power spectrum as applied to ocean waves, the reader should consult the work by Kinsman (1965).

In its simplest form, the energy spectrum allocates the amount of energy of the sea surface according to frequency. For small amplitude sinusoidal waves of the form

$$\eta(t) = A \cos (kx - \omega t),$$

where $\eta(t)$ = the fluctuating water-surface elevation

A = amplitude
k = wave number
ω = wave frequency
x = dummy variables in the space and time dimensions

the mean total energy per unit surface area is

$$E = 1/2 \rho g A^2 = 1/8 \rho g H^2 \qquad \text{(A-8)}$$

where H = the wave height measured from crest to trough.
ρ = density of water
g = acceleration of gravity

One of the fundamental premises of the spectral approach is that irregular waves are the result of the superposition of an infinite number of simple sine waves of small amplitudes and having a continuous frequency distribution. This process can be approximated with a finite number of small-amplitude sine waves having discrete frequencies. Under these conditions, the mean total energy per unit surface area is given by

$$E = \frac{\rho g}{8} (H_1^2 + H_2^2 + H_3^2 + \ \ldots + H_n^2 + \ldots) \qquad \text{(A-9)}$$

where H_n is the wave height associated with a given discrete frequency.

For purposes of illustration, Figure A-2 is presented which shows the contribution of energy according to frequency from the individual component waves. The reader's attention is directed to the note accompanying this figure. Indications of height are intended merely to point out that the spectral energy density for a specified frequency is the result of an identifiable wave (or waves) having that frequency. Wave records can be synthecized from the superposition of a finite number of discrete wave forms. Amplitude spectra obtained from such synthetic records are indistinguishable from amplitude spectra derived from real ocean wave records. In many cases, such synthetic records are adequate for engineering purposes.

The determination of spectra from wave records depends crucially upon certain details of the analytic procedure, among the most important of which are; length of record, sampling interval, degree and type of filtering and smoothing, and length of autocovariance function if the autocovariance method is utilized. In general, some compromise between stability, confidence, resolution, and practical limitations of available computer time and budget must be achieved. Additional details on the various analytic procedures are available from the appropriate references.

In order to make the spectral expressions useful for practical purposes, the mean wind must be taken into account. Although there are some minor differences in the coefficients as presented by various investigators, the general form of the wave energy spectrum is

$$S_{H^2}(\omega) = A\omega^{-m} e^{-B\omega^{-n}} \qquad \text{(A-10)}$$

where coefficients A, B, m and n define the spectrum. The most widely used energy spectrum formulations take m = 5 and n = 4. The coefficient A is proportional to the square of the significant wave height (H_s^2) and inversely proportional to the fourth power of the significant or average wave period (T_s^4 or T_o^4). The coefficient B is inversely proportional to the fourth power of the significant wave period (T_s^4). The energy contained in the spectrum is the area under the curve.

A.4 Describing the Energy Spectrum

There are significant differences between the various spectrum formulas which appear in the open literature. These differences, which are not always obvious, include terminology, symbolic notation, and other spectral descriptions. The suggested descriptions for identification of the different spectral density ordinates, as shown in Figure A-3, are due to Michel (1967).

With regard to the abscissa of the energy spectrum; cyclic frequency (f=1/T), angular circular frequency ($\omega = 2\pi/T$), and period (T) have been used. Period (T) when used appears usually with one of the above frequencies, and of the two types of frequency, circular frequency has the more common usage.

The ordinate of the energy spectrum, generally called spectral density, is presented in various terminologies and designated by various symbolic notations. The average energy of the wave system is related to the sum of the square of the wave component heights or amplitudes (area under the energy spectrum) and is also related to the average of all the squared values of wave-component heights (H^2) or amplitudes in an irregular ocean wave record. Michael (1967) has suggested the following terminology for identifying the various spectral density ordinates, the corresponding spectra, and relation to the significant wave height.

1. Amplitude Spectrum. This is the spectral form which was first presented by Pierson (1952) and which appears in Pierson, Neumann and James (1955). The spectral density is a function of the square of the wave amplitude, and the significant wave height is related to the area (in this case twice the variance) under the spectral curve by the following relation:

$$H_s = 2.83\sqrt{2 \text{ x Variance}} = 2.83\sqrt{(\text{Area})_1} \qquad \text{(A-11)}$$

2. Amplitude Half Spectrum. Some investigators found it more convenient to work with the variance directly rather than to take twice the variance. When this is done, the spectral density is related to one-half of the square of the wave amplitudes and the significant wave height is related to the area (in this case the variance) under the spectral curve by the following:

$$H_s = 2.83\sqrt{2 \text{ x Variance}} = 4\sqrt{\text{Variance}}$$
$$H_s = 4\sqrt{(\text{Area})_2} \qquad \text{(A-12)}$$

Obviously, in comparing equations A-11 and A-12, $(\text{Area})_2$ = Variance where as $(\text{Area})_1$ = 2 x Variance.

3. Height Spectrum. Other investigators have found it even more convenient to work with heights. Under these conditions, the area under the spectral function is four times the area obtained when amplitudes are used, since height is twice the amplitude and $(\text{Height})^2 = 4 \text{ x } (\text{Amplitude})^2$. Therefore, the spectral density is a function of the square of the wave height, and the significant wave height relation to the area under the spectral curve is obtained as follows:

$$H_s = 2.83\sqrt{2 \text{ x (Variance)}}$$
$$H_s = 2.83\sqrt{(\text{Area})_1}$$
$$H_s = 1.414\sqrt{4 \text{ x } (\text{Area})_1}$$
$$H_s = 1.414\sqrt{(\text{Area})_3} \qquad \text{(A-13)}$$

4. Height Double Spectrum. Finally, some investigators found that, by taking twice the $(\text{height})^2$ rather than the $(\text{amplitude})^2$, the constant relating significant wave height and the square root of the area under the resulting spectrum could be made equal to unity. Accordingly, the area is eight times that given when amplitudes are used, and the significant height relation is as follows:

$$H_s = 2.83\sqrt{(\text{Area})_1}$$

$$H_s = \sqrt{8 \times (\text{Area})_1}$$

$$H_s = \sqrt{(\text{Area})_4} \qquad \text{(A-14)}$$

There are a number of precautions to be observed in working with wave energy spectrum. First, the units of the spectral density ordinates (i.e., English units of ft^2-sec or Metric units of m^2-sec and of the values of frequency on the abscissa (i.e., cyclic (f) or circular (ω)) should be established. Second, determine the specific formulation appropriate to the ordinate values. Sometimes this can be done by examining the symbolic notation; otherwise the Raleigh distribution factor used in conjunction with the area to determine significant wave height (or wave amplitude) should be inspected. Finally, the reader can readjust the spectral formulation into a form most convenient for his purposes.

A.5 Comparison of Wave Energy Spectra

There are a number of different wave-energy spectra formulations. These wave-spectra formulations derive from their corresponding wave generation relationships. There are two general methods in use: the wave-spectrum method and the significant-wave method. Although the topic of wave generation is beyond the scope of this Appendix, a minimum amount of background information is necessary. Although both methods are based on certain theoretical concepts, both require sufficient data for calibration purposes, and therefore, may be termed semi-empirical or curve fitting approaches. When sufficient data is available for calibration for both methods, then both methods yield generally accurate estimates of the wave parameter.

Wave-spectra formulations that are derived from the wave-spectrum method are termed wind-speed spectra since wind speed is directly related to the spectral density values. Those that derive from the significant-wave method are termed height/period spectra since they use significant height and significant or mean apparent period to derive the spectral density ordinates.

Again, it is to be emphasized that both methods utilize the distribution functions derived theoretically by Longuet-Higgins (1952) which is supported by the empirical relations derived from an analysis of ocean wave data by Putz (1952). Also, both methods are based on actual wave data, although the data was not the same for both methods, and the methods of analysis of the data were somewhat different.

All of the spectral formulations for both categories have the same general form:

$$2h^2(\omega) = A\omega^{-m} e^{-B\omega^{-n}} \qquad \text{(A-15)}$$

where the coefficients A, B, m and n define the spectrum and govern the total energy (area under the spectrum) and the distribution of energy at various frequencies.

In the following subsections, some of the more popular wave spectra are compared. These spectra will be discussed in terms of the height-double spectrum with a base of circular frequency (ω). All spectra will be in English units and will use the notation $2h^2(\omega)$ for the ordinate of spectral density and H_s^2 for the area under the spectrum curve.

A.6 Wind-Speed Energy Spectra

One of the fundamental concepts governing the wave spectrum formulations is that waves generate from the high frequency end of the spectrum due to a transfer of energy from the wind field to the wave system. With this premise, Neumann (1952) developed the co-cumulative power spectra from which the variance could be predicted. From the latter, the generated wave energy can be determined. In essence, the wave-spectrum formulations permit predictions of the wave spectrum directly from a knowledge of the wind field. With the wave spectrum, the significant height as well as the statistical distribution of waves can be determined.

Therefore, in order to predict the spectrum, it is necessary to know the history of the wind field, or to assume that a steady wind of a specified magnitude has been acting on a particular fetch of the ocean surface for a given time duration.

Three commonly used wind-speed spectra, with wind speed (V) expressed in knots are:

1. Neumann spectrum (as used in the Pierson Neumann James wave prediction technique):

$$2h^2(\omega) = 400\omega^{-6} e^{-(725/V^2)\omega^{-2}} \qquad \text{(A-16)}$$

$$H_s = 0.00436V^{2.5} \qquad \text{(A-17)}$$

2. Neumann spectrum, (as modified by Roll-Fischer and others):

$$2h^2(\omega) = 400\omega^{-5} e^{-(725/V^2)\omega^{-2}} \qquad \text{(A-18)}$$

$$H_s = 0.0195\ V^2 \qquad \text{(A-19)}$$

3. Pierson-Moskowitz spectrum

$$2h^2(\omega) = 135\omega^{-5} e^{-(97{,}000/V^4)\omega^{-4}} \qquad \text{(A-20)}$$

$$H_s = 0.0187\ V^2 \qquad \text{(A-21)}$$

A.7 Significant-Wave Method Formulations (Height/Period Energy Spectra)

In contrast to the wave spectra method, the significant wave method requires a knowledge of the significant wave height and period from which the spectrum can be determined. Estimates of the significant wave height and period are obtained from the wind speed, fetch length, and wind duration by reference to empirical data.

The significant-wave method does not require consideration of the history of wind speed, duration, and fetch length, and therefore, there is a belief that it is somewhat less laborious to apply than is the wave spectrum method. Establishing these wave generation parameters is the most subjective and difficult procedure in the art of wave forecasting. These wave properties are sometimes more readily determinable, at least to a suitable degree of accuracy, for a region under consideration than is a complete history of wind characteristics.

For practical applications, if a wind speed is given then any wave prediction technique desired may be chosen to determine fully developed or partially developed wave characteristics. The wave spectrum can then be determined from these wave characteristics. If sea conditions are given in terms of a significant height and period then the wave spectrum can be computed directly.

Three commonly used height/period spectra are:

4. Bretschneider spectrum (used in conjuntion with the Sverdrup-Munk-Bretschneider wave prediction technique):

$$2h^2(\omega) = (4200\ H_s^{\,2}/T_s^{\,4})\omega^{-5} e^{-(1050/T_s)\omega^{-4}} \qquad \text{(A-22)}$$

$$H_s = 0.025\ V^2 \text{ (for fully developed seas only)} \qquad \text{(A-23)}$$

5. ISSC spectrum (International Ship Structures Congress modification of the Bretschneider Spectrum):

$$2h^2(\omega) = (2760\ H_s^{\,2}/T_s^{\,4})\omega^{-5} e^{-(690/T_s^{\,4})\omega^{-4}} \tag{A-24}$$

$$H_s = 0.025\ V^2 \text{ (for fully developed seas only)} \tag{A-25}$$

6. Roll-Fischer spectrum as used by some model basins for fully developed seas (its spectrum curve is nearly the same as equation A-18):

$$2h^2(\omega) = (5050\ H_s^{\,2}/T_s^{\,4})\omega^{-5} e^{-(50.26/T^2)\omega^{-2}} \tag{A-26}$$

$$H_s = 0.0195\ V^2 \tag{A-27}$$

The Roll-Fischer height/period spectra, 6, is equivalent to the Roll-Fischer Neuman spectrum, 2, except the formulation is in a more convenient form. The above list of spectra is not meant to be complete, but only to include typical spectra encountered in model testing.

A.8 Which Spectrum Should Be Used

It should be remembered that these spectra, and other such spectra represented by formula, are mathematical descriptions of curves fit through a number of spectra derived from actual wave records which were measured in generally similar sites and environments. They do not represent point by point each measured spectra but only the mean of the points of the family of actual spectra. In other words, they are simply curve fitting formulas and in no sense can be considered as "theoretical spectra". Hoffman, who has compared many actual spectra with formula spectra, has stated the actual spectra height may show a sigma variation of 30% or more from applicable formula spectrum.* A spectrum derived from an actual wave record may fit any one of several formula spectra, depending on the experience of the investigator.

Michel (1967) has compared the six spectra described above (reproduced in Figure A-4) for the case of a typical fully developed 30 ft. (significant wave height) sea. Each of the spectra is associated with a particular wave prediction (hindcasting) technique which relates a given significant wave height and significant or mean period to a particular wind speed acting over some fetch distance for some length of time. To achieve a fully developed 30 ft. sea, the six spectra require wind speeds ranging from 34 to 40 knots.

