Mooring Equipment Guidelines

Second Edition 1997

Oil Companies International Marine Forum

WITHERBY

OCIMF

Mooring Equipment Guidelines

(Second Edition – 1997)

The OCIMF mission is to be recognised internationally as the foremost authority on the safe and environmentally responsible operation of oil tankers and terminals.

Oil Companies International Marine Forum

Issued by the
Oil Companies International Marine Forum
First Published 1992
Second Edition 1997

ISBN 1 85609 088 4

The **Oil Companies International Marine Forum** (OCIMF) is a voluntary association of oil companies having an interest in the shipment and terminalling of crude oil and oil products. OCIMF is organised to represent its membership before, and consult with, the International Maritime Organization and other governmental bodies on matters relating to the shipment and terminalling of crude oil and oil products, including marine pollution and safety.

British Library Cataloguing in Publication Data
Mooring Equipment Guidelines.
I. Oil Companies International Marine Forum
627.98
ISBN 1 85609 088 4

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Printed & Published by
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London EC1R 0ET. England

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Glossary of Terms and Abbreviations

BIGHT	A loop formed by doubling back a rope upon itself.
BITTS	Vertical steel posts or bollards mounted in pairs around which a line can be secured.
BOLLARD	A vertical post to which the eye of a mooring line can be attached.
BREAST LINES	Mooring lines leading ashore as nearly perpendicular as possible to the ship's fore and aft line.
BS	British Standard.
BSI	British Standard Institution.
CHAIN STOPPER	A fitting for securing a chain, consisting of two parallel vertical plates mounted on a base, with a pivoting bar or pawl which drops down to bear on a chain link.
CHOCK	A guide for a mooring line which enables the line to be passed through a ship's bulwark or other barrier (See also FAIRLEAD).
DEADWEIGHT-DWT	The carrying capacity of a ship, including cargo, bunkers and stores, in metric tonnes. Strictly speaking, it can be given for any draft, but here it is used to indicate summer deadweight at summer draft.
ELASTICITY	The elastic (non permanent) elongation of a unit length of an element caused by a unit load. May refer to a material or a composite structure such as a mooring line.
ELONGATION	Refers here to the total elongation (elastic and plastic) of a line.
FLAKE DOWN	Laying a rope in long bights on the deck with each bight clear of the adjacent one, so that it can be paid out quickly and free from turns.
FAIRLEAD	A guide for a mooring line which enables the line to be passed through a ship's bulwark or other barrier, or to change direction through a congested area without snagging or fouling.
FLEET ANGLE	The angle between the mooring line and a plane perpendicular to the axis of the winch drum.
FIRE WIRE	A wire rigged to the waterline over the off-berth side of a ship to facilitate towing off in an emergency.
FIRST-ASHORE LINE	A line (usually fibre) put ashore first to help in hauling the ship into berth.
HAWSER	Synthetic or natural fibre rope or wire rope used for mooring, warping and towing.
HEAD LINES	Mooring lines leading ashore from the fore end or forecastle of a ship, often at an angle of about 45 degrees to the fore and aft line.
HEAVING LINE	A very light line that is thrown between the ship and the berth, and is used to draw the messenger line ashore.
HOCKLE	A knot-like twisting of individual strands of a twisted rope.
HTS	High Tensile Steel.

IMO	International Maritime Organization.
INDEPENDENT WIRE ROPE CORE (IWRC)	A type of construction of wire rope.
ISO	International Organization for Standardization.
JIS	Japanese Industrial Standard.
kDWT	Deadweight in thousands of tonnes.
LEAD	The direction a mooring line takes up whilst being handled or when made fast.
LENGTH BETWEEN PERPENDICULARS (LBP)	The length of a ship, generally between the stem at the design loadline and the centre of the rudder stock.
LENGTH OVERALL (LOA)	The extreme length of a ship.
LIGHTENING OR LIGHTERING	The process of transferring cargo from a tanker to another ship.
LOADING ARMS	Oil transfer units between ship and shore for discharge and loading; may be articulated all-metal arms (hard arms) or a combination of metal arms and hoses.
MANDEL SHACKLE	A special shackle used to connect a wire mooring line to a synthetic tail.
MESSENGER LINES	A light line attached to the end of a main mooring line and used to assist in heaving the mooring to the shore or to another ship.
MINIMUM BREAKING LOAD (MBL)	The minimum breaking load of a mooring line as declared by the manufacturer for a new line. It does not include allowance for splicing or for wear and tear.
MOORING RESTRAINT	The capability of a mooring system to resist external forces on the ship.
MULTI-BUOY MOORINGS (MBM)	A facility whereby a tanker is usually moored by a combination of the ship's anchors forward and mooring buoys aft and held on a fixed heading. Also called conventional buoy moorings (CBM).
N	Newton (unit of force); 1kN = 1000 N.
PANAMA TYPE FAIRLEAD	A non-roller type fairlead mounted at the ship's side and enclosed so that mooring lines may be led to shore with equal facility either above or below the horizontal. Strictly pertains only to fairleads complying with Panama Canal Regulations, but often applied to any closed fairlead or chock.
PEDESTAL ROLLER FAIRLEAD	A roller fairlead usually operating in a horizontal plane. Its purpose is to change the direction of lead of a mooring or other line on a ship's deck.
PRE-TENSION	Additional load applied to a mooring line by a powered winch over and above that required to remove sag from the main run of the line.
SAFE WORKING LOAD (SWL)	A load less than the yield or breaking load by a safety factor defined by a code, standard or good engineering practice.
SEA ISLAND	A pier structure with no direct connection to the shore, at which tankers can berth. Berthing can take place either on one or both sides of the structure.
SEICHE	Very long waves of small height generated by resonant oscillation within a partly closed harbour or other body of water. Strong horizontal currents can also be set up which may cause ship surging in adverse circumstances.

SHIP-TO-SHIP TRANSFER OPERATIONS (STS)	Transfer of crude oil or petroleum product between two ocean-going ships made fast alongside at anchor or underway. The transfer of petroleum to barges and estuarial craft is specifically excluded.
SINGLE POINT MOORING (SPM)	A facility whereby the tanker is secured by the bow to a single buoy or structure and is free to swing with the prevailing wind and current.
SMIT BRACKET	A fitting for securing the end link of a chafing chain, consisting of two parallel vertical plates mounted on a base with a sliding bolt passing through the plates.
SPRING LINES	Mooring lines leading in a nearly fore and aft direction, the purpose of which are to prevent longitudinal movement (surge) of the ship while in berth. Headsprings prevent forward motion and backsprings aft motion.
STERN LINES	Mooring lines leading ashore from the after end or poop of a ship, often at an angle of about 45 degrees to the fore and aft line.
STOPPER	A device for securing a mooring line temporarily at the ship whilst the free end is made fast to a ship's bitt. A Carpenter's stopper is a device with opening jaws to receive wire and shaped wedges to hold line when tension is applied.
STS	See SHIP-TO-SHIP TRANSFER OPERATIONS (STS) above.
SUMMER DEADWEIGHT	The deadweight of a ship when loaded to summer marks.
SUMMER MARKS	The summer loadline mark and centre of Plimsoll disc on the ship's side.
t	Metric ton(s) or tonne(s); unit of mass, but often also used for forces (sometimes expressed as 'tf'); 1tf = 9.81kN.
TAIL	A short length of synthetic rope attached to the end of a wire mooring line to provide increased elasticity and also ease of handling.
VERY LARGE CRUDE CARRIER (VLCC)	For purposes of these guidelines, used to describe ships with deadweight between 140,000 tonnes and 400,000 tonnes.
ULTRA LARGE CRUDE CARRIER (ULCC)	In these guidelines, used for ships with a deadweight greater than 400,000 tonnes.
UNIVERSAL FAIRLEAD	A fairlead with three or more cylindrical rollers.

Bibliography

Reference 1	OCIMF publication "Recommendations for Equipment Employed in the Mooring of Ships at Single Point Moorings".
Reference 2	OCIMF publication "Recommendations for Oil Tanker Manifolds and Associated Equipment".
Reference 3	OCIMF publication "Prediction of Wind and Current Loads on VLCCs".
Reference 4	OCIMF/ICS publication "Ship to Ship Transfer Guide (Petroleum)".
Reference 5	OCIMF/SIGTTO publication "Prediction of Wind Loads on Large Liquefied Gas Carriers".
Reference 6	OCIMF/ICS/IAPH publication "International Safety Guide for Oil Tankers and Terminals".
Reference 7	Marsh, F.W. and Thurston, R.C.A., "Investigation of Stress Distribution in Stud Link Anchor Cable", Report IR61-46, Department of Mines and Technical Surveys, Ottawa, Canada.
Reference 8	Buckle, A. K., "Anchoring and Mooring Equipment on Ships", 1974, Royal Institute of Naval Architects.
Reference 9	OPTIMOOR Mooring Analysis Computer Program Users Guide (1997)

Introduction

Numerous standards, guidelines and recommendations concerning mooring practices, mooring fittings and mooring equipment exist throughout the worldwide marine industry. These various requirements lack uniformity, and vital information, such as strength of fittings, is often missing. For example, Classification Society rules provide empirically-based tables for hawsers as a guide, but not as a condition of class. These are derived from an equipment numeral obtained using displacement and dimensional criteria. Winches are required only to conform to an applicable national standard or code of practice. Very little guidance is provided in the classification rules on the design and installation of bitts, bollards, fairleads or chocks. No information is given on safe working loads, strength of materials, structural reinforcement, foundations or method of attachment of these fittings or mooring winches. Where guidance is given, it often falls short: the inconsistency in recommending a number of hawsers and their breaking strength without any advice on mooring winch pulling force or brake holding capacity is one instance.

The result is that a wide variation in the number of mooring lines, type of mooring equipment, capacity of winches, etc., can be found. At the one extreme, moorings may even be inherently unsafe for certain environmental conditions. Many tanker loading berths are situated at exposed locations and existing mooring equipment on some ships may not permit their maintaining a safe moor at these berths during all prevailing environmental conditions. At the other extreme, an incomplete understanding of mooring requirements could equally well result in a system which is over-designed and unnecessarily expensive for the purpose intended.

The shipping industry has always been concerned with safe mooring practices. A fundamental aspect of this concern entails the development of mooring systems which are adequate for the intended service, with maximum integration of standards across the range of tanker types and sizes. To further this, the Oil Companies International Marine Forum set up a task group to investigate current standards and usage and to develop and promote guidance for the safe and efficient design and operation of mooring equipment.

The International Association of Independent Tanker Owners (INTERTANKO), the International Chamber of Shipping (ICS) and the International Association of Classification Societies (IACS) have participated in the task group activity and from this broadly-based industry group a set of criteria has been developed which, it is hoped, will find wide industry acceptance.

While the study initially focussed on tankers, it was soon realised that many of the principles adopted and the equipment strength and installation criteria proposed could be applied to many other types of ship moorings. Hence the task force adopted the convention that where the contents of these guidelines were applicable to a range of ships, the term ship is employed and, where the provisions were specific to a ship type, the terms 'tanker' or 'gas carrier' have been used.

In 1996 an editorial task force updated the original work in the light of new coefficients in the OCIMF publication "Prediction of Wind and Current Loads on VLCC's" 1994. Opportunity has also been taken to incorporate updated extracts from the "Guidelines and Recommendations for the Safe Mooring of Large Ships at Piers and Sea Islands" first published in 1978, and OCIMF Recommendations on Equipment for The Towing of Disabled Tankers, 1981, which are now no longer in print.

These guidelines represent best known mooring technology and practice. It is recognized that it may not always be practical to retrofit all aspects of this technology to existing mooring systems. For existing ships, where the mooring arrangement does not meet the recommendations described in these guidelines, both ship and terminal operators should be made aware of the limitations of the mooring system and contingency plans drawn up to deal with them. The contingency plans should include (but not be limited to) predetermined environmental limits for berthing, stoppage of cargo loading or unloading, and departure from the berth.

This publication attempts to refine, unify and update selected existing guidelines and to add essential information which has either been omitted or poorly defined. Care has been taken to ensure that design performance of equipment is optimised, while not overlooking the equally important factors of ease of handling and safety of personnel. These guidelines represent a recommended minimum requirement, and are intended to be useful to ship designers and surveyors as well as ship and terminal operators. They are not intended to inhibit innovation or future technological advances.

Section 1.0

Principles of Mooring

1.1 GENERAL

The term "mooring" refers to the system for securing a ship to a terminal. The most common terminals for tankers are piers and sea islands, however, other shipboard operations such as mooring at Single Point Moorings (SPM's), Multi-Buoy Moorings (MBM's), emergency towing, tug handling, barge mooring, canal transit, lightening and anchoring may fall into the broad category of mooring and thus require specialised fittings or equipment. Anchoring equipment is covered by Classification Society rules and is therefore not included in these guidelines.

Figure 1.1 shows a typical mooring pattern at a tanker terminal.

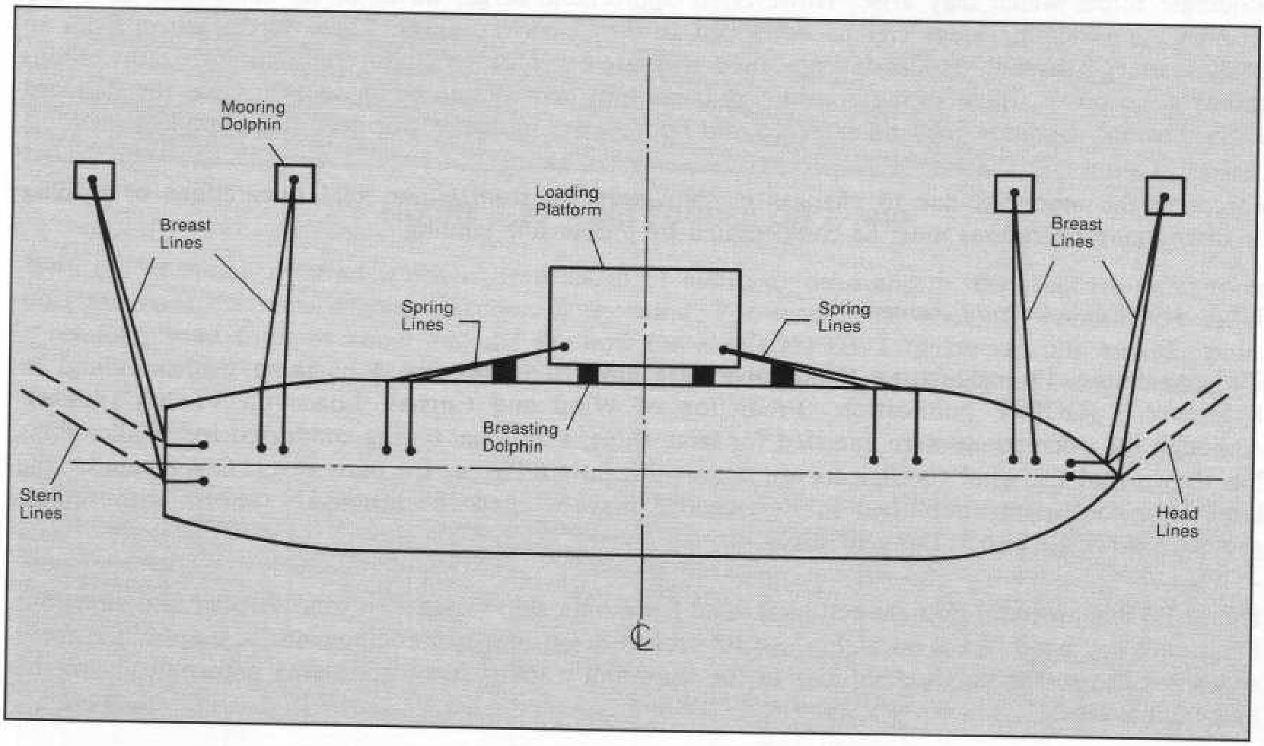


FIGURE 1.1: TYPICAL MOORING PATTERN

The use of an efficient mooring system is essential for the safety of the ship, her crew, the terminal and the environment. The problem of how to optimise the moorings to resist the various forces will be dealt with by answering the following questions:

- What are the forces applied on the ship?
- What general principles determine how the applied forces are distributed to the mooring lines?
- How can the above principles be applied in establishing a good mooring arrangement?

Since no mooring outfit has unlimited capability, in order to address these questions it will be necessary to understand precisely what the moorings of a ship are expected to achieve.

1.2 FORCES ACTING ON THE SHIP

The moorings of a ship must resist the forces due to some, or possibly all, of the following factors:

- Wind
- Current
- Tides
- Surges from passing ships
- Waves/Swell/Seiche
- Ice
- Draft changes

This Section deals mainly with the development of a mooring system to resist wind, current and tidal forces on a ship at a conventional berth. Normally, if the mooring arrangement is designed to accommodate maximum wind and current forces, reserve strength will be sufficient to resist other moderate forces which may arise. However, if appreciable surge, waves or ice conditions exist at a terminal, considerable loads can be developed in the ship's moorings. These forces are difficult to analyse except through model testing, field measurements or dynamic computer programs. Ships calling at terminals where such extraordinary conditions exist should be made aware that the standard environmental condition may be exceeded and appropriate measures will need to be implemented.

Forces in the moorings due to changes in ship elevation from either tidal fluctuations or loading or discharging operations must be compensated by proper line tending.

1.2.1 *Wind and Current Forces*

The procedures for calculating these forces are covered in Section 2 of these guidelines and in Reference 3 (OCIMF publication "Prediction of Wind and Current Loads on VLCCs", 1994). Although the calculations were intended for large ships, additional testing conducted for smaller ships has shown that the wind coefficients are not significantly different for most cases. Consequently, the large ship coefficients published in Reference 3 may be used for bridge-aft tankers with similar geometry down to 16,000 DWT in size.

Figure 1.2 demonstrates how the resultant wind force on a ship varies with wind velocity and direction. For simplicity, wind forces on a ship can be broken down into two components: a longitudinal force acting parallel to the longitudinal axis of the ship, and a transverse force acting perpendicular to the longitudinal axis.

Wind force on the ship also varies with the exposed area of the ship. Since a head wind only strikes a small portion of the total exposed area of the tanker, the longitudinal force is relatively small. A beam wind, on the other hand, exerts a very large transverse force on the exposed side area of the ship. For a given wind velocity the maximum transverse wind force on a VLCC is about five times as great as the maximum longitudinal wind force. For a 50-knot wind on a light 250,000 DWT tanker, the maximum transverse and longitudinal forces are about 320 tonnes (3138 kN) and 60 tonnes (588 kN), respectively.

If the wind hits the ship from any quartering direction between the beam and ahead (or astern), it will exert both a transverse and longitudinal force, since it is striking both the bow (or stern) and the side of the ship. For any given wind velocity, both the transverse and longitudinal force components of a quartering wind will be smaller than the corresponding forces caused by the same wind blowing abeam or head on.

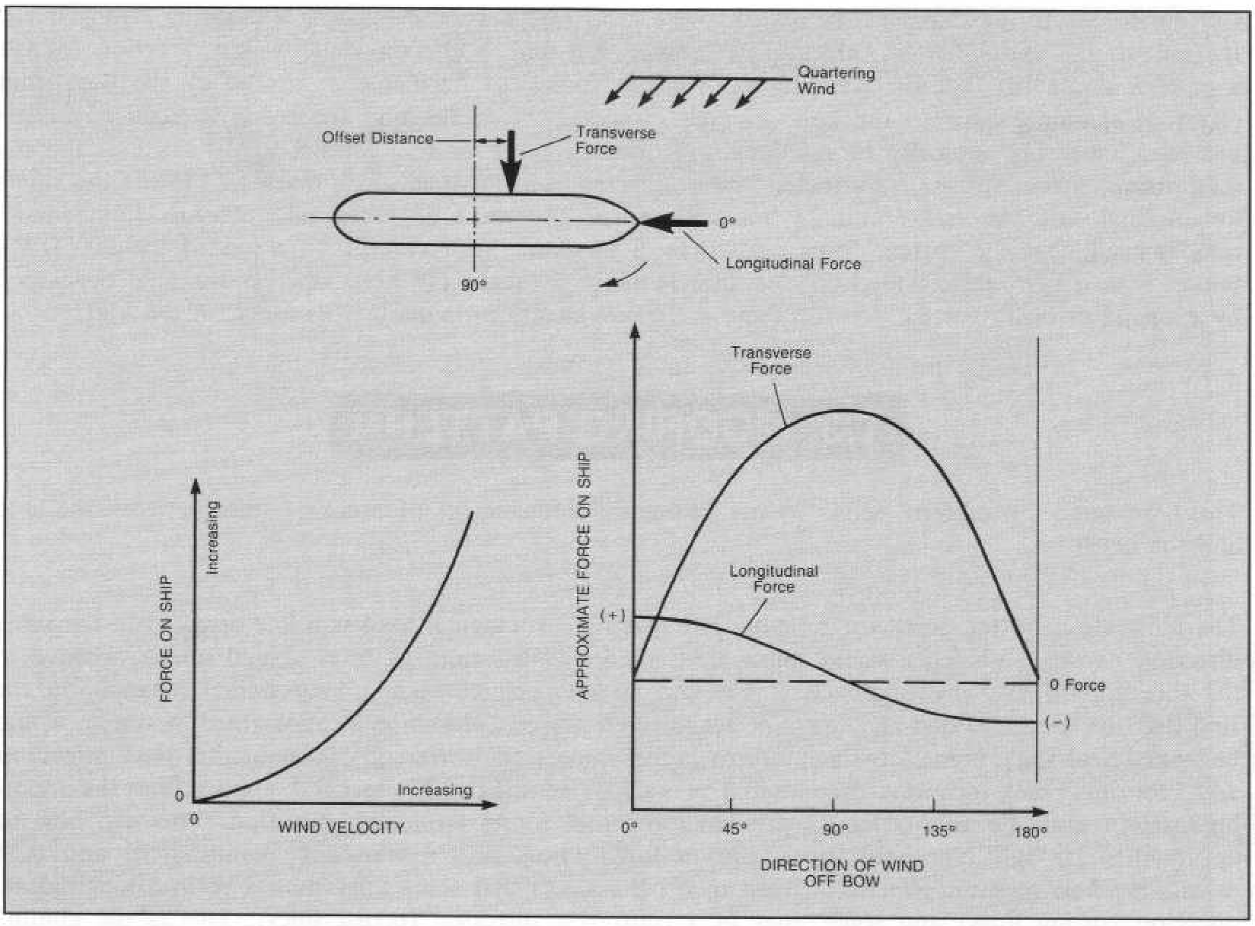


FIGURE 1.2: WIND FORCES ON A SHIP

With the exception of wind which is dead ahead or astern or dead abeam, the resultant wind force does not have the same angular direction as the wind. For example, a wind 45° off the bow yields a resultant wind force of about 80° off the bow for a 250,000 DWT tanker. In this case the point of application of the force is forward of the transverse centre line, producing a yawing moment on the ship.

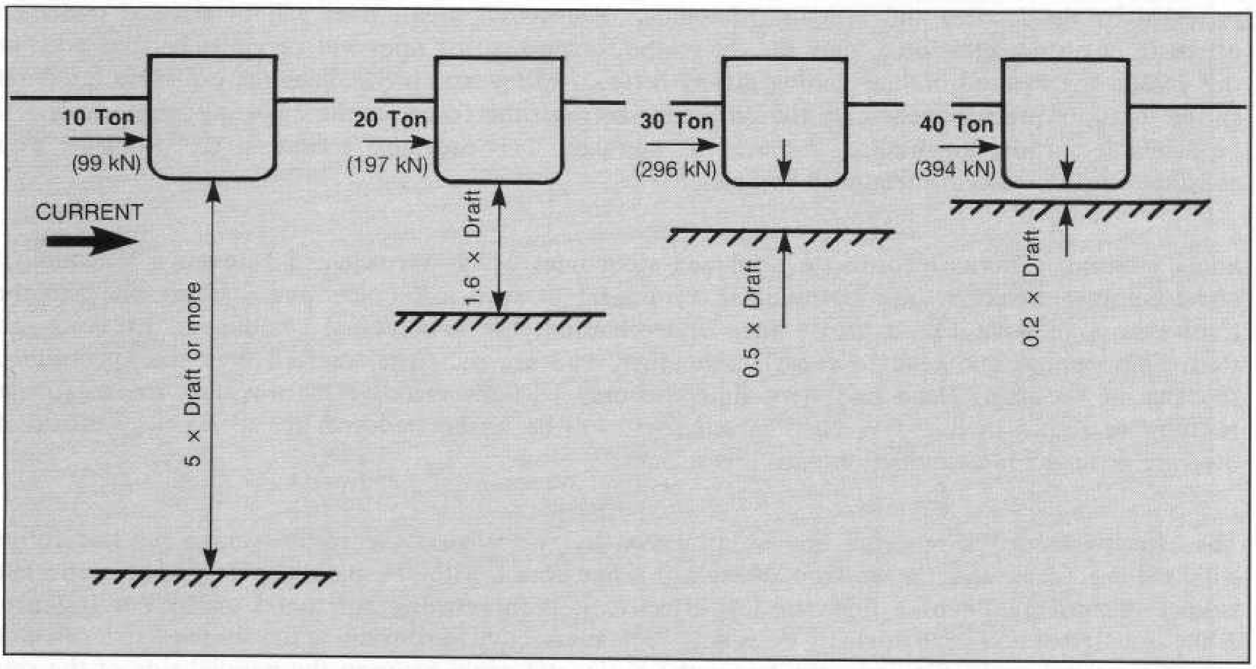


FIGURE 1.3: EFFECT OF UNDERKEEL CLEARANCE ON CURRENT FORCE

Current forces on the ship must be added to the wind forces when evaluating a mooring arrangement. In general, the variability of current forces on a ship due to current velocity and direction follows a pattern similar to that for wind forces. Current forces are further complicated by the significant effect of clearance beneath the keel. Figure 1.3 shows the increase in force due to reduced underkeel clearance. The majority of terminals are oriented more or less parallel to the current, thereby minimizing current forces. Nevertheless, even a current with a small angle (such as 5°) off the ship's longitudinal axis can create a large transverse force and must be taken into consideration. Model tests indicate that the current force created by a one-knot head current on a loaded 250,000 DWT tanker with a two metre underkeel clearance is about 5 tonnes (49 kN), whereas the load developed by a one-knot beam current for the same underkeel clearance is about 230 tonnes (2268 kN).

1.3 MOORING PATTERN

The term 'mooring pattern' refers to the geometric arrangement of mooring lines between the ship and the berth.

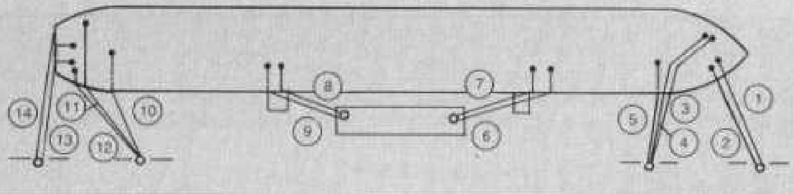
The most efficient line 'lead' for resisting any given environmental load is a line oriented in the same direction as the load. This would imply that, theoretically, mooring lines should all be oriented in the direction of the environmental forces and be attached at such a longitudinal location on the ship that the resultant load and restraint act through one and the same location. Such a system would be impractical since it has no flexibility to accommodate the different environmental load directions and mooring point locations encountered at various terminals. For general applications the mooring pattern must be able to cope with environmental forces from any direction. This can best be approached by splitting these forces into a longitudinal and a transverse component, and then calculating how to most effectively resist them. It follows that some lines should be in a longitudinal direction (spring lines) and some lines in a transverse direction (breast lines). This is the guiding principle for an effective mooring pattern for general application, although locations of the actual fittings at the terminal will not always allow it to be put into practice. The decrease in efficiency by deviating from the optimum line lead is shown in Figs. 1.4. and 1.5 (Compare Cases 1 and 3 in Fig 1.4 where the maximum line load increases from 57 (559 kN) to 88 tonnes (863 kN)).

There is a basic difference in the function of spring and breast lines which must be understood by designers and operators alike. Spring lines restrain the ship in two directions (forward and aft); breast lines restrain in only one direction (off the berth), restraint in the on-berth direction being provided by the fenders and breasting dolphins. Whereas all breast lines will be stressed under an off-berth environmental force, only the aft or the forward spring lines will generally be stressed. For this reason the method of line-tending differs between spring and breast lines (as explained later). If spring lines are pretensioned, only the difference between the forces in the opposing spring lines will be available for the longitudinal restraint of the ship. This fact also relates to the problems with constant tension winches mentioned in Section 7.

Some mooring patterns incorporate head and stern lines which are oriented between a longitudinal and transverse direction. The longitudinal component of such a line acts like a spring line and the transverse component like a breast line. Under tension, the longitudinal components of head and stern lines oppose and tend to cancel each other, and are therefore ineffective in the longitudinal restraint of the ship. Head and stern lines are only partially effective in providing the transverse restraint as shown in Fig. 1.5. Their effectiveness will be further reduced due to elasticity effects if they are arranged in combination with breast lines.

The effectiveness of a mooring line is influenced by two angles: the vertical angle the line forms with the pier deck and the horizontal angle the line forms with the parallel side of the ship. The steeper the orientation of a line, the less effective it is in resisting horizontal loads. For instance, a line oriented at a vertical angle of 45° is only 75% as effective in restraining the ship as a line oriented at a 20° vertical angle. Similarly, the larger the horizontal angle between the parallel side of the ship and the line, the less effective the line is in resisting a longitudinal force.

CASE 1
Idealised All Wire
All lines 42mm
MBL 115 tonnes



All loads are in tonnes

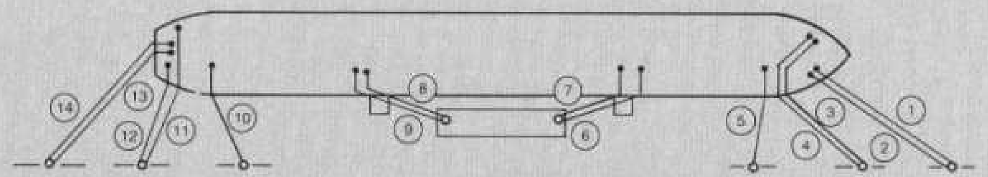
Line number	1	2	3	4	5	6	7	8	9	10	11	12	13	14
60 knot head wind	8.6	11.3	0	0	0	0	0	39.0	39.5	0	0	0	0	0
60 knot wind 45° off bow	56.7	57.1	34.5	34.9	39.0	5.9	5.9	10.9	11.3	25.9	25.4	34.0	24.9	23.6
60 knot beam wind	56.7	56.7	39.5	39.9	44.9	13.2	13.2	6.3	6.3	43.5	42.6	57.1	51.2	47.6

CASE 2
Idealised
Mixed Moorings

Mooring arrangements as above
except that lines 2, 4, 11 and 13 are polypropylene

Line number	1	2	3	4	5	6	7	8	9	10	11	12	13	14
60 knot head wind	15.9	5.0	0	4.1	0	0	0	39.4	39.4	0	3.6	0	2.7	0
60 knot wind 45° off bow	91.6	6.8	54.4	5.9	62.6	7.7	7.3	14.5	15.0	37.6	5.4	50.3	5.4	33.6
60 knot beam wind	91.2	6.8	61.2	5.9	69.8	17.2	16.8	9.5	9.5	67.1	5.9	88.0	6.3	73.0

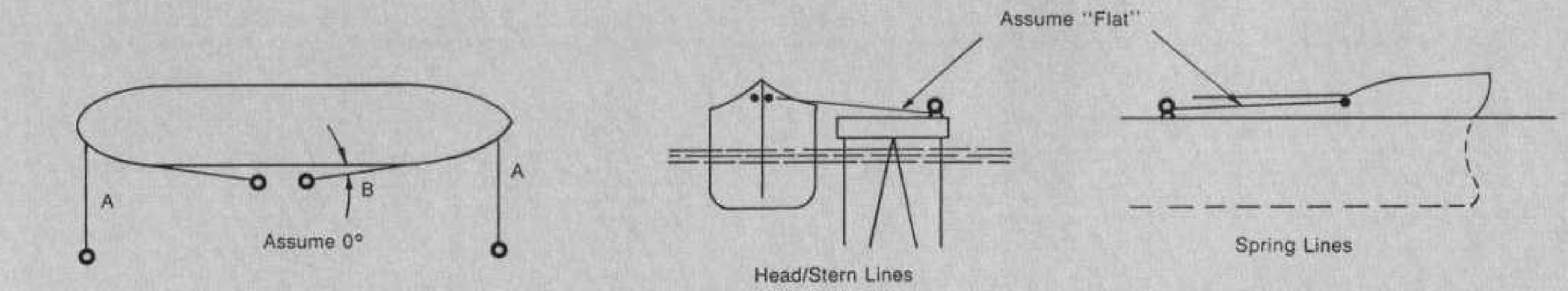
CASE 3
Non-Idealised
All Wire Moorings



Line number	1	2	3	4	5	6	7	8	9	10	11	12	13	14
60 knot head wind	10.4	11.8	5.4	8.2	0	0	0	28.6	28.6	0.9	0	0	0	0
60 knot wind 45° off bow	52.6	49.9	48.5	43.5	83.9	19.5	19.0	5.0	5.0	36.7	30.4	40.8	24.9	24.0
60 knot beam wind	56.2	54.0	53.1	48.1	88.4	17.7	17.2	11.8	12.2	70.3	49.9	70.3	46.3	45.8

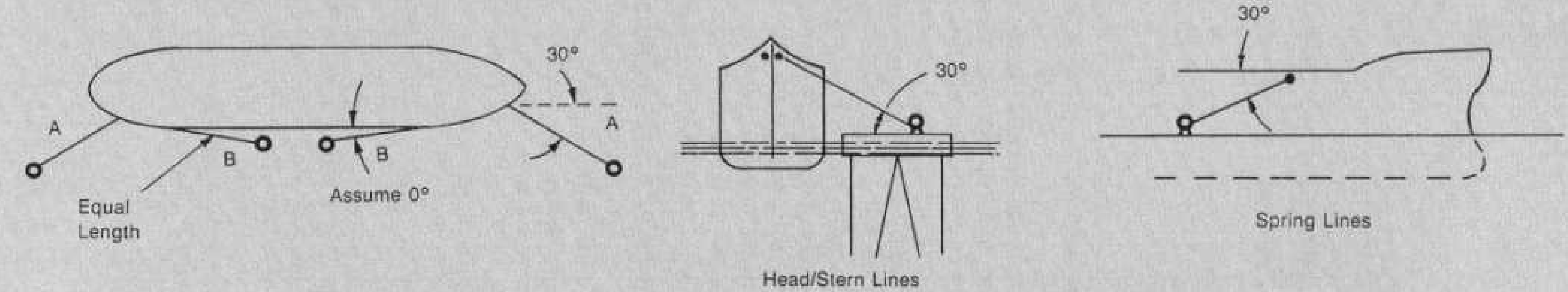
Note: Computer Programme assumes line does not yield or break. Examples are based on ballasted 250,000 dwt ship. Loads are for conditions shown. Should wind shift, lines without loads, as shown above, would assume some loadings, hence all lines should be tended at all times.

FIGURE 1.4: MOORING PATTERN ANALYSIS



Transverse Restraint Capacity $= 2 \times A = 2A^*$
Longitudinal Restraint Capacity $= 1 \times B = 1B^*$

***A = Allowable Working Load in Head/Stern Lines**
***B = Allowable Working Load in Spring Lines**



Transverse Restraint Capacity $= 2 \times A \times \text{SIN } 30° = 0.87A$
** Longitudinal Restraint Capacity $= 1 \times B \times \text{COS } 30° + 1 \times A \times \text{COS } 30° \times \text{COS } 30°$
$= 0.87B + 0.75A$

** Longitudinal Restraint Capacity under a longitudinal force only; if a transverse force is present the longitudinal restraint will be further reduced due to opposition of forces in head and stern lines. Also, elasticity effects have been neglected in this example which may further reduce the longitudinal restraint capacity.

FIGURE 1.5: EFFECT OF HAWSER ORIENTATION ON RESTRAINT CAPACITY

1.4 ELASTICITY OF LINES

The elasticity of a mooring line is a measure of its ability to stretch under load. Under a given load, an elastic line will stretch more than a stiff line. Elasticity plays an important role in the mooring system for several reasons:

- High elasticity can absorb higher dynamic loads. For this reason, high elasticity is desirable for ship-to-ship transfer operations, or at terminals subject to waves or swell.

- On the other hand, high elasticity means that the ship will move further in her berth and this could cause problems with loading arms or hoses. Such movement also creates additional kinetic energy in the mooring system.

- A third and most important aspect is the effect of elasticity on the distribution of forces among several mooring lines. The simple four-line mooring pattern shown in the upper portion of Fig. 1.5 is insensitive to the elasticity of the lines but is suitable only for boats or very small ships. Due to size limitations on individual lines, many more lines must be used for larger ships. The optimum restraint is generally accomplished if all lines, except spring lines, are stressed to the same percentage of their breaking strength. Good load-sharing can be accomplished if the following principles are understood:

The general principle is that if two lines of different elasticity are connected to a ship at the same point, the stiffer one will always assume a greater portion of the load (assuming the winch brake is set) even if the orientation is exactly similar. The reason for this is that both lines must stretch an equal amount, and in so doing, the stiffer line assumes a greater portion of the load. The relative difference between the loads will depend upon the difference between the elasticities, and can be very large.

The elasticity of a mooring line depends upon the following factors:

- Material
- Construction
- Length
- Diameter

Figure 1.6 demonstrates the significance of each of the above factors on load distribution. The most important points to note are the appreciable difference in elasticity between wire lines and fibre ropes and the effect of line length on elasticity. Cases A) and B) in Fig. 1.6 are examples of mooring arrangements that should be avoided, while Case C) shows an acceptable mooring where each rope is stressed to approximately the same percentage of its breaking strength.