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*From personal communication with D. Hoffman, April 27, 1977.

Comparing the wave spectra on the basis of equal wave heights instead of equivalent wave-generation parameters shows three of the six spectra generally similar in form and two others, Bretschneider and ISSC, different from the others. The comparison is made on the basis of fully developed seas. A Bretschneider spectrum for a partially developed sea is shown as 4A in Figure A-4. That spectrum is very close to the Pierson Moskowitz spectrum and close to the Neumann and Roll-Fischer spectra. Had the spectra been compared on the basis of equivalent wave generation parameters they would have illustrated pronounced differences primarily due to differences in significant wave height which would be manifest by different areas under the curve. The wave heights which would be predicted for [illegible]0 knot winds for each spectrum range from 30 ft. (for 1) to 40 ft. (for 4 and 5). It is to be emphasized that these differences originate with and are directly related to the wave prediction technique and are not due to the specific spectral formulations.

Three spectra are compared with a spectrum produced in a model basin in Figure A-5. The model basin spectrum has a lower wave height and lower mean period because the basin did not produce the high-frequency components of the spectrum, a common deficiency of model basin spectra. Note, however, that either the Roll-Fisher spectrum, 2/6, the Pierson Moskowitz spectrum, 3, or the Bretschneider spectrum, 4A, fit reasonably well with the model basin spectrum.

Within the limitations of present wave prediction technology, it is generally not justified to exert considerable effort in determining what formula to apply to the spectrum. Several different spectrum may adequately fit the waves common at the site, and may also serve reasonably well as prototypes for the spectrum to be modeled. If considerable wave-record data are available for a particular site, then this wave data and a spectrum developed from it will better serve for purposes of modeling and analysis than a formula spectrum predicted for the site. Therefore, within the present state-of-the-art, it is difficult to justify a highly sophisticated analysis for most projects. However, when the consequences of failure are so great as to truly constitute a potential disaster, then advanced approaches to the description of the waves may be necessary.

A.9 Second-Order Wave Phenomena

Attention is now being given to long-period wave phenomena and their possible effects on mooring systems. These long-period phenomena are generally referred to as seiche, swell, and wave grouping. These phenomena will not usually be evident in wave spectra, but are recognizable in long wave records. In a typical irregular-wave record, waves of many different frequencies are randomly superimposed to create a record such as the top record in Figure A-6.

The second record in the figure shows an irregular wave with a slowly varying mean. The slow variation in mean water level may be due to seiche, that is the movement of water back and forth across a basin, or to a very long-period swell. A complete spectral analysis of this record would show two peaks; one encompassing the band of the irregular waves, and another at the frequency of the slowly varying mean water level. Normally, the very long-period wave would be overlooked in a wave spectrum unless special attention is devoted to it in the analysis.

The third record in Figure A-6 shows an example of wave grouping in which several waves of slightly different period are superimposed. Wave grouping is characterized by successive occurrence of waves of high amplitude. Wave grouping has been found in some wave records, but it is not known how common it is in ocean waves. Several firms, including ER&E and NSMB, are now investigating certain aspects of wave grouping.

Because the variation in mean water level associated with wave grouping is negligible if present at all, there will be no power in a routine amplitude-spectral analysis at the period of wave grouping. A phase-spectral analysis, however, may indicate definite phase relationships between the component waves. A spectral analysis taken at very fine frequency-band widths may show two closely spaced peaks. These peaks correspond to the frequencies of the superimposed waves which are components of the wave grouping. The amplitude of the resultant wave varies at a frequency equal to the differences between the two superimposed frequencies.

$$f_g = f_1 - f_2 \qquad \text{(A-28)}$$

where f_1 and f_2 are the frequencies of the superimposed waves and f_g is the frequency of the wave grouping.

Wave grouping can be detected through a spectral analysis of the envelope of the wave record. A curve drawn through the peaks of the wave record will form this envelope as shown in the figure. A spectral analysis of this curve will reveal the presence of wave grouping.

Long-period wave phenomena, such as seiche, swell, and wave grouping may affect the motions of a moored vessel as much as the short period wave phenomena depicted in the conventional wave spectrum. Therefore, analysis should be performed to determine if these long-period phenomena exist at the mooring site. If seiche, swell, or wave grouping are expected at the mooring site then they should be accounted for as much as possible in model testing and analysis. If they are not expected to be present at the mooring site, then checks should be made during model testing to assure they are not present in the model basin and affecting the results of the tests.

WAVE-HEIGHT STATISTICAL CORRELATIONS
ACCORDING TO VARIOUS RESEARCHERS

TABLE A-1

Reference		Source	H_s/H_o	$H_{1/10}/H_s$	H_{max}/H_s
Munk	(1940)	Field data	1.53	-	-
Seiwell	(1948)	Field data	1.57	-	-
Wiegel	(1949)	Field data	-	1.29	1.87
Barber	(1950)	Theoretical	1.61	-	1.50
Putz	(1950)	Field data	1.63	-	-
Longuet-Higgins	(1952)	Theoretical	1.60	1.27	1.77
Putz	(1952)	Theoretical	1.57	1.29	1.80
Darbyshire	(1952)	Field data	1.60	-	1.50
Hemada, et.al	(1953)	Experimental	1.35	-	-

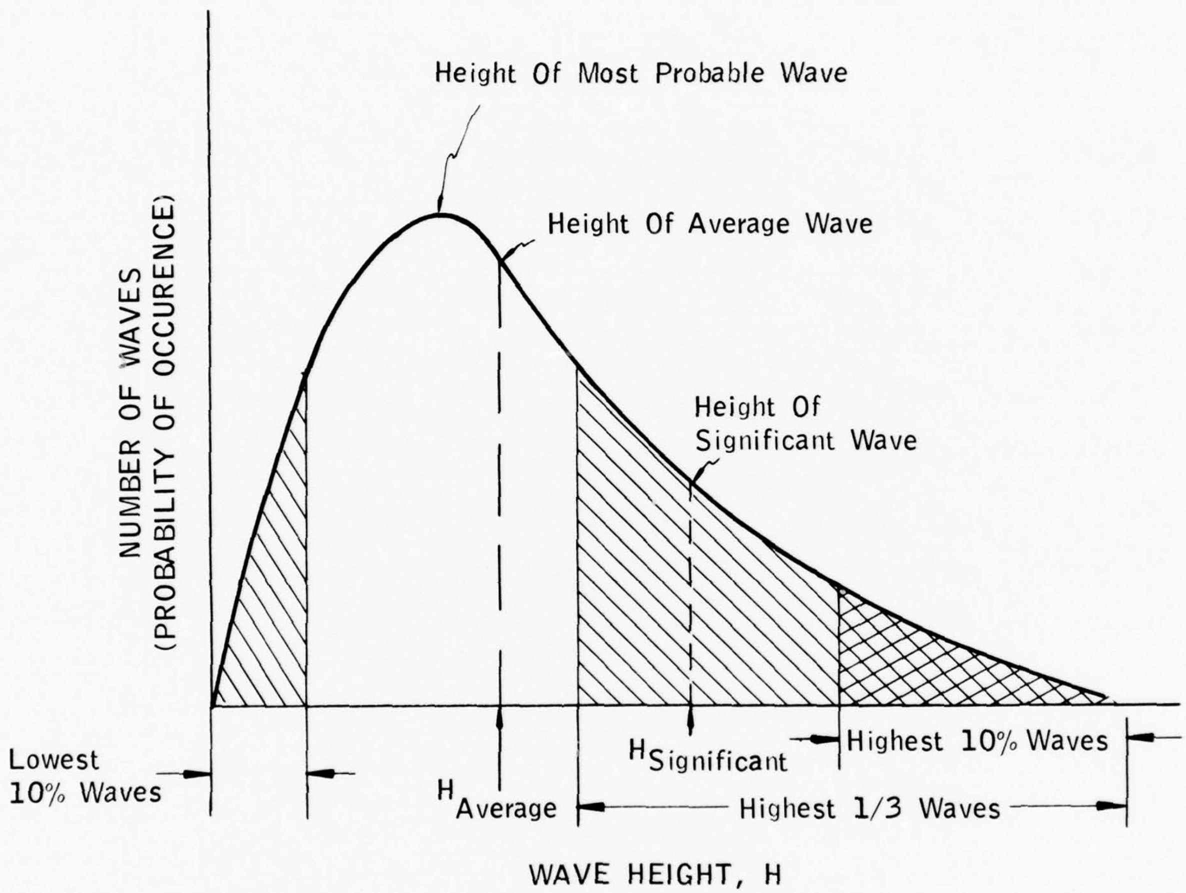


Figure A-1 - DISTRIBUTION OF WAVE HEIGHTS IN ACCORDANCE WITH RALEIGH DISTRIBUTION

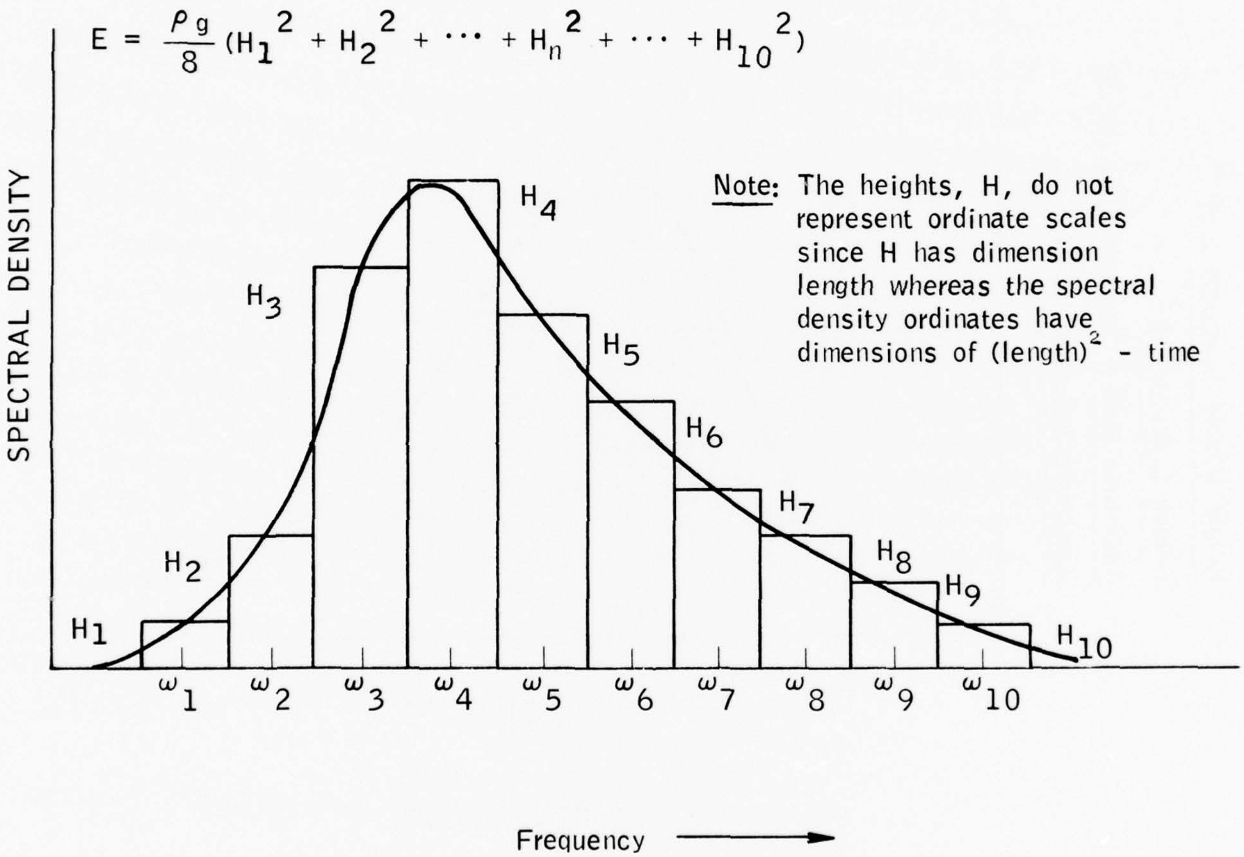


Figure A-2 SIMPLIFIED REPRESENTATION OF WAVE SPECTRUM DEVELOPMENT

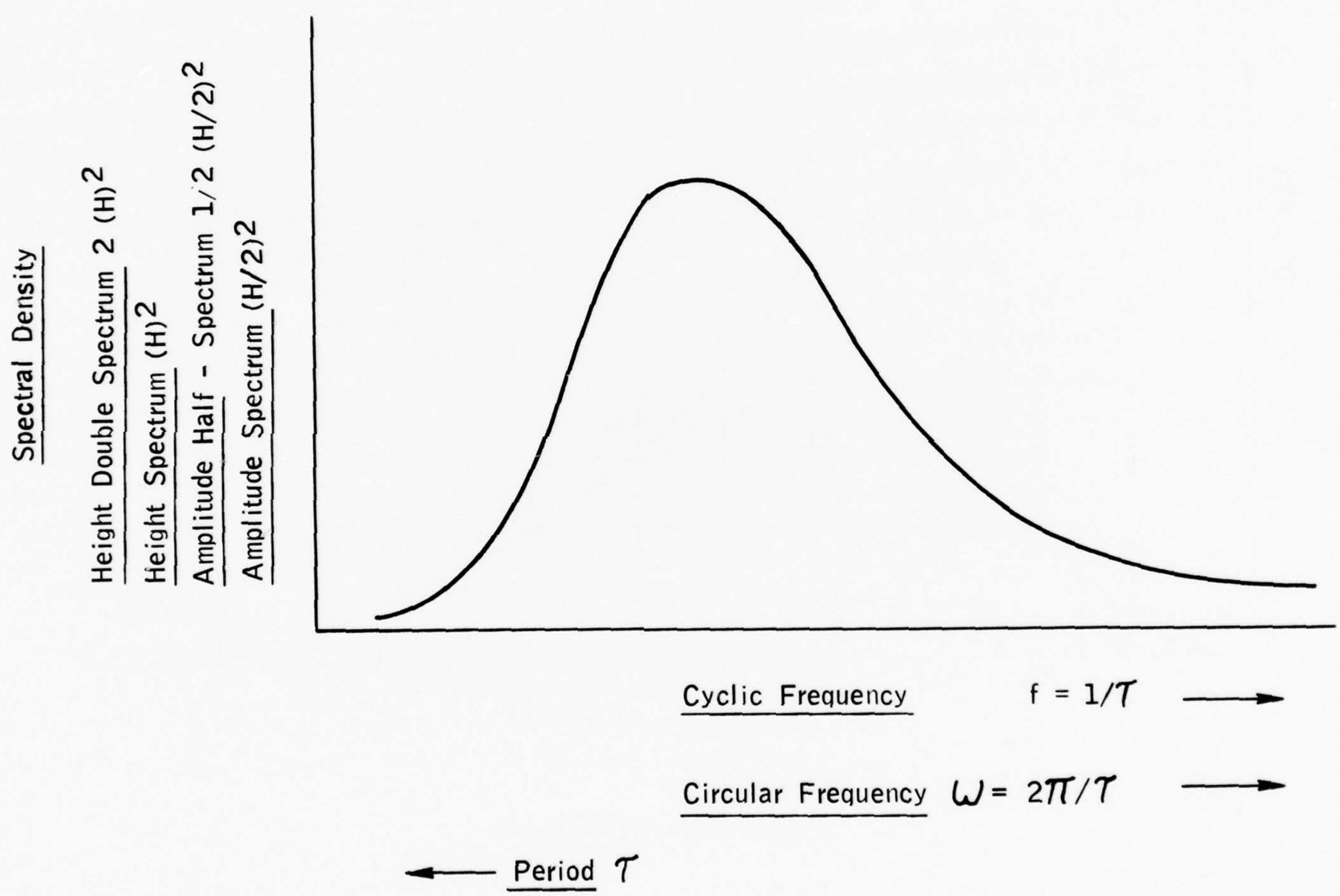
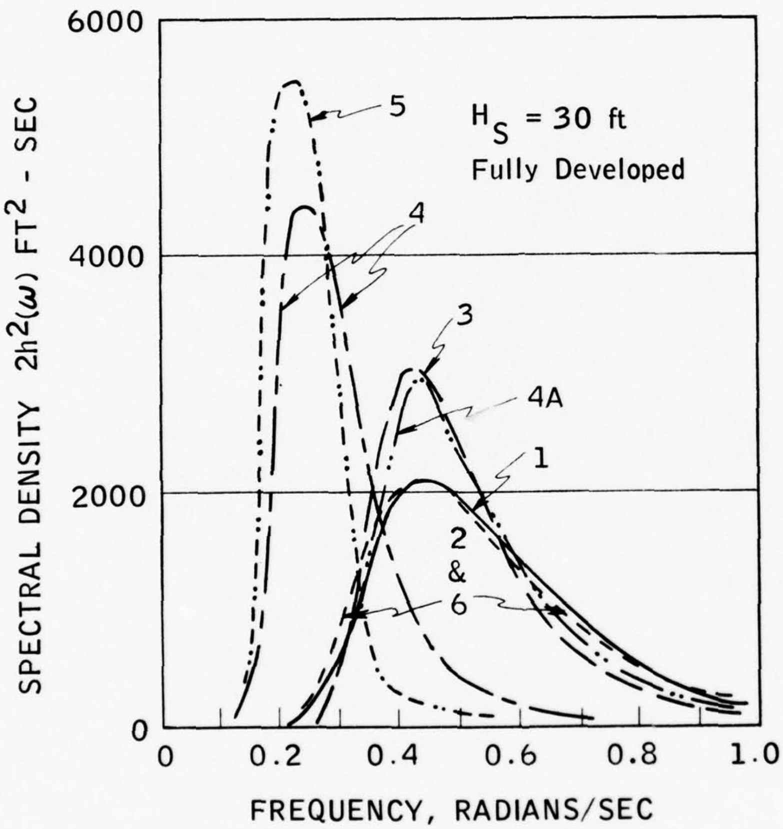


Figure A-3 VARIOUS NOMENCLATURE USED WITH WAVE SPECTRA



1. Neumann Spectrum - 34.2 Knots
2. Neumann Modified By Roll-Fischer Spectrum - 39.3 Knots
3. Pierson Moskowitz Spectrum - 40 Knots
4. Bretschneider Spectrum - 34 Knots
4A. Partially Developed Bretschneider Spectrum - 36.1 Knots
5. ISSC Spectrum - 34.6 Knots
6. Same As 2 Above

Figure A-4 COMPARISON OF VARIOUS WAVE SPECTRA
(Modified From Michel, 1967)

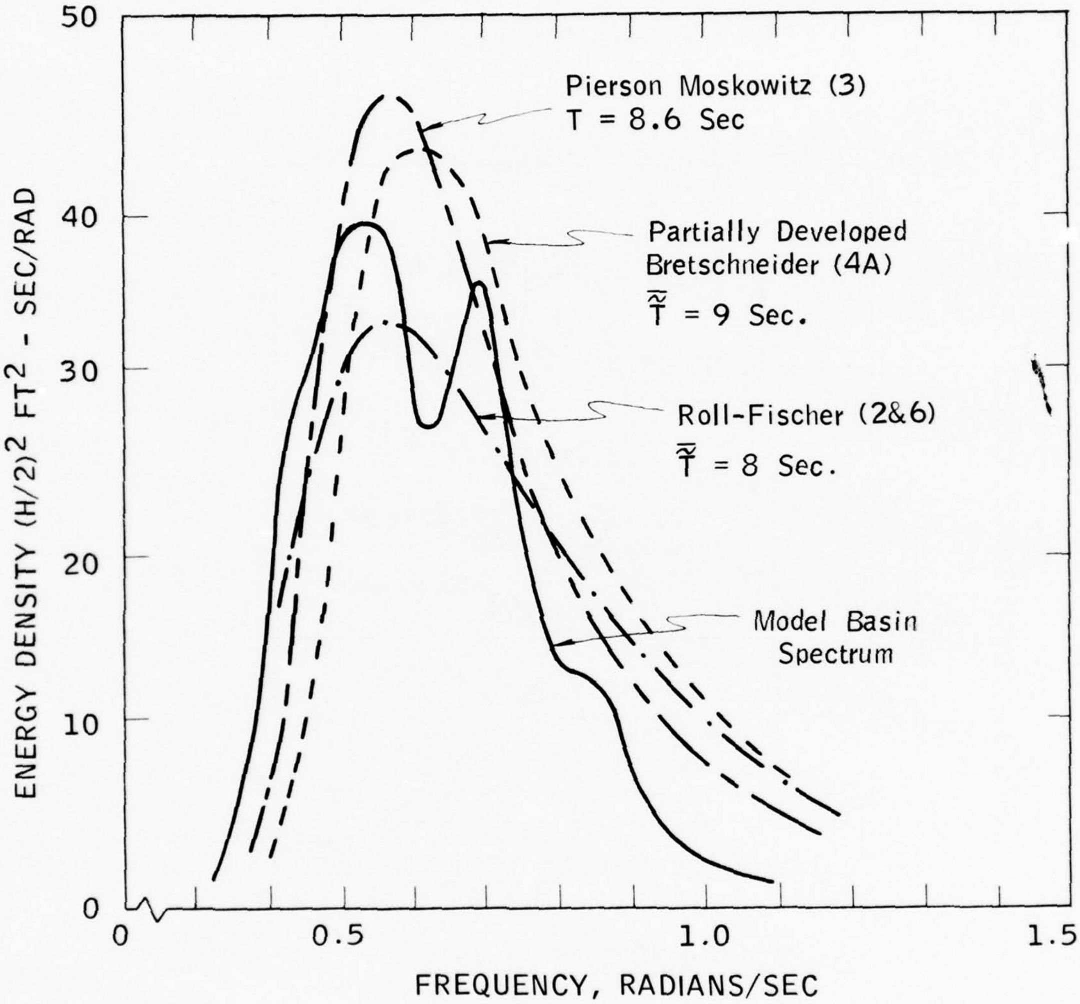


Figure A-5 - MATCH OF MODEL BASIN SPECTRUM WITH THREE FORMULA SPECTRA

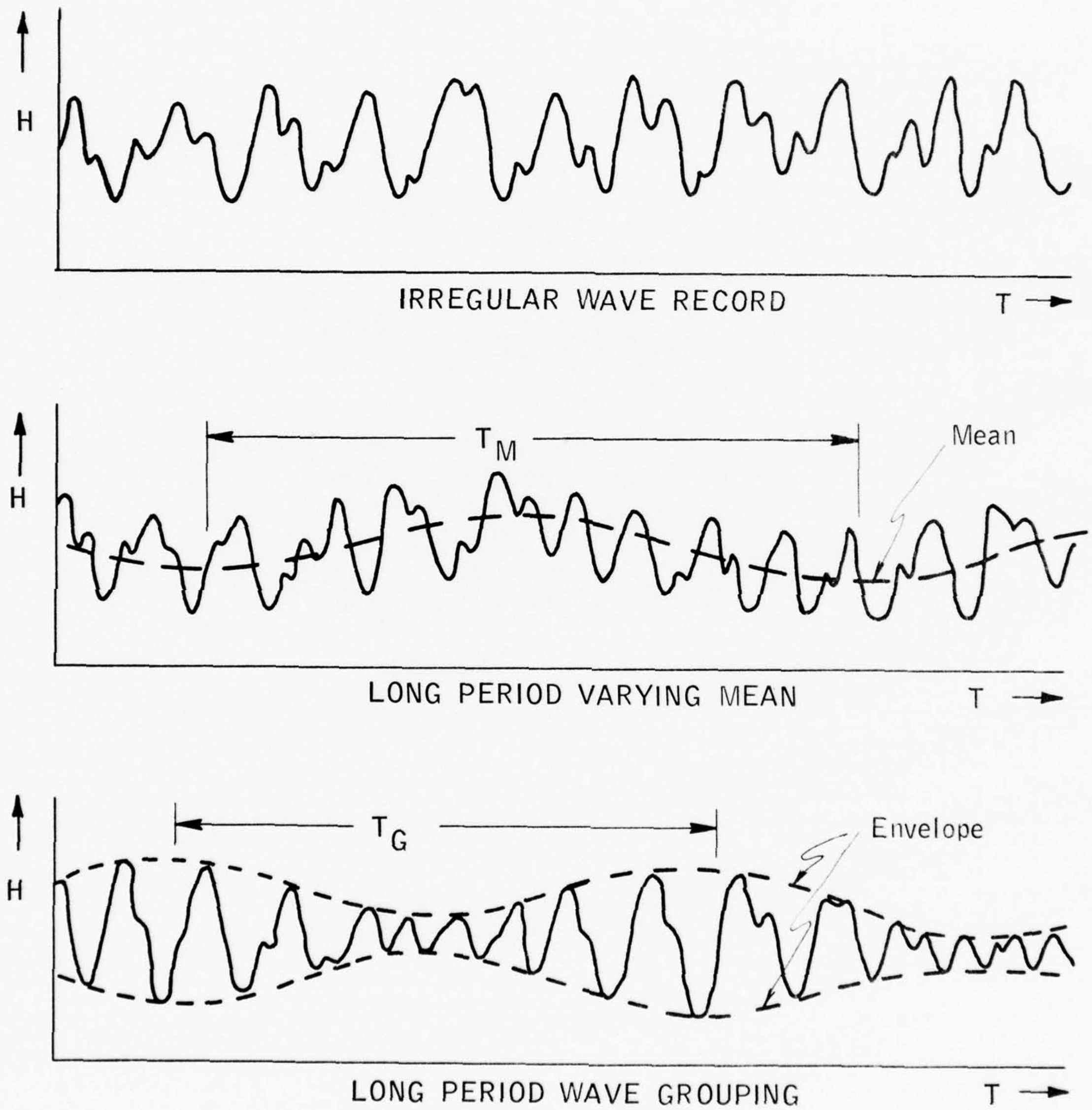


Figure A-6 - LONG PERIOD WAVE PHENOMENA

APPENDIX B

THE ENERGY THEORY OF SPM MOORING LOADS

This appendix describes a theory which relates the elasticity of the SPM system to the mooring loads which are experienced in irregular waves. The theory is presented here not as a required or recommended method of predicting mooring loads at SPMs, but as an aid in understanding the influence of mooring system elasticity on mooring loads.