Wire mooring lines are very stiff. The elongation for a 6 × 37 construction wire line at the loading at which the material begins to be permanently deformed is about one percent of wire length. (For a more complete discussion, see Section 6). Under an equivalent load a polypropylene rope may stretch ten times as much as a wire. Thus if a wire is run out parallel to a fibre line, the wire will carry almost the entire load, while the fibre line carries practically none. Elasticity also varies between different types of fibre lines and, although the difference is generally not as significant as that between fibre line and wire, the difference will affect load distribution. Aramid fibre lines for example have much less elasticity than other synthetic fibre lines and would carry the majority of the load if run out parallel to conventional synthetic lines.

The effect of material on load distribution is critical and the use of mixed moorings for similar service, e.g. forward springs, is to be avoided. In some cases the fibre lines may carry almost no load, while at the same time some of the wires are heavily loaded, possibly beyond their breaking strength. The same could be true of mixed fibre lines of varying elasticity, although the differences would generally not be as great.

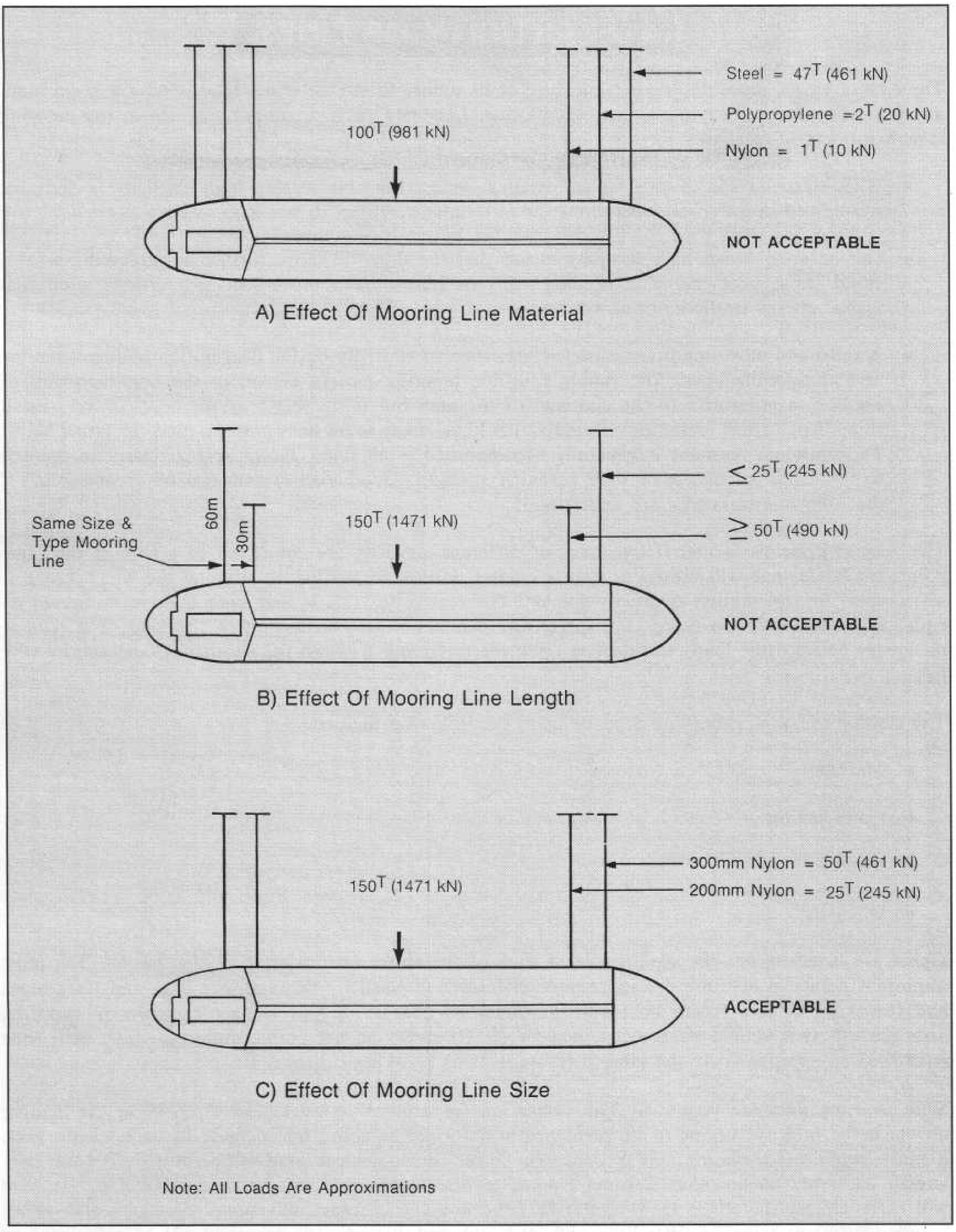


FIGURE 1.6: EFFECT OF MOORING ELASTICITY ON RESTRAINT CAPACITY

The effects of mixing wire and synthetic fibre lines are shown in Fig. 1.4, by comparison of Cases 1 and 2. (Note the low loads in fibre lines 2, 4, 11 and 13 and the increase in wire loads from a maximum of 57 tonnes (559 kN) to a maximum of 88 tonnes (863 kN)).

The effect of line length on load distribution must also be considered. Line elasticity varies directly with line length and has a significant effect on line load. A wire line 60 m long will assume only about half the load of a 30 m parallel and adjacent line of the same size, construction and material.

Elasticity of a given type of line also varies with its diameter and construction. Usually this factor is not an important consideration since the load relative to a line's strength is the governing factor rather than the absolute load.

1.5 GENERAL MOORING GUIDELINES

Consideration of the principles of load distribution in 1.4 lead to the following mooring guidelines. These assume that the moored ship may be exposed to strong winds or current from any direction.

- Mooring lines should be arranged as symmetrically as possible about the midship point of the ship. (A symmetrical arrangement is more likely to ensure a good load distribution than an asymmetrical arrangement.)
- Breast lines should be oriented as perpendicular as possible to the longitudinal centre line of the ship and as far aft and forward as possible.
- Spring lines should be oriented as parallel as possible to the longitudinal centre line of the ship.

Head and stern lines are normally not efficient in restraining a ship in its berth. Mooring facilities with good breast and spring lines allow a ship to be moored most efficiently, virtually 'within its own length'. The use of head and stern lines requires two additional mooring dolphins and decreases the overall restraining efficiency of a mooring pattern when the number of available lines is limited. This is due to their long length and consequently higher elasticity and poor orientation. They should only be used where required for manoeuvring purposes or where necessitated by local pier geometry, surge forces or weather conditions. Obviously, small ships berthed in facilities designed properly for larger ships may have head and stern lines because of the berth geometry.

- The vertical angle of the mooring lines should be kept to a minimum.

The 'flatter' the mooring angle, the more efficient the line will be in resisting horizontally-applied loads on the ship.

A comparison of Cases 1 and 3 in Fig. 1.4 demonstrates that a ship can usually be moored more efficiently within its own length. Although the same number of lines are used in each situation, Case 1 results in a better load distribution, minimising the load in any single line.

- Generally, mooring lines of the same size and type (material) should be used for all leads. If this is not possible due to the available equipment, all lines in the same service, i.e. breast lines, spring lines, head lines, etc. should be the same size and type. For example, all spring lines could be wire and all breast lines synthetic.

First lines ashore can be synthetic lines, even though the main mooring lines are wire. This is acceptable as long as it is realized that the fibre lines will not add to the final restraining capacity of the system unless all lines in that group are of the same material.

- If tails are used on the wires, the same size and type of tail should be used on all lines run out in the same service.

Synthetic tails are often used on the ends of wire lines to permit easier handling and to increase line elasticity. The addition of an 11 m nylon tail would increase line elasticity of a 45m long wire line by five to sixfold (see also Section 6.2.7).

- Mooring lines should be arranged so that all lines in the same service are about the same length between the ship's winch and the shore bollard. Line elasticity varies directly with line length and shorter lines will assume more load.

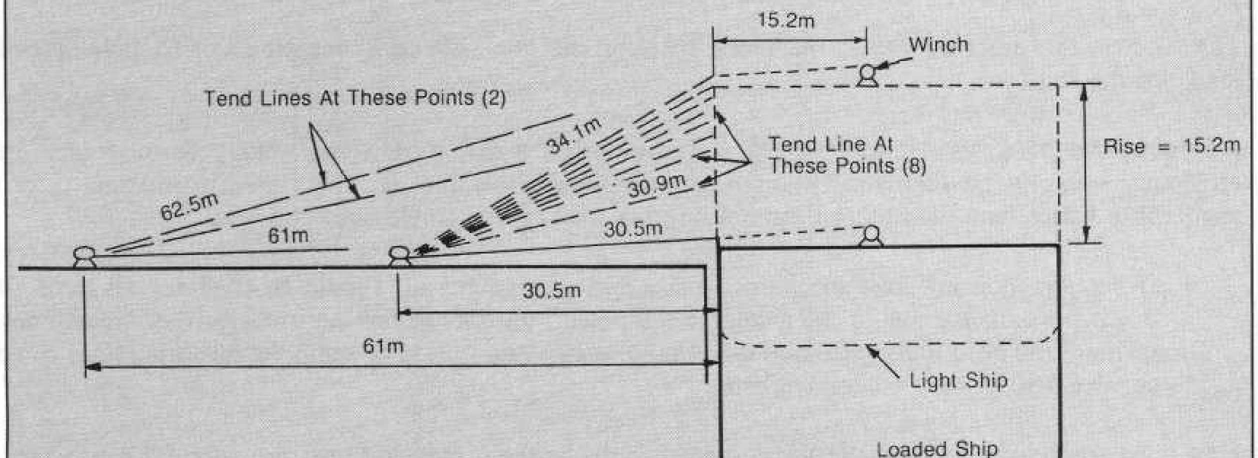
1.6 OPERATIONAL CONSIDERATIONS

The above mooring guidelines were developed to optimize load distribution to the moorings. In practice, final selection of the mooring pattern for a given berth must also take into account local operational and weather conditions, pier geometry and ship design. Some pilots, for example, desire head and stern lines to assist ships moving into, along, or out of a berth, while others may use spring lines for this purpose. Head and stern lines would be advantageous at berths where the mooring points are too close to the ship and good breast lines cannot be provided, or where the bollards are located so that the spring lines will have an excessive vertical angle in the light condition. These excessive angles would result in considerably reduced restraint capability.

High winds and currents from certain directions might make it desirable to have an asymmetrical mooring arrangement. This could mean placing more mooring lines or breast lines at one end of the ship.

The other factor to consider is the optimum length of mooring lines. It would be desirable to keep all lines at a vertical angle of less than 25°. For example, if the ship's chock location is 25m above the shore mooring point, the mooring point should be at least 50m horizontally from the chock.

Long lines are advantageous both from standpoint of load efficiency and line-tending. But where fibre ropes are used, the increased extension can be a disadvantage by permitting the ship to move excessively, thereby endangering loading arms. Figure 1.7 illustrates the effects of line lengths on line-tending requirements.



Distance From Bollard To Ship Side	Minimum No. Tendings Due To 15.2m Ship Rise
30.5m	8
45.7m	4
61m	2

Notes:
- Assume Wire Lines. Yield Strain = 1% Length = 0.46m for 45.7m Hawser
- Assume No Slack in Wires
- Assume Lines Tended when they reach Maximum Allowable Load

FIGURE 1.7: EFFECT OF LINE LENGTH ON TENDING REQUIREMENTS

1.7 TERMINAL MOORING SYSTEM MANAGEMENT

Good mooring management requires the application of sound principles, well maintained equipment, trained personnel and, most importantly, proper co-ordination and interaction between ship and shore.

While the safety of the vessel and hence its proper mooring is the prime responsibility of the Master, the terminal, because of its knowledge of the operating environment at its site and its equipment, should be in the best position to advise the Master regarding mooring line layout and operating limitations.

The responsibilities and arrangements for mutual checking of moorings, cargo transfer and other aspects of the ship shore interface are covered under the ship/shore check list.

Mooring equipment of existing tankers varies widely, ranging from synthetic mooring ropes, mixed moorings (synthetic ropes and wire lines), all wire moorings (with and without synthetic tails) to modern synthetic "high modulus" systems. Rated brake capacities, winch and fairlead locations can vary significantly from ship to ship. Ship crews will have varying degrees of expertise in mooring matters and varying philosophies concerning maintenance and/or replacement of critical items of mooring equipment.

The terminal can utilize a number of concepts in modern mooring management to reduce the possibility of ship break-out. These are:

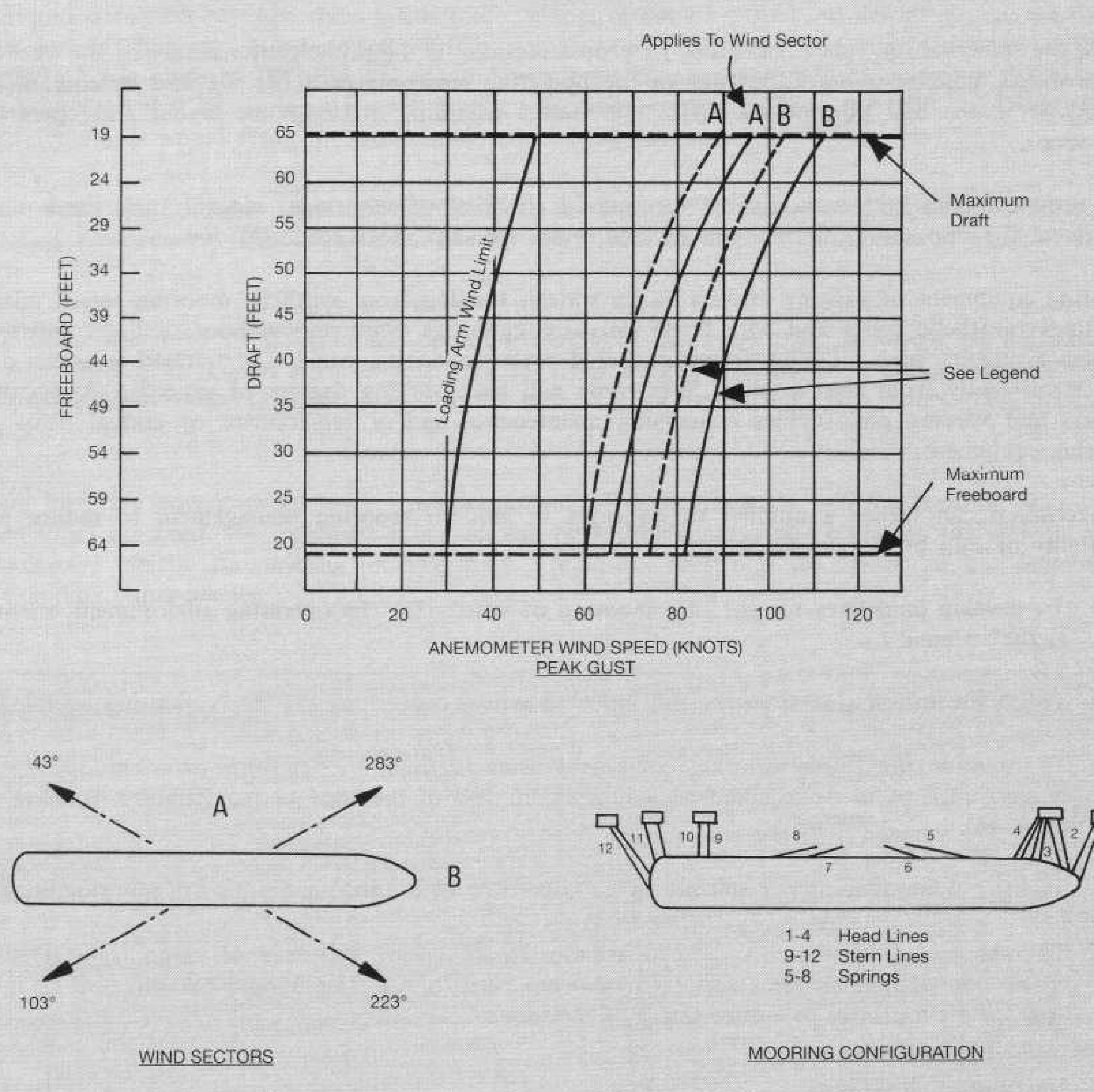
- To develop guidelines for the safe mooring of vessels for the operating environment existing at the terminal.
- To obtain information from the ship prior to arrival concerning the ship's mooring equipment
- To examine the ship's mooring equipment after berthing to determine what modification, if any, must be made to standard guidelines in view of the state of maintenance, training of crew, etc.
- To inspect line tending periodically either visually or by the instrumentation of mooring hooks.
- To take whatever action is deemed appropriate to ensure stoppage of cargo transfer, disconnection of loading arms and removal from berth of the ship should the ship fail to take appropriate measures to ensure safety of mooring.

1.7.1 *Operating Limits*

Another important aspect in restraining the ship at its berth is the movement of the ship. No simple formula can be offered for the ship movement, although this is generally included in the output of computer calculations. Movement of the ship due to environmental loads can exceed loading arm operating limits before the strength limits in the mooring lines are reached. Similarly limits and requirements may apply to gangways, particularly shore-based equipment incorporating a tower or a long span from the jetty to the ship. This is especially true for synthetic line systems. Under worsening environmental conditions the loading arms and gangways may therefore have to be disconnected at lesser wind and current conditions than those used as a design basis for the mooring system.

1.7.2 *Operating Guidelines/Mooring Limits*

In the past, operating guidelines have generally been developed empirically. With the advent of computers and more accurate wind and current coefficients, guidelines can be developed systematically which can provide the limits for various classes of ships with varying mooring capabilities. At facilities which are located in climes where the operating environment is other than mild, it is desirable to have this done.



Legend

———— Rated Winch Brake Capacity ≥55 tonnes (539 kN)
Mooring Line Breaking Strength ≥85 tonnes (834 kN)

– – – – – – Rated Winch Brake Capacity ≥40 tonnes (392 kN)
Mooring Line Breaking Strength ≥72 tonnes (706 kN)

Notes
- Loading Arm Wind Limits Are Governed By Vessel Motions.
- Limits Shown Are Based On Pretension Of 2-3 Tonnes (20-29 kN) And Use of Nylon Lines.
- Limits Applicable To A Flood Current Condition From Stern.
- Lines 1, 2, 3, 4, 11, 12 Are Led Around Shore Based Pulleys Back To Ship's Bitts.

FIGURE 1.8: OPERATING WIND LIMITS FOR 250,000 DWT TANKERS WITH ALL NYLON MOORING ROPES

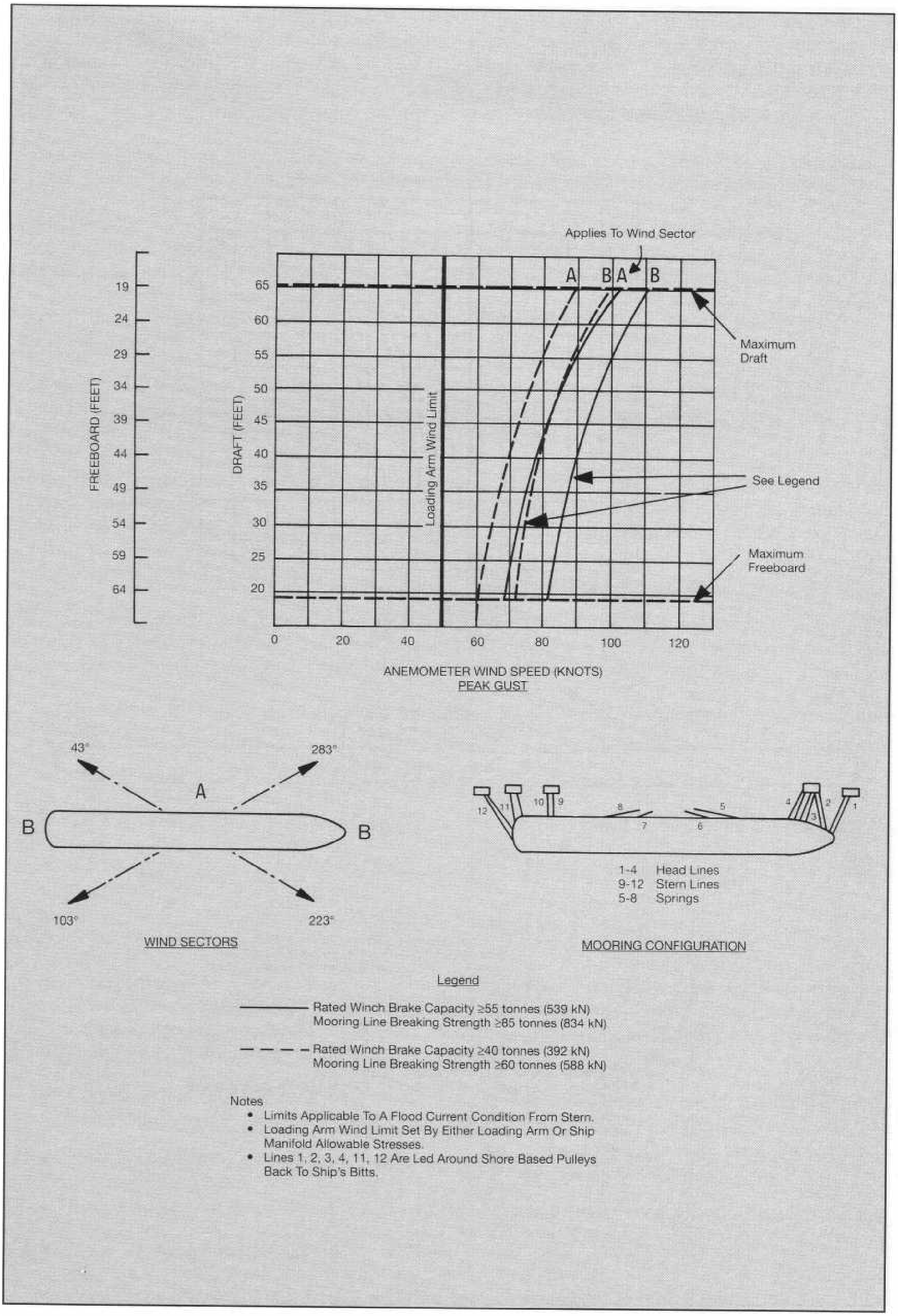
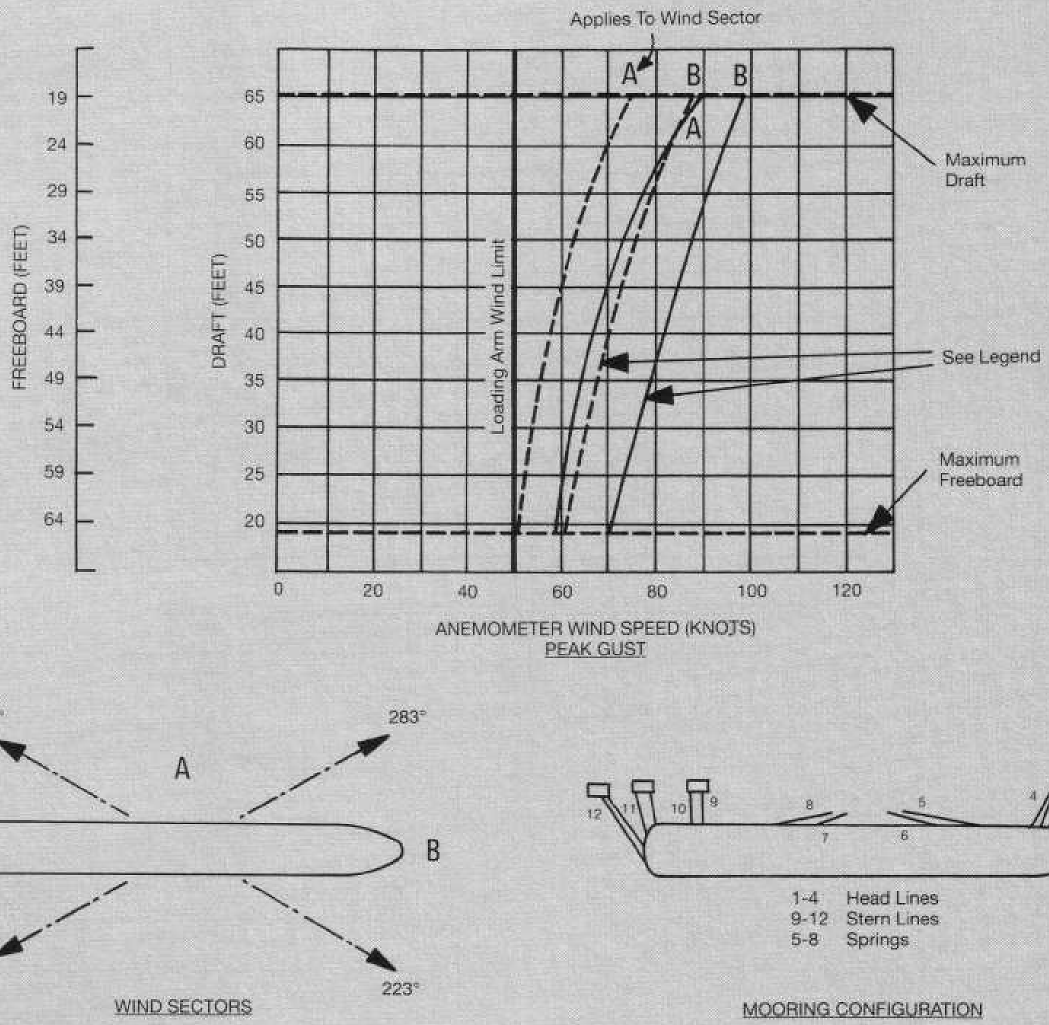


FIGURE 1.9: OPERATING WIND LIMITS FOR 250,000 DWT TANKERS WITH ALL WIRE MOORING LINES



Legend

——— Rated Winch Brake Capacity ≥55 tonnes (539 kN)
Mooring Line Breaking Strength ≥85 tonnes (834 kN)

– – – Rated Winch Brake Capacity ≥40 tonnes (392 kN)
Mooring Line Breaking Strength ≥60 tonnes (588 kN)

Notes

- 2 Nylon Lines In Each Set Of Head-Stern Lines.
- Limits Shown Are Based On Pretension Of 2-3 Tonnes (20-29 kN) And Use of Nylon Lines.
- Limits Applicable To Flood Current Condition From Stern.
- Loading Arm Wind Limit Set By Either Loading Arm Or Ship Manifold Allowable Stresses.
- Lines 1, 2, 3, 4, 11, 12 Are Led Around Shore Based Pulleys Back To Ship's Bitts.

FIGURE 1.10: OPERATING WIND LIMITS FOR 250,000 DWT TANKERS WITH MIXED MOORINGS

A number of terminals have developed information to be used by their operators as guidelines. Examples of the type of information that would be valuable to the jetty operator for mooring 250 kdwt tankers are shown in Figures 1.8, 1.9 and 1.10. These examples are for mooring configurations as shown and for wind variations, wind only and for flood tide. They have been developed for the following mooring systems: nylon rope, all wire lines and mixed wire and nylon moorings.

Inclusion of the mixed mooring case as an example should not be construed to be an endorsement of this system. Other sites will probably have other criteria on which to base operating limits.

With 18 nylon lines Figure 1.8 shows that at maximum freeboard (or minimum draft) the loading arms should be disconnected at wind speeds of approximately 30 knots peak gust. As the draft increases, the permissible wind speed increases to 50 knots peak gust. Loading arm limits are governed by vessel motion. For the winch brake capacities shown and for the wind directions indicated, mooring line loads become excessive in at least one line at the drafts and wind speeds indicated by the curved lines.

For an 18 wire with tails case, Figure 1.9, the loading arm wind limit is constant at 50 knots peak gust. This limit is no longer a function of vessel movement but of the allowable stress levels in the arm or ships manifold. Allowable wind speeds for mooring are as shown.

Figure 1.10 demonstrates a mixed mooring situation wherein 14 wire and tail mooring lines and 4 nylon mooring ropes are used as indicated. In this case as with the all wire case the loading arm wind limits are established by the stresses in the arms or ship's manifold. However, a comparison of the two figures shows there is a marked reduction in the wind speeds that can be tolerated by the mixed mooring wire and rope system.

The foregoing limitations can be created for varying combinations of wind and current, vessel draft, mooring line combinations and configurations and various ship winch design brake capacities. The information thus generated can be used for a number of purposes:

- To decide whether a given ship can be moored at a given berth under the expected weather conditions.
- To determine when to discontinue cargo transfer and to disconnect loading arms.
- To advise the ship when it would be desirable to take on ballast to reduce its freeboard.
- To advise the ship when it would be desirable to have tugs available to assist in maintaining the ship's position at the jetty.

1.7.3 *Joint Terminal/Ship Meeting and Inspection*

As soon as practicable after berthing, it is recommended that terminals have their representative board the vessel to establish contact with the Master or his designated representative. At this meeting the Terminal Representative should provide information relating to shore facilities and procedures. In addition he should in concert with the Ship Representative:

- Complete the Ship/Shore Safety Check List in line with guidance given in ISGOTT and, where appropriate, physically check items before ticking off.
- Obtain details of moorings and winches, including state of maintenance.
- Review forecasted weather and arrange for the Master to be advised of any expected changes.
- Assess freeboard limitations.
- Assess type and condition of ship mooring equipment and its ballasting ability.
- Determine the conditions at which cargo transfer will be discontinued and loading arms and hoses be disconnected and precautions to be taken under high mooring load situations.

1.7.4 *Instrumented Mooring Hooks or Visual Inspection of Mooring Lines*

The terminal should monitor the ship's line tending activity by visual inspection of the mooring lines, particularly during cargo transfer and periods of changing environmental conditions.

In addition to the above, and dependent on the physical environment at the berth, it may be desirable to install mooring line load measurement apparatus where an appropriate need has been identified. This equipment is now available and has been installed at a number of large tanker berths and at many LNG berths. It measures the line loads and has a central read-out in the terminal operation's control room. Should the line loads become high or the lines become slack, the terminal operator can advise the ship accordingly.

In many terminals mooring tension information is transmitted to a shipboard fixed or portable display for direct access by ships staff. In any case the terminal should inspect lines periodically. If poor line tending by ships staff is observed, the terminal should notify the ship.

1.8 SHIP MOORING MANAGEMENT

Good ship mooring management requires a knowledge of good mooring principles, information about the mooring equipment installed on the ship, proper maintenance of this equipment, and good, seamanlike line tending.

Officers in charge of line tending and personnel assigned to tend lines should be aware of the capabilities of the equipment installed on their ship. Specifications should be available on the winch drum to show the design holding capacity and the torque required on the hand wheel or lever to achieve this. Specifications of the mooring lines should also be available.

Recommendations concerning the proper direction of reeling of the wire on the drum should be followed and the drum should be marked accordingly to prevent any possibility of error.

1.8.1 *Line Tending*

The objective of good line tending is to ensure that all lines share the load to the maximum extent possible and to limit the ship's movement off the berth or alongside the berth. Pretensioning of lines (that is loading a line with a winch prior to the application of environmental forces) reduces ship movement and improves the load distribution when lines of different lengths and elasticities are being used. Figure 1.11 demonstrates how pretensioning affects the load distribution and the movement of the ship.

It is very important to tend spring lines differently from breast lines. Tending head or stern lines presents a special problem (which is one more reason why they are not recommended). They must be tended like either spring or breast lines depending on whether longitudinal or transverse restraint is more critical. For example, if a high longitudinal current on the bow is expected, the bow line should be pretensioned while the stern line is tensioned only to take up any slack. The following general rules apply to line-tending.

- Slack lines should be hauled in first. Slack lines may permit excessive movement of the ship when there is a sudden change in the environment.
- Only one line should be tended at a time. Any time a line is tended, it temporarily changes the load in other lines and may increase it. The simultaneous tending of two lines may therefore give erratic results or even an overload.
- Whenever a spring line is tended, the opposite spring must also be tended. Otherwise, rendering or heaving-in on one spring line may cause excessive movement of the moored ship along the pier face.

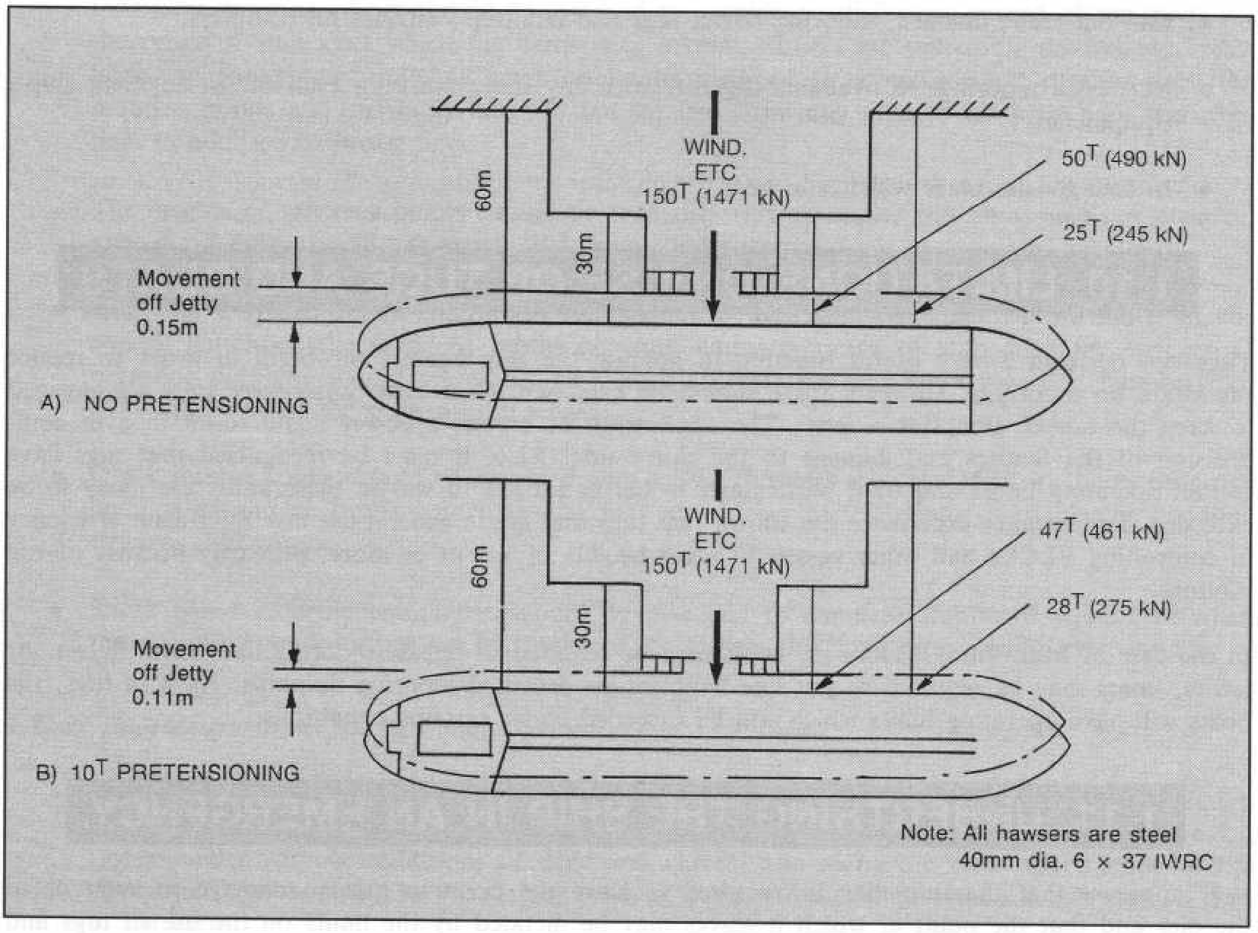


FIGURE 1.11: EFFECT OF PRETENSIONING

Generally only the forward or aft spring lines will be tensioned by environmental loads. The tensioned lines should be tended first, but only if necessary to correct longitudinal positioning of the ship. The opposing spring lines should then be adjusted to remove slack only.

- Fender compression should be observed during discharge or during a rising tide. Fender compression may be caused by over-tight breast lines. If there is high fender compression which is not caused by on-shore winds or currents, the breast lines must be slackened.

1.9 PRECAUTIONS APPLICABLE IN HIGH MOORING LOAD CONDITIONS

Overloading of mooring lines is evidenced in a number of ways; for example, by direct measurements of mooring line loads, by direct observation of the moorings by experienced personnel, or by predictions made by those having a knowledge of the effects of wind and current on the ship mooring system or by winch slippage.

The following precautions are likely to apply:

- Harden-up on the winch brakes
  Do not release brakes and attempt to heave in
- Discontinue cargo operations
- Reduce freeboard by taking-on ballast if loads are due to high wind conditions.
- Disconnect loading arms and gangways.

- Call out crew, linemen, mooring boats, tugs and put ship's engines on readiness.
- Run extra moorings as available together with any shore mooring available to augment ship's equipment.
- In emergencies place winches in gear.

1.10 LIMITATIONS ON USE OF TUGS AND BOATS

Tugs can perform a very useful function in holding the ship against the berth in order to reduce the strain on moorings. However, care should be exercised when high horsepower tugs are engaged to keep the tanker alongside a jetty. The application of excessive power could result in over compression of the fenders and damage to the ship's side. Also, it must be recognized that tugs have certain operating limits and that, particularly in berths subject to waves, these limits are likely to be exceeded. For instance experience has shown that tugs that are in general use lose significant efficiency in controlling VLCCs and other vessels in wave heights of 1.5 m or more, primarily because of tug motions.

In the case of Multi-Buoy Moorings, boats may be required to release mooring lines from buoys. At jetties, boats may be required to put line handlers on detached mooring dolphins. As with tugs, the boats will have operating limits which will be exceeded under extreme conditions.