Through the analysis of data from model tests and some full-scale tests of tankers moored to SPMs, Exxon Research and Engineering has developed several empirical design techniques. One of the tools used in these techniques is known as the significant energy technique. From analysis of mooring load data it was discovered there is a relationship between the area under the mooring elasticity curve up to the significant force and the test environment and tanker-size parameters. The mooring elasticity properties and the tanker size or mass constitute a dynamic system which is excited by the environment. The energy theory may be thought of as an analog for understanding the dynamic inter-relationships of these parameters.

The following is a brief disucssion of the major principals of the significant energy theory.

B.1 Elasticity of the Mooring

The elasticity of an SPM is the sum of the non-linear elasticities of the buoy anchoring system and the bow-hawser system. From model test and field experience it has become evident that the mooring system elasticity should not be too non-elastic or stiff because the mooring forces caused by waves would then be extremely high. It is also evident from observations that a very elastic or soft mooring system allows the tanker excessive freedom to move; thus, the tanker acquries substantial kinetic energy. This kinetic energy eventually transfers to the mooring system in the form of potential energy as the motion is arrested. This potential energy is evident as strain in the mooring system. In the design of an SPM a proper mooring system elasticity must be established which is stiff enough to minimize the motions of the moored vessels and the resultant forces due to these motions and yet not be so stiff as to cause high mooring loads.

The elasticity of the mooring system is influenced by various parameters, as discussed in Section 2 of this report. Different SPMs may have different elasticity characteristics. The elasticity characteristic of an SPM is important in determining the mooring loads which will be experienced at that mooring.

B.2 Significant Mooring Force

The siginficant mooring force is defined as the average of the highest one-third peaks in the bow-hawser mooring-load record. It is similar in concept to the significant wave height (see Appendix A) and can be calculated by the same methods.

The significant force is a statistical value which remains essentially constant regardless of the length of the record, provided the record is of sufficient length and other parameters are not varied. It may be used directly as a means of comparing the relative merit of several proposed mooring systems. The probable maximum force for a given duration of the record may be determined from the force by the statistical methods discussed in Section 2.

B.3 The Significant Energy Concept

Figure B-1 shows mooring elasticity curves for two mooring systems SPM A and SPM B. If a given tanker were to be moored in a given environment to SPM A, a significant mooring force, F_a would be experienced. If the same tanker were to be moored in the same environment to SPM B, a different significant force F_b would be experienced. The area under the elasticity curve for SPM A up to force F_a has the physical equivalence of energy stored in this system when it is deflected by the significant force F_a, and therefore, it is refered to as the significant energy E_a. In the same manner significant energy E_b may be defined for SPM B.

From the analysis of many model test records for SPMs having different elasticities, it has been discovered that for a given tanker size and a given environment the energy under the elasticity curve up to the significant force is essentially equal for different SPMs of not too dissimilar designs. For the example given here, energy E_a would equal energy E_b, because the tanker size and environment are the same even though the moorings are different.

If a different tanker were moored at SPM A, a different significant force F_a' would be measured and a different significant energy E_a' would be determined. If the same tanker were to moor at SPM A in a different environment yet another significant force F_a'' would result, thus establishing another significant energy E_a''. From these tests at SPM A the measured significant energy E_a' for the different tanker size, and the measured significant energy E_a'' for the different environment, would be equal to the respective significant energies E_b' and E_b'' for SPM B.

A series of curves may be developed giving the relationship between tanker size and environment and the significant energy levels based on the results of a large number of model tests. Figure B-2 is an example of such a significant-energy curve.

The designer of a system, knowing tanker size and environment may use such a significant-energy curve to determine the significant energy corresponding to the significant force. Knowing the elasticity curve for an SPM under study, the designer may then determine the significant hawser force corresponding to the significant energy under the elasticity curve.

The significant-energy theory should be used with caution in predicting mooring loads, especially when the method is based on load data for a different SPM. The description given here does not deal with the effects of wind and current in combination with waves. The effects of certain differences in the mooring system, such as hawser length and water depth, must be accounted for in other ways when applying the energy technique. The use of the energy technique as an empirical method for predicting mooring loads should only be done when sufficient mooring load data for the same or similar SPM systems are available and when the effects of the various parameters are understood.

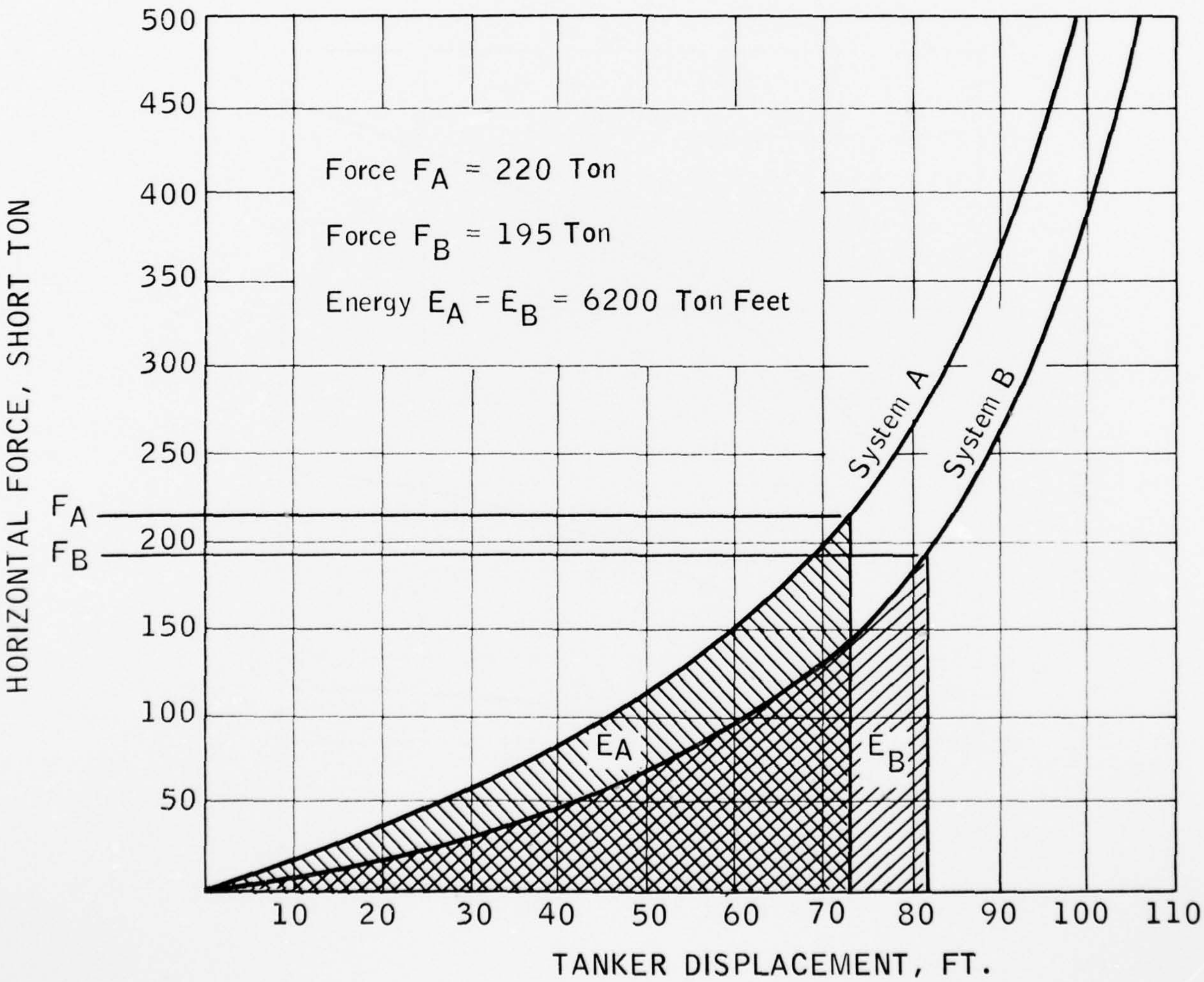


Figure B-1 - SPM ELASTICITY CURVES SHOWING APPLICATION OF ENERGY THEORY

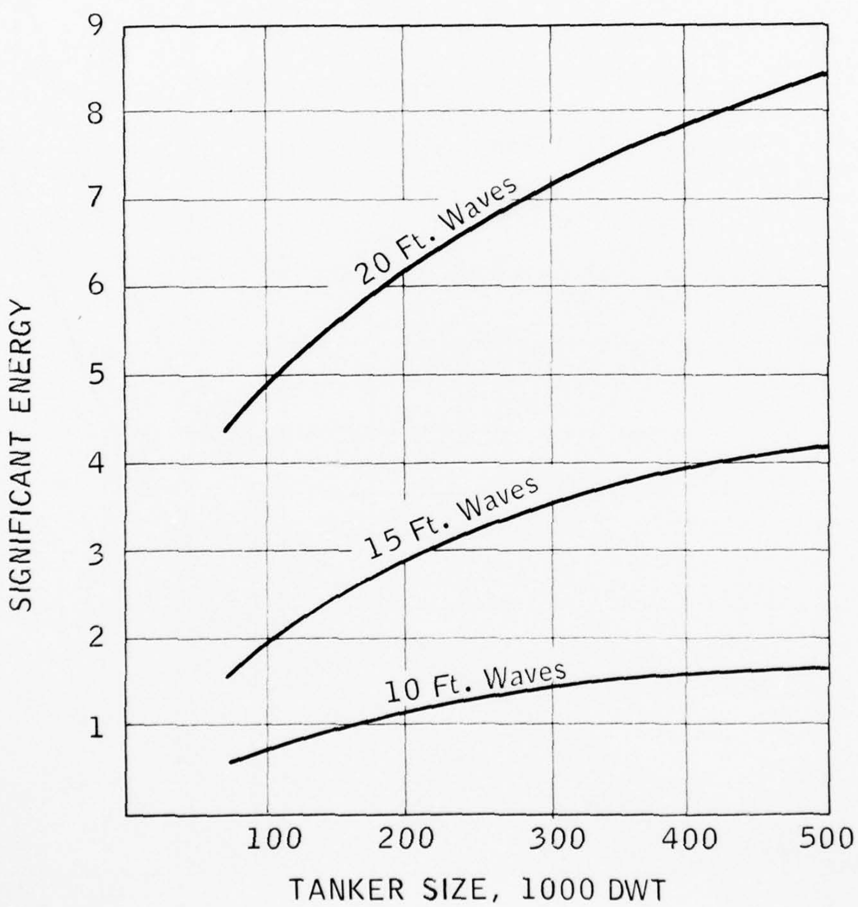


Figure B-2 - EXAMPLE ENERGY PREDICTION CURVE

APPENDIX C

MODEL TEST BASINS

DANISH SHIP RESEARCH LABORATORY
Skibsteknisk Laboratory
Hjortekaevsvej 99
DK-2900
Lyngby (Copenhagen), Danmark

Test Basin

The DRSL test basin is a towing tank 240 m (800 ft) long, 12 m (40 ft) wide, and 5.5 m (18 ft) deep. The towing carriage has a maximum velocity of 15 m/s (50 ft/s) and can tow models that weigh up to 10,000 kg (22,000 lb) and are 8 m (26 ft) long. The towing carriage has an onboard computer having a maximum logging frequency of 17,000 Hz.

Wave Generation

DRSL uses a pneumatic wave generator capable of creating regular and irregular waves. The generator is controlled by an 8-hole paper tape which can contain any type of wave signal.

Current Generation

The tank has no capability to generate a current, but it may be possible to tow models through deep water.

Instrumentation

Force - No information available

Displacement - Horizontal large-amplitude planar-motion mechanism, and a Selsport optical tracker. Verticle small-amplitude planar motion mechanism

Velocities - Differentiation of displacement

Acceleration - Double-differentiation of displacement and accelerometers

Water Level - Wave probes

C.2

HYDRONAUTICS SHIP MODEL BASIN, INC.
7710 Pindell School Road
Laurel, Maryland 20810

Test Basin

The test basin at HYDRONAUTICS is a towing tank 127 m (418 ft) long and 7.6 m (25 ft) wide. The maximum water depth is 4 m (13 ft). Although the water level may be lowered, the existing wave generator (as described below) and beach do not extend to the total depth of the tank. Therefore, the existing equipment is not able to produce waves in shallow water. There is a 3.4 m (11 ft) diameter, 13 m (43 ft) deep pit located near the center of the basin. The main towing carriage has a maximum velocity of 6 m/s (20 ft/s) and the high speed carriage can reach velocities of 9 m/s (30 ft/s). A data acquisition and computer center is onboard the main carriage.

Current Generation

HSMB can simulate current only by towing the model through the basin with the carriage. Thus, the simulated current can only be parallel to the generated waves.

Wind Generation

Wind in line with waves can be generated by fans. HSMB will have in the near future a 1 m (3.3 ft) diameter axial-flow fan with a variable speed drive. However, it would be difficult to model wind at angles to the waves because the building walls are very close to the edge of the tank.

Wave Generation

The wave generator is a vertical-moving wedge driven by hydraulic power. The hydraulic drive mechanism is controlled by a bank of 10 potentiometers. Each potentiometer controls the amplitude and frequency of a sinusoidal signal generator. These 10 signals are combined to form a 10-component irregular signal for wave generation. This control system permits the generation of very long non-repeating wave trains.

Instrumentation

Force - Reluctance block guages and Shavits Load Cells.

Displacement - A two-arm horizontal-linkage mechanism for surge, sway, yaw, and heave, and gyroscopes for pitch and roll

Velocity - Differentiation of displacement

Acceleration - Double-differentiation of displacement or accelerometers

Water Level - Wave probe

Water Velocity - Measure carriage velocity

NETHERLANDS SHIP MODEL BASIN
Haagsteed 2
Post Box 28
6700 4A Wageningen
The Netherlands

Test Basins

- WAVE AND CURRENT BASIN

The Wave and Current Basin is 60 m (200 ft) long, 40 m (133 ft) wide, and 1.0 m (3.3 ft) deep. This basin is used to study systems requiring waves and current acting simultaneously at angles to each other, such as drilling platforms, single-point moorings, open sea dredging, harbor entrances, etc. The water depth is adjustable. A maximum 0.6 m/s (2.0 ft/s) current velocity can be created parallel to the short edges of the basin at a water depth of up to 0.5 m (1.6 ft). The current direction is reversible. Wave generation flaps (described below) are located on one long side and one short side. Regular or irregular waves can be generated at any angle to the current.

- SHALLOW WATER BASIN

The Shallow Water Basin is a towing tank which has a length of 216 m (710 ft), a width of 16 m (52 ft) and a variable water depth of zero to 1 m (3.3 ft). At the center of the basin, there is a 3 m (9.8 ft) deep pit suitable for testing larger structures. The Shallow Water Basin carriage has a maximum velocity of 3 m/s (10 ft/s) and has onboard analog data recording devices. A mechanical wave generator, described below, is at one end of the basin and a wave absorbing beach is opposite the wave generator. There are no means to generate a current in this basin other than by towing the model.

- SEAKEEPING BASIN

The Seakeeping Basin is primarily used for vessel seakeeping tests. The tank is 100 m (328 ft) long, 24 m (78.7 ft) wide, and 2.4 m (7.9 ft deep). The towing carriage has a maximum velocity of 4.5 m/s (15 ft/s) and has onboard digital data recording equipment. Wave generators (described below) line two adjacent walls of the basin to enable creation of regular or irregular waves from any direction. Wave damping beaches are installed on the basin edges opposite the wave generators. There are no means to generate a current in the basin other than towing the model.

Current Generation

Current is generated in the Wave and Current Basin by pumping water through inlet orifices, across the basin, and out through outlet orifices. The pumping units have a maximum capacity of 15 m^3/s (530 cfs). The other basins can simulate current only by towing the model.

Wave Generation

The wave generators for the Wave and Current Basin, Shallow Water Basin, and Seakeeping Basin are snake-type wave makers consisting of banks of flaps hinged at the bottom. Long crested irregular waves are generated when the flaps are oscillated with varying frequency at a constant stroke. The range of oscillation frequency corresponds to the frequency range of the irregular waves, and thus, waves of different frequencies are sequentially generated.

The sequence of wave frequencies is generated by an electronic control system, of which the main part consists of a set of 96 potentiometers, each controlling the frequency of one wave component.

Wind Generation

Wind is created by banks of axial flow fans. Each fan is controlled by a potentiometer.

Instrumentation

Force - Full bridge strain gauges or reluctance block gauges--as appropriate to the force level and required resolution.

Displacement - Pantograph system and an optical system of servo-controlled cameras which follow a halogen light. The optical system measures heave, surge, and sway. Gyroscopes measure pitch, roll and yaw. The gyro-optical system allows for vessel to be completely free of the instrumentation except for wire leads.

Velocity - Differentiation of displacement, rate gyros, tachmeters

Accelerations - Double-differentiation of displacement or accelerometers

Water Level - Wave probe, servo-controlled wave followers

Water Velocities - Screw-type current meters

Wind Velocities - Cup-type anemometer

OFFSHORE TECHNOLOGY CORPORATION
578 Enterprise Street
Escondido, California 92025

Test Basin

The OTC test basin is 90 m (295 ft) long, 14.6 m (48 ft) wide, and 4.6 m (15 ft) deep. The water depth in the basin can be varied and for shallow water tests a special wave generator is used. Approximately two-thirds the way down the basin is a square pit 4.6 m (15 ft) deep (below basin floor) which measures 5.5 m (18 ft) square at the top and 4.6 m (15 ft) square at the bottom. An unmanned carriage runs as wheels along the length of the basin. The carriage has a maximum velocity of 3 m/s (10 ft/s). Another carriage on wheels is used to support instrumentation at fixed positions over the basin. The data acquisition and computer center is located in a room adjacent to the basin. Sixty-four channels of digital data can be recorded.

Wind Generation

Wind can be generated by two large ducted fans, each powered by a 37 kW (50 hp) motor or by ten 3 ft diameter fans powered by 1 hp motor. The fans are placed beside the basin and can be either aimed directly at the model, or air can be channeled through ductwork to the vicinity of the model. Wind speed is controlled by varying fan-blade pitch or varying the suction orifice. Wind can be blown at angles to the waves only when the basin is at maximum water depth. At shallow water depths, the basin side walls will interfere with cross winds.

Current Generation

OTC can generate a current up to about .3 m/s (1 ft/s) parallel to the waves by setting up a circulation pattern in the basin. Water is pumped through nozzles in the side of a submerged pipe.

Wave Generation

Cassette tapes are generated by the computer to control the irregular wave generation system. The signal on the tape is calculated to generate a wave train with the designated spectral shape. The main wave generator is a large steel flap pivoted at the bottom and driven by a hydraulic cylinder. The shallow-water wave generator is a set of four flaps pivoted at the top and driven by four small hydraulic cylinders.

Instrumentation

Force - Load cells

Displacement - Six-degree-of-freedom mechanical motion sensor

Velocities - Differentiation of displacements

Acceleration - Accelerometers and double-differentiation of displacement

Water Level - Capacitance wave probes

Water Velocity - Savonis rotor, electromagnetic probe

Wind Velocity - Annemometer

Pressure - Pressure sensors

U.S. NAVAL ACADEMY HYDROMECHANICS LABORATORY
Richover Hall
Annapolis, Maryland

Test Basin

The model tank basin is presently under construction. It is 116 m (380 ft) and 7.9 m (26 ft) wide. The depth of water is 4.9 m (16 ft) deep. Although the water level can be reduced, the wave maker and wave absorber beach would be ineffective at shallow water depths. The tank will be equipped with a high speed carriage capable of speeds up to 7.6 m/s (25 ft/s).

Wave Generation

Wave generation is controlled by an independent PDP 11-05 computer. The desired wave amplitude spectrum is specified and the computer generates control signals having random phasing. The control signals are fed to the wave making flap. The generated waves are analyzed and compared with the specified spectrum by the computer. The computer then corrects the control signals. This process is repeated until the desired wave spectrum is attained. A given wave train may be repeated as often as desired, and the control signals can be recorded so that the exact wave train can be re-established in the future.

Current Generation

There is no capability to generate a current in the basin. The model must be towed to simulate current effects.

Wind Generation

No provision has been made for producing wind. Wind could be produced in line with the waves. However, the adjacent walls of the building do not make it possible to produce winds at angles to the waves.

Limitations on Use of the Facilities

The primary function of the Hydromechanic Laboratory is instruction of midshipmen. Private firms could make use of the facilities only under extraordinary circumstances.

OTHER MODEL TEST BASINS

The following basins were not visited during the course of this study, but are understood to have facilities and capabilities similar to those described above and may be suitable for deepwater port model tests.

BRITISH HOVERCRAFT CORPORATION, LTD.
Experimental and Electronics Laboratory
East Cowes
Isle of Wright, England

BRITISH HYDRAULIC RESEARCH STATION
Wallingford, Berkshire
England

CANADIAN CENTER FOR INLAND WATERWAYS
867 Lakeshore Blvd.
Box 5050
Burlington, Ontario L7R4A6

CHICAGO BRIDGE AND IRON COMPANY
901 West 22nd Street
Oak Brook, Illinois 60521

NAVAL SHIP RESEARCH DEVELOPMENT CENTER
Canderock Laboratory
Bethesda, Maryland 20034

SHIP RESEARCH INSTITUTE OF NORWAY
Novges Skipfforskningsinstituft
N-7034
Trondheim, Norway

SOGREAH, INC.
Avenue Leon-Blum
Grenoble, France

C.8

STEVENS INSTITUTE OF TECHNOLOGY
Davidson Laboratories
Castle Point Station
Hoboken, New Jersey 07030

APPENDIX D

ROPE TESTING ORGANIATIONS

COORDINATED EQUIPMENT COMPANY

P.O. Box 1244
Wilmington, California 90744

Rope Test Facilities

Coordinated Equipment Co (CEC) has acquired ten very large hydraulic cylinders and five high-strength beams which were surplus from the GLOMAR EXPLORER submarine salvage project. There are three different cylinder designs. The load capaciaties and strokes of these cylinders are as follows:

Cylinder Type	Stroke		Maximum Force Compression		Maximum Force Tension	
	(mm)	(in)	(MN)	(Short Tons)	(MN)	(Short Tons)
I	2972	117	8.3	924	14.2	1588
II	1651	65	18.9	2121	15.4	1722
III	1295	51	18.9	2121	15.4	1722

CEC plans to construct a test bed with an overall length of 53m (174 ft). With cylinders Type I in place, this bed will accommodate a maximum test specimen length of 34.7 m (114 ft) and will be capable of applying a maximum stroke of 5.9 m (19.5 ft) to the specimen. The maximum load applied in this set-up is 14.2 MN (1588 short tons). Other set-ups and capacities are summarized in Figure D.1.

Experience in Testing Large Ropes

Since CEC's equipment is not yet operational, they have no experience in its use. However, CEC plan to have the new test machine installed and operational by the end of 1977. This test machine will be adequate for static break testing large-diameter hawsers used on SPMs. The maximum load capacity is more than adequate to test the largest hawsers now made. The stroke and bed length are adequate for testing rope specimens of sufficient length to include typical splice or end effects. The stroke may be sufficient to test halves of used SPM hawsers with original eye splices.

CEC has extensive experience in testing smaller synthetic ropes, as well as wire ropes and other load-carrying components. One of the large cylinders is now temporarily installed on a 15 m (50 ft) long test bed for use in testing short lengths of large wire and synthetic rope.

ENGINEERING MECHANICS LABORATORY
National Bureau of Standards
Gaithersburg, Maryland

Rope Testing Facilities:

- A 54 MN (6000 short ton) static testing machine capable of accommodating rope specimens of overall length up to 17.7 m (58 ft). The working stroke of the machine is 1.5 m (5 ft). (See NBS Special Publicaton 355 for further details on this machine.)

Experience in Testing Large Ropes:

- No known experience in testing large ropes.

FRITZ LABORATORY
Lehigh University
Lehigh, Pennsylvania

Rope Test Facilities

- A 22 MN (2500 short ton) static testing machine capable of accommodating rope specimens of overall length up to 12.2 m (40 ft). The hydraulic working stroke of the machine is 0.9 m (3 ft). The maximum rate of specimen loading is 76 mm/min (3 in/min) when using the hydraulic working stroke and up to 610 mm/min (24 in/min) with the mechanical screws.

- A 3.5 MN (400 short ton) static testing machine capable of accommodating rope specimens of overall length up to 6.9 m (22.5 ft) at speeds of about 28 mm/min (1.1 in/min) under no load and about 10 mm/min (0.4 in/min) under maximum load.

Experience in Testing Large Ropes

- Static break testing program for Samson Cordage Works on new double-braided nylon ropes of up to 120 mm diameter (15 inch circumference).

- Several static break testing programs for Columbia Rope Company on new three-strand nylon ropes of up to 120 mm diameter (15 inch circumference).

NATIONAL COAL BOARD
Mining Research and Development Establishment
Mechanical Testing Branch
Stanhope Bretby
Burton on Trent, Darbishire, England

Rope Test Facilities:

- A 4.5 MN (500 short ton) static testing machine with a 15.2 m (50 ft) test bed and a working stroke of 3.0 m (10 ft).

Experience in Testing Large Ropes

- Static break testing program for British Ropes on new 144 mm diameter (18 inch circumference) double-braided nylon ropes.

NATIONAL ENGINEERING LABORATORY
Department of Trade & Industry
East Kilbride, Glasgow, Scotland

Rope Test Facilities:

- A 10 MN (1120 short ton) static testing machine capable of accommodating rope specimens of overall length up to 8.0 m (26 ft). The working stroke of the machine is 1.5 m (5 ft) and the maximum crosshead speed is 150 mm/min. (6 in./min.). For loads below 5 MN (560 short ton), the crosshead speed can be doubled.

- A 1.2 MN (134 short ton) static or cyclic loading testing machine capable of accommodating rope specimens between 0.6 and 7.0 m (2-23 ft) long at speeds from 0 to 400 mm/min. (0-16 in./min.). This machine is equipped to control and measure such parameters as load, velocity of travel, and rope elongation.