1.11 UNBERTHING THE SHIP IN AN EMERGENCY

It is apparent that the ship may be required to leave the berth to prevent damage to itself or to the pier and that the point at which it leaves may be dictated by the limits on the use of tugs and work boats and not by mooring line loads or ship movement. It must be emphasized, however, that the ship Master who is responsible for the safety of the ship must decide whether it is safe to do so or whether by making a hurried unberthing manoeuvre he will in fact place his ship in greater jeopardy. There are also certain berths where tidal conditions or manoeuvring areas may be such as to prevent the unberthing of the ship at certain times.

1.12 GENERAL RECOMMENDATIONS

1.12.1 *Recommendations for Berth Designers*

- The mooring facilities provided at the berth should be such as to permit the largest vessel which is to be accommodated to remain safely moored alongside.
- The wind and current forces on the ship should be calculated for the wind and current conditions under which the ship may remain moored at the berth, using the procedures covered in Section 2 of these guidelines and in Reference 3.
- Allowable loads in any one mooring line should *not* exceed 55% of its Minimum Breaking Load (MBL).
- The following principles should be applied when designing the layout of mooring facilities on the berth:

  Mooring points shall be disposed as nearly as possible symmetrically about the centre point. Breast moorings shall be provided such that they will emanate from points near the fore and aft ends of the ship and as nearly as possible perpendicular to the fore and aft line of the ship.

  The length of mooring lines should be within the range 35 to 50 m and, where intended for the same service and practicable, be equal.

Sufficient mooring points should be installed to provide a satisfactory spread of moorings for the range of ship sizes which the berth is to accept. VLCCs are preferably moored by breast lines and spring lines only, although on berths designed to accept a range of ship sizes, the mooring points will inevitably be such that smaller ships may need to use head lines and stern lines in addition to breast lines.

The heights of mooring points should be such that vertical angles will be as small as practical and, if possible, should not exceed 30° from the horizontal.

- Breasting dolphins should preferably be positioned at a distance apart of one third of the overall length of the ship. At berths accommodating a range of ship sizes the spacing of breasting dolphins should preferably be located so that they provide a breasting face between 25% and 40% of the ships overall length.

- Quick release hooks should be provided with a SWL not less than the MBL of the largest rope anticipated and be supplemented by capstans or winches and fairleads to enable the handling of large ship's moorings.

- Shore based mooring equipment should be provided to augment shipboard equipment when the operating conditions at the berth exceed the Design Environmental Conditions.

1.12.2 *Recommendations for Terminal Operators*

- Terminal Operators should have a good understanding of mooring principles, of the design of the mooring system for the berth, of the loads likely to be experienced in the mooring system under varying conditions of wind and current and to have a clear appreciation of the operating limits applying to the various types of ships and mooring systems which may be used in the berth.

- They should recognize the problems likely to arise from the use of mixed moorings and be aware of the need for effective application of winch brakes and good mooring management while the ship is in the berth.

- Ship-to-shore liaison should be established by the Terminal Operator prior to arrival. A joint agreement is required with the ship on the way in which the ship will actually be moored, and on continuing liaison on mooring matters during the time the ship is in the berth; particular attention being paid to the procedures to be followed in the event of emergencies.

1.12.3 *Recommendations for Ship Designers*

- The mooring facilities provided in the ship should be such as to permit the vessel to remain safely moored under the Standard Environmental Criteria alongside a berth which is provided with a standard arrangement of mooring points.

- Wind and current forces on the ship should be calculated applying the Standard Environmental Criteria and the coefficients determined by OCIMF (reference 3) and by using the methods described in this book. This calculation will determine the number, size and disposition of moorings required on board.

- Loads in any one mooring line should not exceed 55% of the MBL.

- Mixed moorings, comprising full length synthetic ropes used in conjunction with wires, are not recommended.

- Wire ropes should be the standard mooring equipment for all large tankers and it is recognised that wire ropes greater than 44 mm diameter may require special handling arrangements in terminals.

- Synthetic ropes may be used as the first line ashore for positioning the ship at either end, preferably by means of handling and storage winches. These ropes should not be considered as contributing to the restraint of a vessel moored principally with wires.

- When nylon tails are fitted to wire ropes they should have an MBL at least 25% in excess of the wire, have a length of about 11 m, and be subject to rigorous examination and renewal procedures, as recommended in Section 6.

- Winches for handling the wire ropes may be either of the split drum or undivided drum type; the relative merits of the two types are described in Section 7.3.

- Automatic winches are not recommended, but if fitted must have a capability to disengage the automatic operational features.

- Winch brakes should provide a minimum holding capacity of 60% of the MBL of the wire on the first layer of wire of a split drum winch and on the normal working layer of an undivided drum winch. They should be maintained during service in order to retain their efficiency.

- The layout of moorings should be such as to provide:

  — symmetry about the mid length and to provide the design numbers of moorings on each side of the ship,

  — breast lines sited as near as possible to the end of the ship,

  — moorings used in the same service to be as nearly as possible of the same length inboard of the ship,

  — suitable chocks and fairleads to be provided in order to ensure correct alignment of moorings,

  — bitts to be positioned for supplementary moorings.

- Minimum safety factors listed in Table 4.3 are based upon the appropriate design criteria and loading assumptions, and should be incorporated in all new equipment and mooring fittings.

- All equipment and fittings should be clearly marked with their SWL.

1.12.4 *Recommendations for Ship Operators*

- The principles of good mooring, including the dangers associated with mixed moorings, should be understood by ship operators. Particular attention should be given in ship's instructions to the proper application of winch brakes, the maintenance of moorings and winch brakes, good line tending procedures and the practices to be observed in the case of mooring emergencies.

- Each ship should be provided with information on the design of the mooring system, with examples to show the loads likely to be experienced under particular conditions and to illustrate those situations under which the limit of the system is likely to be reached.

Section 2.0

Mooring Restraint and Environmental Criteria

2.1 GENERAL CONSIDERATIONS

In order to design a ship's mooring system, the environment loads likely to act upon the ship must first be determined. These can be highly variable from terminal to terminal. To ensure a minimum standard is met for mooring equipment on ships engaged in worldwide trades, the Standard Environmental Criteria given below should be assumed. The Standard Environmental Criteria apply to the design of the ship mooring system and are not criteria for pier design nor a required operating capacity for a pier/ship mooring plan. These parameters are not intended to cover the worst possible conditions, since this would be neither practical nor reasonable. In situations where the Standard Environmental Criteria are exceeded, such as during hurricanes or at a river berth with extremely strong currents, additional measures must be taken such as doubling mooring lines, requesting tug assistance or leaving the berth.

Environmental loads acting upon a ship due to wind and current should be derived from specific model test data for that ship design or from the general non-dimensional force coefficients for oil tankers and gas carriers contained in Reference 3 and Reference 5.

The wind and current coefficients contained in Reference 3 were defined originally for VLCC size ships above 150,000 tonnes deadweight. More recent model test data on the modern tanker forms which have all accommodation superstructures aft confirms that these same coefficients are, in most cases, sufficiently accurate when applied to smaller ships, and that they may therefore be used for a range of ships down to approximately 16,000 tonnes deadweight.

For gas carriers the wind coefficients quoted in Reference 5 are recommended. These apply to membrane and spherical tank designs in the range 75,000 m^3 to 125,000 m^3. For smaller gas carriers, generally acceptable wind coefficient data is not available and, unless model tests are to be carried out in each case, a conservative extrapolation from large gas carrier data must be adopted.

The current coefficients quoted in Reference 3 for VLCCs may be used for all sizes of tankers with similar geometry down to 16,000 tonnes deadweight and for gas carriers of an equivalent length and draft.

It must be recognised that the coefficients contained in Reference 3 and Reference 5 will yield accurate results only if the vessel geometry and drafts are similar to those of the models used in establishing the coefficients. Reference 3 contains some general guidelines for adjustments if the actual vessel geometry is outside the model range. For instance, oil tanker models used correspond to typical pre-MARPOL vessels with a ratio of overall length to freeboard of 50–60, and a ratio of ballast freeboard to full load freeboard of 3.1:1 Reference 3 suggests that wind coefficients for post-MARPOL (SBT and double hull) vessels (which generally have higher freeboard than pre-MARPOL vessels) be obtained by interpolation or extrapolation on the basis of midships freeboard. The transverse coefficients will generally be higher for newer vessels in full load condition and may be higher in ballast conditions depending on actual ballast draft and trim. Likewise, current coefficients contained in Reference 3 are based on vessels with a length to beam ratio of 6.3–6.5:1. Some vessels have a lower ratio, and Reference 3 indicates that the longitudinal current coefficient may be 25% to 30% higher with a length to beam ratio of 5.0.

The appropriate formulae for mooring restraint requirements per Section 2.6 are less sensitive to freeboard changes (since the breast line requirements are based on the transverse coefficient at the aft perpendicular, where the differences between full load and ballast coefficients are much smaller than for the total lateral coefficients). Therefore, these formulae may be used also for newer vessels with higher freeboard.

2.2 STANDARD ENVIRONMENTAL CRITERIA

For all tankers above 16,000 tonnes deadweight intended for general worldwide trading, the mooring restraint available onboard the ship as permanent equipment should be sufficient to satisfy the following conditions:

60 knots wind from any direction simultaneously with either:

3 knots current at 0° or 180°;

or

2 knots current at 10° or 170°;

or

0.75 knots current from the direction of maximum beam current loading.

Water depth to draft ratios for these conditions are to be taken as 1.1:1 when loaded and 3.0:1 when in ballast.

While a number of terminals have a minimum depth to draught ratio alongside the berth of 1.05:1 this ratio will inevitably prevail around low slack water when average current velocities would be less than when the water level is at a depth to draught ratio of 1.1:1. It is therefore suggested that the average velocities previously recommended be used with the 1.1:1 ratio.

Obviously when a terminal designer is reviewing the need for shore augmentation, he will find it necessary to be more precise and should use actual site data for his calculations.

The ballast condition is to correspond to the mean IMO ballast draft given approximately by $0.02L_{BP} + 2.0$m.

Wind velocity is the velocity measured at the standard datum height of 10 m above ground/water surface and is representative of a 30 second average mean velocity. The selection of the 30 second wind is based on the time it takes the forces in a mooring system to respond to wind velocity changes. Thirty seconds is a typical value for a ballasted VLCC. Smaller vessels will respond more quickly; and a fully laden VLCC may require 60 seconds to respond. However, for consistency, a 30 second average period is suggested for all vessel sizes and loading conditions.

The current velocity is to be taken as the average velocity over the draft of the ship.

For gas carriers above 150 metres in length the same standard environmental criteria should be applied. However, the water depth to draft ratio should be taken as 1.1:1 for all conditions, since the draft of a gas carrier changes little during normal cargo transfer operations.

The above criteria are intended to cover conditions that could readily be encountered on worldwide trade, but they cannot possibly cater for the most extreme combination of environmental conditions at every terminal worldwide. Particularly exposed terminals, or those where for some reason the criteria are likely to be exceeded, are expected to supplement ships' mooring restraint with appropriate shore-based equipment.

Where a ship is operating exclusively on a dedicated route using terminals whose specific environmental data is available the recommended criteria may be revised to suit the local conditions.

Although dynamic effects are not explicitly included in the above criteria, the design margins are considered conservative and may allow for some dynamics. At terminals where dynamic effects from passing ships, wave action, etc. are significant, shore-based mooring equipment may have to be deployed to supplement the ship's equipment.

2.3 COMPUTATION OF ENVIRONMENTAL LOADS FOR ANY WIND AND CURRENT COMBINATION

Environmental loads induced by wind and current can be computed with the procedures described in Reference 3 which are also listed in this section for ease of reference. The forces and moments generated are suitable for a computer analysis of the required mooring restraint. The following non-dimensional coefficients are used in the calculation of design loads:

C_x — longitudinal force coefficient
C_y — lateral force coefficient
C_{xy} — yaw moment coefficient

Additional subscripts w and c are used to distinguish between wind and current.

The sign convention and coordinate system adopted are illustrated in Fig. 2.1. Force and moment coefficients are a function of wind and current angle of attack. The wind and current coefficients are based upon data obtained from model tests.

Instead of a lateral force coefficient and a yaw moment coefficient, some sources quote a lateral force coefficient at the forward perpendicular (C_{yF}) and a lateral force coefficient at the aft perpendicular (C_{yA}). This is generally more convenient for hand calculations, but either convention can be converted to the other with the following equations:

(a) From total force/yaw moment system to forces at perpendicular system:

$$C_{yF} = \tfrac{1}{2}C_y + C_{xy}$$

$$C_{yA} = \tfrac{1}{2}C_y - C_{xy}$$

(b) From forces at perpendicular system to total force/yaw moment system:

$$C_y = C_{yF} + C_{yA}$$

$$C_{xy} = \frac{C_{yF} - C_{yA}}{2}$$

The wind forces/moment acting on a moored ship are calculated using the following equations:

Longitudinal wind force
$$F_{xw} = C_{xw}\left(\frac{\rho_w}{7600}\right) V_w^2 A_T \quad (1)$$

Lateral wind force
$$F_{yw} = C_{yw}\left(\frac{\rho_w}{7600}\right) V_w^2 A_L \quad (2)$$

Wind yaw moment
$$M_{xyw} = C_{xyw}\left(\frac{\rho_w}{7600}\right) V_w^2 A_L L_{BP} \quad (3)$$

The current forces/moment acting on a moored ship are calculated using the equations:

Longitudinal current force
$$F_{xc} = C_{xc}\left(\frac{\rho_c}{7600}\right) V_c^2 T L_{BP} \quad (4)$$

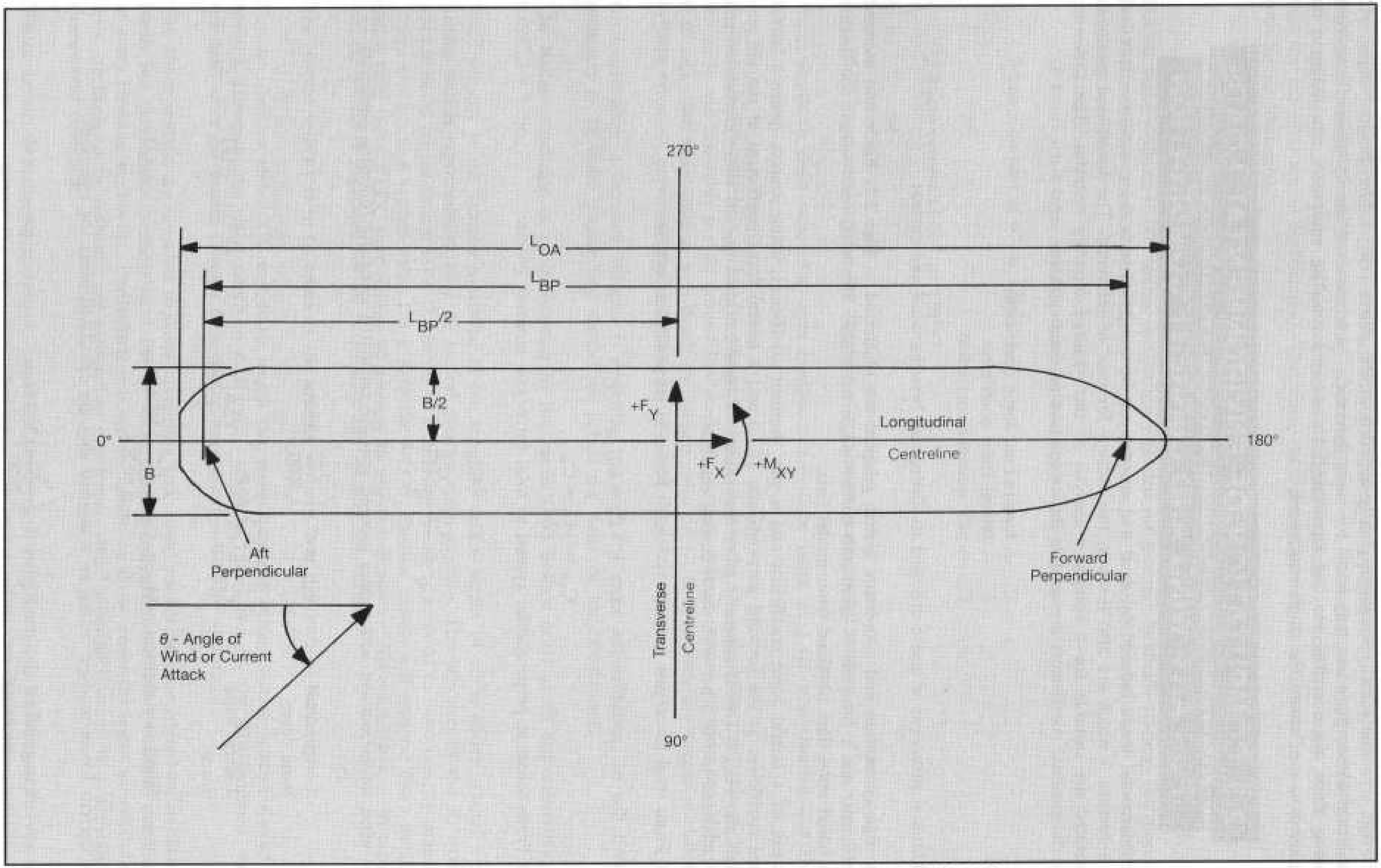


FIGURE 2.1: SIGN CONVENTION AND CO-ORDINATE SYSTEM

Lateral current force	$F_{yc} = C_{yc}\left(\frac{\rho_c}{7600}\right) V_c^2 T L_{BP}$	(5)
Current yaw moment.	$M_{xyc} = C_{xyc}\left(\frac{\rho_c}{7600}\right) V_c^2 T L_{BP}^2$	(6)

Where:

C_{xw} = non-dimensional longitudinal wind force coefficient

C_{yw} = non-dimensional transverse wind force coefficient

C_{xyw} = non-dimensional wind yaw moment coefficient

ρ_w = density of air = 1.223 Kg/m^3 at 20°C

V_w = velocity of wind at 10m elevation in knots

A_T = transverse (head-on) above water area in metre^2 (for condition investigated)

A_L = longitudinal (broadside) above water area in metre^2 (for condition investigated)

L_{BP} = length between perpendiculars in metres

F_{xw} = longitudinal wind force in kN

F_{yw} = lateral wind force in kN

M_{xyw} = wind yaw moment in kN metres

C_{xc} = non-dimensional longitudinal current force coefficient

C_{yc} = non-dimensional transverse current force coefficient

C_{xyc} = non-dimensional current yaw moment coefficient

ρ_c = density of sea water = 1025 Kg/m^3 at 20°C

V_c = average current velocity acting over the draft of ship in knots

T = ship draft in metres (for condition investigated)

F_{xc} = longitudinal current force in kN

F_{yc} = lateral current force in kN

M_{xyc} = current yaw moment in kN metres

For the convention using lateral forces and coefficients at the perpendiculars, the formulae (2) and (5) above apply with the substitution of F_y with either F_{yF} or F_{yA} and C_y with either C_{yF} or C_{yA}.

The shape of the bow may have some influence on the wind and current coefficients. Reference 3 shows two separate values where the coefficients are different. A "conventional" bow represents a bulbous bow, and "cylindrical" bow represents a bow without a bulb and rounded waterlines in the stem area. The different bow shapes are shown in Figure 2.2.

Figures 2.3 through 2.8 are provided for use with hand calculations. Figures 2.6 through 2.8, and Figure 2.2 are taken directly from Reference 3.

Figure 2.3 provides the lateral wind coefficients at the forward and aft perpendicular (C_{yaw} and C_{yfw}) for loaded and ballasted oil tankers. Figure 2.4 provides the longitudinal wind force coefficient for loaded and ballasted oil tankers. Figures 2.3 and 2.4 provide these coefficients for various wind attack angles. Figure 2.5 provides the lateral current force coefficients (C_{YAC} and C_{YFC}) and Figure 2.6 the longitudinal current force coefficients (C_{xc}) for water depth to draught ratios of 1.1 in full load condition and various current attack angles. Figure 2.7 provides the lateral current force coefficients (C_{yAc} and C_{yFc}) in the ballast condition with a water depth to draught ratio of 1.5. Coefficients for the standard ratio of 3.0 are not available.

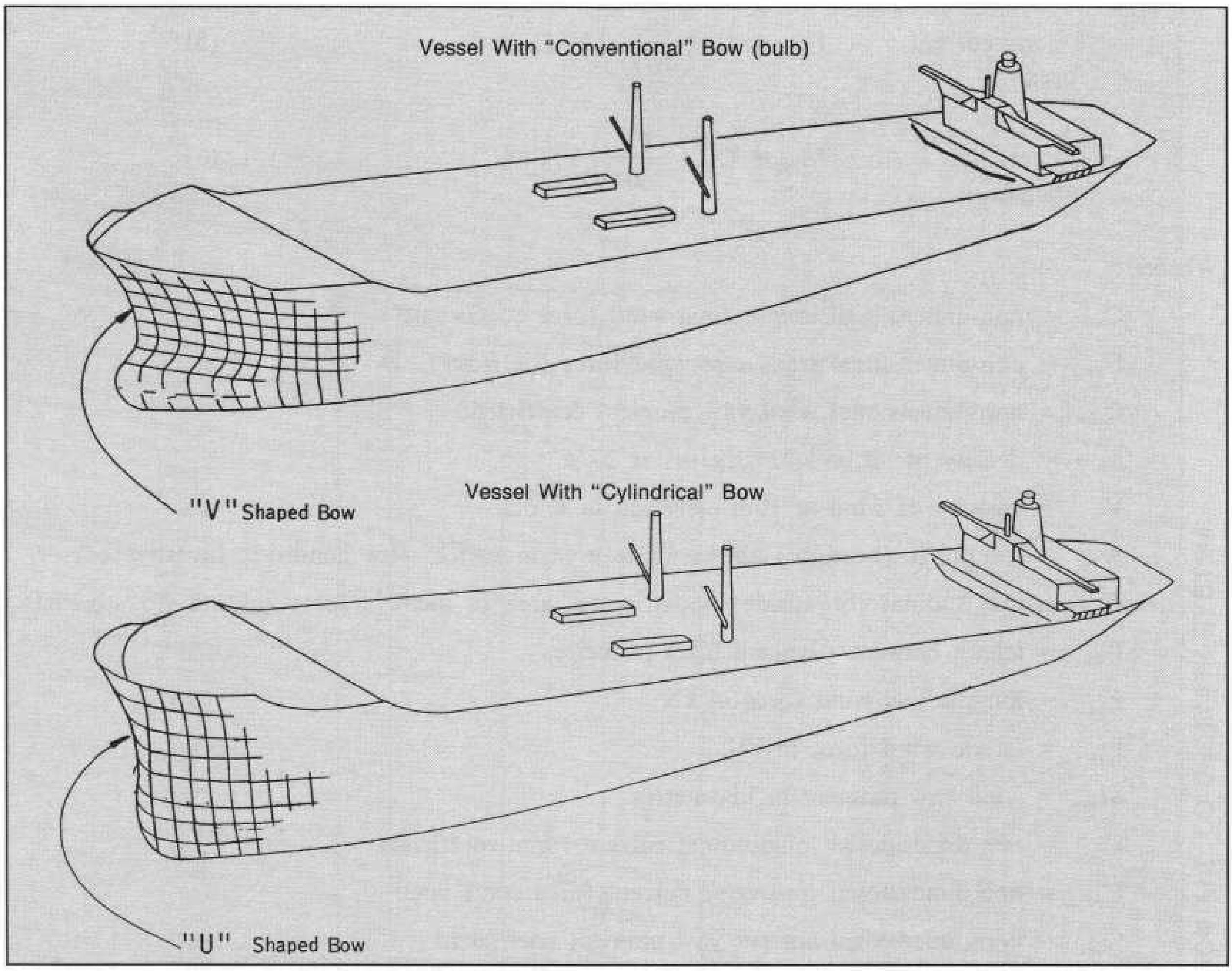


FIGURE 2.2: VARIATION IN BOW CONFIGURATION

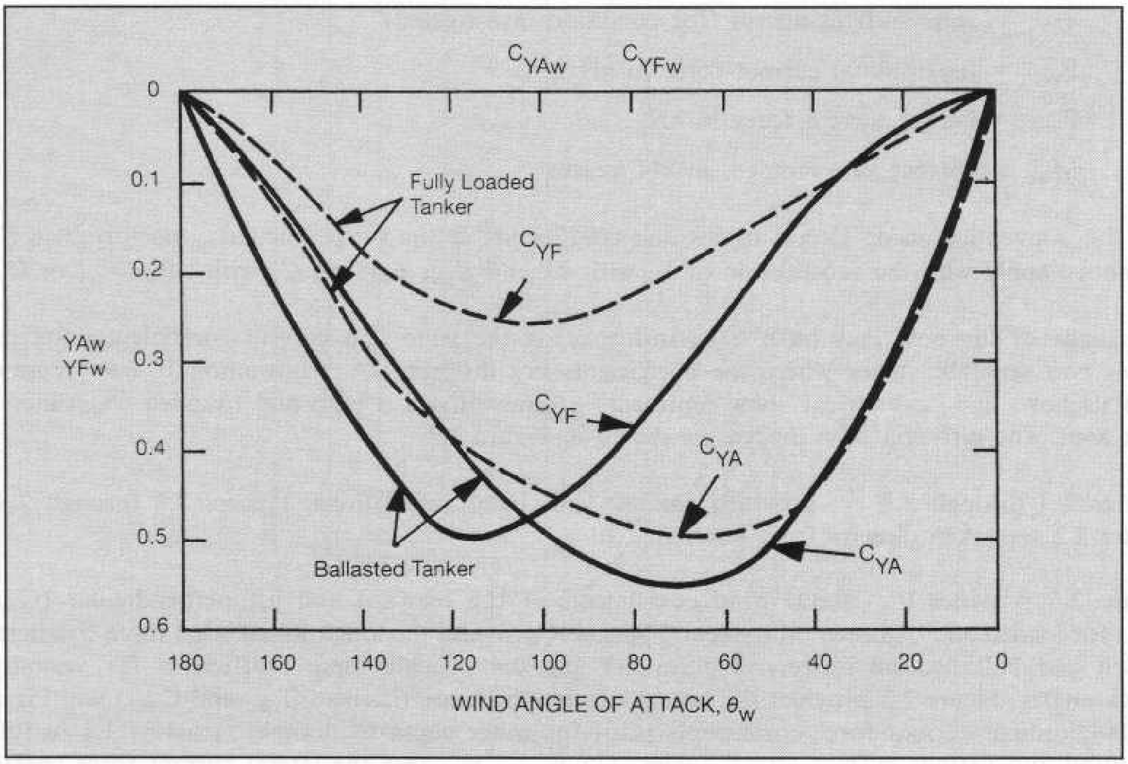


FIGURE 2.3: LATERAL WIND FORCE COEFFICIENT AT THE FORWARD AND AFT PERPENDICULARS

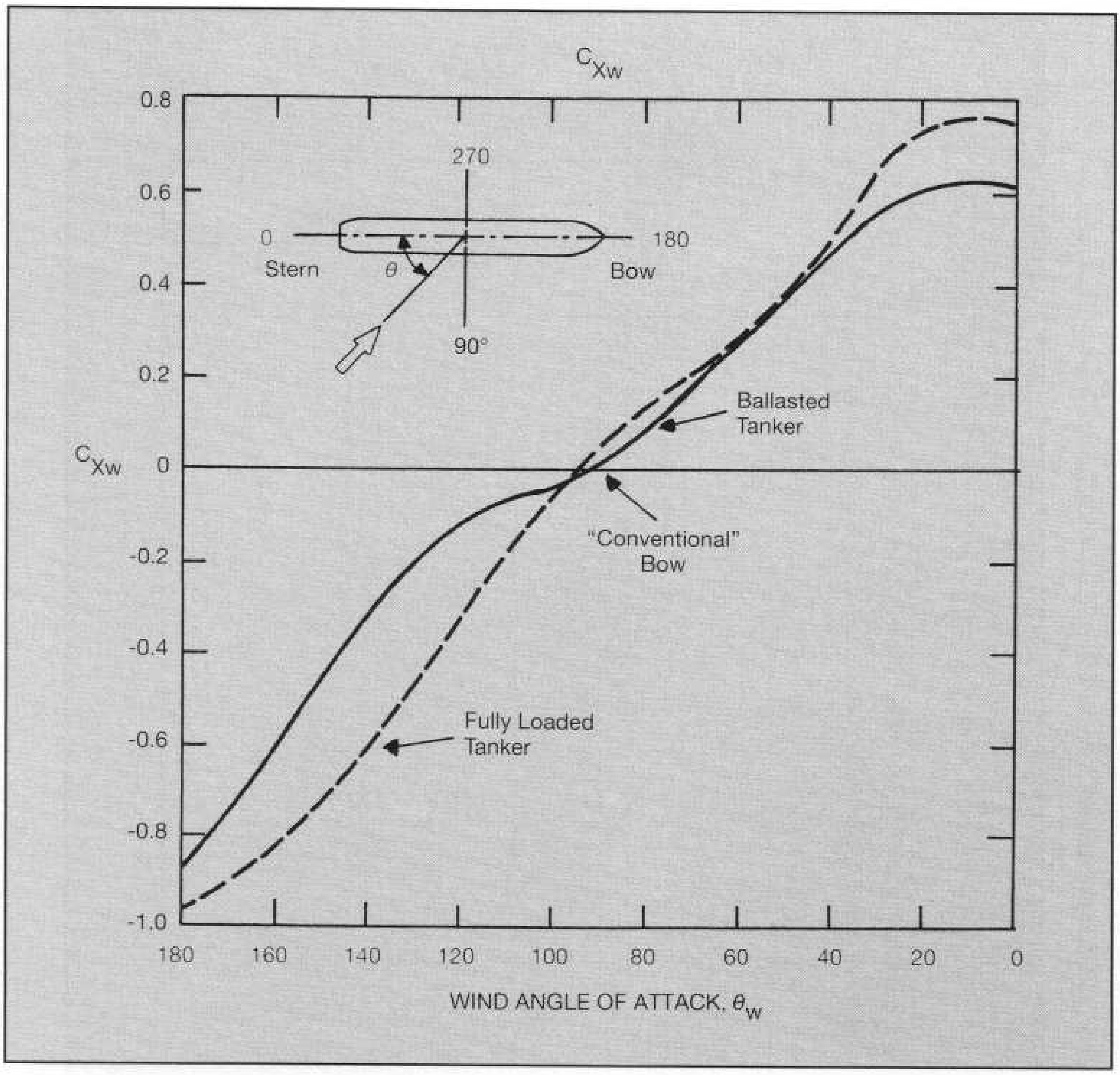


FIGURE 2.4: LONGITUDINAL WIND FORCE COEFFICIENT

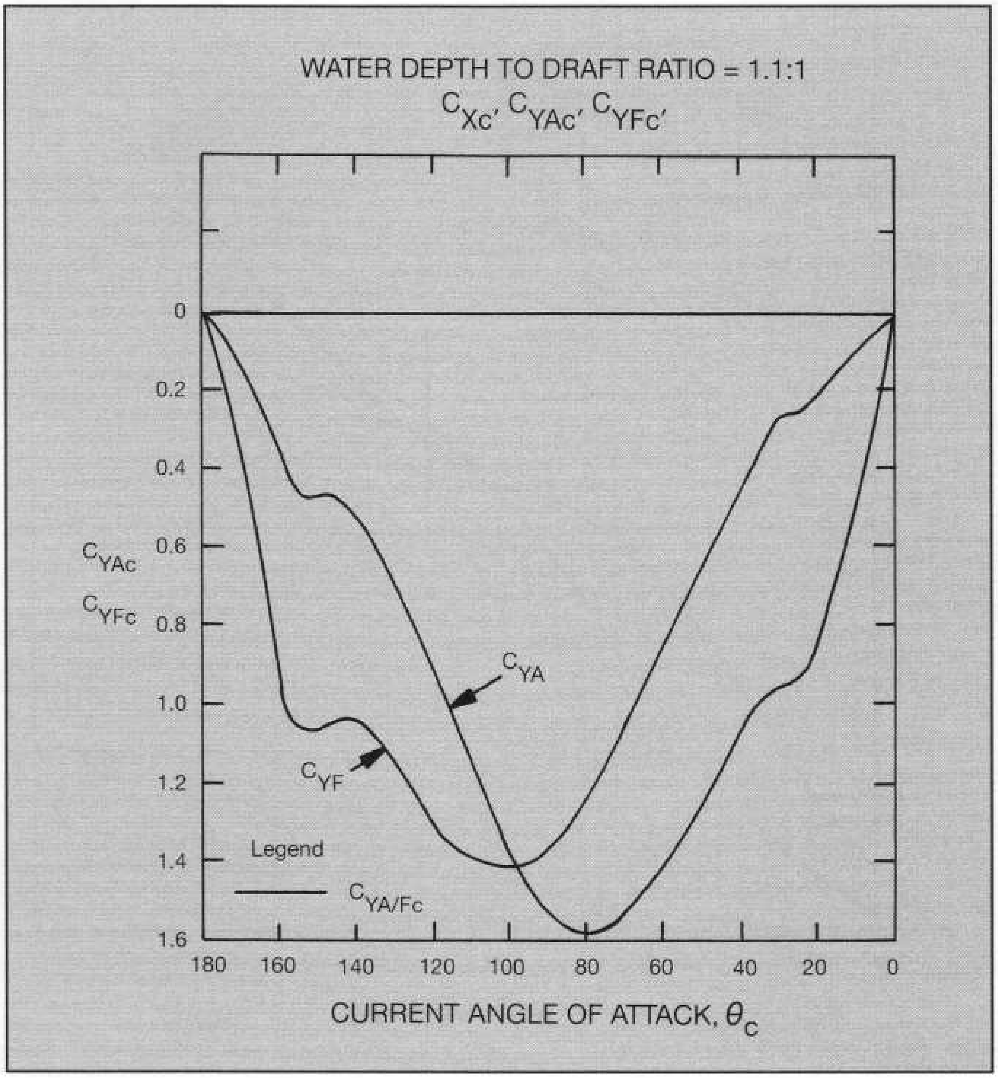


FIGURE 2.5: LATERAL CURRENT FORCE COEFFICIENT AT THE FORWARD AND AFT PERPENDICULARS—LOADED TANKER

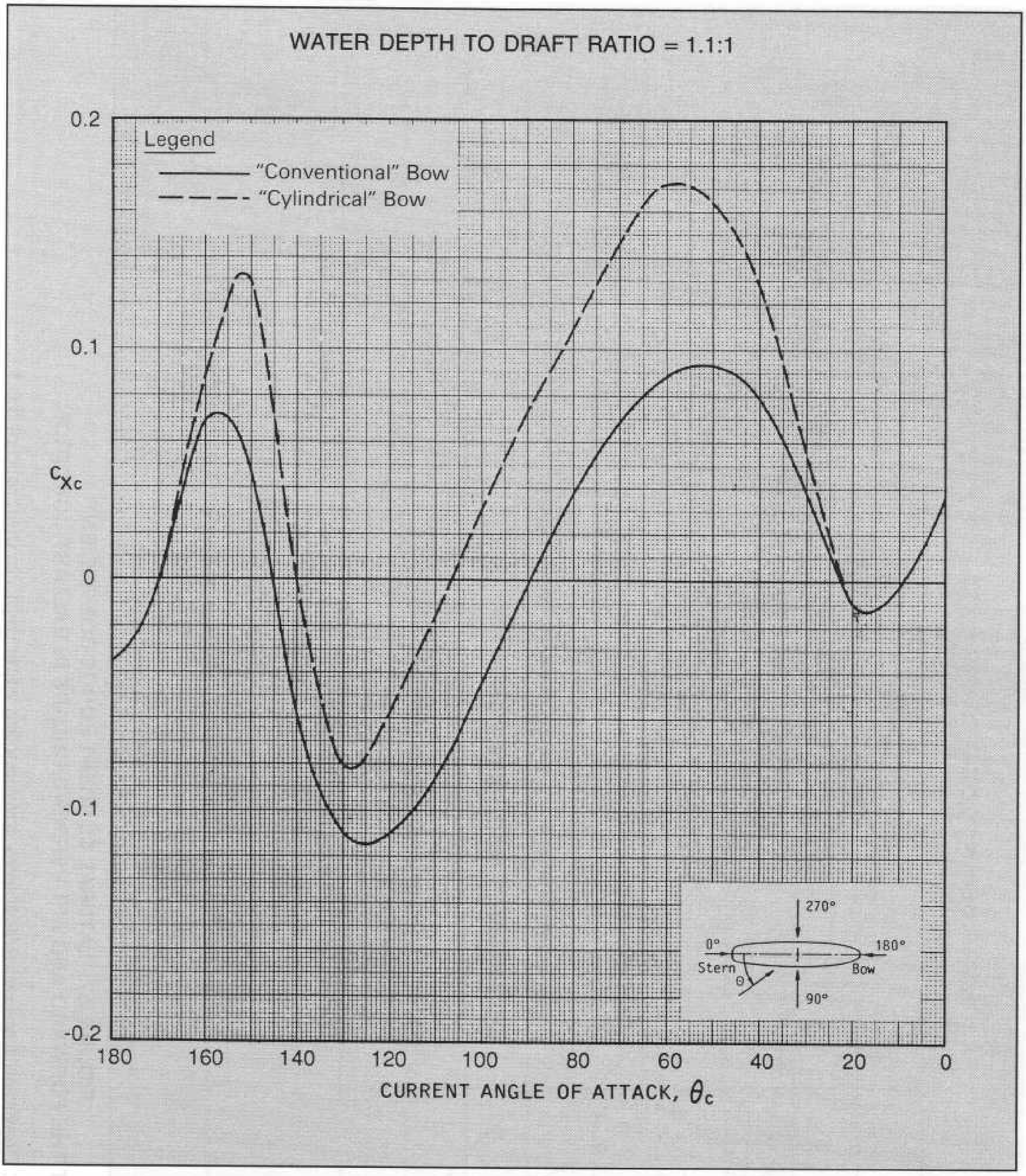


FIGURE 2.6: LONGITUDINAL CURRENT FORCE COEFFICIENT—LOADED TANKER

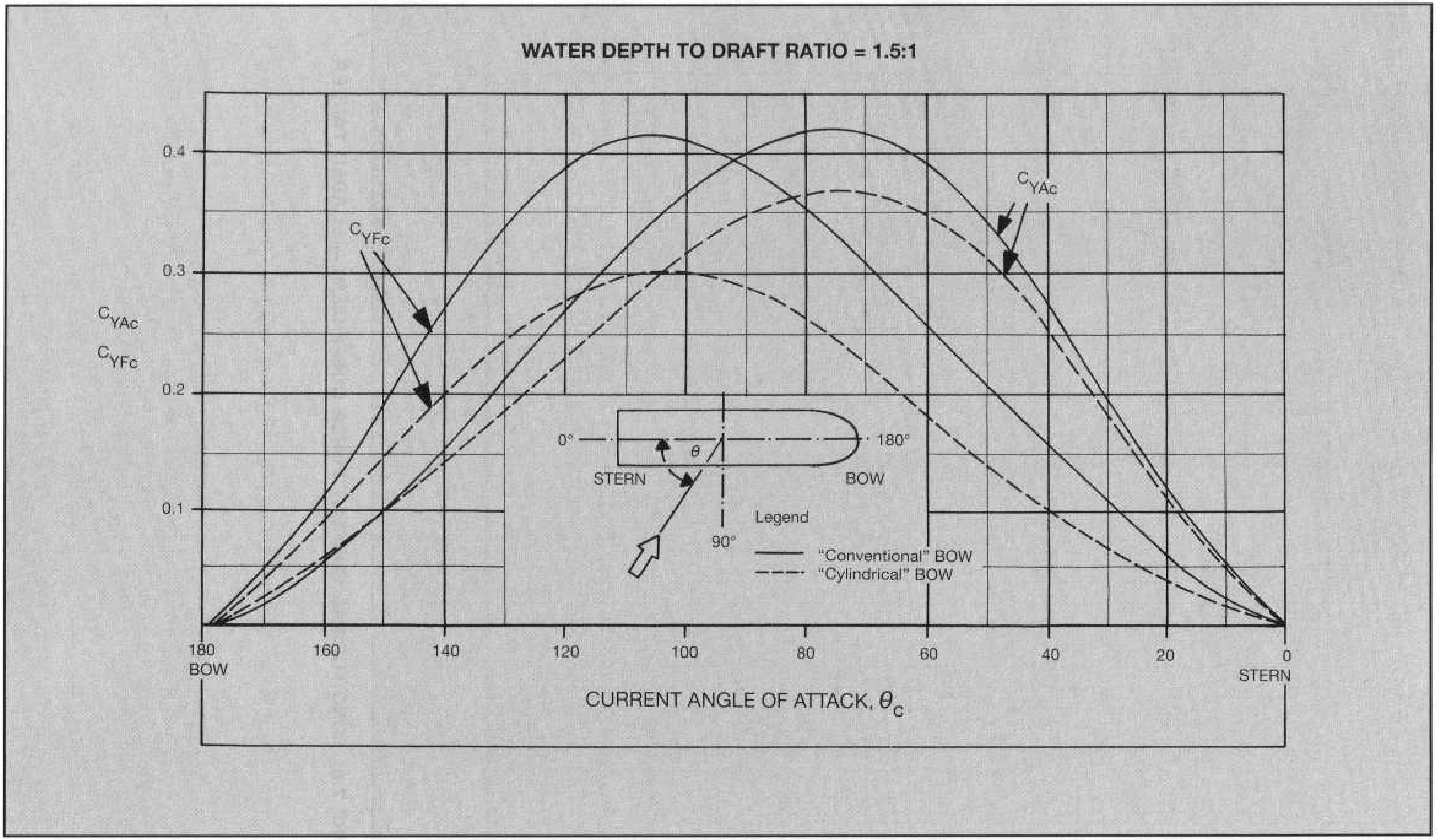


FIGURE 2.7: LATERAL CURRENT FORCE COEFFICIENT AT THE FORWARD AND AFT PERPENDICULARS IN BALLAST CONDITION

NOTE: Coefficients are not available for the standard water depth to draft ratio of 3.0:1.