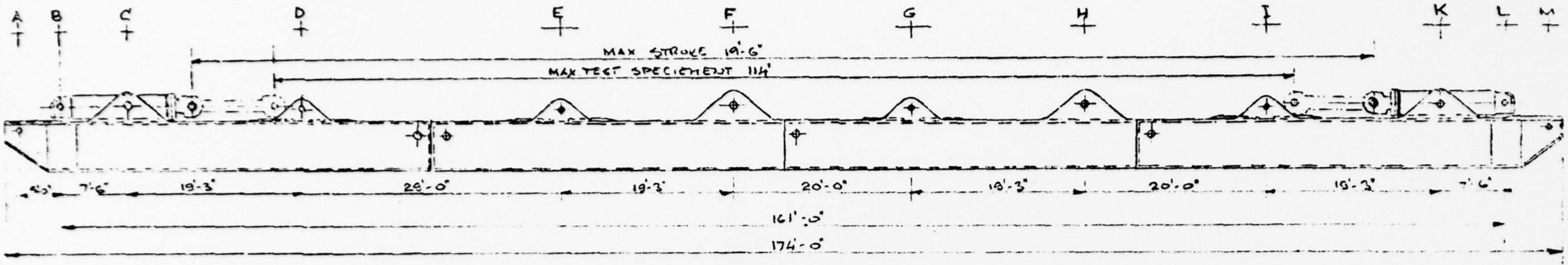
- A dynamic load test facility which simulates shock loading conditions on ropes. In this unit impact energy of 155 kJ (114,000 ft-lbs) is achieved by a large linear induction motor and speeds up to 17.8 m/s (58.3 ft/s) are reached as the 100 kg (220 lbs) carriage, supported on air bearings, is propelled along parallel rails to pick up the free end of a rope and break it under impact conditions. At these speeds sufficient energy is stored to break ropes of all types of construction in synthetic or natural fibers up to about 150 kN (17 short ton) breaking load.

Neither of the latter two machines mentioned above is suitable for testing of large-diameter ropes because of strength limitations. The 10 MN static testing machine has been used to carry out a few tests on large diameter ropes in an endless grommet configuration. However, due to specimen length and stroke restrictions this machine is probably not suitable for testing large diameter ropes with eye splices.

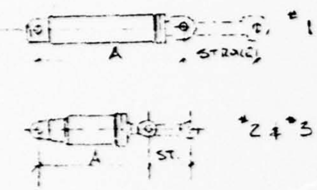
NEL has developed tentative plans for the construction of a new testing machine. The new machine would have a 10 MN (1120 short tons) capacity and be able to accommodate rope specimens of up to 30-60 m (100-200 ft) long. The working stroke of the machine would allow for about a 50% stretch of the specimen. Plans for the testing machine are currently on a "hold status" until NEL can ascertain the level of interest in use of the machine by industry as well as possible construction funding by industry.

Experience in Testing Large Ropes

- Static break testing program for British Ropes on new 128 mm diameter (16 inch circumference) double-braided nylon ropes.
- Static break testing program for ARAMCO on used 120 mm diameter (15 inch circumference) double-braided nylon ropes.
- Static break testing and cyclic tests for Exxon Research and Engineering Co. on new and used 64 and 72 mm diameter (8 and 9 inch circumference) nylon ropes. New ropes were all three-strand construction. Used ropes were of three-strand, eight-strand and double-braided construction.



HYDRAULIC CYLINDERS



	#1	#2	#3
A	176	148	133
ST	117	65	51
$F_{comp} \times 10^6$	1.347	4.241	4.241
$F_{TE} \times 10^6$	3.176	3.443	3.443

NOTE #1 COMP. 3000 PSI
#2,3 - 6000 PSI

234"-STROKE 3.175×10⁶-LBS		117"-STROKE 3.175×10⁶-LBS		130"-STROKE 3.443×10⁶-LBS		116"-STROKE 3.443×10⁶-LBS		102"-STROKE 3.443×10⁶-LBS		65"-STROKE 3.443×10⁶-LBS		51"-STROKE 3.443×10⁶-LBS	
B-L	112-2	B-M	141-7	C-K	110-6	C-K	112-11	C-K	115-3	C-M	140-9	C-M	[illegible]
B-K	104-8	B-L	136-7	C-I	91-3	C-I	93-8	C-I	96-1	C-L	135-9	C-L	138-2
C-K	97-2	B-K	124-1	C-H	71-3	C-H	73-8	C-H	76-1	C-K	128-3	C-K	138-8
B-H	65-5	B-I	104-10	K-E	62-3	K-E	64-8	K-E	67-1	C-I	109-0	C-I	111-5
C-H	57-11	M-E	85-0	C-G	52-0	C-G	54-5	C-G	56-10	C-H	89-0	C-H	91-5
F-K	35-8	B-H	84-10	K-F	43-0	K-F	45-5	K-F	47-10	K-E	80-0	K-E	82-5
B-F	26-2	M-F	66-7	C-F	32-0	C-F	34-5	C-F	36-10	C-G	69-9	C-G	72-2
C-F	18-8	B-G	65-7	K-G	23-0	K-G	25-5	K-G	27-10	K-F	60-9	K-F	[illegible]
		M-G	46-7	C-E	12-9	C-E	15-2	C-E	17-7	C-F	49-9	C-F	52-2
		B-F	45-7	K-H	3-9	K-H	5-5	K-H	8-7	K-G	40-3	K-G	43-2
		M-H	27-4							C-E	30-6	C-E	32-11
										K-H	21-6	K-H	23-11
										C-D	1-6	C-D	3-11

MAX. SPACING [FT-IN] BETWEEN CROSS-HEADS OR CROSSHEAD & DEAD END RESPECTIVELY
WITH REDUCED STROKE SPACING COULD BE INCREASED.
LOAD CAPABILITIES BASED ON 3000 PSI OPERATING PRESSURE - FOR HIGHER LOAD REQUIRMENTS
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NK. 11 20-76

FIGURE D.1 - COORDINATED EQUIPMENT COMPANY'S PROPOSED TEST BED

APPENDIX F

BIBLIOGRAPHY

American Bureau of Shipping, Rules for Building and Classing Single Point Moorings, 45 Broad Street, New York, 1975

American Bureau of Shipping, Rules for Building and Classing Steel Vessels, New York, published yearly

American Petroleum Institute, API Recommended Practice for Planning, Designing and Constructing, Fixed Offshore Platform, Washington, D.C., Eighth Edition 1977

American Petroleum Institute, API Specification for Mooring Chain, Washington, D.C., Second Edition, 1977

Baptist, C., Tanker Handbook for Deck Officers, Brown, Son, and Ferguson, Ltd., Glasgow, 1969.

Barber, N. F., "Ocean Waves and Swell", Lecture Published by Institute of Civil Engineers, London, 1950

Batchelor, G. K., Fluid Dynamics, Cambridge University Press, Cambridge, England 1967

Benham, F. A., "Visit to Continental Oil's Tetney Terminal Facilities", ER&E Trip Memorandum, December 9, 1976

Benham, F. A., "Visit to National Engineering Laboratory Concerning Rope Testing", ER&E Trip Memorandum, November 24, 1976

Benham, F. A., "Visit to Samson Cordage", ER&E Trip Memorandum, December 17, 1976

Benham, F. A., S. S. Fang, C. H. Fetters, "Wind and Current Shape Coefficients for Very Large Crude Carriers" 1977 Offshore Technology Conference Proceedings, OTC 2739, pp. 97-108, May, 1977

Berteaux, H. O., Buoy Engineering, John Wiley and Sons, New York, 1976

Black, John, "The Single Point Mooring System", Marine Engineers Review, pp. 28-32, May, 1974

Borwick, G. R., and G. Elliot, "Dynamic Properties of Marine Fibre Ropes", 1972 Second Annual Tug Conference, Proceedings, Kenneth D. Troup, ed., Thomas Reed Industrial Press, Ltd., London, 1972

British Ropes, Ropes Made From Man-Made Fibers, 1970 Addendum, British Ropes Limited, London, 1970

British Ropes, Ropes Made from Man-Made Fiber, British Ropes Limited, London, 1966

British Standards Institution, Specification for Polyester Filament Ropes, B.S. 3758:1964, London, 1964

British Standards Institution, Specification for Polymid (Nylon) Filament Ropes, B.S. 3977:1968, London, 1968

British Standards Institution, Specification for Polypropylene Ropes, B.S. 4928:1973, London, 1973

Buckle, A. K., "Anchoring and Mooring Equipment on Ships", Royal Institution of Naval Architects, April 25, 1974

Celander, Ivar, "Preload Influence on Fatigue Characteristics of Chain Cable Exposed to Salt Water and Atmospheric Conditions", 1972 Offshore Technology Conference Proceedings, OTC 1578, pp. 645-647, 1972

Chislett, M. S., "Wind Simulation in Model Tests With Moored Offshore Tanker Terminals", OTC Paper 2740, 1977 Offshore Technology Conference Proceedings, pp. 109-117, 1977

H. Clarkson & Company, The Tanker Registry, London, (published yearly)

Coast Guard, Department of Transportation, Recommended Procedures For Developing Deepwater Ports Design Criteria, Deepwater Ports Project, Office of Marine Environment and Systems, Washington, D.C., 1975

Coast Guard, Department of Transportation, Guide to Preparation Of Environmental Analysis For Deepwater Ports, Deepwater Ports Project, Office of Marine Environment and Systems, Washington, D.C., 1975

Coast Guard, Department of Transportation, Environmental Impact Statement Deepwater Port Regulations, Deepwater Ports Project, Office Of Marine Environments and Systems, 1975

Coast Guard, Department of Transportation, "Deepwater Ports, Regulations on Licensing Procedures and Design Construction, Equipment and Operations Requirements", Federal Register, Part III, Vol. 40, No. 217, pg. 52540, Nov. 10, 1975

Coast Guard, Department of Transportation, "Deepwater Port Site Evaluation, Proposed Requirements", Federal Register, Part III, Vol. 40, No. 217, pg. 52518, Nov. 10, 1975

Coast Guard, Department of Transportation, Guidelines for Preparation of a Deepwater Port Operations Manual, Deepwater Ports Project, Office of Marine Environment and Systems, Washington, D.C., 1975

Comstock, J. P., ed., Principles of Naval Architecture, Society of Naval Architects and Marine Engineers, New York, 1967

D'Archangelo, A. M., ed., Ship Design and Construction, Society of Naval Architects and Marine Engineers, New York, 1969

Darbyshire, J., "The Generation of Waves by Wind", Proceedings Royal Society of London, Serial A, 1952 de Lavel, Gilbert, "Fatigue Tests on Anchor Chain-Cable", 1971 Offshore Technology Conference Proceedings, OTC 1503, pp. 855-858, 1971

Dickson, Capt. A. F., "Single Buoy Moorings", 16th A.P.I. Divisional Transportation Annual Tanker Conference, pp. 105-125, 1971

Dunn, B. J., and H. H. Hulley, "Ropes and Wire Ropes for Marine Towage", 1972 Second Annual Tug Conference, Proceedings, Kenneth D. Troup, ed., Thomas Reed Industrial Press, Ltd., London, 1972

E. I. duPont de Nemours and Co., Properties of Ropes of Dacron and DuPont Nylon, DuPont Technical Bulletin X-226, Wilmington, Deleware, February, 1969

E. I. duPont de Nemours and Co., Resistance of Fibers to Bleaching and Stripping Agents, DuPont Technical Bulletin X-226, Wilmington, Deleware

Feizy, G., and J. E. McDonald, "Pazarg--World's Largest Crude Oil Storage Barge", 1972 Offshore Technology Conference Proceedings, OTC 1526, pp. 183-190, 1972

Filson, John J., "Single-Point Mooring Units Analyzed", Oil and Gas Journal, pp. 47-51, Sept. 2, 1974

Flessner, M. F., C. D. Pike, S. S Weidenbaum, "Creep Tests on Synthetic Mooring Link", 7th Annual Conference of Marine Technology Society, August, 1971

Flory, John F., John Mascenik, and Kristen I. Pedersen, "The Single Anchor Leg Mooring", 1972 Offshore Technology Conference Proceedings, OTC 1644, pp. 259-266, 1972

Flory, J. F., "Verto-United Rope Works Large Diameter Hawsers", ER&E Trip Memorandum, October 27, 1976

Flory, J. F., "ARAMCO SPM Mooring Line Experience", ER&E Trip Memorandum, April 18, 1977

Flory, J. F., "Trip Report - Coordinated Equipment Co.", ER&E Trip Memorandum, January 10, 1976

Flory, J. F., "Trip Report- Tension Member Technology", ER&E Trip Memorandum, January 27, 1977

Flory, J. F., "Oil Ports on the Continental Shelf", Oceanus, Vol. 19, No. 1, Fall, 1965

Flory, J. F., and Peter F. Poranski, "The Design of Single Point Moorings", 1977 Offshore Technology Conference Proceedings, OTC 2824, 1977

Gault, John A., and William R. Cox, "Method for Predicting Geometry and Load Distribution in an Anchor Chain from A Single Point Mooring Buoy to a Buried Anchorage", 1974 Offshore Technology Conference Proceedings, OTC 2062, pp. 309-318, 1974

Gruy, Robert H., and W. L. Kiely, "The World's Largest Single Point Mooring Terminals: Design and Construction of the SALM System for 750,000 DWT Tankers", 1977 Offshore Technology Conference Proceedings, OTC 2826, 1977

Gumbel, Emil J., "Statistical Theory of Extreme Values and Some Practical Applications", National Bureau of Standards, Applied Mathematics Series, No. 33, 1953

Gumbel, Emil J., Statistics of Extremes, Columbia University Press, New York, 1958

Hamada, Tokvichi, Hisashi Mitsuyasu, and Maoki Hase, "An Experimental Study of Wind Effect Upon Water Surface", Transportation Technical Research Institute Report, Tokyo, June, 1953

Haring, R. E., R. C. Adams, R. A. Beazley, K. L. Kipp, "Application of Probability to the Design of Single-Point Moorings", Society of Petroleum Engineers Journal, pp. 219-227, September, 1970

Haring, R. E., R. B. Adams, R. A. Beazley, K. L. Kipp, "Design of Single-Point Mooring Systems for the Open Ocean", 1969 Offshore Technology Conference Proceedings, OTC 1022, pp. 211-222, 1969

Haring, R. E., "Single-Point Tanker Mooring Measurements in the North Sea", 1976 Offshore Technology Conference Proceedings, OTC 2711, pp. 1057-1065, 1976

Holt, Maurice, "Dimensional Analysis", Section 15 of Handbook of Fluid Mechanics, V.L. Streeter, ed., McGraw-Hill Book Company, New York, 1961

Hood, H. A., A New Look in Rope Making, The Parallel Yarn Approach, Wall Rope Works, Beverly, N.J., July, 1975

International Organization for Standardization, Ropes--Determination of Certain Physical and Mechanical Properties, International Standard 2307, 1972

Ippen, Arthur T., ed., Estuary and Coastline Hydrodynamics, McGraw-Hill Book Company, New York, 1966

Kiely, William L., K. I. Pedersen, and R. H. Gruy, "Design, Fabrication, Installation, and Operation of a Single Anchor Leg Mooring (SALM) Tanker Terminal in 300 Feet of Water", 1975 Offshore Technology Conference Proceedings, OTC 2213, pp. 703-713, 1975

Kinsman, Blair, Wind Waves, Their Generation and Propagation on the Ocean Surface, Prentice Hall, Inc. Englewood Cliffs, New Jersey, 1965

Langeveld, J. M., "Design Criteria for Single Point Mooring Terminals, "ASCE Journal of Waterways Harbors, and Coastal Engineering Division, WW4, pp. 305-323, November 1975

Langhaar, H. L., Dimensional Analysis and the Theory Of Models, John Wiley and Sons, New York, 1951

Laureti, R. J., Get the Most Out of Your Synthetic Ropes, Wall Product Bulletin PB15A, Wall Rope Works, New York, 1963

Lee, R. D., "Ju'aymah Crude Oil Loading Terminal", ASCE National Water Resources and Ocean Engineering Convention, Preprint 2642, 1976

Linehan, Joseph, David Ingamells, and David C. Vodden, "Instrumentation of 1000 Ton Nylon Single Point Mooring Hawser", 1975 Offshore Technology Conference Proceedings, OTC 2210, pp. 647-659, 1975

Longuet-Higgins, M. S., "On the Statistical Distribution of the Heights of Sea Waves", Journal of Marine Research, Vol. 11, No. 3, pp 345-366, 1952

Maari, Roger, "Design Aspects of Single Point Moorings", Ocean Industry Magazine, March, 1976, pp. 37-40

Maari, Roger, Offshore Mooring Terminals, (Layout Studies, Bazzoli), Monoco

MacKenzie, K. B., and G. C. Lee, "Design and Construction of Single Point Mooring Buoy Systems", ASME Petroleum Mechanical Engineering Conference, Los Angeles, 73-PET-38, 1973

Maddox, Notley Roger, "An Energy Basis for the Design of Open-Ocean Single Point Moorings", 1972 Offshore Technology Conference Proceedings, OTC 1536, pp. 255-266, 1972

McCormick, Michael E., Ocean Engineering Wave Mechanics, John Wiley & Sons, New York, 1973

Michel, W. H., "How to Calculate Wave Forces and Their Effects," Ocean Industry, June, 1967, pp 49-54

Muga, Bruce J., and James F. Wilson, Dynamic Analysis of Ocean Structures, Plenum Press, New York, 1970

Muga, Bruce J., and M. A. Freeman, "Computer Simulation of Single Point Moorings", 1977 Offshore Technology Conference Proceedings, OTC 2829, 1977

Munk, W. H., "Proposed Uniform Procedure for Observing Waves and Interpreting Instrument Records", Scripps Institute of Oceanography Wave Report No. 26, 1944

Myers, John J., Carl H. Holm, and R. F. McAllister, Handbook of Underwater Engineering, McGraw-Hill Book Company, New York, 1969

Neumann, Gerhard, "On Ocean Wave Spectra and a New Method of Forecasting Wind-Generated Sea", Technical Memorandum No. 43, Beach Erosion Board, U.S. Army Corps of Engineers, 1952

Neumann, Gehard, and Willard J. Person, Jr., Principles Of Physical Oceanography, Prentice-Hall Inc., Englewood Cliffs, New Jersey, 1966

Nisbet, A., and Strickland, J. C.,"Ropes For Offshore Services", 2nd International Offshore Craft Conference, (Date Unknown)

Nolte, K. G., and F. G. Hsu, "Statistics of Ocean Wave Groups", 1972 Offshore Technology Conference Proceedings, OTC 1688, pp. 637-644, 1972

Ocean Wave Spectra, Proceedings of the conference sponsored by U.S. Naval Oceanographic Office and National Academy of Science, Prentice-Hall, Englewood Cliffs, New Jersey, 1963

Oil Companies International Marine Forum, Prediction Of Wind and Current Loads on VCCs, OCIMF, London, 1977

Owen, D. G., and B. T. Linfoot, "The Development of Mathematical Models of Single Point Mooring Installations", 1976 Offshore Technology Conference Proceedings, OTC 2490, pp. 689-700, 1976

Pheonix, S. L., "Probabilistic Theories of Time Dependent Failure of Fiber Bundles", Oceans '76 Conference, 1976

Pierson, W. J. Jr., "A Unified Mathematical theory for the Analysis, Propagation and Refraction of Storm Generated Ocean Surface Waves", Parts I and II. N.Y.U. College of Engineering Research Division, Dept. of Meteorology and Oceanography, New York, N.Y., 1956

Public Law 93-627 "Deepwater Port Act of 1974", 93rd Congress, H.R. 10701 January 3, 1975

Putz, R. R., "Wave Height Variability; Prediction of Distribution Function", Univ. of California, Berkeley, Calif. Tech. Rep. No. HE 116-318 Series No. 3, Issue No. 318, 1950

Putz, R. R., "The Generation of Waves by Wind", Proceedings Royal Society of London, Serial A, 1952

Raleigh, Lord, "On the Resultant of a Large Number of Vibrations of the Same Pitch and of Arbitrary Phase", Phil. Mag., Series 5, Vol. 10 (60), 1880

Remery, G.F.M., and A.J. Hermans, "The Slow Drift Oscillations of a Moored Object in Random Seas", 1971 Offshore Technology Conference Proceedings, OTC 1500, pp. 829-838, 1971

Remery, G.F.M., and G. van Oortmerssen, "The Mean Wave, Wind and Current Forces on Offshore Structures and their Role in the Design of Mooring Systems", 1971 Offshore Technology Conference Proceedings, OTC 1741, pp. 169-184, 1973

Remery, G.F.M., and R. Kokkeel, "On the Origin of Wave Induced Motions of Mooring Buoys", 1976 Offshore Technology Conference Proceedings, OTC 2468, pp. 479-500, 1976

Rendle, Capt. J.D., "Tankers and Buoy Moorings", Shell International Marine, Ltd.

Rye, Henrik, Stein Rynning, and Hermann Moshagen, "On the Slow Drift Oscillations of Moored Structures", 1975 Offshore Technology Conference Proceedings, OTC 2366, pp. 299-310, 1975

Samson Cordage Works, Shirley Mass., Samson Rope Manual, 1973 edition

Samson Cordage Works, Shirley, Mass., "Comparative Cycling Tests for Esso Engineering", Project No. Tech Report, Report No. 6/69, Feb. 18, 1969

Samson Cordage Works, Shirley, Mass., "Tests on 2-in-1 Nylon Ropes at -65 Degrees F", Project No. RFD #251, Report No. TR 20-69, July 2, 1969

Samson Cordage Works, Shirley Mass., "5 1/2-Inch Circ. Nylon Model of Conoco 21" Whipped Loop Sling with Thimbles", Project No. RFD #412, Report No. TR 12-74, September 3, 1974

Samson Cordage Works, Shirley Mass., "Load-Elongation Data for Various Samson and Competitors' Ropes", Project No. RFD #380, Report No. TR 6-74, June 7, 1974

Seiwell, H. R., "Results of Research On Surface Waves of The Western North Atlantic", Papers in Physical Oceanography and Meteorology, Vol. 10, No. 4, pp 30-51, 1948

Smith, A. W., Ropes Made From Man-Made Fibers (Addendum), British Ropes, Limited, London, Publication 555, Addendum, 1970

Smith, A. W., Ropes Made From Man-Made Fibers, British Ropes Limited, London, Publication 555, 1966

Stoker, J. J., Water Waves, Interscience Publishers, New York, 1957

Streeter, Victor, Fluid Mechanics, McGraw-Hill Book Company, New York, 1966

Sverdrup, H. V. and W. H. Munk, Wind, Sea and Swell; Theory of Relations for Forecasting Hydrographic Office Publication No. 603, U.S. Navy Hydrographic Office, Washington, D.C., 1947

Synodis, Steve T., and John F. Flory, "Six Years Experience with the Brega Single Anchor Leg Mooring", 1977 Offshore Technology Conference Proceedings, OTC 2824, 1977

U.S. Army Coastal Engineering Research Center, Shore Protection Manual, Fort Belvoir, Virginia, 1973

Van Houten Associates, Report on Design of Single-Buoy Mooring Terminals, Department of the Army, Office of the Chief of Engineers, Washington, D.C., 1976

Vanderbilt, Hendrikcis H., and Wuang Tran, "Acoustic Emissions from Synthetic Rope", Naval Engineers Journal, December, 1971

Vellozzi, Joseph, and Edward Cohen, "Gust Response Factors", ASCE Structural Division, ST6, pp. 1295-1313, June 1968

"Veritas Establishes Offshore Loading Rules", Offshore, pp. 129-131, December 1976

Watkins, M. T., Strength Retenion of Used Synthetic Tails, for Exxon Research and Engineering Company by National Engineering Laboratory, East Kilbride, Glascow, Scotland, March, 1976

Wichers, J.E.W., "On the Slow Motions of Tankers Moored to Single Point Mooring Systems", 1976 Offshore Technology Conference Proceedings, OTC 2548, pp. 283-300, 1976

Wiegel, R. L., "An Analysis of Data Wave Recorders On the Pacific Coast of the United States", Transactions American Geophysical Union, Vol. 30, No. 5 pp 700-704, 1949

Wiegel, Robert L, Oceanographical Engineering, Prentice-Hall, Englewood Cliffs, New Jersey, 1964

Wilbourn, J. Phil, "SPM Model Testing for Proposed U.S. Superports", ASCE National Water Resources and Ocean Engineering Convention, San Diego, Preprint 2685, 1976

Wills, A. E., "Tow Ropes", 1972 Second Annual Tug Conference Proceedings, Kenneth D. Troup, ed., Thomas Reed Industrial Press, Ltd., London, 1972

Wilson, James F., and Nabil G. Awadalla, "Computer Simulation of Nonlinear Motions of Moored Ships", NATO Advanced Study Institute on Analytical Treatment of Problems in the Berthing and Mooring of Ships, Walingford, England, May, 1973

Woehleke, S. P., "Meeting With DuPont", ER&E Trip Memorandum, January 27, 1977

Woehleke, S. P., "Meeting With Wall Rope Works", ER&E Meeting Report, February 2, 1977

Woehleke, S. P., "Visit to British Ropes Ltd.," ER&E Trip Memorandum, December 10, 1976

Woehleke, S. P., "Visit to Hawkins and Tipson Ropemakers", ER&E Trip Memorandum, December 10, 1976

APPENDIX E

GLOSSARY

Aft Perpendicular	A vertical line passing through the rudder post.
Anchor Windlass	A machine mounted on the forecastle of a ship for deck hoisting or lowering the ship's anchor.
Ballasted Tanker	A tanker empty of cargo and partially filled with ballast. More generally a tanker having less than roughly one half of its loaded capacity of cargo or ballast onboard.
Bitts	A pair of vertical steel posts mounted on a baseplate on the deck of a vessel and used to secure lines and ropes.
Braided Rope	Rope constructed by braiding or interwearing strands together.
Bollard	A single vertical post mounted on a pier and used to secure lines and ropes. Also another name for a bitt.
Bow Chock	A large chock mounted at or near the vessel centerline at the bow.
Breaking Length	A measure of the strength of a rope in relation to its weight per unit length; the length of rope having a weight equal to the breaking load of the rope; the length of rope which can be hung free without breaking due to its own weight.
Breaking Strength	The axial load applied to a rope which results in rupture.
Bulbous Bow	A special bow design having a bulb-shaped protrusion extending forward beneath the water line.
Carrier	The bobbin or spool on which the strand or group of strands are carried on the ropemaking machine during the braiding or twisting of the rope; also sometimes used to refer to the strand or the group of strands which are wound on a carrier.