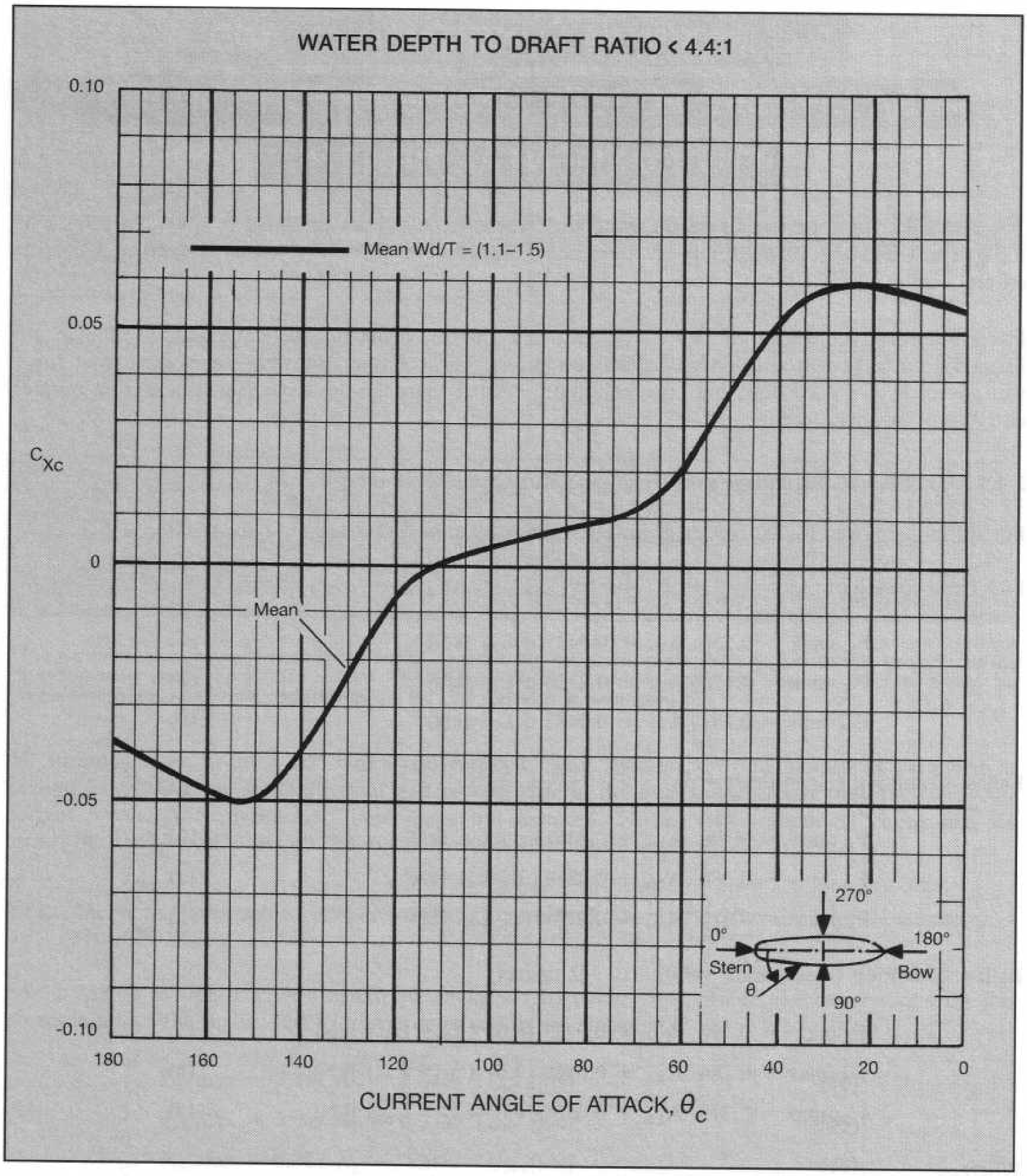


FIGURE 2.8: LONGITUDINAL CURRENT FORCE COEFFICIENT—BALLASTED TANKER

Figure 2.8 provides the longitudinal current coefficient (C_{xc}) in the ballast condition. According to Reference 3 these coefficients are not sensitive to the water depth to draft ratio, and may be used for all ratios, other than deep water cases. NB. note that the attack angle is anti-clockwise from the stern. Also, the positive X and Y directions are forward and up, respectively.

An example for predicting wind and current forces to be used in determining numbers of mooring lines with hand calculations is provided in Appendix D.

Section 2.4 provides information on the maximum combined forces on a vessel when subjected to the Standard Environmental Criteria, both for oil tankers and two types of gas carriers.

2.4 MAXIMUM LONGITUDINAL AND TRANSVERSE FORCES UNDER THE STANDARD ENVIRONMENTAL CRITERIA

The following formulae are based on force and moment coefficients provided in the 1994 edition of Reference 3 and the 1985 edition of Reference 5 and are valid for the standard wind and current criteria outlined in Section 2.2 above.

For oil tankers subject to the standard wind and current environmental conditions, formulae are listed for both the full and loaded ballast conditions. The condition showing the highest total force will govern. For gas carriers, only one condition is listed since the draft changes little during normal cargo transfer operations.

2.4.1 *Maximum Longitudinal and Transverse Forces on Oil Tankers*

For oil tankers over 16,000 tonnes deadweight the maximum forces are as follows:

Full Load Condition:

$$F_x \max = 0.550\ A_{TF} + 0.0425\ d\ L_{BP}\ [kN] \quad (7)$$

$$F_{yF} \max = 0.1506\ A_{LF} + 0.2320\ d\ L_{BP}\ [kN] \quad (8)$$

$$F_{yA} \max = 0.2839\ A_{LF} + 0.2482\ d\ L_{BP}\ [kN] \quad (9)$$

Ballast Condition (mean draft $= d_B$)

$$F_x \max = 0.498\ A_{TB} + 0.04613\ d_B\ L_{BP}\ [kN] \quad (10)$$

$$F_{yF} \max = 0.284\ A_{LB} + 0.02595\ d_B\ L_{BP}\ [kN] \quad (11)$$

$$F_{yA} \max = 0.319\ A_{LB} + 0.02617\ d_B\ L_{BP}\ [kN] \quad (12)$$

Ballast Condition (mean draft $= 0.02\ L_{BP} + 2$ metres)

$$F_x \max = 0.498\ A_{TB} + 0.0923\ (0.01\ L_{BP}^2 + L_{BP})\ [kN] \quad (13)$$

$$F_{yF} \max = 0.284\ A_{LB} + 0.0520\ (0.01\ L_{BP}^2 + L_{BP})\ [kN] \quad (14)$$

$$F_{yA} \max = 0.319\ A_{LB} + 0.0524\ (0.01\ L_{BP}^2 + L_{BP})\ [kN] \quad (15)$$

Where:

d = full load (summer) draft in metres

A_{TF} = transverse above water area in the full load condition, in square metres

A_{TB} = transverse above water area in the ballast condition, in square metres

A_{LF} = longitudinal above water area in the full load condition, in square metres

A_{LB} = longitudinal above water area in the ballast condition, in square metres

The maximum forces are based on the following wind and current coefficients.

FORCE / CONDITION	WIND			CURRENT			
	COEFF. —	DIRECT. °	SPEED KNOTS	COEFF. —	DIRECT. °	SPEED KNOTS	APPLICABLE WATER DEPTH TO DRAFT RATIO
F_x / FULL LOAD	−.95	180	60	−.035	180	3.00	$\frac{WD}{T} = 1.10$
F_{yF} / FULL LOAD	+.26	100	60	+.43	170	2.00	$\frac{WD}{T} = 1.10$
F_{yA} / FULL LOAD	+.49	65	60	+.46	10	2.00	$\frac{WD}{T} = 1.10$
F_x / BALLAST	−.86	180	60	−.038	180	3.00	$\frac{WD}{T} < 4.4$
F_{yF} / BALLAST	+.49	115	60	+.342*	110	0.75	$\frac{WD}{T} = 3.0$
F_{yA} / BALLAST	+.55	70	60	+.345*	75	0.75	$\frac{WD}{T} = 3.0$

*The transverse current coefficients in the ballast condition are estimated values since no test data is available for the standard water depth to draft ratio of 3.0:1. The accuracy of these coefficients generally does not affect the breast line requirements since these are governed by the full load condition in most cases.

All formulae may be used for both "conventional" and "cylindrical" bow shapes as described in Section 2.3. Under the standard wind and current criteria the bow shape affects only the transverse current forces. A "conventional" bow shape was assumed. Forces with "cylindrical" bows may be lower, but the overall effect on the maximum transverse force is not significant.

2.4.2 *Maximum Longitudinal and Transverse Forces on 75,000 m³ Gas Carrier with Membrane or Prismatic Tanks*

For a 75,000 m^3 gas carrier (membrane tanks or prismatic tank, where the cargo tanks do not protrude above the upper deck) the maximum combined forces are:

$$F_x \text{ max} = 0.557\ A_T + 0.0425\ TL_{BP}\ [kN] \qquad (16)$$

$$F_{yF} \text{ max} = 0.331\ A_L + 0.2320\ TL_{BP}\ [kN] \qquad (17)$$

$$F_{yA} \text{ max} = 0.387\ A_L + 0.2482\ TL_{BP}\ [kN] \qquad (18)$$

2.4.3 *Maximum Longitudinal and Transverse Forces on 125,000 m³ Gas Carrier with Spherical Tanks*

For a 125,000 m^3 gas carrier (spherical tanks) the forces are:

$$F_x \text{ max} = 0.597\ A_T + 0.0425\ TL_{BP}\ [kN] \qquad (19)$$

$$F_{yF} \text{ max} = 0.345\ A_L + 0.2320\ TL_{BP}\ [kN] \qquad (20)$$

$$F_{yA} \text{ max} = 0.365\ A_L + 0.2482\ TL_{BP}\ [kN] \qquad (21)$$

Symbols and units are the same as used in Section 2.3. Since a gas carrier's draft changes little during normal cargo transfer operations. the coefficients for a water depth to draft ratio of 1.1:1 have been used throughout. The above formulae for gas carriers should only be used for gas ships that are geometrically similar to those listed in Reference 5.

2.5 MOORING RESTRAINT REQUIREMENTS

Having determined the environmental forces acting on the ship it is necessary to calculate the strength and number of mooring lines required to balance these forces.

Computer programmes are widely available to carry out this task for any combination of wind and current speeds and angles. A hand calculation method is described in Appendix D. For either method the three-dimensional coordinates of all ship and terminal mooring points must be known or assumed, as well as the elastic characteristics of the mooring lines. Computer calculations are especially suitable to explore the adequacy of the mooring system of an existing or planned ship at a terminal known to have unusual environmental conditions or mooring geometry.

Hand calculations are not always adequate for use by the terminal designer in evaluating either mooring limitations or the need for, and amount of, shore augmentation which may be required for a given vessel at a given berth for given operating environmental conditions. In this case it is necessary to evaluate various mooring arrangements which cater for the particular combinations of forces in order to determine that arrangement which is the best for that berth. The simplifying assumptions (or the laborious, time-consuming and costly effort to avoid them) may in this case render a hand calculation method impractical. A more detailed discussion of the basic principles of mooring calculations is provided later.

Static and dynamic computer programs are available. Thus, in designing terminal facilities, the engineer is advised to use the tools and technology which will aid in the design of a safe berth.

2.5.1 *Basic Principles of Mooring Calculations*

A tanker at a jetty or sea island is held in the berth by a combination of mooring lines and breasting dolphins which resist the forces applied to the tanker by wind, current, waves and other environmental factors. There are multiple mooring lines which connect from several locations on the ship to several mooring points on the jetty or sea island. It is a system in which the forces in the lines cannot be calculated solely by the principle of static equilibrium, so the solution must consider the elasticity of the components. Basically, the tanker and mooring system together may be considered as a two-dimensional elastic system such that when a load is applied to the tanker, the tanker will move a small but determinable amount. By determining the amount of movement of the tanker, the forces in the mooring lines and breasting dolphins can be determined.

For steady-state forces on the tanker (current and wind forces are considered steady-state forces for the purposes of the analysis), the forces in the mooring lines are determined using the following basic principles or characteristics:

2.5.1.1 *The Principle of Static Equilibrium*

The sums of the components of the forces in the mooring lines and breasting dolphins in each principal direction and the moment of forces about the centre of the tanker are equal and opposite to the sums of the components of the applied forces (current and wind) and the moment of these forces. The principal directions are ahead (or astern) and abeam, and the moment is a yawing moment.

2.5.1.2 *The Load/Deflection Characteristics of each Mooring Line and Breasting Dolphin*

For each there is a relationship between its elongation (for mooring lines in tension) or inward deflection (for breasting dolphins in compression) and the force in the member.

For mooring lines, the load/deflection characteristic is dependent on the material and construction of the line, its diameter and the loaded length (i.e., from the ship's winch to the mooring point on the jetty or sea island). Mooring lines become stiffer (less stretch for a given load) with use, and the characteristic for used line is normally employed for calculating loads rather than the characteristic for new line. The characteristics can be obtained from line manufacturers or suppliers.

For breasting dolphins, the characteristics for manufactured fender units are available from the manufacturer. Deflection of the dolphin structure, if significant, can be calculated from the properties of the structure.

2.5.1.3 *The Geometrical Relationship Between the Parts of the System*

The elongation of each mooring line and deflection of each breasting dolphin can be calculated from the amount of surge, sway and yaw at the centre of the tanker. Since the tanker is essentially a rigid element, each chock through which a line passes effectively moves in relation to the mooring point thus changing the distance between the chock and the mooring point on the jetty or sea island.

Using the above principles or system characteristics, the forces in the mooring lines for wind and current forces are determined by the following general procedure:

1. Calculate the applied forces in the fore/aft direction and the beam direction and the yaw moment for wind and current as described in Section 2.3 on wind and current force predictions.

2. Determine the elasticity of the entire mooring system from the load/deflection characteristics of each component and the geometry of the system. The elasticity of the system is expressed in terms of amount of surge, sway and yaw per unit force in each principal direction and per unit yaw moment.

3. Calculate the total amount of surge, sway and yaw at the centre of the tanker by multiplying the amount per unit force and moment determined in Step 2 by the applied forces and moment calculated in Step 1. Then calculate the new location of each chock point.

4. Determine the force in each mooring line and breasting dolphin by calculating the stretch in the line (or compression of the dolphin) for the movement of the tanker calculated in Step 3 and finding the corresponding force from the load/deflection characteristic.

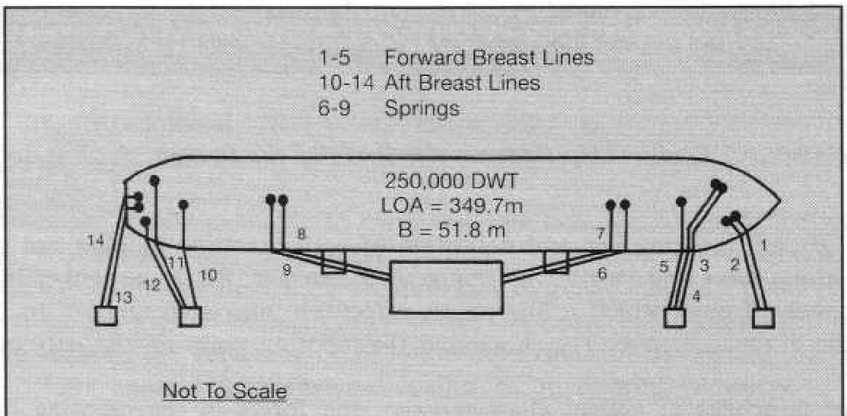
The above calculations are complicated by the fact that the elasticity of the system varies with the actual position of the tanker because for any particular position some lines may not be in tension and some breasting dolphins not in compression. These must be omitted in calculating the elasticity of the system in that position. Also the load/deflection characteristics of the components may not be linear. This is the case for synthetic fibre lines and most elastomer render units. Consequently, the solution involves an interactive process in which the elasticity of the system is first determined for an assumed position of the tanker. A new position is then determined by the above procedure and the elasticity adjusted. This process is repeated until the desired accuracy of the calculated line forces is obtained.

2.5.2 *Computer Calculations*

Because of the complexity of the calculations, they are generally performed using a computer. A computer program can be written incorporating the above procedure and using matrix techniques for the solution. When a computer or computer program is not available, mooring line loads can be approximated by the hand-calculation method described later. Since the hand-calculation method does not incorporate all the principles noted above, the calculated values will only be rough approximations.

The above method can be used for calculating mooring line loads for other steady-state forces such as drift due to waves and ice. Once the applied forces have been established, the loads in the lines can be determined using Steps 1 to 4.

For dynamic loadings such as the oscillatory forces from waves, swell, seiche and surges from passing vessels, the maximum forces in the lines and breasting dolphins may be calculated from the maximum movement of the tanker in surge, sway and yaw as determined from model tests or analytical techniques. Once the maximum movement of the tanker is determined, the maximum forces in the mooring lines are determined as in Step 4 above.



Inboard Lengths Of Lines

Line No.	Length In Metres	Line No.	Length In Metres
1	6	8	35
2	6	9	35
3	42.5	10	32
4	44	11	32
5	35	12	7.5
6	35	13	6
7	35	14	6

FIGURE 2.9: MOORING LAYOUT USED FOR COMPUTATIONAL PURPOSES

Pre-tensioning of the mooring lines may be taken into account in determining the mooring line forces. When this is done, the force calculated in Step 4 is added to the pre-tension force. The mooring line force calculated in Step 4 may be negative reducing the pre-tension provided that the net load is never negative. The pre-tension forces in the lines and the forces in the breasting dolphins should be in equilibrium before applying forces due to wind, current and other environmental factors.

2.5.3 *Hand Calculation Procedures*

The hand calculation method for assessing the restraint capabilities of mooring layouts is intended for use when circumstances preclude the application of more sophisticated computer methods. In addition, it can be used to improve the quantitative understanding of good mooring principles.

The method is described in detail in Appendix D.

It uses the basic principles outlined earlier in this chapter but includes a number of assumptions and approximations to yield a workable procedure. As a result it is reasonably accurate provided certain limitations are observed. The mooring layout should comply with the Principles of Mooring given in Section 1, notably the separation of lateral and longitudinal restraint functions, symmetry of layout, avoidance of poorly angled head and stern lines, similarity of line type and size. If such criteria are observed the answers produced are within 10% of those from computer methods. Bearing in mind the accuracy of other input data, it is considered that the method has quite adequate validity if applied with discretion.

Appendix D includes a worked example and the results are compared with a computer method solution. The particular mooring layout used is well-conditioned and thus the results agree quite closely with those of the computer method.

Another way to determine a vessel's general mooring line requirement without the knowledge of a specific terminal geometry is to base the requirement on the maximum components of the environmental forces, and an efficiency factor for each line group. Such method is the basis for the formulae listed in Section 2.6. For instance, assuming an efficiency of 90% for either forward or aft spring lines, and 70% for either forward or aft breast lines, and further assuming that the maximum allowable load in a line is 55% of a line's minimum breaking load, the requirement would be as follows:

Spring lines, either forward or aft: $S \times MBL = \dfrac{F_x \text{ max.}}{0.9 \times 0.55} = 2.0\ F_x \text{ max.}$

Forward breast lines: $BR \times MBL = \dfrac{F_{yF} \text{ max.}}{0.7 \times 0.55} = 2.6\ F_{yF} \text{ max.}$

Aft breast lines: $BR \times MBL = \dfrac{F_{yA} \text{ max.}}{0.7 \times 0.55} = 2.6\ F_{yA} \text{ max.}$

Symbols are the same as used in Sections 2.3 and 2.6, except that the number of lines (S, BR) relates to the individual groups listed above.

The maximum force components (F_x max., F_{yF} max., F_{yA} max.) under the standard environmental criteria can be calculated by formulae (7) to (15) in Section 2.4.1 for oil tankers, and by formulae (16) to (21) in Section 2.4.2 for gas carriers.

This method assumes that the mooring layout complies with good principles as mentioned above for the more involved hand calculation method.

2.5.4 *Standard Restraint Requirements*

To obtain a uniform standard of mooring equipment for tankers not designed for a specific trade or terminal, it is recommended that a ship designer:

1. Follow the principles provided in Section 3 regarding the placement of winches, chocks, and fairleads.

2. Assume breast lines to be at an angle of 75° to the longitudinal axis of the ship. A horizontal angle of 10° to the side of the vessel should be assumed for spring lines. Maximum vertical angles of 25° should be assumed for the lightest ballasted condition. These criteria therefore determine the position of mooring points for an idealised mooring line layout. Figure 2.10 illustrates the layout.

3. Calculate the number of breast lines and spring lines that would be required for the "Standard Environmental Criteria" and for the idealised mooring layout as discussed in 1 and 2 above.

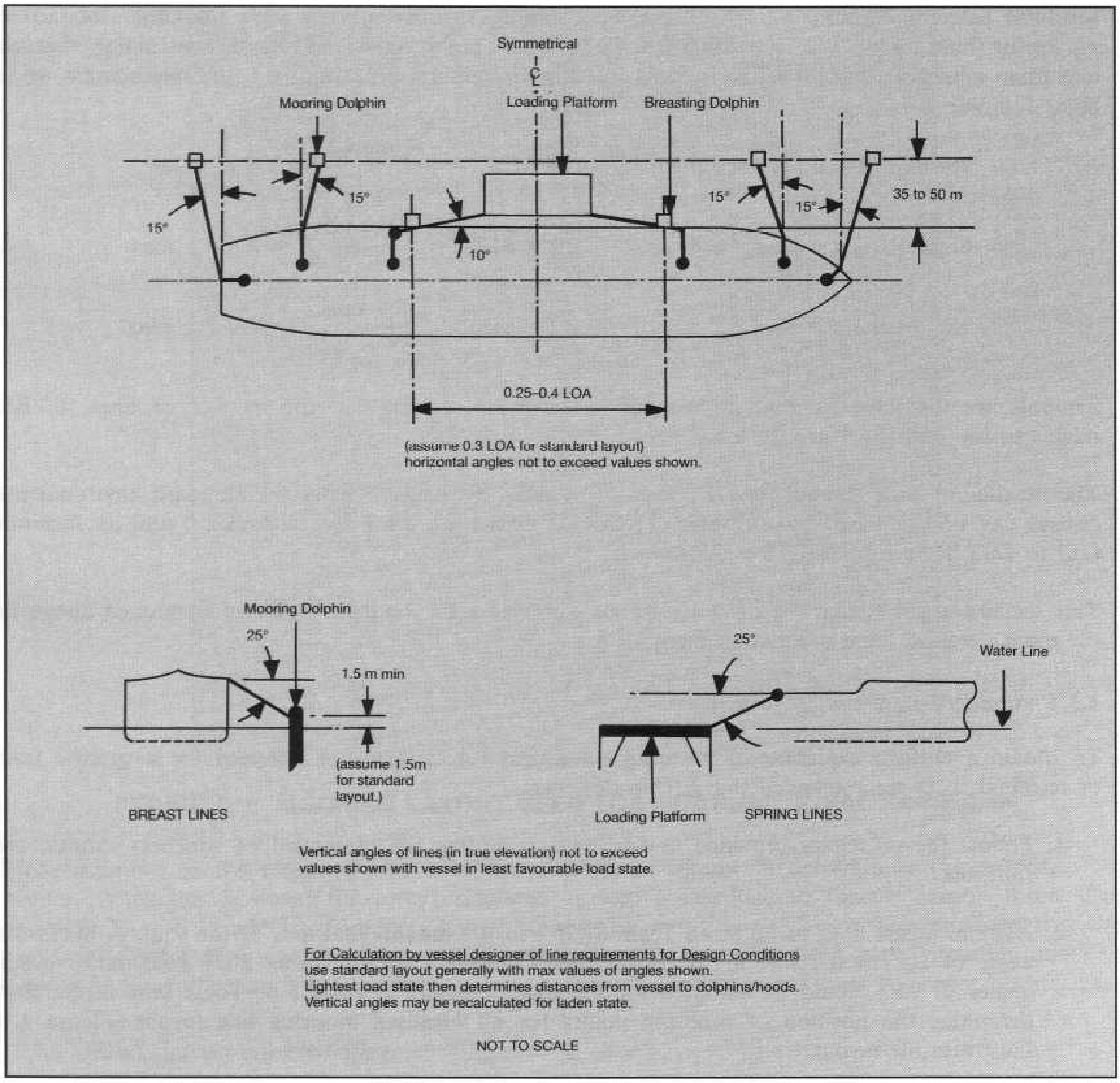


FIGURE 2.10: IDEALISED MOORING LINE LAYOUT

2.6 APPROXIMATE METHODS FOR DETERMINING MOORING RESTRAINT REQUIREMENTS FOR TANKERS AND GAS CARRIERS UNDER STANDARD ENVIRONMENTAL CRITERIA

In practice it is often necessary to evaluate a ship's mooring equipment without the detailed knowledge of all ship and berth parameters. For this purpose the formulae quoted in this Section may be used, providing the underlying assumptions on which they are based are applicable.

The formulae are developed on the philosophy that spring line restraint should be proportional to the maximum longitudinal force on the ship due to the standard environmental loads, and the breast line restraint should be proportional to the maximum transverse forces at the ship perpendiculars. Although this approach does not consider the variations in actual mooring patterns, it provides a rational "scale" for mooring restraint systems of different size ships, and as such is more realistic than the present Classification Society guidelines. This is because Classification Society guidelines base the mooring equipment on the IACS equipment numeral. While this is a good basis for anchoring equipment, it does not adequately reflect the transverse loads imposed upon a restrained ship.

The formulae for oil tankers are based on formulae (9) and (13) and assume that the maximum longitudinal force occurs in the ballast condition and the maximum transverse force at the aft perpendicular in the full load condition. The validity of this assumption should be checked by use of the formulae listed in Section 2.4.1.

2.6.1 *Mooring Restraint Requirements for Oil Tankers above 16 kDWT*

2.6.1.1 *Wire systems*

Spring lines:

$$S \times MBL = 1.992\ A_{TB} + 0.369(0.01\ L_{BP}^2 + L_{BP}) \quad (22)$$

Breast lines:

$$BR \times MBL = 1.476\ A_{LF} + 1.291\ TL_{BP} \quad (23)$$

Where:

S = total number of spring lines
BR = total number of breast lines
MBL = minimum breaking load of each line in kN
T = maximum (summer) vessel draft

Assumptions:

(a) Environmental conditions correspond to the standard criteria outlined in Section 2.2. Maximum forces on the ship are derived by formula (9) for the longitudinal direction and by formula (13) for the transverse direction.

(b) The mooring layout follows good mooring principles (as per Section 1). A layout efficiency factor based upon experience with good layouts has been used.

$$\text{'Efficiency'} = \frac{\text{restraint capability of line group}}{\text{max. load in any one line} \times \text{no. of lines in group}}$$

Line groups are broken down into spring lines, forward breast lines and aft breast lines.

One hundred percent efficiency would represent equal length breast lines at 90° to the longitudinal axis of the ship and equal length spring lines parallel to the longitudinal axis, all in the horizontal plane.

The following efficiencies are assumed:

Spring lines 0.90 = 90%
Breast lines 0.70 = 70%

(c) All mooring lines are of steel.

(d) The maximum allowable tension in any one line is 55% of the minimum breaking load of the line, and the winch brakes must be capable of holding this load.

(e) Spring and breast lines are arranged symmetrically about midships so that the forward number of lines is equal to the aft number.

(f) The minimum draft of oil tankers corresponds to the IMO draft, which is approximately $0.02\ L_{BP} + 2.0$ m.

2.6.1.2 *Synthetic Line Systems*

Due to the lower recommended allowable loads in synthetic lines (refer to Section 4), it is recommended that the values of '$S \times MBL$' and '$BR \times MBL$' obtained by formulae (22) and (23) be increased by 20% for nylon and by 10% for other conventional synthetic mooring lines. In case of newer, low stretch, synthetic materials, such as Aramid, specific recommendations have not been made at this time due to the limited experience.

2.6.2 *Rough Spring Line Restraint Determination for Oil Tankers above 16 kDWT when the Ship's Transverse Wind Area is Unknown*

Where a ship's length, beam and depth are known, a very approximate determination of spring line requirements for tankers with wire mooring lines can be made using the following formula:

$$S \times MBL = 1.992DB - 0.0398\, L_{BP} + 19.9B + 40 + 0.369\, (0.01L_{BP}^2 + L_{BP}) \qquad (24)$$

The formula is derived from formula (22) by substituting the following for A_{TB}

$$A_{TB} = BD - 0.02\, L_{BP} + 10B + 20$$

The superstructure and house transverse area is roughly approximated by "10B + 20".

2.6.3 *Rough Breast Line Restraint Determination for Oil Tankers above 16 kDWT when the Ship's Lateral Wind Area is Unknown*

Where a ship's length, summer draft and depth are known, a very approximate determination of breast line requirements for tankers with wire mooring lines can be made using the following formula:

$$BR \times MBL = L_{BP}[1.550\,(D - T) + 1.291\,T + 2.0664] + 517 \qquad (25)$$

This formula is derived from formula (23) by substituting the following for A_{LF}:

$$A_{LF} = 1.05\, L_{BP}\,(D - T) + 1.4\, L_{BP} + 350$$

where '$D - T$' is the freeboard midships in the full load condition.

The term "$1.4\, L_{BP} + 350\ m^2$", constitutes a rough statistical approximation of the superstructure and house lateral area. Even though this value can vary considerably from tanker to tanker, the inaccuracy arising from variance in superstructure and house area is generally small. This is due to the relatively far greater importance assigned to the hull area.

A formula of this type may be used by a terminal operator to assess the suitability of individual ship moorings at his berth rather than relying on a general set of requirements based solely on deadweight or displacement.

2.6.4 *Mooring Restraint Requirements for Gas Carriers*

The required mooring restraint for gas carriers may be approximated by the method described in Section 2.6.1. The same assumptions as listed in Section 2.6.1.1 apply, except that maximum forces to be restrained are based on formulae (16) and (19) for the longitudinal direction and formulae (18) and (21) for the transverse direction.

The following approximate mooring line requirements for gas carriers apply to wire rope systems. For synthetic line systems refer to Section 2.6.1.2.

For a 75,000 m^3 gas carrier:

$$S \times MBL = 2.228\ A_T + 0.170\ TL_{BP} \quad (26)$$

$$BR \times MBL = 2.012\ A_L + 1.291\ TL_{BP} \quad (27)$$

For a 125,000 m^3 gas carrier (spherical tanks):

$$S \times MBL = 2.388\ A_T + 0.170\ TL_{BP} \quad (28)$$

$$BR \times MBL = 1.898\ A_L + 1.291\ TL_{BP} \quad (29)$$

2.6.5 *Example Calculations*

The following examples show the application of the approximate formulae.

- *For an oil tanker of 90 kDWT with segregated ballast tanks*

(a) Ship particulars:

L_{BP} = 234.0 m

B = 39.6 m

D = 20.6 m

d = 14.0 m

A_{TF} = 755 m^2 (at design draft d); A_{TB} = 1044 m^2 (at ballast draft)

A_{LB} = 4200 m^2 (at 6.7 m draft = 0.02 L_{BP} + 2m); A_{LF} = 2492 m^2 (at design draft d)

(b) Maximum forces on the ship, by use of formulae (7), (8), (9), (13), (14) and (15)

Full Load:

F_x max = 415 + 139 = 554 kN

F_{yF} max = 375 + 760 = 1135 kN

F_{yA} max = 707 + 813 = 1520 kN

Ballast (IMO draft):

F_x max = 520 + 72 = 592 kN

F_{yF} max = 1193 + 41 = 1234 kN

F_{yA} max = 1340 + 41 = 1381 kN

(c) Spring line requirements, by use of formula (22):

$$S \times MBL = 2080 + 288 = 2368 \text{ kN}$$

Assuming the ship will have a total of four spring lines (S = 4), the minimum breaking strength required for each line will be:

$$MBL = \frac{2368}{4} = 592 \text{ kN } (= 60 \text{ tonnes force})$$

(d) Breast line requirements, using formula (23):

$$BR \times MBL = 3678 + 4229 = 7907 \text{ kN}$$

Assuming that the ship has a total of eight breast lines (B = 8) symmetrically arranged (four forward and four aft), the minimum breaking strength required for each line will be:

$$MBL = \frac{7907}{8} = 988 \text{ kN } (= 101 \text{ tonnes force})$$

(e) Spring line requirement by use of formulae (24):

$$S \times MBL =$$
$$1.992 \times 20.6 \times 39.6 - 0.0398 \times 234 \times 39.6 + 19.9 \times 39.6 + 40 + 0.369(0.01 \times 234^2 + 234)$$
$$= 1625 - 368.8 + 788 + 40 + 288.4$$
$$= 2373 \text{ kN}$$

with $S = 4$

$$MBL = \frac{2373}{4} = 593 \text{ kN } (= 60 \text{ tonnes force})$$

(f) Breast line requirement by use of formula (25):

$$BR \times MBL = 234[1.550(20.6 - 14.0) + 1.291 \times 14 + 2.0664] + 517$$
$$= 7624 \text{ kN}$$

with $BR = 8$

$$MBL = \frac{7624}{8} = 953 \text{ kN } (= 97 \text{ tonnes force})$$

(g) Actual lines provided:

Since the wind area for this ship is known, the more accurate formulae (22) and (23) should be used in calculating line requirements. [The example ship has a forecastle and poop with a larger wind area than the statistical average]. Also, as recommended, all lines are of the same size and material. This ship is therefore provided with a total of 12 lines (four spring lines, eight breast lines) of 40 mm diameter wire rope with independent wire core, each having a minimum breaking strength of 1008 kN (= 103 tonnes force) .