Catenary Anchor Leg Mooring (CALM)	A single point mooring system in which the vessel is moored to a buoy which is anchored by multiple anchor legs extending in catenaires to anchor points on the sea floor some distance from the buoy. Generally the mooring swivel is mounted on the deck of the buoy and the cargo swivel is mounted concentric with and above the mooring swivel.
Chafing Chain	A short length of chain attached to the end of the hawser assembly and connected to a strong point on the SPM or the moored vessel, intended to reduce chafing against the deck and fairleads.
Chain Stopper	Two parallel vertical plates mounted on the vessel deck with a pivoted bar or pawl which drops down to bear on the exterior of a chain link to secure the chafing chain.
Chain-Support Buoy	A buoy used to support the chafing chain at the tanker end of an SPM hawser when tankers are not moored.
Chance of Exceedance	The probability that a given value will be exceeded in a defined duration or number of occurrences.
Chance of Non-Exceedance	The probability that a given value will not be exceeded in a defined duration or number of occurences.
Cycle	The distance parallel to the axis of the rope in which a strand makes one complete spiral, same as lay length and pitch.
Cylindrical Bow	A special ship-bow design having a rounded shape that is essentially constant with depth and lacks a bulb-shaped projection.
Deadweight Tonnage (dwt)	The weight of a full load of cargo together with all fuel, water, stores, and supplies.
Denier	A measure of size of a fiber; the weight in grams of 9,000 meters of filament, the lower the denier the finer the yarn.
Design Environment	The maximum combination of waves, wind, and current in which a vessel may remain moored without exceeding the design load.

Design Load	The maximum load which should be applied to a rope consistant with the factor of safety for the type of rope use.
Double-Braid Rope	A rope consisting of a hollow core of many braided strands enclosed in a cover of many braided strands.
Eight-Strand Rope	A rope consisting of two pairs of strands twisted to the right and two pairs of strands twisted to the left and braided together such that pairs of strands of opposite twist alternately overlay one-on-another.
End	The largest twisted sub-component of the rope, same as strand.
Factor of Safety	The ratio of the rated breaking strength of a new rope to the design load.
Fairleads	Fittings used to change the direction of lead of a rope or line on the deck of a vessel. Usually a vertically mounted roller. Also used to refer to chocks.
Fiber	A single filament or group of filaments furnished on bobbins or spools by the fiber manufacturer.
Fid	A tapered pin, sometimes hollow at one end, used to separate the strands of a rope and to insert the ends of strands when splicing rope.
Filament	The smallest unit or component of the rope, a single extruded element of synthetic material.
Film	A generally flat extruded element whose width is much greater than its thickness, where the thickness is usually less than 0.25 mm (0.01 in).
Forecastle	The deck near the bow of a ship, usually one level above the main deck.
Forward Perpendicular	A vertical line passing through the foremost point of the bow at the loaded-water line.

Froude's Law Scaling	Dynamic similitude modeling in which the ratio of the square of velocity to the product of the gravitational constant and a length parameter is kept constant, generally used for SPM model testing and other studies where gravitational effects predominate.
Grommet	An endless rope, spliced end-to-end at one or more places, sometimes with thimbles at each end to form a hawser of two parallel ropes. Same as strop.
Gypsy Head	A cylinder-like fitting on the end of a winch or windlass shaft used to haul or slack ropes or hawsers by winding a few turns of the line around it and holding the free end of the line taut as the gypsy head turns.
Hawser	The mooring rope between an SPM and a moored vessel. Generally any large rope 40 mm or more in diameter (greater than 5 inches circumference).
Hawser Assembly	The assembly, including the hawser, thimbles, chafing chain, and floatation, used to moor a vessel to an SPM.
Hawser Eye	A loop at the end of a hawser or strop used for connection purposes.
Hawser Float	A floatation device attached to a polyester or nylon hawser to support the hawser when tankers are not moored.
Hawser-Lay Rope	A three-strand rope in which the direction of twist of the yarns making up the strands is opposite the direction of twist of the strands. More generally any three-strand rope.
Heave	Vertical translational movement of a vessel.
Hockle	A defect in a rope in which strands are kinked or twisted contrary to the direction of their normal lay. Usually caused by improper handling of the rope. Resembles a knot in the rope.
Irregular Waves	Ocean waves made up of many waves of different frequencies. Generally non-sinusoidal waves.

Lashing	Wrapping or binding the two legs of a strop together with small rope. Also wrapping a rope with smaller rope to prevent abrasion or to secure or prevent a collar from slipping.
Lay	The direction in which the strands of a rope are twisted.
Lay Length	The length along the axis of a rope in which a strand makes one complete spiral around the axis, same as cycle and pitch.
Left-Hand Lay	Twisted as a left-hand screw thread, with twist in direction of fingers of fist of left hand will advance in direction of thumb.
Length Between Perpendiculars	The distance between the forward and aft perpendiculars.
Line	Stranded assembly made from metallic wires. Also used to refer to synthetic rope especially small rope.
Loaded Tanker	A tanker full or almost full of its capacity of cargo.
Manifold	A cargo hose or piping connection point on a tanker; usually located on the main deck at midships. On an SPM normally located at the end of the under-water pipeline and may also be located on a buoy, base, or mooring tower.
Maximum Operating Environment	The maximum combination of waves, wind, and current in which a vessel may remain moored without exceeding the design load.
Maximum Wave Height	The maximum difference in elevation between successive wave troughs and wave crests. Alternatively, the maximum crest elevation above the still water level.
Messenger Line	A small rope lowered from the vessel to a launch for the purpose of raising the pickup rope or hawser to the vessel's deck.
Monofilament	A coarse circular filament, in excess of 25 to 50 denier.

Mooring Bracket	Two parallel vertical plates mounted on the vessel deck with a sliding bolt passing through the plates which is used to secure the end link of a chafing chain.
Mooring Tower	A single point mooring system in which the vessel is moored to a rigid or semi-rigid tower extending above the sea surface and rigidly secured to the sea floor. Generally, the mooring swivel is mounted on the upper deck of the tower and the cargo swivel is mounted concentric with and above the mooring swivel.
Multifilament	A fine filament less than 25 to 50 denier per filament.
Nine-Strand Rope	A rope consisting of three 3-strand ropes twisted together in a lay opposite to the direction of lay of the 3-strand rope.
Nylon	Generic name for long-chain polymeric amide molecules in which recurring amide groups are part of the main polymer chain.
Nylon 6.6	A long-chain polymer of hexamethylene diamine and adipic acid.
Nylon 6.0	A long-chain polymer of epsilon amino caproic acid.
Open Link	A chain without a stud, normally placed on thimbles and the ends of chafing chains to allow the components to be shackled.
Overlay Finish	Coating or process applied to synthetic fibers to improve fiber properties.
Panama Chock	A single large chock mounted in the center of the bow.
Picks per inch	A count of the number of strands on the surface of the rope in a distance of one inch parallel to the rope axis.
Pickup Rope	A rope, usually smaller than the hawser, attached to the vessel end of the hawser assembly for the purpose of raising the hawser assembly to the vessel forecastle and drawing it through the chocks.

Pitch	Rotational movement of a vessel about the transverse centerline.
Pitch	The length along the axis of a twisted rope in which one-strand makes one complete spiral around the axis, same as lay length and cycle.
Plaited Rope	Same as eight-strand rope.
Polyamide	Another name for nylon.
Polyester	A thermosetting synthetic resin made by esterification of polybasic organic acids with polyhydric acids.
Polyethylene	A thermo-plastic material composed of polymers of ethylene.
Polypropylene	A crystalline, thermo-plastic resin made by the polymerization of propylene, C_3H_6.
Proof Load	A test load applied to a chain to prove its quality and ability to take load, normally approximately two-thirds of the breaking strength of the chain.
Realization Factor	The ratio of the breaking strength of a rope to the product of the strength of an individual strand or yarn times the number of strands or yarns in the rope. Sometimes used by manufacturers and classification societies to determine the strength of a rope from tests conducted on sample strands.
Reference Load, P	A low-level load applied to samples of ropes to remove sag and slack when measuring length, circumference, and certain other properties; frequently the load in kilonewtons equivalent to 29 times the diameter in millimeters squared, $P\ (kN) = 29\ D^2\ (mm^2)\ [P\ (lb) = 200\ D^2\ (in^2)]$.
Right-Hand Lay	Twisted as right-hand screw thread, with twist in direction of fingers of fist of right hand with advance in direction of thumb.

Roll	Rotational movement of a vessel about the longitudinal centerline.
Roller Chock	A steel frame mounted at the edge of the deck with two or three vertical steel rollers designed for use with wire lines alongside piers.
Rope	Strand assembly made from natural or synthetic fibers. (Also used to refer to wire lines.)
'S' Lay	Another name for left-hand lay.
Scale, $1/\lambda$	The fractional ratio of the dimension of the model divided by the dimension of the prototype, the reciprocal of scale factor.
Scale factor, λ	The ratio of the dimension of the prototype to the dimension of the model, unless otherwise specified, the dimension is normally length.
Seas	Wind driven waves, more generally any short or intermediate period generally irreglar waves which are not swell.
Seizing	Wrapping a small rope around a large rope to protect against damage, to prevent a splice from working loose, or to prevent an accessory, such as a float, from slipping along the rope.
Sewing	Passing a small rope through the structure of a large rope to secure a splice from working loose, or to secure an accessory, such as a float, from slipping along the rope.
Shackle	A fitting used as a connecting link between SPM hawsers and chafing chains, usually a "D" shaped fitting with a removable pin forming the straight leg of the "D".
Siech	The resonant oscillation of water back and forth across a basin.
Significant Mooring Force, F_S	The average of the highest one-third of the peaks in a mooring force record.
Significant period, T_S	The average of the periods of the highest one-third of the waves in a wave record.

Significant wave height, H_S	The average of the highest one-third of the peak-to-trough waves in a wave record.
Single Anchor Leg Mooring (SALM)	A single point mooring system in which the vessel is moored to a buoy which is anchored by a single anchor leg to a base or anchor on the sea floor. Generally, the anchor leg is maintained under tension by the buoy, the mooring swivel forms a part of the anchor leg, and the cargo swivel is concentric with the anchor leg or its point of attachment to either the base or the buoy.
Single Buoy Mooring (SBM)	A single point mooring in which the vessel is moored to a single buoy at the sea surface. generally another name for single point mooring.
Single Point Mooring (SPM)	A mooring and cargo transfer system for vessels including a mooring swivel and a cargo transfer swivel, in which at one point either the cargo transfer swivel is concentric with the mooring-load-carrying system or the mooring swivel is concentric with the cargo transfer system such that the moored vessel may swing completely around the mooring point while transferring cargo.
Smit Bracket	Another name for mooring bracket.
Snotter	A short line or grommet made of wire or synthetic rope used to connect chafing chain to bitts on the vessel deck; if a grommet, it normally has no thimbles; if a single line, it normally has a thimbled eye at one end and an unthimbled eye at the other end.
Strand	A component of rope composed of several twisted yarns; the direction of twist of strands is usually opposite to that of the yarns.
Stiching	Same as sewing.
Stitch Length	The length along the axis of a braided rope in which one yarn or strand makes one complete spiral around the axis.
Strop	Another name for grommet.

Surge	Longitudinal (fore and aft) translational movement of a vessel.
Sway	Transverse (sideways) translational movement of a vessel.
Swell	Waves which are no longer driven by wind, more generally any long period, generally regular waves.
Tenacity	A measure of unit breaking stress, the breaking strength of the fiber in grams divided by the denier.
Terylene	Another name for polyester.
Tex	A unit of fiber fineness assessed by the weight in grams of 1000 meters of yarn; the lower the number the finer the yarn, 1 tex = 9 denier.
Thimble	A circular or pear-shaped barrel with a contoured groove on its exterior to fit the eye of a rope, usually with a hole through its center for a shackle pin, and usually with covers around part or most of the groove to protect the rope.
Thimble Encapsulation	Material placed between SPM hawsers and thimbles to prevent hawser abrasion.
Three-Strand Rope	A rope consisting of three strands twisted together in a spiral pattern.
Towing Bracket	Another name for mooring bracket.
Turn	Another name for cycle.
Very Large Crude Carrier (VLCC)	A tanker larger than about 150,000 dwt, sometimes 400,000 dwt is defined as an upper limit to this tanker size classs with tanker larger than 400,000 dwt, referred to as ultra-large crude carriers (ULCC).
Wave Amplitude	Generally the elevation of the wave crest above the still water level.

Wave Grouping	A tendency of irregular waves to vary in amplitude in a periodic or non-random manner.
Wave Height	Generally the difference in elevation between the lowest point on a wave (the wave trough) and the highest point on a wave (the wave crest). For irregular waves, several different wave heights having different probabilities of occurrence may be defined. See significant wave height, maximum wave height.
Wave Spectrum	The distribution of wave energy with wave frequency, the power spectrum of the waves.
Wide-Beam Tanker	A tanker having a relatively shallow loaded draft and relatively wide beam in relation to its cargo capacity, generally a tanker with a length to beam ratio of approximately 5.
Yarn	Next to the smallest component of rope, composed of twisted fibers or filaments.
Yaw	Rotational movement of a vessel about the vertical axis passing through the transverse and longitudinal centerlines.
'Z' Lay	Another name for right-hand lay.

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