- *For a 75,000m³ gas carrier (prismatic tanks)*

(a) Ship particulars:

L_{BP} = 220.0 m

D = 22.4 m

T = 8.0m (average ballast draft)

A_T = 1003 m^2 (at average ballast draft)

A_L = 3844 m^2 (at average ballast draft)

(b) Maximum forces on the ship, by use of formulae (16), (17) and (18):

F_x max = 559 + 75 = 634 kN

F_{yF} max = 1272 + 408 = 1680 kN

F_{yA} max = 1488 + 437 = 1925 kN

(c) Spring line requirements,. by use of formula (26):

$$S \times MBL = 2235 + 299 = 2534 \text{ kN}$$

Assuming the ship has a total of four spring lines (S = 4), the minimum breaking strength required for each line will be:

$$MBL = \frac{2534}{4} = 634 \text{ kN } (= 65 \text{ tonnes force})$$

(d) Breast line requirements, using formula (27):

$$BR \times MBL = 7734 + 2272 = 10006 \text{ kN}$$

Assuming that the ship has a total of 10 breast lines (BR = 10) symmetrically arranged (five forward and five aft), the minimum breaking strength required for each line will be:

$$MBL = \frac{10006}{10} = 1000 \text{ kN } (= 102 \text{ tonnes force})$$

- *For a 125,000m^3 gas carrier (spherical tanks)*

(a) Ship particulars:

L_{BP} = 274.0 m

D = 25.0 m

T = 9.0 m (average ballast draft)

A_T = 1382 m^2 (at average ballast draft)

A_L = 7122 m^2 (at average ballast draft)

(b) Maximum forces on the ship, by use of formulae (19), (20) and (21):

$$F_x \max = 825 + 105 = 930 \text{ kN}$$

$$F_{yF} \max = 2457 + 572 = 3029 \text{ kN}$$

$$F_{yA} \max = 2600 + 612 = 3212 \text{ kN}$$

(c) Spring line requirements, by use of formula (28):

$$S \times MBL = 3300 + 419 = 3719 \text{ kN}$$

Assuming the ship has a total of four spring lines (S = 4); the minimum breaking strength required for each line will be:

$$MBL = \frac{3719}{4} = 930 \text{ kN } (= 95 \text{ tonnes force})$$

(d) Breast line requirements, using formula (29):

$$BR \times MBL = 13518 + 3184 = 16702 \text{ kN}$$

Assuming that the ship has a total of 14 breast lines (B = 14) symmetrically arranged (seven forward and seven aft), the minimum breaking strength required for each wire will be:

$$MBL = \frac{16702}{14} = 1193 \text{ kN } (= 122 \text{ tonnes force})$$

Section 3.0
Mooring Arrangements and Layouts

3.1 PRINCIPAL OBJECTIVES

The objective of a good shipboard mooring arrangement is to provide and arrange equipment to accomplish the following:

- provide for an efficient mooring pattern at conventional piers and sea islands, as described in Section 1.
- facilitate safe and quick mooring, unmooring and line-tending operations with minimum demand on manpower.
- enable safe and efficient mooring at anticipated non-conventional terminals such as SPMs and MBMs.
- facilitate safe and efficient handling of tugs.
- permit safe and efficient conduct of other customary tanker operations such as hose-handling and mooring alongside of fuel barges.
- allow safe and efficient specific anticipated operations such as ship-to-ship transfers or canal transits.
- provide for emergency situations such as excessive winds requiring doubling of lines, emergency towing of disabled ships, or shipboard fires requiring the ship to be towed off the berth quickly without shipboard assistance.

3.2 REQUIREMENTS AT PIERS AND SEA ISLANDS

The primary concern in the shipboard mooring arrangement is suitability for mooring at conventional piers and sea islands, since this is the requirement most commonly encountered. The principles for an efficient and safe mooring operation at these terminals are covered in Section 1. These principles apply to ships of all sizes and may be summarised as follows:

- Mooring arrangements should be symmetrical.
- Breast lines should be as perpendicular as possible to the longitudinal centre line of the ship.
- Spring lines should be as parallel as possible to the longitudinal centre line of the ship.
- Mooring lines in the same service should have about the same length between the vessel's winch and the jetty mooring points.

In addition to the foregoing principles, the following general guidelines should be kept in mind in laying out the mooring equipment:

- Keep mooring areas as clear as possible.
- Locate mooring operations as far forward and aft as possible.

- Locate bow and stern fairleads as far forward and aft and as low as possible on the ship.
- Locate spring line fairleads as far forward and aft on the main deck as possible to provide adequate line lengths to spring mooring points on the berth.
- Stress the need for correct alignment between fairleads (or chocks) and winch drums.
- Locate winch control positions to provide a clear view of the mooring operations and the officer-in-charge of mooring.
- Mooring lines in the same service should have about the same length between the vessel's winch and its chocks.
- All mooring lines should be capable of being run to either side of the vessel.

3.2.1 *Number and Size of Lines*

Before any mooring layout can be considered, the number, material and size of lines must be determined. This can best be done by computer analysis. However, Section 2 provides approximate methods to determine the product of line quantity × line breaking load for standard environmental conditions. This is given for breast lines and spring lines separately, as 'B × MBL' and 'S × MBL', where 'B' is the total number of breast lines and 'S' the total number of spring lines. Once the product of either 'B × MBL' or 'S × MBL' is determined. 'B', 'S' and 'MBL' must be balanced using the following guidelines:

- Select the most appropriate material on the basis of strength, elasticity, durability and handling characteristics. Section 6 provides general guidelines for the selection of mooring line materials and line construction.
- Maximum flexibility is provided if all lines are of the same size and material (as mentioned in Section 1.5).
- Select the largest line that can safely be handled by ship and terminal personnel. For wire rope, 44 mm diameter is considered a working maximum based on operators' experience, although 48 mm diameter wire ropes are used on occasion. For fibre rope, 80 mm diameter (10 inches in circumference) is considered a practical maximum for ship-supplied hawsers. There are practical minimums for the number of lines as given in the next paragraph, and there is no need to select an MBL higher than required to comply with the 'B × MBL' or 'S × MBL' criteria.
- To provide a symmetrical arrangement about midships, 'B' and 'S' should be even numbers. Four is considered a practical minimum for the number of spring lines (to provide two lines in each direction). Likewise, four is a practical minimum for breast lines to provide two lines each at the bow and the stern. If an uneven number of breast lines is utilized, the extra line should generally be attached to the stern, since the standard environmental criteria given in Section 2 produces an aft transverse force that is about 10% higher than the forward one.
- The considerations discussed above assume that mooring lines can be issued at either port or starboard side of the ship and that all lines are permanently stowed on winch drums. If the arrangement of winches and fairleads does not allow this, or the terminal or trading pattern dictates otherwise, additional lines (and winches) would be required.
- In addition to the recommendations on mooring line sizes and quantities listed above, the designer as well as the ship operator must consider the generalised mooring equipment requirements stipulated by terminals. Sometimes these requirements are based on past experience with inefficient mooring equipment (such as mixed material moorings) and may demand more lines than required for a ship with efficient and well-maintained mooring equipment. In such cases, the ship owner complying with the recommendations of this Guide may strongly represent to the terminal operator that his ship provides an outfit able to securely moor the ship in specific conditions, citing the Guide as grounds for its acceptance.

3.2.2 *Arrangements for Breast Lines*

Breast lines are effective in holding the ship against transverse forces; they also are most effective in restraining the yawing tendency of a vessel which is induced by wind, current, etc., acting on the ship. However, to be most effective in restraining the yawing tendency, issue points for breast lines should be as far forward and aft as possible. The lead from the winch drum to the shipside fairlead should be as direct as possible, preferably avoiding the use of pedestal fairleads. If pedestal fairleads are used, the change in rope direction should be kept to a minimum in order to reduce the loads on the fairlead. With limited deck space, a good arrangement can often be accomplished by placing winches in a diagonal or transverse pattern as shown in Figs. 3.1 and 3.4, respectively. Details of the mooring layouts for two sizes of ships are shown in Figs. 3.2, 3.3, 3.5 and 3.6. In these examples all mooring lines can be issued on either side and only two pedestal fairleads are required on the forecastle deck to accomplish this (the other pedestal fairleads shown do not relate to mooring at piers and sea islands).

Another point that must be considered is the shore lead of the lines issued at the extreme ends of the ship. For instance, the aftermost two lines shown in Fig. 3.7, which can be issued only from chocks located at the transom, would chafe on the transom if the shore mooring dolphin is forward of the transom. (This is not very common, but occasionally occurs when a large ship moors at a berth designed for smaller ships.) The arrangements shown in Figs. 3.1 through 3.6 provide more flexibility in accommodating different shore mooring point locations.

Some arrangements incorporate a 'first-ashore' line. This line is used only to assist the ship during docking manoeuvres and is generally a polypropylene line because of its buoyancy (refer to Section 6). An 80 mm diameter line of 370m length would be suitable for large ships. Two such lines are provided, one forward and one aft. Several configurations are in use to handle these lines. One is by use of powered twin grooved drums where the line is led back and forth between the drums in a figure-of-eight fashion (this device can be compared to a bollard with powered barrels). The inboard end of the line is taken up by a powered stowage reel, usually located below the weather deck. Another method is to use a conventional winch, and a third method is to use a warping head of a winch in combination with a powered take-up reel.

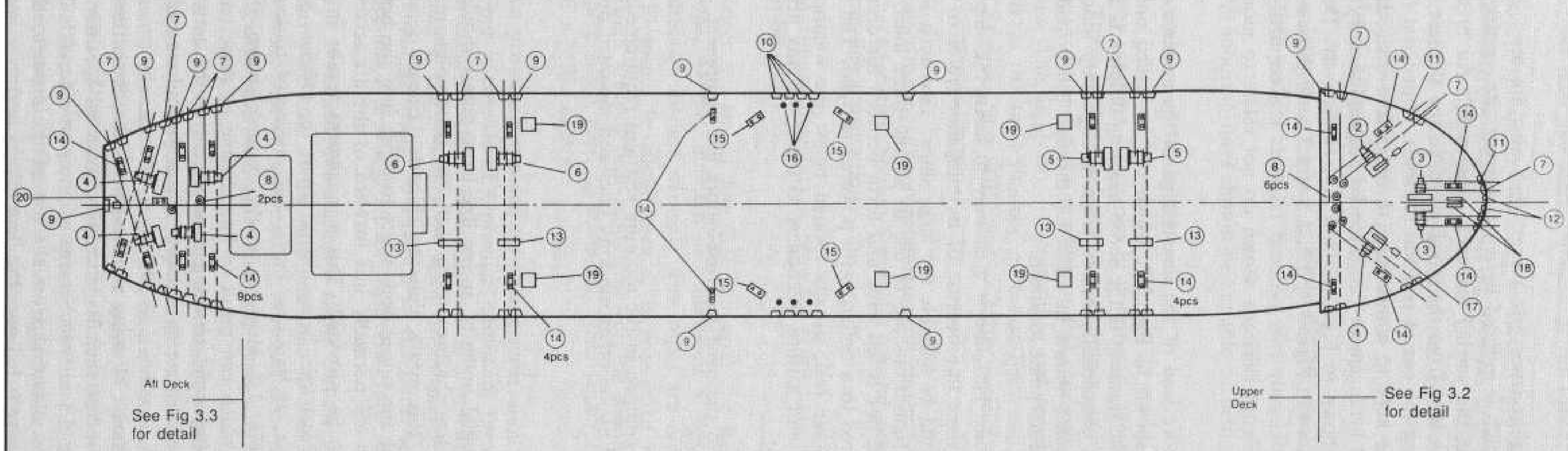
There is no consensus among ship operators as to the need for first-ashore line equipment, and no special equipment is shown in Figs. 3.1 through 3.6.

If first-ashore lines are used, they should not be counted in the mooring restraint requirements given in Section 2 unless they are of the same material as the other mooring lines and mounted on drums equipped with brakes as recommended in Section 7.

3.2.3 *Arrangements for Spring Lines*

In order to provide an efficient lead to the terminal bollards, spring line issue points should be as far forward and as far aft as possible. To avoid line chafing on the shell, the issue points must also be within the parallel body. In practical terms, this means that the shipside fairleads serving the forward headsprings should be at the point where the upper deck starts to taper into the bow area. The shipside fairleads serving the aft backsprings are normally just forward of the aft accommodation house where a direct lead to the winch can be provided. This arrangement results in the aft spring winches and the winches serving the aft breast lines being too far apart for efficient manning during docking and undocking. To overcome this, at least one owner has attempted to locate the aft spring winches on the aft deck as shown in Fig. 3.7. However, in the example shown, the shipside fairleads are aft of the parallel deck area, which can result in line chafing at some terminals. Nevertheless, with proper coordination of hull shape and mooring arrangement at the early design stage, this concept may be workable and could contribute to reduced manning requirements.

Figs. 3.1 and 3.4 show the conventional arrangement. Mooring winches should be placed directly in line with the fairleads as shown. Since the ship's centreline is generally obstructed by deck piping, main deck winches are moved to the side. In Fig. 3.1 all winches are on the port side with the wire leading from the bottom of the drum to the port side and from the top of the drum to the starboard side. In case of docking to starboard, the wire is led over the deck piping and over horizontal



ITEM	DESCRIPTION
1, 2	Combined Anchor Windlass/Mooring Winch
3, 4, 5 & 6	Single Drum Mooring Winch with Warping Head
7	Universal Roller Fairlead
8	Pedestal Fairlead
9	Panama-type Fairlead, 600 × 300mm opening, deck mounted
10	Closed Fairlead, 450 × 300mm opening, deck mounted
11	Panama-type Fairlead, 600 x 300mm opening, bulwark mounted
12	Panama-type Fairlead, 600 x 450mm opening, bulwark mounted
13	Overhead Roller Fairlead
14	Vertical Bollard, 600mm diameter
15	Vertical Bollard, 350mm diameter
16	Cruciform Bollard, 300mm diameter
17	Chain Stopper for Anchors
18	SPM Chain Stopper
19	Tugger Winch
20	Towing Fitting

FIGURE 3.1: MOORING FITTING ARRANGEMENT OF 200 kDWT TANKER WITH 12 MOORING WIRES (42 MM DIA)

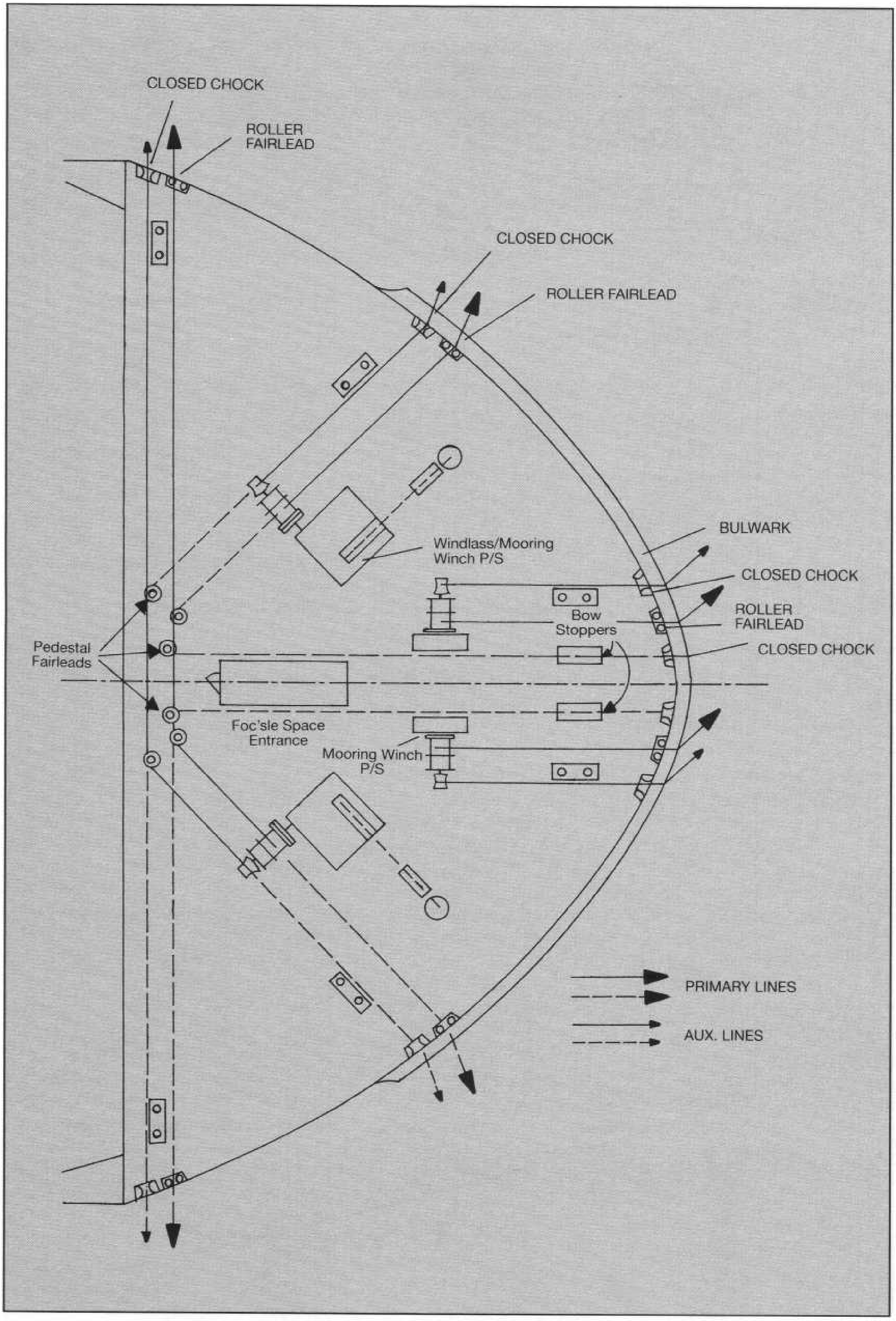


FIGURE 3.2: 200 kDWT TANKER MOORING ARRANGEMENT ON THE FORECASTLE DECK

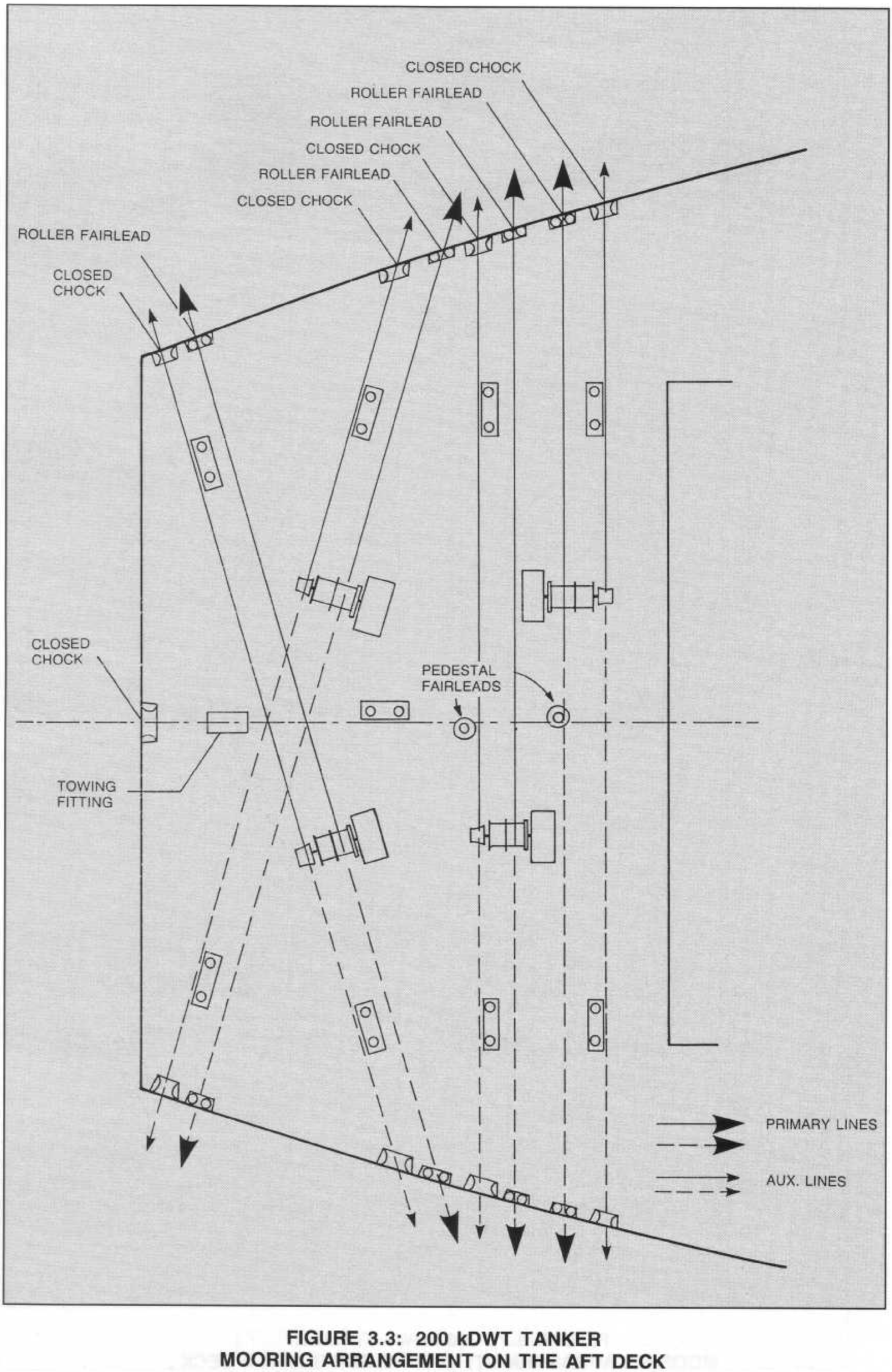
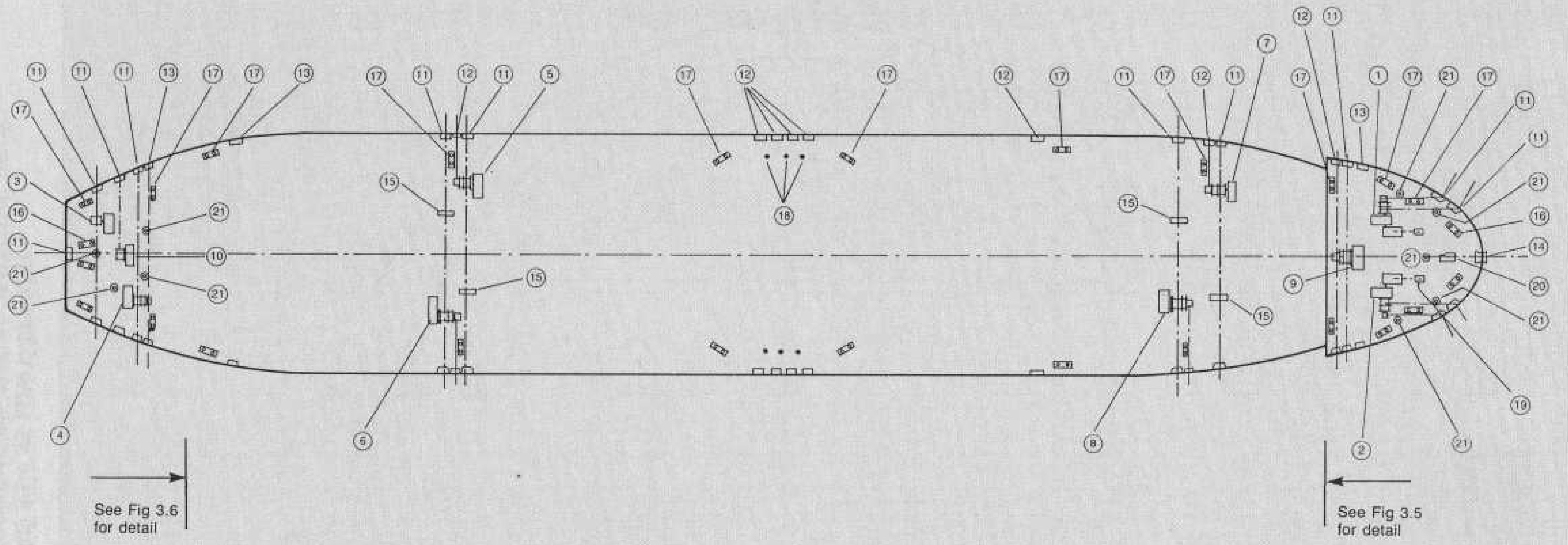


FIGURE 3.3: 200 kDWT TANKER
MOORING ARRANGEMENT ON THE AFT DECK



ITEM	DESCRIPTION
1, 2	Combined Anchor Windlass/Mooring Winch
3, 4, 5, 6, 7, 8, 9	Single Drum Mooring Winch with Warping Head
10	Single Drum Mooring Winch without Warping Head
11	Panama-type Fairlead, 500 × 350mm opening, bulwark mounted
12	Closed Fairleads, 400 × 250mm opening, deck mounted
13	Panama-type Fairlead, 350 x 250mm opening, bulwark mounted
14	Panama-type Fairlead, 600 x 450mm opening, bulwark mounted
15	Overhead Roller Fairlead
16	Vertical Bollard, 450mm diameter
17	Vertical Bollard, 300mm diameter
18	Cruciform Bollard, 300mm diameter
19	Chain Stopper for Anchors
20	Bow Stopper
21	Pedestal Fairlead

FIGURE 3.4: TYPICAL MOORING FITTING ARRANGEMENT OF 35 kDWT TANKER WITH 10 MOORING WIRES

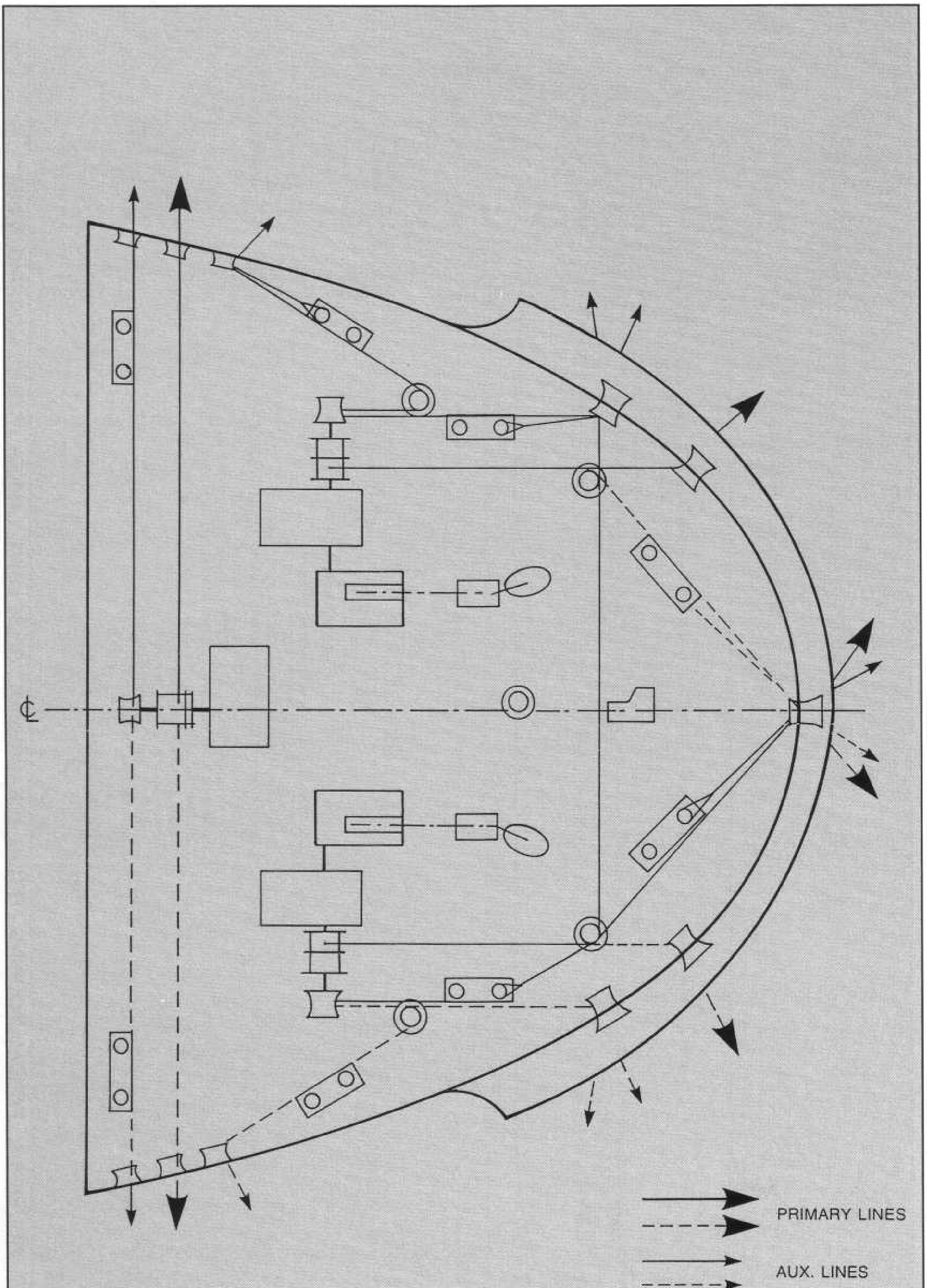


FIGURE 3.5: 35 kDWT TANKER
MOORING ARRANGEMENT ON THE FORECASTLE DECK

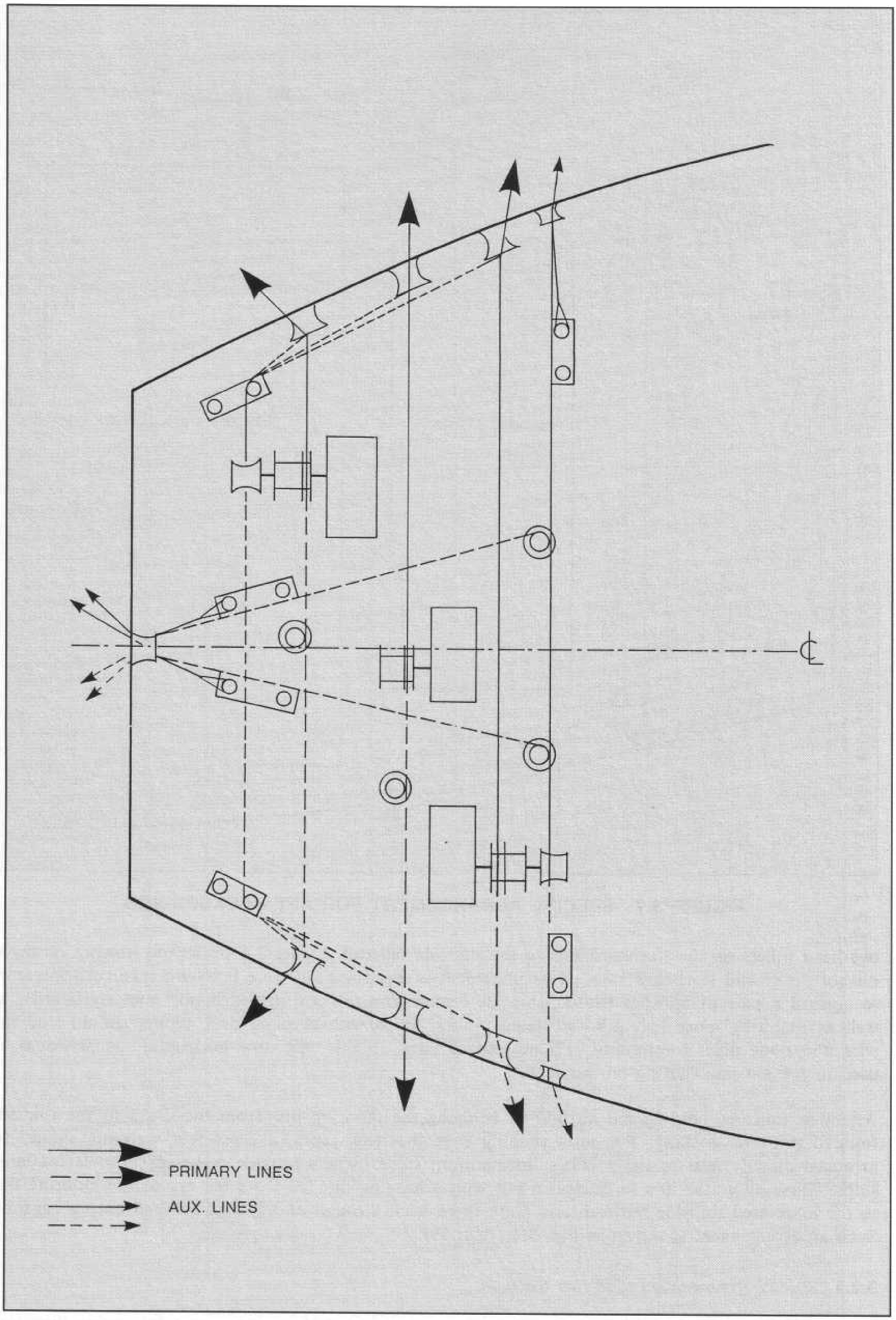


FIGURE 3.6: 35 kDWT TANKER
MOORING ARRANGEMENT ON THE AFT DECK

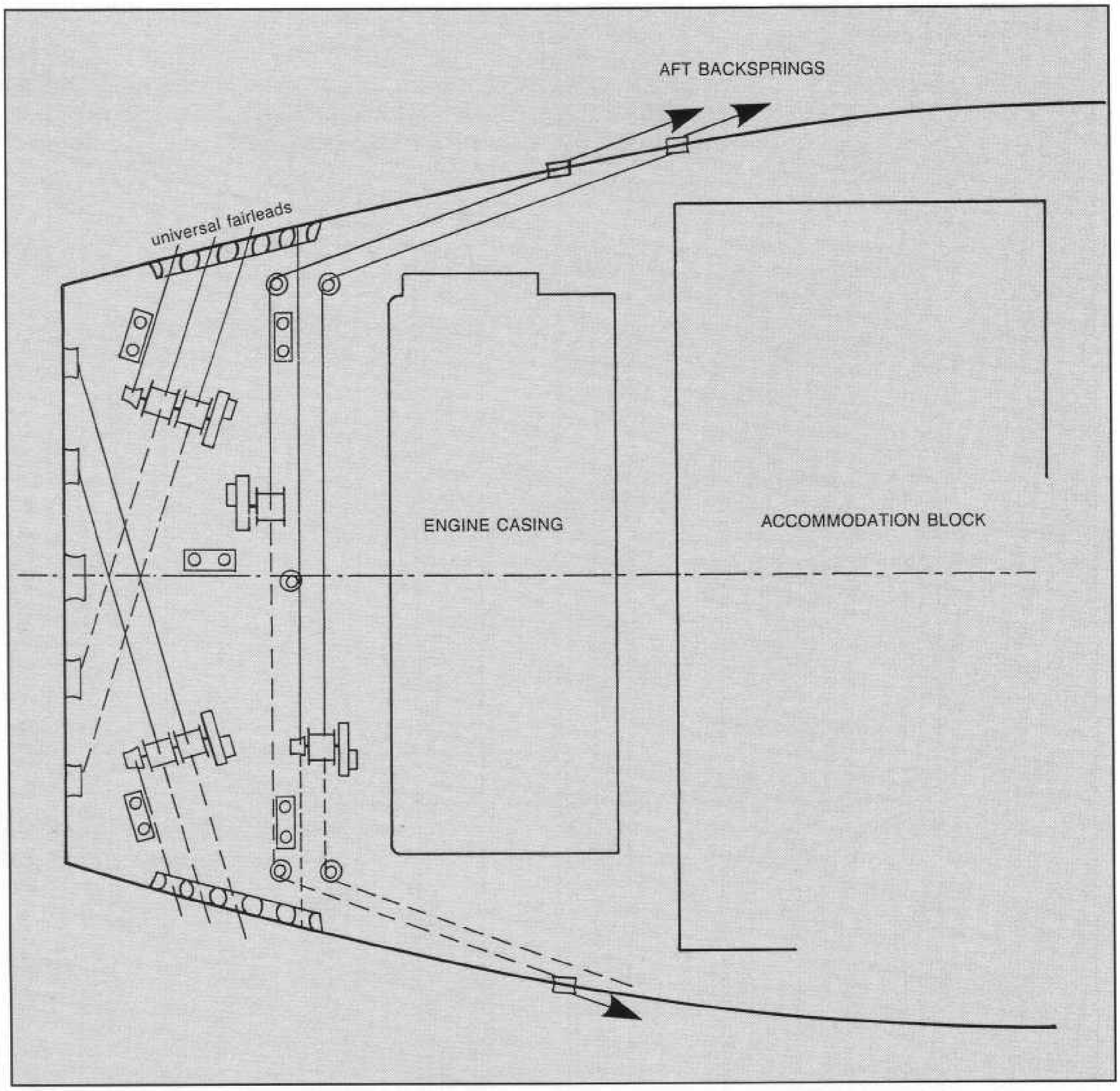


FIGURE 3.7: SPECIAL ARRANGEMENT FOR AFT BACKSPRINGS

overhead rollers on the starboard side to the shipside fairlead. Figure 3.4 shows the winches arranged on both port and starboard sides. This placement is less common since it would take two operators to control a pair of winches (unless shipside remote controls are provided port and starboard), as well as requiring more hydraulic or steam piping. Ships with elevated deck piping should lead the wire under the pipes to simplify bringing out of lines. In this case, low horizontal rollers would be used to prevent line chafing on the deck.

Attention must be paid to the method of bringing the mooring line from the drum to the shipside fairlead prior to docking. For lines running over the deck pipes, a cross-over walkway should be arranged directly next to the winches. In addition, some owners provide powered capstans for large ships. These allow the line to be drawn out with a heaving line led from the eye of the mooring line to the associated shipside fairlead, and from there back through an adjacent fairlead onto a capstan. Such an arrangement is shown in Fig. 3.1 (Item 19).

3.2.4 *Special Arrangements for Gas Carriers*

For flush deck LNG and LPG carriers (membrane type, etc.) the arrangement for spring lines and associated winches may be similar to those adopted for oil tankers. For spherical tank LNG carriers and lobe/cylindrical tank LPG carriers it is usually not practicable to incorporate main deck winches

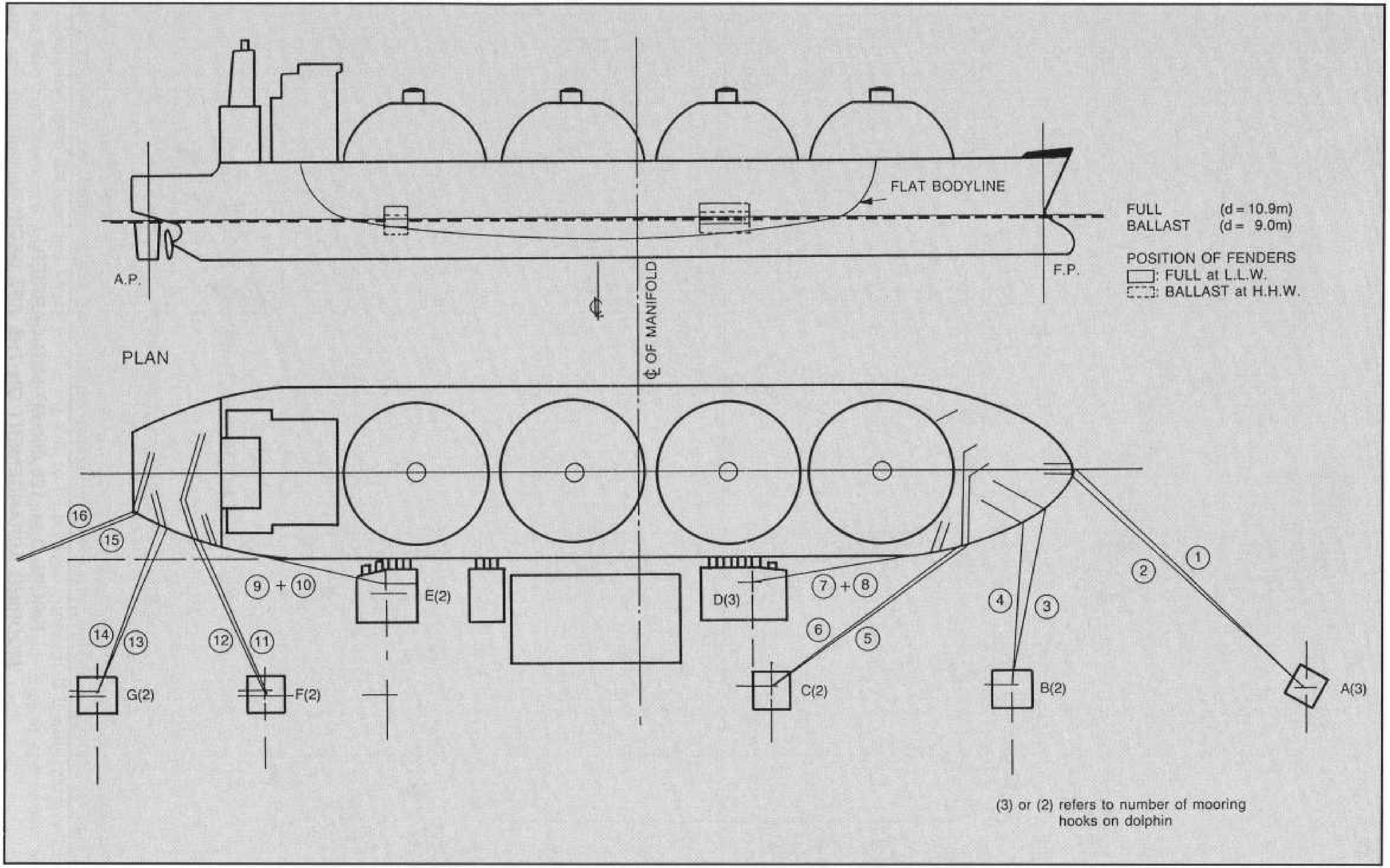


FIGURE 3.8: TYPICAL MOORING PATTERN—LNG CARRIER

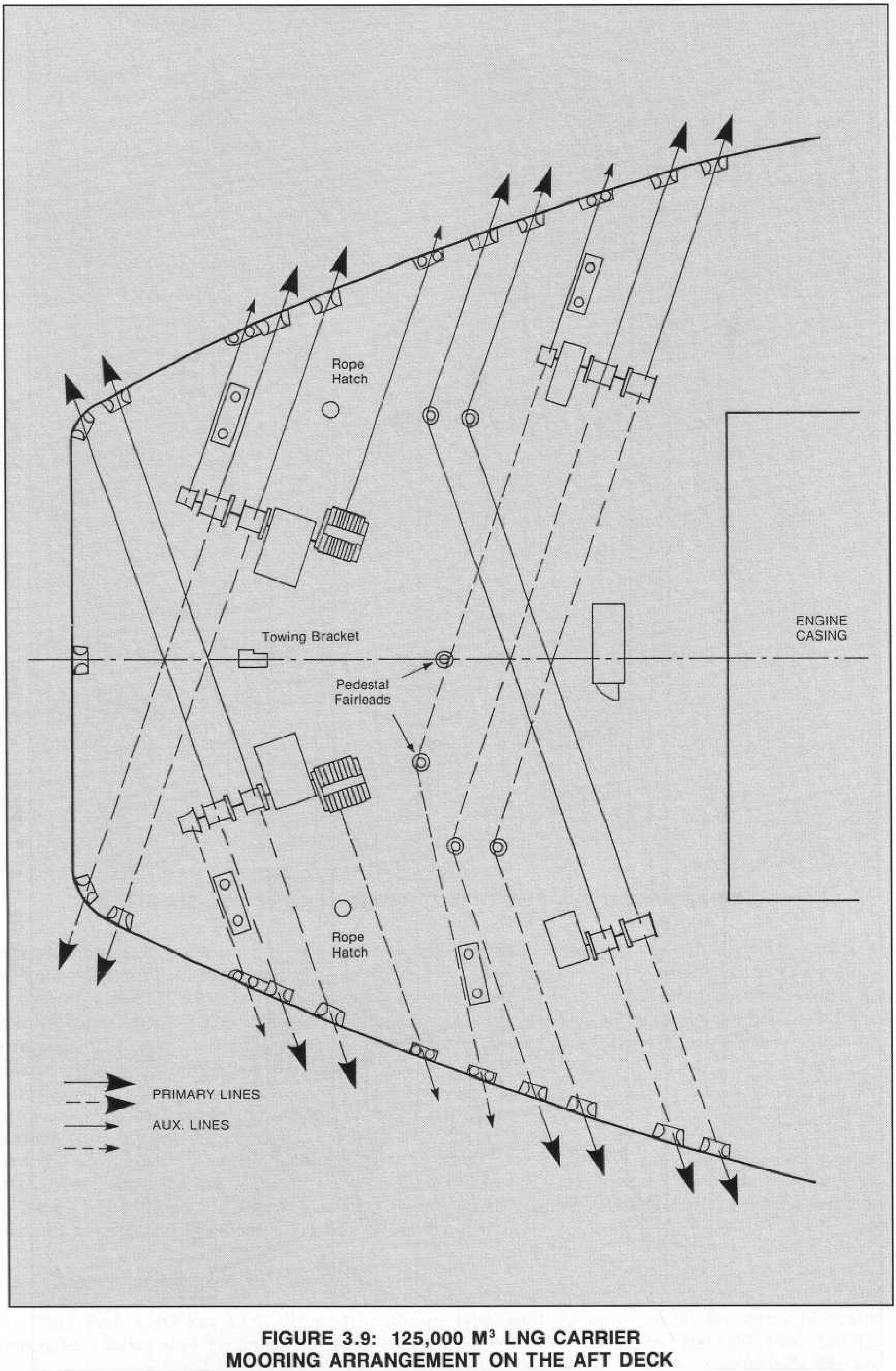


FIGURE 3.9: 125,000 M^3 LNG CARRIER MOORING ARRANGEMENT ON THE AFT DECK

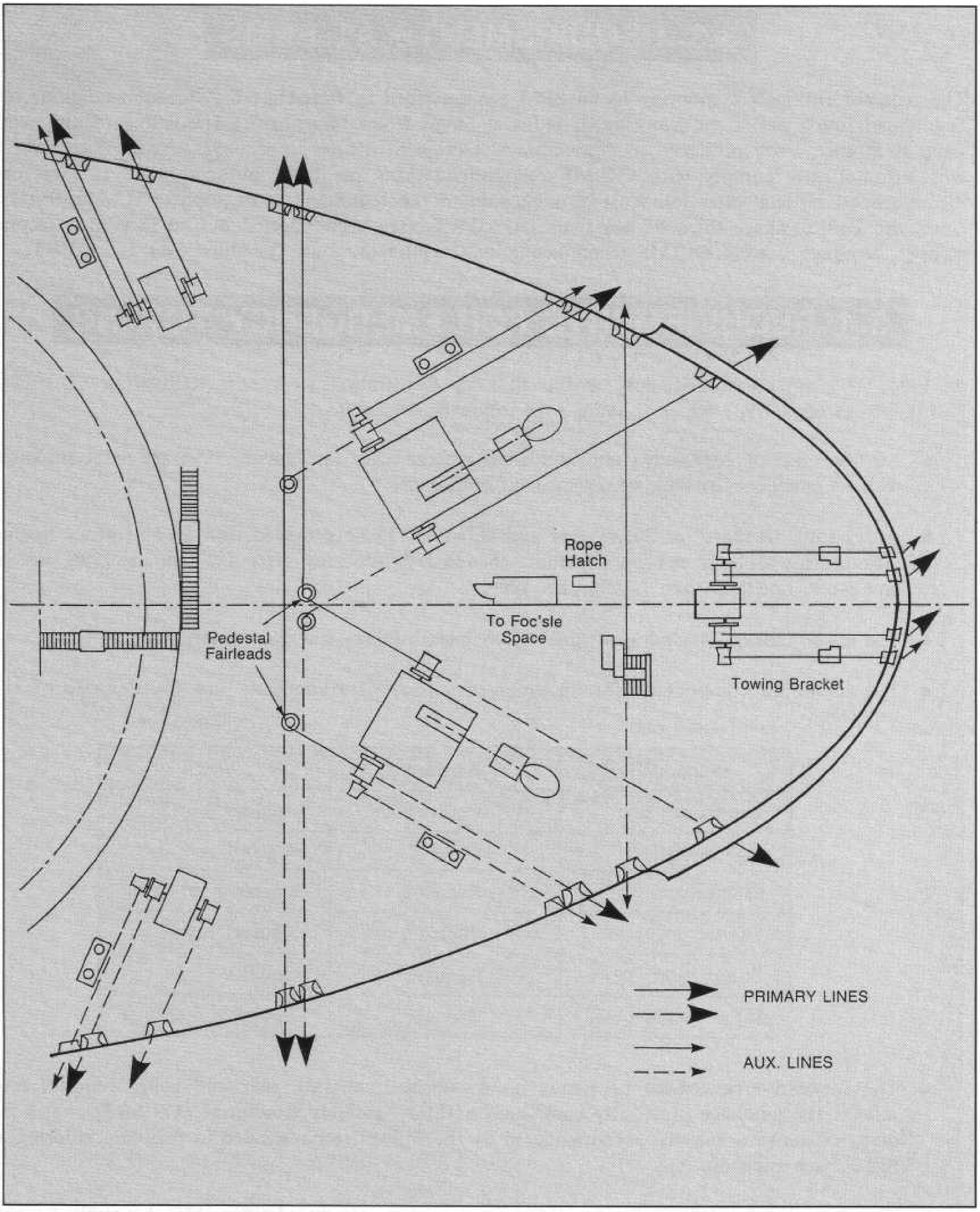


FIGURE 3.10: 125,000 M³ LNG CARRIER MOORING ARRANGEMENT ON THE FORWARD DECK

and the lead of springs must be from aft of the accommodation deckhouse and from the forward main deck or the forecastle. A typical arrangement is shown in Figs. 3.8, 3.9 and 3.10.

The lack of main deck winches on this type of ship can mean that capstans are required close to tug positions for handling tug lines. These should be rated at 1.0 to 1.5 tonnes (10 to 15 kN) depending upon the size and length of the tug line to be brought onboard. It is also common practice on such high freeboard ships to utilize towing chocks recessed into the side shell at appropriate heights for easy handling of the line from the tug.

3.3 REQUIREMENTS AT SPMS

The required fittings for mooring to an SPM are described in Reference 1 ("Recommendations for Equipment Employed in the Mooring of Ships at Single Point Moorings".) The fitting requirements from Reference 1 are produced in Appendix A. Design of fittings in accordance with Appendix A will automatically comply with OCIMF recommendations for SPM mooring, and is therefore recommended. Special attention must also be paid to the requirements of specific SPM operators. Cases are known where ships of less than 150 kDWT have been rejected due to lack of a second stopper, although current OCIMF recommendations require this only for ships over 150 kDWT.

3.4 REQUIREMENTS FOR EMERGENCY TOWING

In 1994 IMO agreed amendments to the SOLAS Convention as a new Regulation Ch V/15–1 (Ch II–1/3–4 from 1/7/98), which contained the following provisions:

- All "tankers" of 20,000 dwt and above constructed after 1st January 1996 are to be provided with an emergency towing arrangement at both ends.
- All existing "tankers" of 20,000 dwt and above are to be provided with an emergency towing arrangement at both ends at the first scheduled dry-docking after 1st January 1996, but in any event, not later than 1st January 1999.
- The term "tankers" includes oil tankers, chemical tankers and gas carriers.
- The minimum components for an emergency towing arrangement are to comprise of the following:

Component	Forward	Aft
Towing pennant	Optional	Required
Pick-up gear	Optional	Required
Chafing gear	Required	Dependent on design
Fairlead	Required	Required
Strong point	Required	Required
Roller pedestal lead	Required	Dependent on design

- The forward arrangement of strong point, fairlead, chafing gear and roller pedestal lead reflects the guidance previously contained in IMO Assembly Resolution A.535(13), which on many oil tankers, may be accommodated by the fittings recommended to facilitate mooring at SPM's (see appendix A).
- The arrangement aft contains a major new provision introduced since IMO Assembly Resolution A.535(13) was developed, namely the requirement for the ship to carry a pre-rigged towing pennant incorporating pick-up gear. The pick-up gear must be capable of being deployed manually by one person and the pennant must be demonstrated to be capable of full deployment within 15 minutes under harbour conditions.

3.5 REQUIREMENTS FOR MULTI-BUOY MOORINGS

Multi-buoy mooring (MBM) consists of tying up a ship to several permanently anchored buoys in conjunction with the ship's own anchors. It is also called conventional buoy mooring or 'CBM'. A typical five-buoy configuration is shown in Fig. 3.11. In rare cases the ship is moored to buoys only.

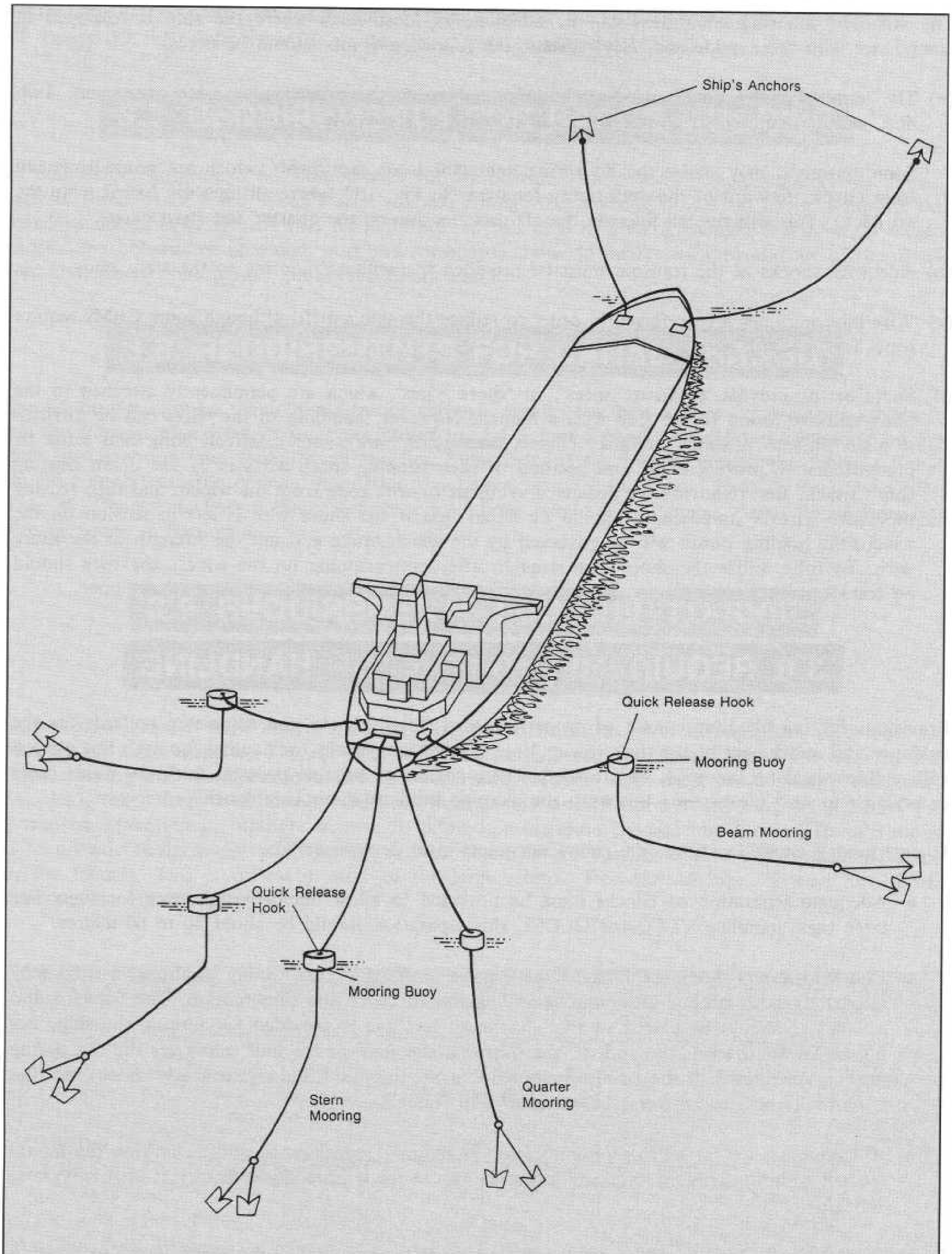


FIGURE 3.11: MULTI-BUOY MOORING (MBM)
(5-BUOY MOORING SHOWN)

Multi-buoy moorings are usually located in areas where weather and sea conditions are mild to moderate. This is because the mooring restraint is limited due to the requirement to pay out the mooring lines on both port and starboard sides, in contrast to mooring at piers and sea islands where the lines are paid out on one side only.

The standard mooring equipment will be adequate in most cases where the ship is equipped in compliance with these guidelines. Nevertheless, the following points should be noted:

(a) The terminal will normally require the ship to provide the necessary mooring equipment. Two lines may be required to be run out to all or some of the buoys.

Some terminals may utilize the aft spring lines (for beam moorings), which are generally issued from chocks forward of the deck house (contrary to Fig. 3.11 where all lines are issued from the aft deck). This will free all lines on the aft deck for use on the quarter and stern buoys.

(b) Adequate chocks at the transom must be provided to facilitate mooring to the stern buoy.

(c) Wire mooring lines are preferred in order to reduce the ship's drift, although some CBMs require ropes for handling purposes.

(d) Some berths provide 'preventer wires', or 'shore wires', which are permanently attached to the buoy and are towed to the ship with a launch. However, handling of the wires can be difficult if made fast to a ship's bollard. This is because the wires are relatively long and must be pretensioned to prevent drift. One method of pretensioning shore wires is by use of an existing ship's winch, first removing the dedicated synthetic or wire rope from the winch, and then reeving the shore wire in its place. It should be noted that if the shore wire is left in tension on the winch, the holding power will be governed by the winch brake and not the strength of the shore wire. To fully utilize the shore wire strength after pretensioning on the winch, the wire should be transferred to a suitable set of bitts or bollards using a chain or carpenter's stopper.

3.6 REQUIREMENTS FOR TUG HANDLING

Provisions for tug handling consist of properly placed closed chocks and associated bollards for the guidance and attachment of the tug's towing line. In addition, means for hauling the tug's line aboard with a ship's heaving line must be provided. These consist of suitable pedestal fairleads, guide posts or bollards to lead the heaving line onto the warping head of a mooring winch.

In determining chock locations, the following points must be considered:

- Adequate separation of chocks must be provided to allow manoeuvring space for tugs. For large tugs, handling VLCCs or ULCCs, this separation should be about 50 to 60 metres.

- Chock locations should be in the same transverse plane as tug-pushing locations, as tugs may alternately push or pull from the same location to check the ship motion. The forward and aft chocks should be placed so that maximum leverage is provided for turning the ship, but not be so far towards the ends of the ship that the flare of the hull endangers the tug during pushing operations. It should also be noted that the tug push (and consequently chock) location is normally near a transverse bulkhead or web frame.

- An alternate neutral pull or push location is required midships to allow checking the lateral motion without applying a turning moment. The chock is generally located just aft of the hose saddle.

For VLCCs and ULCCs, the above requirements generally result in five push/pull locations on each side of the ship. For smaller ships, where adequate separation of five tugs cannot be provided, three locations on each side will suffice.

If a bollard is used exclusively for securing harbour tug lines, the size should be related to the bollard pull of the tug, but need not exceed 500mm. If the bollard is intended for multiple applications, it is recommended that the size and strength be determined in combination with Table 4.3 in Section 4 and Table 8.1 in Section 8, assuming a SWL equivalent to the MBL of the ship's mooring line. The 'eye' belaying case may be assumed for bollards. For example, a ship having mooring lines of

40 tonnes (392 kN) MBL would require 315mm diameter bollards and one with mooring lines of 92 tonnes (902 kN) or more MBL would require 500mm diameter bollards.

3.7 REQUIREMENTS FOR BARGE MOORING

In many cases barges can be moored with fittings provided for other mooring requirements. Nonetheless, some VLCCs and ULCCs lack suitable fittings for mooring a fuel oil barge alongside the midships manifold or the aft fuel oil manifold. In this case it is recommended that a set of closed chocks and bollards be provided, port and starboard, about 35 metres forward and aft of the manifold and, where appropriate, the aft bunkering station.

3.8 REQUIREMENTS FOR CANAL TRANSIT

Special fittings may be required for transit through canals. The best known requirements are those for the Panama Canal where ships are pulled in and out of locks by shoreside locomotives having their own mooring lines mounted on winches. Ships suitable for transit through the Panama Canal should comply with the detailed regulations contained in the U.S. Central Federal Register (CFR), Title 35, Chapter 1, Part 4. Some reference to fittings required by Panama Canal Regulations can also be found in Section 8.

3.9 REQUIREMENTS FOR SHIP-TO-SHIP (STS) TRANSFER

Ship-to-ship transfer requires the mooring alongside offshore of two different size ships for the purpose of cargo transfer. The mooring arrangements adopted will depend upon the sizes of the ships carrying out the operations and the difference between their sizes. As a general guideline, Fig. 3.12, taken from Reference 4, illustrates a recommended and proven mooring arrangement for a transfer operation in offshore waters. A prime consideration in mooring during STS operations is to provide fairleads for all lines without the possibility of chafing against each other, the ships or the fenders. This is critical in view of the large relative freeboard changes between the ships, illustrated in Fig. 3.13.

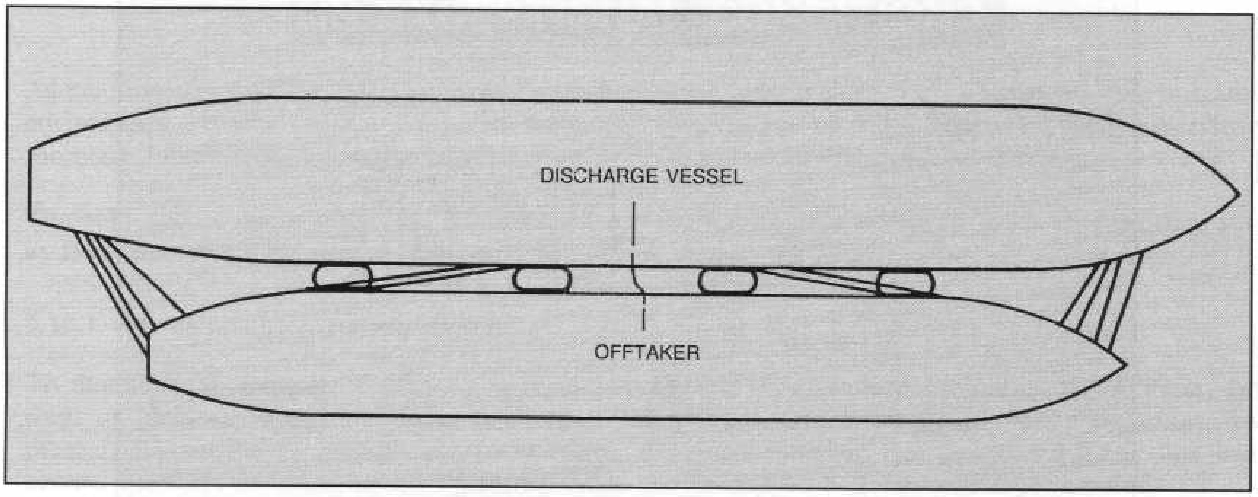


FIGURE 3.12: MOORING PATTERN DURING SHIP-TO-SHIP TRANSFER

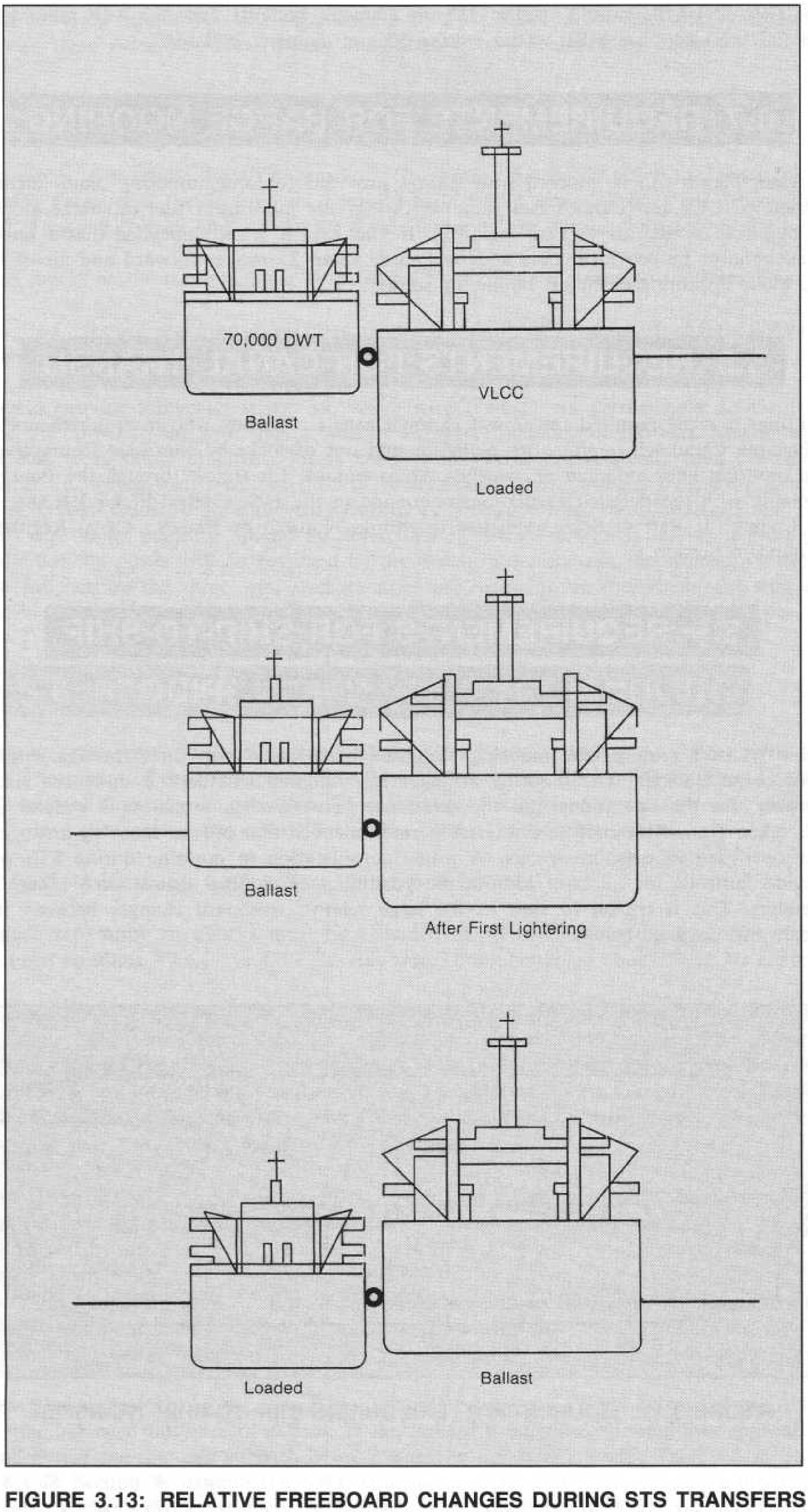


FIGURE 3.13: RELATIVE FREEBOARD CHANGES DURING STS TRANSFERS

3.9.1 *Requirements for Offtaker*

The offtaker's (smaller ship's) standard mooring equipment is generally suitable for STS transfers. Ships equipped with wire moorings should fit synthetic rope tails to introduce more elasticity and electrical discontinuity, and to permit cutting in an emergency. It is recommended that the fairleads are of the enclosed type, since the relative freeboard between the two ships will change significantly during the STS transfer operation. If the lines are of wire rope, the opening of the closed chocks must be large enough to permit easy passage of the special shackle connecting the tail to the wire rope.

3.9.2 *Requirements for Discharge Ship*

The discharge ship (larger ship) may require special mooring fittings to allow a proper mooring pattern. Since industry practice is to use the starboard manifold of the discharge ship, special fittings are generally provided on the starboard side only. As can be seen in Fig. 3.12, the offtaker's spring lines terminate on the discharge ship in a location not usually fitted with chocks. For this reason it is recommended that tankers over 160 kDWT be fitted with closed chocks with openings of 500 × 400 mm, about 35 metres forward and aft of mid-length on the starboard side. Since the 500 × 400 mm dimension is not a common standard size, some operators have successfully used two chocks with 500 × 250mm openings arranged in pairs, in place of a single 500 × 400mm chock.

Similarly to requirements for the offtaker, the discharge ship's mooring fairleads should all be of the closed type to avoid difficulties caused by the large relative changes in freeboard. The requirements for closed chocks of adequate size for breast, head and stern lines may already be met by requirements in these guidelines for mooring at piers and sea islands, and by tug handling provisions. But in other cases — especially on ships where closed chocks are not used for the standard mooring lines — special STS transfer closed chocks may be required on the starboard side. In any event, suitable bollards are required inboard of the closed chocks for securing the offtaker's mooring line. Some operators allow only one line to be used in each chock to reduce the possibility of line chafing with changing ship draughts. In this case closed chocks are arranged in pairs and served by a common bollard. Further, it is recommended that means be provided for passing a messenger line (attached to the eye of the offtaker's mooring line) through the chock and onto the warping head of a mooring winch. For this purpose, a bitt or guide post may be used in lieu of a pedestal roller fairlead.

More details on all phases of STS transfer operations may be found in Reference 4.

3.10 MOORING AUGMENTATION IN EXCEPTIONAL CONDITIONS

As mentioned in Section 2, it would not be practical to design all ships for the worst possible operating environment. Where the Standard Environmental Criteria are exceeded, the ship must either leave the berth, obtain continuous tug assistance or arrange for additional mooring restraint.

The ship and terminal must also be prepared to take appropriate action in other emergencies, such as fires, and this may require additional equipment aboard the ship.

3.10.1 *Excessive Environmental Forces*

To augment the mooring system when conditions exceed the Standard Environmental Criteria, two ways of obtaining additional restraint capacity are feasible. One is to provide shore moorings. The other is to provide shore pulleys around which the ship's mooring line is led and made fast back aboard the ship. In either case, shipside fairleads and associated bollards would be required.

It is recommended to provide closed chocks, associated bollards and warping ends for a number of additional lines equal to at least 50% of the ship's standard mooring lines. (Note that the ship shown in Fig. 3.1 provides for 100% augmentation.) The chocks will usually be located next to the fairleads for the standard mooring lines.

This allows heaving aboard of the shore line, or the ship's own line from a shore pulley, via heaving lines led to warping heads of mooring winches. The strength of fairleads and bollards should be based on the MBL of the ship's standard mooring lines.

3.10.2 *Use of Shore Based Pulley*

In its simplest form the shore fitting could consist of a bollard of a sufficient diameter for the size of wire to be employed. A minimum ratio of bollard/wire diameter of 12:1 should be employed.

The provision of a revolving bollard or pulley wheel is recommended to reduce friction on the wire and to ensure that when moorings are adjusted the tension in each part is readily equalized. To provide a release facility under normal operational conditions, the pulley should be incorporated on a quick release mounting so that on the activation of the release mechanism the pulley capsizes and the bight of the wire is released and thrown clear of the jetty. This equipment is available from manufacturers of conventional quick release hooks. The pulley must be designed properly to prevent it from releasing the line inadvertently due to the pulley's weight. Figures 3.14 and 3.14(a) show an instrumental shore based pulley and the method of securing the tail at the shipboard end of the ship's mooring wire.

In considering the design specifications for this equipment it should be remembered that loads sustained will be approximately twice that experienced by a mooring hook or bollard to which a single wire is attached.

A suitable heavy duty winch or capstan should be provided to assist in heaving the bight of the wire on to the fixed mooring structure and securing it over the pulley or bollard.

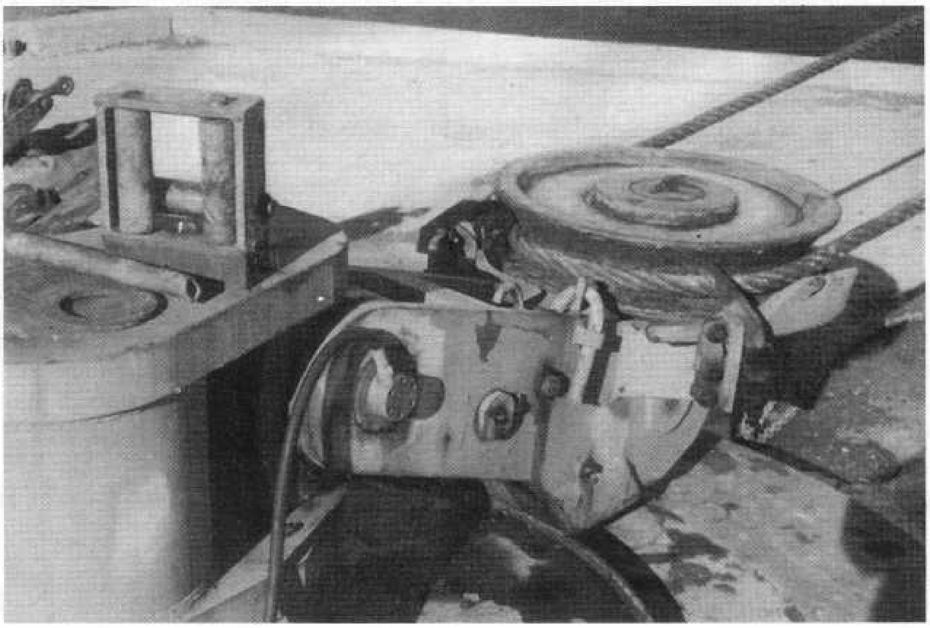


FIGURE 3.14: SHORE BASED PULLEY SYSTEM

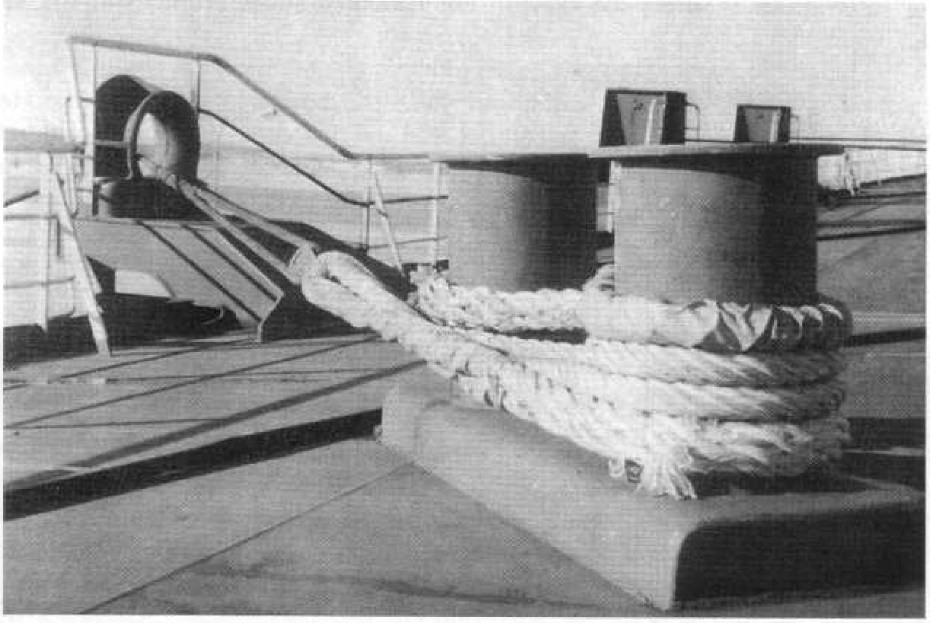


FIGURE 3.14(a): TURNING UP ON SHIP OF SYNTHETIC ROPE TAIL

3.10.3 *Advantage of Pulley System*

The principal advantage of this equipment is that all moorings can be properly tensioned by the vessel's mooring winches in the normal way and manual handling of heavy wires on board is reduced. Also the problem arising from divided control which can occur with the use of shore wires on winches is avoided.

While the use of such a system will reduce the load in a vessel's moorings in proportion to the number of pulleys provided, its complete effectiveness depends on a vessel's mooring equipment being in good condition initially, as with all mooring systems. The terminal should evaluate the effect that ship's tails would have on the loads induced in the mooring lines. Since now there is only one tail for two lines, the elasticity of each is decreased.

3.10.4 *Disadvantage of Pulley System*

The principal disadvantage of this system is the difficulty of handling the length of wire involved by using the bight particularly at marginal jetties where moorings may have long drifts. In addition care must be exercised to prevent abrasion of the tails when they come back aboard ship. Since the tail will usually be in the fairlead, the tail can abrade, especially if the fairlead is not smooth or free of burrs.

To avoid abrasions, some terminals take the tail end to a more distant ship's bitt and others shorten the tail by belaying around the bitt (as in Figure 3.14(a)).

Both of these affect the elasticity of the system and must be considered when calculating restraint capacity.

Larger mooring boats may also be necessary to adequately handle heavy bights associated with this system.

3.11 FIRE WIRES

Terminals require the provision of so-called 'fire-wires' or 'towing-off wires'. These are mooring wires hung over the off-berth side of the ship. They enable tugs to pull the ship away from the pier without the assistance of any crew member in case of a serious fire or explosion. Refer also to Reference 6 ("International Safety Guide for Oil Tankers and Terminals", 4th edition, 1996), Chapter 3.7.2.

A common method is to provide two wires, one near the bow and one near the stern. They are secured to bollards with a minimum of five turns and are led directly to a shipside chock with no slack on deck. The outboard end of the line is provided with an eye to which a heaving line is attached and led back to the deck;. During loading or discharge, the heaving line is periodically adjusted to maintain the eye of the fire wire one to two metres above the water as shown in Figure 3.15. Some terminals require different methods and operators should be aware of local regulations.

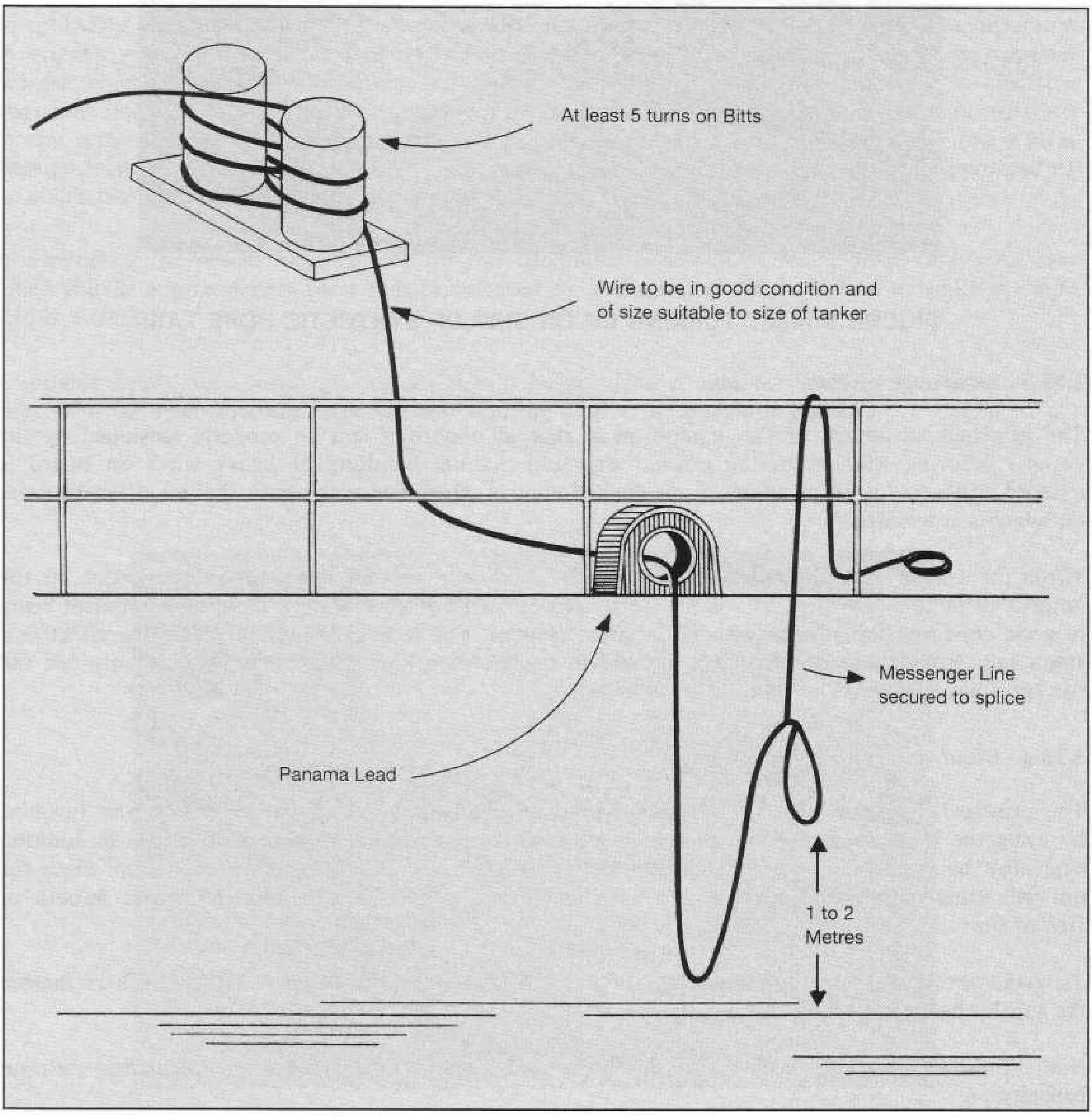


FIGURE 3.15: EMERGENCY TOWING WIRE REQUIREMENTS

When not in use, the wires are preferably spooled onto reels which may be located on or below deck.

Fire wires should be of 6 × 36 IWRC construction and be made of the same type of steel as recommended for standard mooring wires in Section 6. The use of synthetic or natural fibre ropes is not permitted as these would burn in the event of a fire.

The following table gives guidance on minimum diameters and lengths for various ship sizes, however, lengths may vary dependent on positioning of mooring bitts and vessel's freeboard.

kDWT	Diameter (mm)	Length (m)
20-100	28	45
100-300	38	60
over 300	42	70

3.12 COMBINATION OF VARIOUS REQUIREMENTS

The requirement for fittings set forth in Sections 3.2 (piers), 3.3 (SPMs), 3.5 (MBMs), 3.6 (tug handling), 3.7 (barge mooring), 3.8 (canal transit), and 3.9 (STS transfer) do not apply simultaneously. In the interest of reducing cost and complexity, it is desirable during the ship design stage to adjust the location of shipside fairleads slightly so that one fairlead or bollard can serve several requirements. At the same time, all possible line leads for the various requirements must be considered. For instance, when a shipside fairlead designed primarily for use at piers and sea islands in conjunction with mooring winches is utilized for requirements 3.5, 3.6, 3.7 or 3.8 or 3.9 measures may be necessary at the inboard edge if roller fairleads are used. This is especially acute for universal roller fairleads since the inboard fore and aft leads are restricted by the end frames.

3.13 SAFETY AND OPERATIONAL CONSIDERATIONS

For safety reasons, it is highly desirable to lead mooring lines from winch drums directly to the shipside fairlead. If the use of pedestal fairleads cannot be avoided, the winch controls should be located to minimise risk to the operator.

In the interest of manpower savings and speedy mooring and unmooring operations, all mooring lines should be stowed on drums and consideration given to the provision of single drum winches (individual drive for each drum) for all mooring lines. This will eliminate the often difficult task of clutching and declutching drums from a common drive shaft in combination with setting and releasing drum brakes.

3.14 EQUIPMENT AND FITTING LINE-UP

Mooring fittings require adequate clearance for routine operations, and winches have to be arranged to provide an adequate fleet angle for the drum. This is the maximum angle the line deviates from a direction perpendicular to the drum axis. The following are some 'rule-of-thumb' guidelines.

The minimum distance between a bollard and fairlead should be 1.8 metres in order to provide adequate space for the application of rope stoppers (see Figure 8.5 for stopper use).

The minimum distance between a winch drum and the nearest fairlead should be such that the fleet angle does not exceed 1.5°. This means that the minimum distance to the nearest fairlead should be approximately 19 times the drum width if the line rests in one position in the fairlead (such as a pedestal fairlead). If the line position within a fairlead is variable, the distance must be increased to meet the 1.5° requirement in any position.

In the case of split drum winches, only the tension part of the drum should be considered in establishing maximum fleet angles. Figure 3.16 shows the recommended line-up for split drum winches. It must be noted that an unloaded line may be paid out or heaved in directly from any part of either tension or storage drum during the beginning and conclusion of mooring operations. In consequence, the shaded area shown in Figure 3.16 must be kept clear of any obstructions.

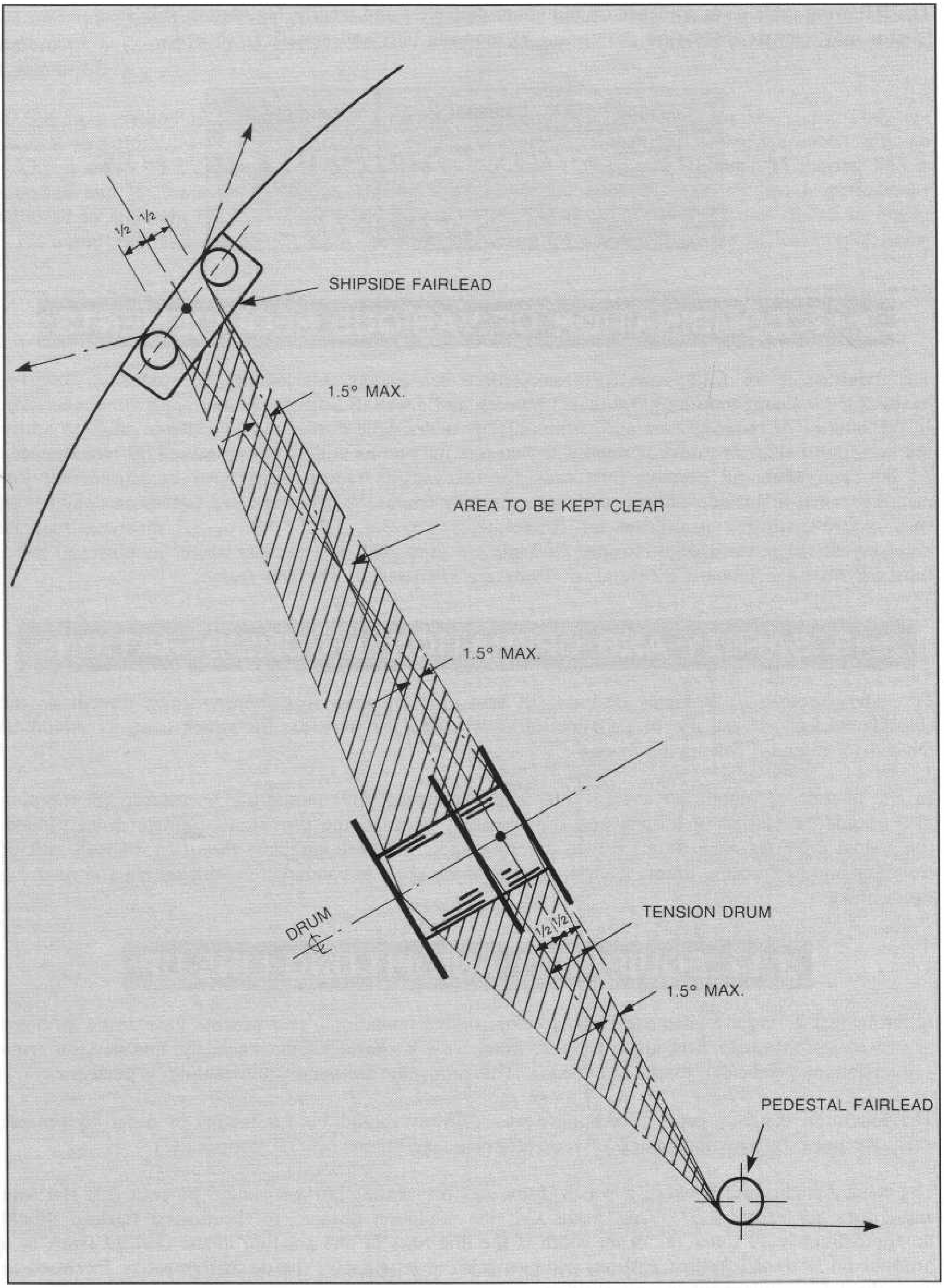


FIGURE 3.16: ALIGNMENT AND MAXIMUM FLEET ANGLE FOR MOORING WINCHES

Section 4.0
Design Loads, Safety Factors and Strength

4.1 GENERAL

These guidelines are intended to assist ship operators, designers and equipment suppliers in outfitting ships with mooring equipment designed to safely accommodate the expected loads.

Mooring equipment means those pieces of equipment mounted onboard a ship to handle the loads needed to temporarily attach the ship to a pier, the ocean bottom or another ship. Mooring equipment includes bollards, mooring winches, anchor windlasses, chain stoppers, fairleads and capstans. Anchoring equipment is not included in these guidelines, since it is adequately covered in Classification Society rules.

In order to define the strength of a fitting or piece of equipment, the following factors must be known or assumed:

- the magnitude, position and direction of application of the most severe service load; this is the safe working load (SWL) of the fitting.
- the safety factor associated with the above load.
- the 'design basis' for the fitting.

The safe working load is the maximum load that should normally be applied in service conditions. The 'design basis' of a fitting provides an additional margin of safety against the rupture of any portion of the fittings or attachments. The stress levels on which the 'design basis' is calculated are in excess of the SWL and represent the load level at which any part or component reaches the minimum yield point of the material. The safety factor may therefore be defined as the ratio of yield load to safe working load, where yield load is the load at which the fitting or equipment will start to deform permanently. Alternatively, the safety factor may be defined as the ratio of breaking load to SWL. The latter definition would be applicable to items such as fibre or wire ropes where the yield load is difficult to establish.

Safety factors account for uncertainties such as additional dynamic loads, normal wear or corrosion of fittings or equipment, small material or welding defects, locked-in weld stress, etc. The choice of the safety factor is also influenced by the consequence of a failure. For example, safety factors must be very high for ropes used to hoist personnel.

The load location and direction, in combination with the line load, determines the total load and stress on a fitting. A line led 180° around a bollard or pedestal fairlead subjects the fitting to twice the rope load; and a line attached to a bollard near the top of the barrel produces a higher stress than one attached close to the base.

4.2 BASIC STRENGTH PHILOSOPHY

Since the wire rope, synthetic rope or chain with a specific breaking strength is used as the link between the ship and the berth, it is convenient to relate the strength of equipment and fittings to the strength of the associated lines or chains.

Industry practice has not been consistent in this respect. Some designers base the strength of fittings and equipment on the maximum line tension anticipated for certain weather criteria; others base the breaking strength of fittings and equipment on the minimum breaking load of mooring line. Neither of these strength criteria is appropriate if damage to the fitting and equipment is to be avoided. Under heavy loads it would be possible for the fitting to become damaged while the mooring line is still intact. A deformed fitting may be unusable, where the overstressing of a line may not be obvious until it breaks.

In terms of repair expense, damage to fittings and equipment is usually more serious than damaging or breaking a line.

The recommended strength standard for mooring fittings and equipment is therefore that the fitting or piece of equipment and its components should be able to withstand, without permanent deformation, loads equivalent to the specified breaking strength of the mooring line.

4.3 EXISTING STANDARDS

Numerous national standards for mooring fittings exist, but they often do not provide sufficient information to establish the actual strength. In some cases a 'SWL' is stated, but no safety factor; in others an 'applicable line' is listed, but no information as to how the line stress relates to fitting stress is given; in yet other cases the line position, direction or quantity may be missing. In comparing two fittings designed to different standards, it is possible that the obviously weaker design lists a higher rated 'load'. Listed 'load' differences between two fittings of equal size may be as much as a ratio of 1 to 10, most of which can be due to different definitions of 'load', safety factors and load application.

Mooring fittings are often specified in nominal sizes such as 300mm or 400mm diameter bollards. Fittings with the same nominal size manufactured to different standards may have widely varying actual strength capabilities and safety factors.

Table 4.1 gives a summary of current mooring equipment safety factors and design criteria as far as these can be determined from some existing standards. As can be seen from the table, these parameters cannot easily be established for all mooring equipment. In the absence of recognized design criteria, manufacturers rely on internal criteria developed during the design, testing and manufacture of mooring equipment. They may also manufacture mooring equipment to recognized standards that specify dimensions, materials of construction, tolerances, quality assurance and testing procedures, as well as the maximum loads or maximum rope or chain sizes that can be accepted. A list of widely used standards (ISO, British and Japanese) for mooring equipment is contained in Table 4.2. These standards can become defacto industry standards as owners experience satisfactory service from a particular manufacturer's equipment. The ISO standards are the recognized international industrial standards on which many national standards are based. However, without universally recognized strength standards, evaluation of new offerings and definition of strength limits by designers and operators is difficult.

When the applicable mooring line size is determined from the existing standards, it is recommended to check the allowable minimum breaking load (MBL) of the rope, if specified, along with the specified applicable rope size because the rope breaking strengths may be different among standards and depending on the grade.

It is to be noted that Regulation 15-1 of SOLAS Chapter V (Ch II-1/3-4 from 1/7/98) requires all emergency towing components to have the specified working strength for all relevant towline angles, i.e., up to 90 degrees from the ship's centreline to port and starboard, and 30 degrees vertical downwards.

Since no conventional closed chocks designed to the current standards will meet the above loading requirements, particular attention should be paid when closed chocks are ordered for compliance with the above regulation, or if closed chocks for SPM will also be used for the emergency towing.

ITEM	STANDARD	DESIGN CRITERIA	SAFETY FACTOR	REMARKS
Bollards	ISO 3913	Design load: MBL of 2 ropes for figure-of-eight fashion, close to base. 2 times MBL of 1 rope on single bitt.	2.0	Safety factor: Design load/Max. Single rope load The bending stress is limited to 85% of the yield stress of the material, and the shear stress to 60% of the yield stress.
	BS MA 12	Ditto except the loading point on single bitt specified at height of 1.2 × bitt diameter.	2.0	Safety factor: Design load/Max. Single rope load The same limits of bending stress and shear stress as by ISO 3913.
	JIS F2001 F2018	Size/strength of applicable rope.	—	The specified MBL of "applicable" rope corresponds to ISO 3913 values.
Closed Chocks	BS MA 19	Size/strength of applicable rope.	—	
	JIS F2005 F2007 F2017	Applicable rope size. Applicable rope size.	— —	
Roller Fairleads	BS MA 22 DIN 81906 JIS F2014	MBL of applicable rope. 180° wrap & SWL specified. Applicable rope size.	— — —	
SPM Chafe Chain	OCIMF	SWL 200/250 tonnes.	2.2 – 2.4	Safety factor based on chain B.S.
SPM Chain Stopper	OCIMF	SWL 200/250 tonnes.	1.5	Safety factor based on yield stress.
Emergency Towing Components	SOLAS	SWL 1000/2000 kN.	—	Tow line angle; up to 90° from ship's centre line and 30° vertical downwards.

TABLE 4.1: MOORING EQUIPMENT DESIGN CRITERIA FROM EXISTING STANDARDS

TITLE	BS MA No.	JIS F No.	ISO
Bollards	12(Pt. 1)	2001-90	3913
Cast Iron Bar Type Chain Cable Stoppers	56	2002-76	6325
Cast Iron Deck End Rollers		2003-90	
Steel Plate Deck End Rollers	21	2004-90	
Closed Chocks		2005-75	
Open Chocks		2006-76	
Mooring Pipes (Closed Chock in Bulwark)		2007-76	
Fairleads (Roller)	22	2014-87	
Cast Steel Bar Type Anchor Cable Stoppers	56	2015-87	
Cast Steel Pawl Type Anchor Chain Cable Stoppers for Grade 2 Anchor Chain Cable	56	2016-87	
Panama Chocks	19	2017-82	
Bollards (Simple Type)	12	2018-76	
Small Size Cast Iron Deck End Rollers	21	2019-79	
Small Size Steel Plate Deck End Rollers	21	2020-79	
Small Size Fairleads	22	2021-76	
Ship's Horizontal Rollers		2022-90	
Ship's Small Size Cast Steel Bar Type Anchor Chain Cable Stopper	56	2023-76	
Ship's Small Size Stand Rollers	22	2024-75	
Cable Clenches		2025-76	
Fairleads with Horizontal Rollers	23	2026-80	
Rollered Pawl Type Anchor Chain Cable Stoppers for Grade 2 Anchor Chain Cable	56	2027-87	
Rollered Bar Type Anchor Chain Cable Stoppers for Grade 2 Anchor Chain Cable	56	2028-87	
Ship's Towing and Mooring Brackets		2029-78	
Single Point Mooring Pipes (Closed Chock in Bulwark)		2030-78	
Cast Steel Pawl Type Anchor Chain Cable Stoppers for Grade 3 Chain Cable	56	2031-87	6325
Rollered Pawl Type Anchor Chain Cable Stoppers for Grade 3 Chain Cable	56	2032-87	6325
Rollered Bar Type Anchor Chain Cable Stoppers for Grade 3 Chain Cable	56	2033-87	6325
Double Type Cross Bitts for Tugboat		2051-76	
Ship's Cross Bitts	12(Pt. 2)	2804-76	

TABLE 4.2: STANDARDS FOR MOORING EQUIPMENT

TITLE	BS MA No.	JIS F No.	ISO
Anchors		3301-90	
Cast Steel Anchor Chain Cables		3302-90	1704
Flash Butt-Welded Anchor Chain Cables		3303-90	1704
Anchor Stoppers		3307-78	
Anchor Stoppers (Small Size)		3310-90	
Chain Stoppers		3406-90	
Ship's Wire Reels		3430-80	
Application of Wire Ropes in Ship		3433-83	
Steel Wire Ropes		JIS G 3525	2408
Manila Ropes		JIS L 2701	
Synthetic Ropes		3434-83 JIS L 2703-2707	1141/1346
Mooring Winches	32	6709-85	3730/7825
Windlasses	35	6714-83	4568
Warping End Profile			6482

TABLE 4.2: (Cont)

4.4 RECOMMENDED STANDARDS

Until accepted standards are developed for all mooring equipment, safety factors and SWL, should be set as given in Table 4.3. These recommendations are based on the basic strength criteria mentioned above and allow for wear and tear or corrosion in service, residual stresses or construction defects during manufacture, and a degree of dynamic loading of the fitting. For all fittings and equipment, the basic requirement is that the fitting or equipment should not suffer any permanent deformation when the associated line or lines are tensioned to their MBL. The criteria for chain stoppers and Smit brackets used at SPMs and for emergency towing are modified to take account of existing standards, which recognize that the actual strength of the associated chafing chain, when bent around a closed bow chock, is less than the nominal chain strength.

When selecting mooring equipment for new ships or conversions, it is recommended that the strength criteria listed in Table 4.3 be specified in addition to the usual information on size and materials. Reference to a specific standard should only be made if all strength details are published and are in general agreement with Table 4.3. Standard fittings of unknown strength may be specified with the proviso that the standards are to be used as a guide to overall dimensions, materials and design concept, but that actual scantlings must be modified, if necessary, to meet the strength requirements in Table 4.3. In this case, compliance with the criteria must be substantiated with detailed calculations and a load test for each type and size of fitting.

4.5 STRENGTH TESTING OF MOORING FITTINGS

Unless a fitting is constructed in strict compliance with a recognised standard that specifies design load, safety factor and load application, a load test may be required. The load test should be performed on one fitting of each type and size. A manufacturer's test certificate is acceptable if the test was witnessed by an independent authority (such as a Classification Society) and the witness certificate lists all details such as test load, load application, dimensions of scantlings and materials.

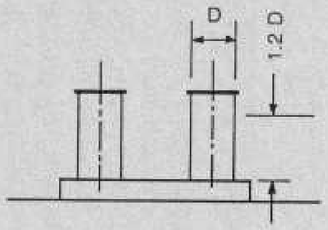
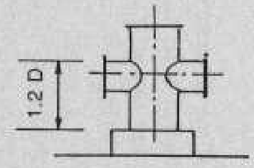
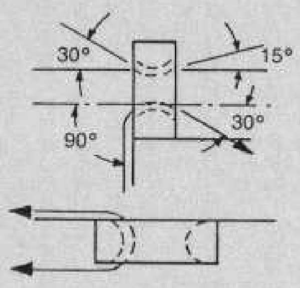
FITTING	SWL (See Footnotes ① and ②)	LOAD POSITION (LINE POSITION)	SAFETY FACTOR ON YIELD (See Footnote ③)	TEST LOAD	NOTES
Double Bollards 	Per ISO 3913 "Single Rope Maximum Loading"	1.2 × barrel dia. (D) above base, max.	2.36	2 × SWL (See also ISO 3913, Part 8.2, for test load position)	1. Scantlings must be as per ISO 3913 2. Barrel loading when figure-of-eight belayed assumed to be twice eye load 3. SWL applies to figure-of-eight belaying
Cruciform Bollards 	Single type: 2 × above load Double type: same as above	1.2 × dia. (D) above base, max.	2.36	2 × SWL (See also ISO 3913, Addendum 1)	1. Scantlings must be as per ISO 3913 2. See note 2 above for double bollard.
Closed Chocks 	MBL of one line $\frac{d}{R} = \frac{1}{4}$ IWRC wire rope with min. strand strength of 1770 N/mm^2 (See Note)	Outboard: horizontal: ± 90° vertical: up 30° down 90° Inboard: horizontal: ± 90° vertical: ± 30°	2.36	2 × SWL	d = rope diameter R = chock surface radius If used for primary mooring line, MBL may be that of actual associated mooring line max. horizontal load = 2 x rope load.

TABLE 4.3: (SHEET 1)
RECOMMENDED SWL, LOAD POSITION, SAFETY FACTOR AND TEST LOAD

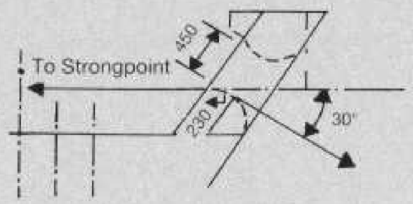
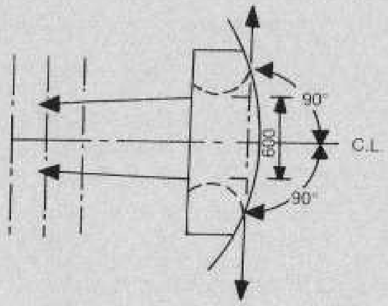
FITTING	SWL (See Footnotes ① and ②)	LOAD POSITION (LINE POSITION)	SAFETY FACTOR ON YIELD (See Footnote ③)	TEST LOAD	NOTES
Closed Chocks for SPMs and Emergency Towing Elevation to Strongpoint   (ships over 20,000 DWT)	20,000 to 50,000 DWT: 1000 kN (stern only) 2000 kN (bow only) 50,000 to 350,000 DWT: 2000 kN (bow and stern) Over 350,000 DWT: 2500 kN (bow only) 2000 kN (stern only)	Outboard: horizontal: ± 90° vertical: up 0° down 30°	1.5 (or 2.0 on ultimate strength)	SWL	Two chocks at bow required for all ships over 150,000 DWT (SPM) Size of stern chock may be reduced for vessels less than 50,000 DWT to suit reduced gear.
Strongpoint for Emergency Towing	Up to 50,000 DWT: 1000 kN Over 50,000 DWT: 2000 kN	towards closed chock	2.0 on ultimate strength	SWL also prototype test required	May be chain stopper, Smit bracket or similar fitting. At bow, SPM chain stopper may be used as strongpoint if chafing chain is not attached

TABLE 4.3: (SHEET 2)

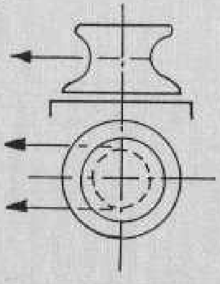
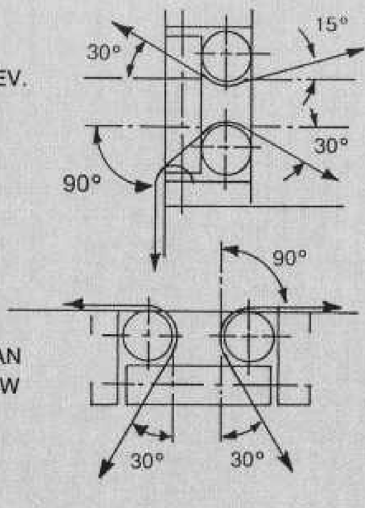
FITTING	SWL (See Footnotes ① and ②)	LOAD POSITION (LINE POSITION)	SAFETY FACTOR ON YIELD (See Footnote ③)	TEST LOAD	NOTES
Pedestal Fairleads and Rollers of Button-Roller Chocks 	0.55 MBL of one line: $\frac{d}{R} = \frac{1}{4}$ IWRC wire rope with min. strand strength of 1770 N/mm^2 (See Note)	180° wrap; at upper end of cylindrical or conical part of throat (at center of roller if radiused throat)	2.02	1.82 × SWL (= MBL)	If used for primary mooring line, MBL may be that of actual associated mooring line. Total load on fitting is equal to twice the line load.
Universal Fairlead 4-roller type ELEV. PLAN VIEW 	0.55 MBL of one line: $\frac{d}{R} = \frac{1}{4}$ IWRC wire rope with min. strand strength of 1770 N/mm^2 (See Note)	Outboard: horizontal: ± 90° vertical: up 30° down 90° Inboard: horizontal: ± 30° vertical: ± 15°	2.02	1.82 × SWL (= MBL)	MBL may be MBL of actual mooring line. Inboard line leads may correspond to actual leads to winch drum. Position of line in aperture to be assumed such as to produce max. stress in part investigated. Assume highest position to calculate stress in frame and roller bearings; at mid-length of roller to calculate bending in roller.

TABLE 4.3: (SHEET 3)

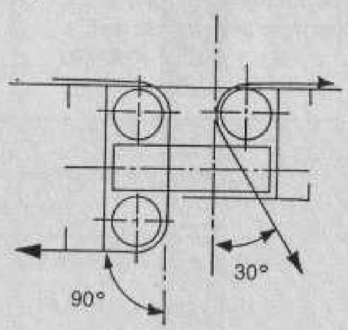
FITTING	SWL (See Footnotes ① and ②)	LOAD POSITION (LINE POSITION)	SAFETY FACTOR ON YIELD (See Footnote ③)	TEST LOAD	NOTES
5-roller type 	As for 4 roller type above	Outboard: horizontal: ± 90° vertical: up 30° down 90° Inboard: horizontal: ± 30° vertical: ± 15°	2.02	1.82 × SWL (= MBL)	See comments for 4-roller type.
Chain Stoppers for use at SPMs	Up to 350,000 DWT: 2000 kN Over 350,000 DWT: 2500 kN	In-line with fitting	1.5 (or 2.0 on ultimate strength)	SWL	SWL to be marked on fitting. Two fittings required for all ships over 150,000 DWT. Bow stopper may be used as strongpoint for emergency towing.
Mooring Winch Drum, Shafts, Bearings.	MBL of mooring line	Split drums: on first layer Single drums: first or top layer, whichever produces highest stress	1.11 (or 2.5 based on rated load whichever is higher	SWL	See Section 7, Table 1 for standard winch sizes and rope B.S.
Mooring Winch Brakes, Frames and Foundations	MBL of mooring line	As above	1.5	SWL	Per BS MA 93, Part 4.3.2.
Mooring Winch Drive Components	Stall load	Not applicable	See Note	See ISO 3730 and Section 7	Safety factor = 1.11 based on stall load; or 2.5 based on rated load; whichever is higher (per ISO 3730, Parts 4.2.2 and 4.2.1 respectively).

TABLE 4.3: (SHEET 4)

FITTING	SWL	LOAD POSITION (LINE POSITION)	SAFETY FACTOR ON DRY MBL	TEST LOAD	NOTES
Mooring Lines	Highest load calculated for adopted standard environmental criteria	Not applicable	STEEL: 1.82 NYLON: 2.2 OTHER SYNTH: 2.0	Test sample to destruction to confirm MBL	The safety factors for ropes quoted should only be applied to primary mooring lines in combination with the SWL defined. For general application, safety factors of ropes must be higher.
Tails for Mooring Lines	As above	Not applicable	POLYESTER: 2.3 NYLON: 2.5	As above	
Chafing Chain for Emergency Towing (if provided as chafing gear) Grade U3 stud link chain (vessels over 20,000 DWT)	Up to 50,000 DWT: 1000 kN Over 50,000 DWT: 2000 kN when bent 90° through chock	Not applicable	2.0 on ultimate strength	Up to 50,000 DWT: 1589 kN Over 50,000 DWT: 3011 kN	Up to 50,000 DWT: 54 mm dia; B.L. = 2265 kN Over 50,000 DWT: 76 mm dia; B.L. = 4295 kN Length outboard: 3.0 m. Required forward; Aft requirement depends on design.
Towing Pennant 6 × 41 W.S. + I.W.R.C. galvanized wire rope (vessels over 20,000 DWT)	Up to 50,000 DWT: 1000 kN Over 50,000 DWT: 2000 kN	Not applicable	2.0 on ultimate strength	Test sample to destruction	Up to 50,000 DWT: 58 mm diameter Over 50,000 DWT: 76 mm diameter Length = 2 × freeboard at chock + 50 m. Pennant required aft; optional forward.
Pick-up Gear (vessels over 20,000 DWT)	225 kN	Not applicable	2.0 on ultimate strength	As above	Size 7 polypropylene line 56 mm dia. Length = 120 m. Required aft; optional forward.

TABLE 4.3: (SHEET 5)

Footnotes

① SWL to be marked on or near the fittings in units of metric tons (letter "t")

② The SWL of foundations and supporting deck structure must be specially considered when siting and rating the capabilities of the fitting. In principle the strength of the supporting structure and connection of the fitting to it should be greater than the fitting itself so that any failure does not result in damage to the ship's structure. Since foundations and connections to the deck are often made in non-ideal conditions onboard the ship rather than factory conditions, and to allow for possible weld imperfections, it is recommended that they be at least 10% stronger than the fitting.

③ Safety Factor $\dfrac{\text{Nominal material yield stress}}{\text{combined tension, compression or bending stress}}$ if only tension compression or bending stress (∂) applied

$\dfrac{\text{Nominal material yield stress}}{\sqrt{3} \times \text{shear stress}}$ if only shear stress (τ) applied

$\dfrac{\text{Nominal material yield stress}}{\sqrt{\partial_x^2 + \partial_a^2 - \partial_x \partial_y + 3\tau^2}}$ if both ∂ and τ applied

The test load should be applied with a rope of adequate strength to allow a line tension equal to the 'test load' listed in Table 4.3. Alternate arrangements are acceptable if the test load is equivalent to the resultant load from a line application.

The load test may be performed aboard the ship after all installation and structural reinforcements are completed. It is recommended that some assurance as to the fitting's strength be obtained at an early stage of the ship's construction, since it is very costly and time-consuming to replace mooring fittings at a time close to the delivery of a new ship.

4.6 MARKING OF MOORING FITTINGS

Marking of mooring fittings is required to provide ship operators with information on the strength of fittings.

> It is recommended that all chain stoppers or Smit brackets used for emergency towing be marked with their SWL (characters at least 5 cm high formed by weld metal deposit on the deck plate beside the chain stopper or Smit bracket).
>
> BS MA 10 requires that oval eyeplates be marked with the number and nominal size of the British Standard (nominal size = SWL in metric tonnes).
>
> BS MA 12 requires that bollards be marked in the centre of one of the caps with the number and nominal size of the British Standard.
>
> BS MA 19 requires that Panama-type fairleads be marked with the number, type and size of the British Standard.

Each fitting must be clearly marked by weld bead outline with its SWL as listed in Table 4.3, in addition to any markings required by other applicable standards. The SWL is expressed in metric tonnes (letter 't') and corresponds to the loading of the associated line or lines, or chain (not the resultant load which may be higher). Note that the unit 't' is recommended rather than the technically correct kN, since some operators may not be fully familiar with the metric system and a fitting may be overloaded if 'kN' is confused with 't'. For double bollards the SWL should correspond to the loading by a wire rope belayed in a figure-of-eight fashion near the base of the bollard.

Since the SWL does not provide information on safety factor, test load or line application, and marking of all data would be impractical, the ship should be provided with all additional relevant information such as actual test load applied, geometry of load application, bollard strength when belayed by eye and higher up on the barrel, maximum size and MBL of applicable line or chain, test certificates, standard drawings, etc. Ideally, such information should be incorporated in a mooring layout plan available or displayed on the ship.

4.7 RECOMMENDATIONS FOR DESIGNERS

Minimum safety factors listed in Table 4.3 are based upon the appropriate design criteria and loading assumptions, and should be incorporated in all new equipment and mooring fittings.

All equipment and fittings should be clearly marked with their SWL and design standard.

4.8 RECOMMENDATIONS FOR OPERATORS

All damaged or deformed equipment or fittings should be treated with suspicion regarding residual strength capabilities and only utilized for low loads, or ideally not used at all until repaired or replaced.

If any doubt exists regarding strength capabilities or rated SWL of a fitting or piece of equipment, it should be proof-tested.

Section 5.0
Structural Reinforcements

5.1 BASIC CONSIDERATIONS

Mooring fittings and equipment must be connected to the ship structure in such a way that no failure will occur under anticipated static and dynamic loadings. Section 4 gives the recommended minimum strength criteria for the fittings or equipment. For heavy equipment such as winches, the weight of the equipment and dynamic loads in a seaway must also be taken into account. It is not usually necessary to add the static and dynamic loads caused by the ship's seaway motion to the loads generated by mooring lines or chains, since the ship will rarely be subjected to excessive motion while moored. Nevertheless, once the static requirements are met, the foundations of heavy equipment should be checked for dynamic forces in the same manner as other main and auxiliary machinery.

In selecting fittings from various standards or vendors, the method of hull attachment must be carefully considered. Less expensive fittings may require elaborate hull reinforcements. For example, some universal fairleads do not provide strength members between the end posts, which results in very high localized reaction loads on the deck.

Fittings or equipment generally apply tension, compression and shear stresses to the deck structure. These stresses must be added to the hull stresses that may exist while the ship is moored. The longitudinal deck stress may be assumed to correspond to the stress generated by the maximum allowable still water bending moment. For fittings in the bow and stern area, this stress may be ignored.

Another consideration for equipment and fittings in the mid-body area is the stress-raising effect that any local reinforcements may have on longitudinal strength members. This applies especially to deck plating and deck longitudinals of high tensile steel (HTS) where the ends of reinforcing members may generate fatigue cracking in the primary structure. For this reason, transverse reinforcing members are strongly preferred over longitudinal reinforcements. Where longitudinal reinforcements cannot be avoided, the ends of the reinforcing members should be very gradually tapered.

Tensile loadings (pull on deck plate) are the most difficult ones to accommodate. If the deck plating is thin in relation to the member on top and the reinforcement below, the heavy welding required could cause tearing of the deck plate. Furthermore, any misalignment between members above and below the deck would result in high deck bending stresses. For this reason, deck insert plates should be provided where the deck plate is thinner than the member welded to it. Another method, especially suitable for bow and stern area equipment such as anchor points for brake bands on mooring winches, is to lead a tension member through a slot in the deck and connect it directly to reinforcing members below the deck. The deck slot generated by this method can be sealed with collar plates.

Special attention must be paid to connections of fittings made from steel of higher strength than the hull steel. If local stresses are high, and adequate compensation cannot be made using the original hull steel quality, then local installation of higher strength steel may be necessary.

5.2 MOORING WINCHES

Mooring winches are bolted to foundations that are welded to the ship's deck. Further details on the safety aspects of deck machinery installation may be found in BS MA-93. The foundation should be designed so that all parts are accessible and hold-down bolts can be fitted from below. Vertical members should be positioned close to bolt holes and generally span the under-deck longitudinals or beams.

Adequate drain holes should be provided to avoid any entrapment of water, which could lead to corrosion damage.

Steel chocks or pourable resin compound may be fitted between the foundation and the machinery bed plate. Resin chocks should be of suitable and proven material.

If steel chocks are used, an area on the foundation top plate may require machining.

If resin chocks are fitted, the top plate should be sized taking into account the necessity of fitting dams to retain the resin.

The surfaces where resin chocks are to be used should be cleaned and the hold-down bolt torque should not exceed the resin chock supplier's recommendation.

Brake anchors should be designed to meet the design criteria given in Section 4. For loads in excess of about 100 tonnes (981 kN), the brake anchors should preferably be carried through the deck; alternatively, brake anchors can be welded to the foundation with adequate toe brackets in line with deck stiffeners. Welding should be full penetration type. Local deck insert plates may also be necessary.

An abutment or end stopper may be welded to the foundation at points predetermined by the machinery manufacturer to reduce the hold-down bolt's shear loading and to reduce the need for fitted bolts.

Foundations of all winches of greater than five tonnes rated pull may require to have abutments fitted.

An underdeck support structure should be provided in line with the foundation above the deck. For winches in the mid-body area, the support should preferably be in a transverse direction and be of adequate size and span to distribute the load into existing deck longitudinals. Reinforcement of existing longitudinals should be avoided if at all possible to prevent fatigue cracking at the ends of reinforcements.

Where tension loads are applied to the deck, it may be necessary for welding above and below the deck to be of the full penetration type. Other type weld sizes should be checked for adequacy. Where foundation members line up with existing deck structure, the standard weld size of deck longitudinals, beams or transverse webs may also have to be increased.

5.3 CHOCKS AND FAIRLEADS

Chocks and fairleads are often welded directly to the hull structure. The outer faces of chocks and fairleads located in the bow and stern areas should butt directly into the shell plating (or a shell plate insert if the shell plate is of insufficient thickness). In the mid-body area, any welding to the sheer strake or the rounded gunnel plate should be avoided if possible. A proper connection can be achieved with a cantilevered foundation as shown in Figs. 5.1 and 5.2. The foundation should be sized so that the two longitudinal members line up with deck longitudinals. Additional transverse reinforcing members below the deck may also be required to spread the load over additional deck and shell longitudinals. The ends of longitudinal foundation members should have well-radiused connections to reduce stress concentrations in the upper deck.

The installation of some universal fairleads, such as the one shown in Fig. 5.3, require special attention, since the supporting structure must not only absorb the overall reactions from a mooring line load, but also the reactions of the two individual end frames. The preferred design is a fairlead with a substantial bottom member that connects the two end frames. The benefits in reduced deck reaction forces and lesser reinforcement requirements are shown in Fig. 5.4.

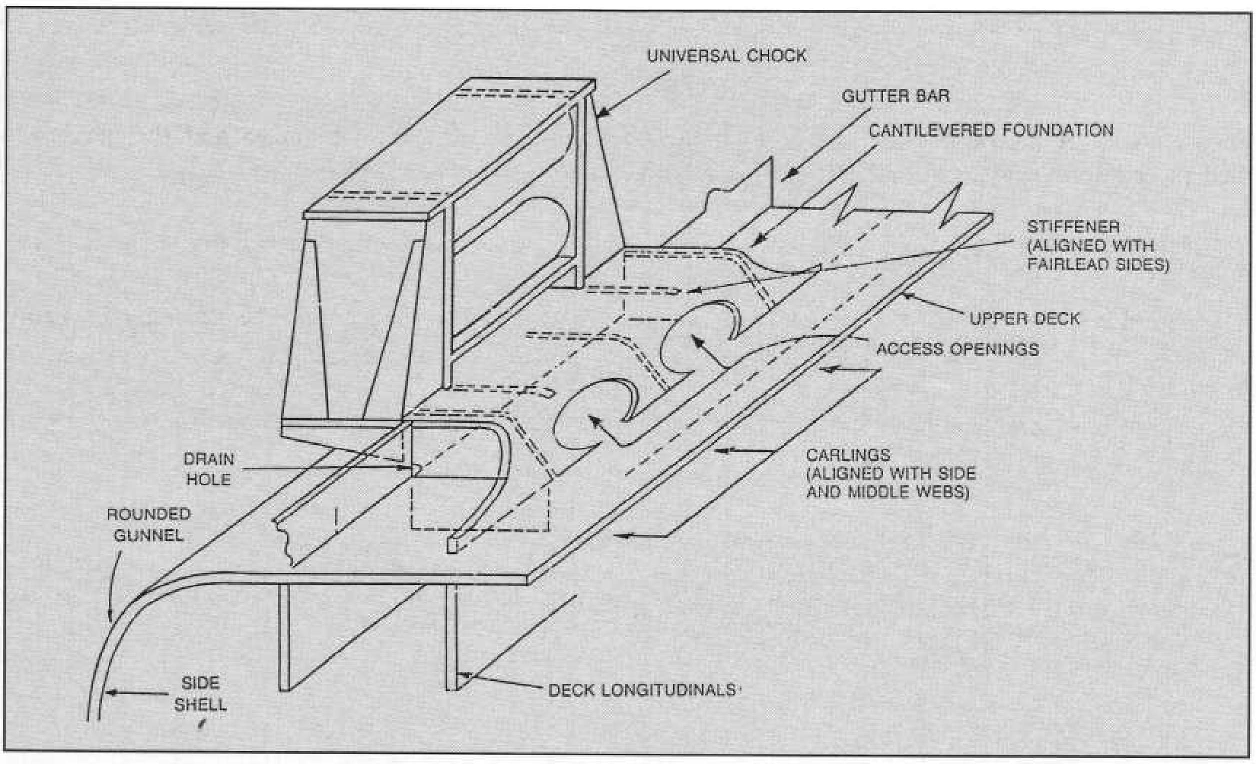


FIGURE 5.1: TYPICAL CANTILEVERED FOUNDATION IN WAY OF ROUNDED GUNNEL

5.4 PEDESTAL FAIRLEADS

The critical connections are the roller pin to pedestal and pedestal to deck attachment. Pedestal fairleads have failed more often in these areas than any other mooring fitting. An example of a good pin-to-pedestal connection and pedestal-to-deck connection is shown in Figs. 5.5 and 5.6.

Most existing pedestals are of cylindrical or conical shape, which makes a proper line-up with reinforcements below the deck difficult. For this reason, a rectangular section with rounded edges (see Figs. 5.5 and 5.6) may be more appropriate in some cases.

5.5 BOLLARDS

Bollards require deck strengthening members in line with all four sides of the bollard base. The members below deck should be of the same thickness as the base, and their welding to the deck should be equal to the weld size between bollard base and deck. Where bollards line up with existing structure, such as deck beams, girders, or transverse webs, the welding of these members to the deck may have to be increased.

5.6 SPM AND EMERGENCY TOWING FITTINGS

Due to the high loading on these fittings, the method of connection to the hull structure should be well thought out. Each chain stopper or Smit bracket should be welded directly to the deck of the vessel or welded or bolted to a plate or pedestal structure which is in turn welded to the deck of the vessel. The stopper or bracket should not be bolted directly to the deck of the vessel. A heavy deck insert plate may be required. In place of one centreline girder, two parallel girders with spacing equal to the distance between the cheek plates of the chain stopper or SPM bracket are recommended. Additional transverse members and pillars may be required to absorb the load. The SPM or towing fitting should be located as close to the deck as possible. However, a small foundation will normally be required to achieve proper alignment between the fitting, the bow chock and the pedestal fairlead. The foundation must be provided with a thrust block capable of absorbing the chain load specified in Section 4.

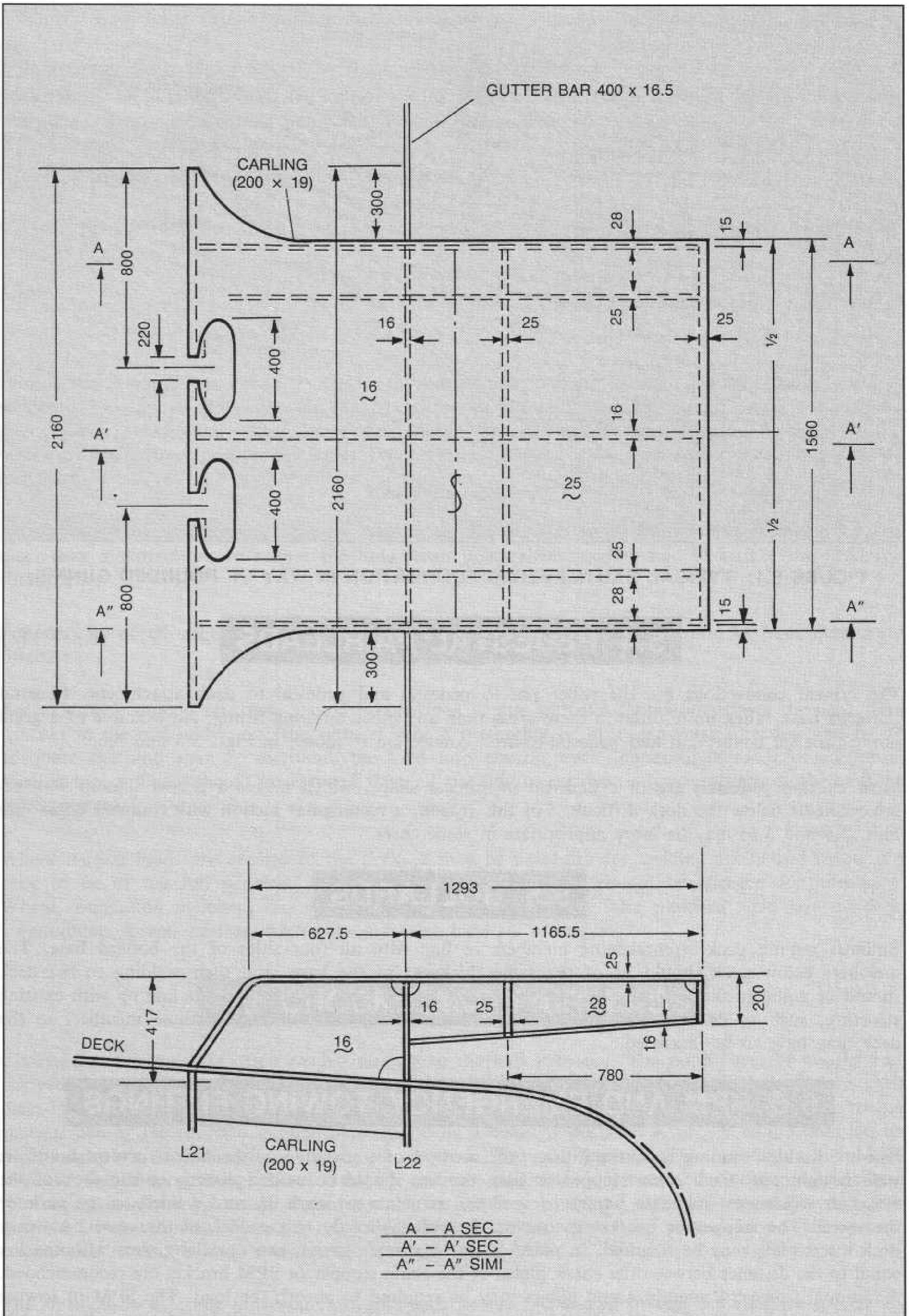
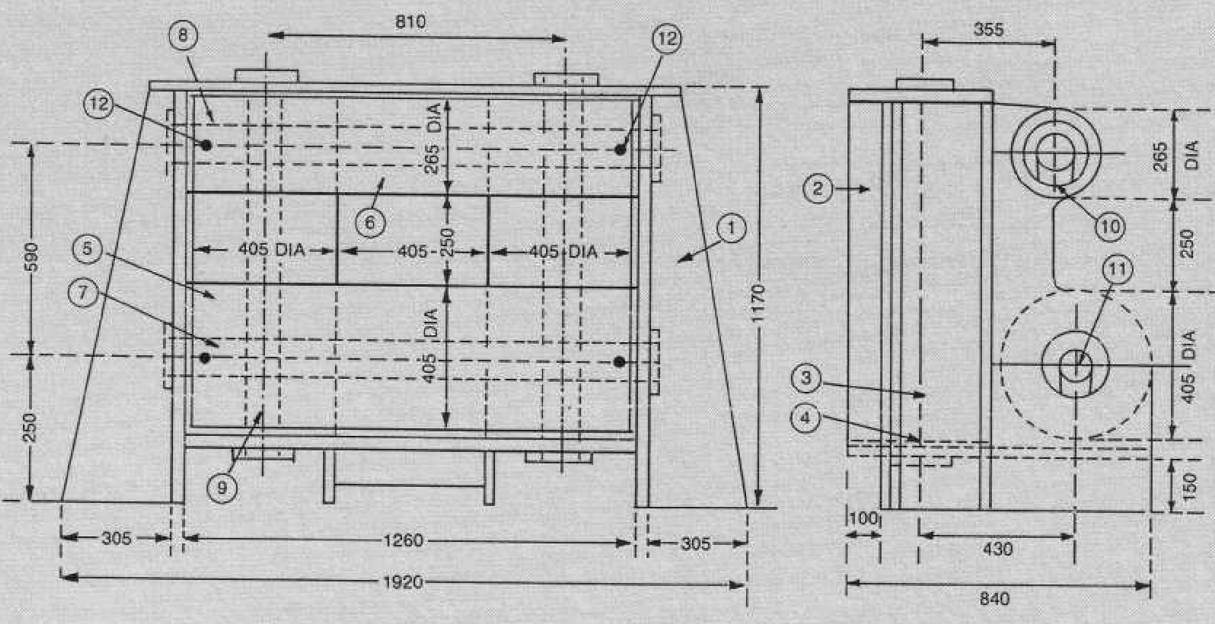


FIGURE 5.2: EXAMPLE OF A CANTILEVERED FAIRLEAD FOUNDATION



ITEM	DESCRIPTION	QTY
1	FABRICATED FRAME	1
2	VERTICAL FRAME	2
3	VERTICAL ROLLER	4
4	THRUST BEARING	2
5	HORIZONTAL ROLLER	1
6	HORIZONTAL ROLLER	1
7	HORIZONTAL PIN	2
8	HORIZ. ROLLER BUSH	4
9	VERTICAL PIN	2
10	TAP BOLT	4
11	WASHER	6
12	GREASE NIPPLE	6

List of material quantities for one fairlead

FIGURE 5.3: UNIVERSAL FAIRLEAD WITH INDIVIDUAL END FRAMES

5.7 SPECIAL CONSIDERATIONS

The following is a discussion of some specific shipboard installation problems:

5.7.1 *Rounded Gunnel Connection*

Most newer ships have a rounded connection between upper deck and side shell in the mid-body area. The rounded plate is of a special steel grade to prevent propagation of major cracks in the hull envelope. Some Classification Societies place restrictions on welding to this plate. Whatever regulatory bodies may have to say on this subject, welding to the plate is discouraged, especially if the deck is of higher tensile steel. Since mooring chocks or fairleads in the mid-body area should be flush with the side shell to avoid line chafing on the rounded gunnel, a cantilevered foundation for all shipside mooring fittings will be required. Figures 5.1 and 5.2 show such an installation.

5.7.2 *Doublers Versus Inserts*

Deck plating in way of mooring fittings may be reinforced by doublers or insert plates. Doublers are usually less expensive but cannot transmit large tensile loads. This is because all loads to the deck are transmitted only through the fillet welds or plug welds of the doubler and these are seldom in line

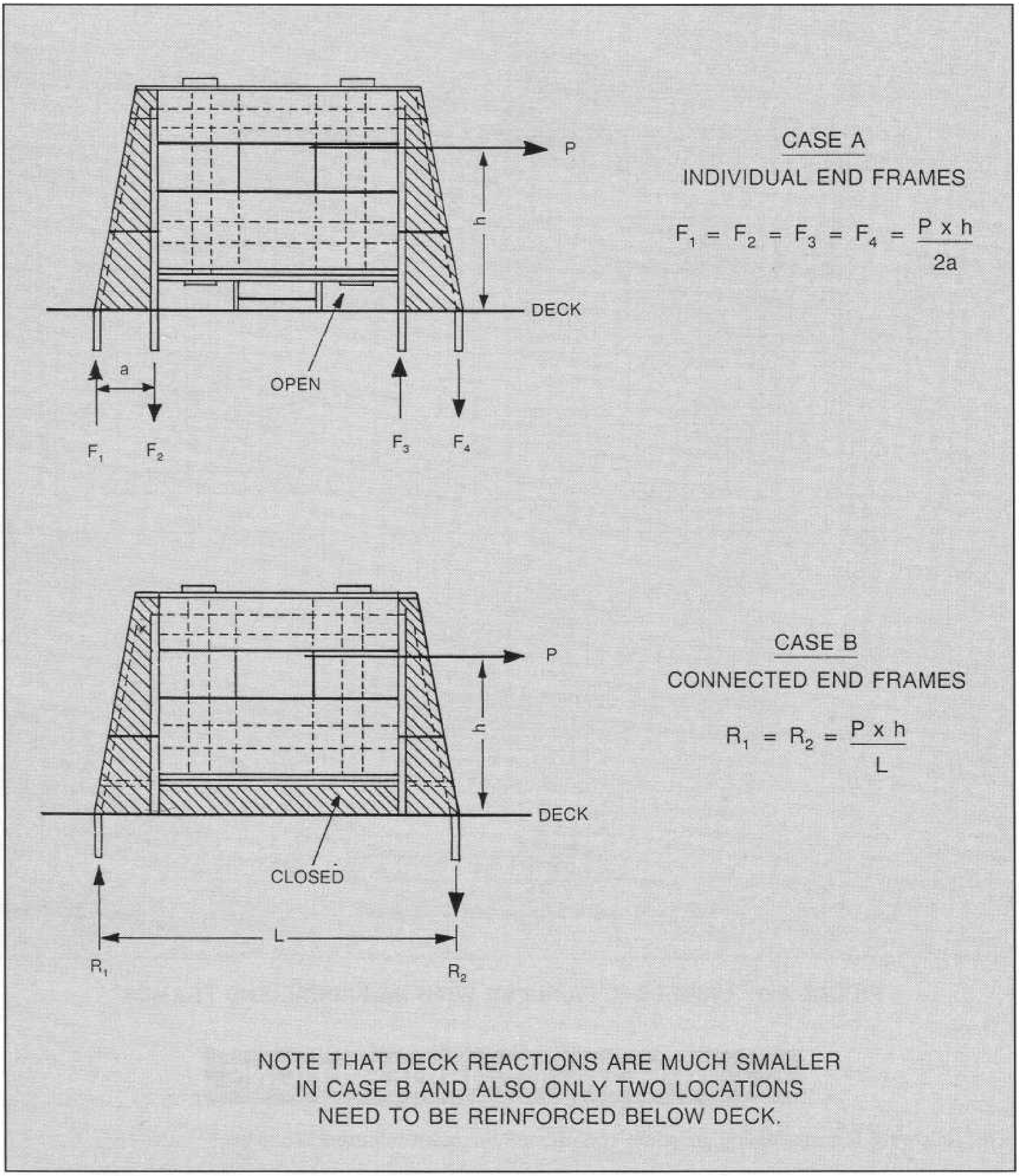


FIGURE 5.4: DECK REACTIONS WITH TWO TYPES OF UNIVERSAL FAIRLEADS

with stiffening below the deck. Doublers are more suitable for small fittings, such as eyes, since a small insert plate in a highly stressed upper deck may lead to crack initiation due to the additional locked-in stress created by the welding. For mooring outfits on the bow and stern where the deck's longitudinal stress is insignificant and the thickness is much less than midships, insert plates should be used.

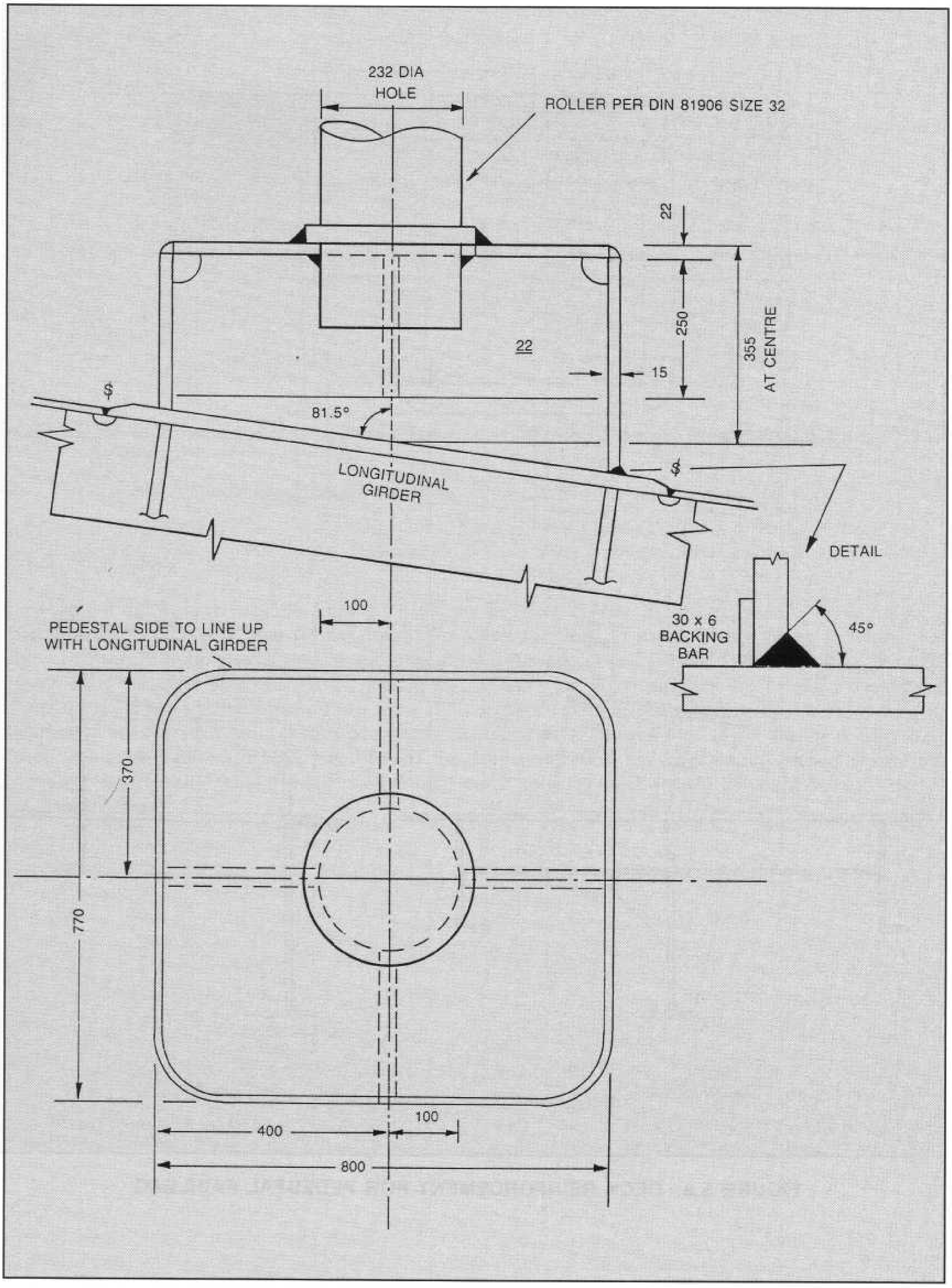


FIGURE 5.5: TYPICAL FOUNDATION FOR PEDESTAL FAIRLEAD

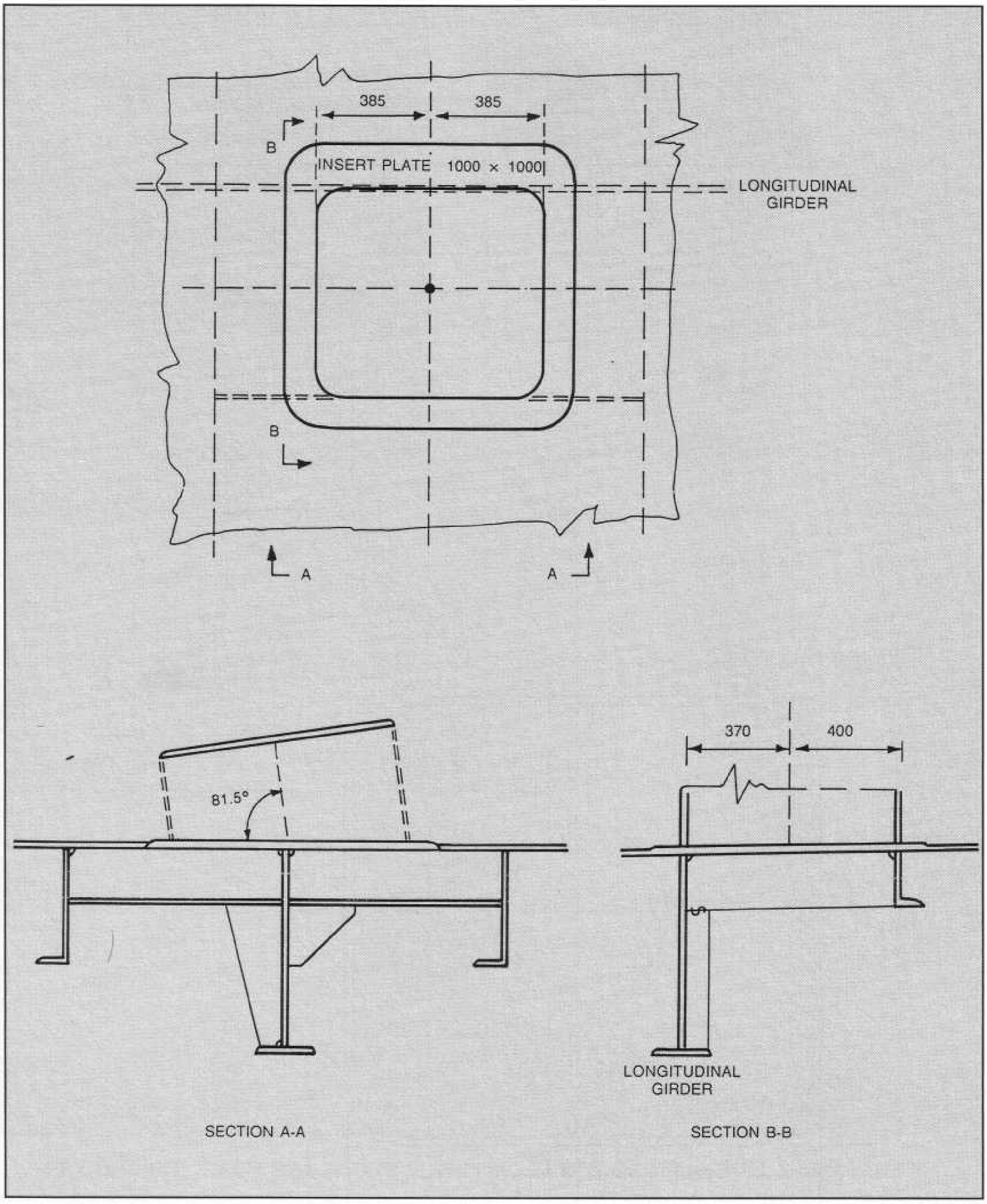


FIGURE 5.6: DECK REINFORCEMENT FOR PEDESTAL FAIRLEAD

Where doublers (or pads) are used, the width should be small to prevent bending under tension loads. Following are typical pad widths and thicknesses which may be used as a guide:

Leg thickness of fittings (mm)	Pad width (mm) (max)	Pad thickness (mm)
Less than 12	50	14
12-13	50	16
14-16	60	19
17-19	70	22
20 and above	75	25

Note: (a) Pad corners should be provided with a minimum of 20 mm radius. The shape of pads should be designed to suit that of legs of fittings.

(b) A greater pad width causes failures (separation of the pad from the deck plate) at lower loads.

5.7.3 *High Strength Steel Fittings*

Some mooring fittings may be built of high tensile steel (HTS) to reduce weight or to improve strength. Connections of such fittings, especially when the deck structure is of a lesser strength steel, should be carefully calculated. Where the maximum stress occurs at the base of the fitting (such as the frame of a universal chock) and the deck is not of HTS or of a sufficient thickness, a deck insert plate of HTS may be required along with HTS strengthening members below the deck. Likewise, existing structures such as deck beams directly in line with HTS members above the deck may have to be locally replaced with HTS members. If the deck connection and reinforcing method is carefully thought out before designing or selecting fittings for a ship, installation will be simplified and overall costs reduced.

Section 6.0
Mooring Lines

6.1 DESIGN STAGE DECISIONS

A major decision must be made at the ship design stage as to whether wire or synthetic fibre mooring lines are to be used. The type of line used will influence such items as winch drum size, types of fairleads, bend radius of fairleads and required deck space.

The principal considerations in selecting a line type are discussed in Section 1. Wire is advantageous where limited movement is required, such as at berths with hard arm equipment and where low dynamic loads are expected. For these reasons wires are recommended on large ships.

Synthetic lines are often favoured for small ships where ease of handling, flexibility of moorings and lower line tension are important criteria. Other factors which may influence the choice of material include cost and the type of outfitting customarily used within a particular trade.

A system utilising wire spring lines and synthetic breast lines, as found on some ships, has certain theoretical advantages. It reduces the fore and aft excursion of the ship while moored which in turn reduces shifting of loads from one breast line to another and limits the motion of loading arms or hoses. Nonetheless, for simplicity and operational flexibility, it is recommended that all lines be of the same material.

Some newer 'high modulus polyethylene' (HMPE) and synthetic aramid fibre ropes with high strength and low stretch characteristics, may be suitable alternatives to steel lines. But they are expensive and have some disadvantages such as low abrasion resistance and being difficult to splice. Retrofitting can also prove problematical as wire line induced mechanical wear on fairleads can result in excessive abrasion of fibre lines.

6.2 WIRE MOORING LINES

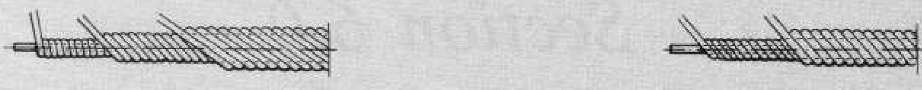
6.2.1 *Material*

To meet the requirements of increased strength for wire mooring lines, manufacturers have developed preformed, drawn galvanized wire with high tensile strengths. The drawn galvanized wire provides strengths of the same magnitude as bright wire and an improvement in wire line quality. To save weight, preformed drawn galvanised wire strands of a minimum tensile strength of 180kg/mm^2 are recommended.

6.2.2 *Construction*

A line should be selected which combines the proper attributes for mooring when reasonable flexibility and high MBL are top priority requirements. The recommended construction is 6×36 or 6×41 (6×37 class) with the wires in each strand of equal lay and the strands of regular (ordinary) right hand lay.

Figure 6.1 illustrates these constructions. Equal lay for lines in each strand is recommended when available because of its higher MBL than cross lay. While Lang lines have a slightly greater MBL than regular lay lines, they have a greater tendency to kink and unlay (or open up the lays of the strands) which is undesirable where grit, dust and moisture are present.



(*a*) Crosslay (*b*) Equal lay

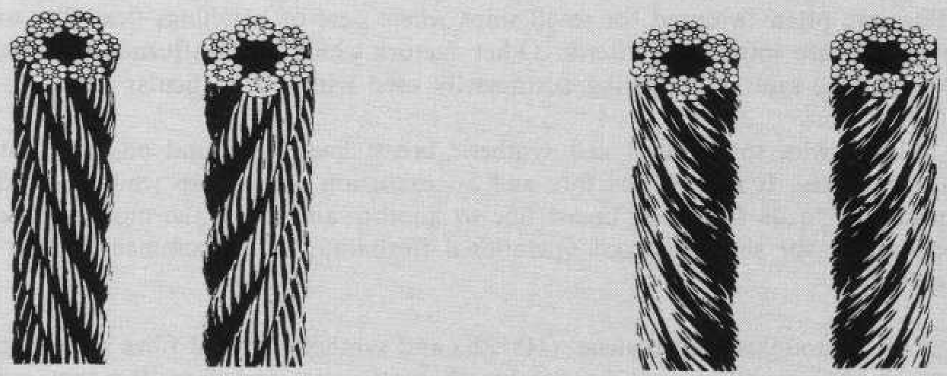
DEFINITIONS

Lay—the twisting of strands to form a rope or wires to form a strand during its manufacture

Crosslay or Equal Lay—terms describing the lay of the wires used to make up the strands. (See (*a*) + (*b*) above)

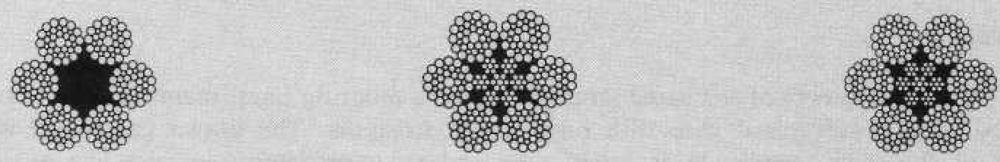
Righthand or Lefthand Lay—the angle or direction of the strands relative to the centre of the rope

Ordinary Lay or Lang's Lay—terms applying to the lay of the strands when making up the rope. (See (*c*) + (*d*) below)



(*c*) *Lefthand* *Righthand*
Ordinary Lay

(*d*) *Lefthand* *Righthand*
Lang's Lay



(*e*) 6 × 36 WS
(1 + 7 + (7 + 7) + 14)
+ Fibre core

(*f*) 6 × 36 WS
(1 + 7 + (7 + 7) + 14)
+ Steel core

(*g*) 6 × 41 WS
(1 + 8 + (8 + 8) + 16)
+ Steel core

This applies to the number of wires in each strand

1 = centre
7 = next layer
7 + 7 = layer with mixed wire diameters
14 = outer layer

FIGURE 6.1: WIRE LINE CONSTRUCTIONS

Steel wire lines with an independent wire rope core (IWRC) are strongly recommended over fibre core steel wire lines for several reasons. An IWRC steel wire line has a much greater resistance to crushing, higher MBL for a given diameter and greater strength retention when bent.

For mooring VLCCs, it is recommended that as a minimum a 42 mm ($1\frac{5}{8}''$) diameter 6×37 class IWRC preformed, heavily drawn galvanized wire line (minimum tensile strength of 180 kg/mm^2) with a typical MBL of 115 mt (1128 kN) be specified.

6.2.3 *Corrosion Protection*

Corrosion protection can be provided by galvanising of individual wires. So-called 'commercial galvanising' is inadequate for marine application. 'Heavy' galvanising should be carried out in accordance with JIS 3525 or DIN E 1548. These standards specify the zinc weight per wire surface as a function of wire diameter. The zinc weight ranges from about 100 g/m for a 1.0 mm^2 diameter wire to 220g/m^2 for a 2.5mm diameter or greater wire. The DIN E 1548 values are slightly higher for normalised wires.

6.2.4 *Bend Radius*

Wire ropes will lose strength when bent over a radius. This is a major factor in the design of shipboard equipment for wire rope, since items such as winch drums and fairleads must have an adequate diameter or surface radius. The recommended minimum values listed in sections 7 and 8 are based on lines with the recommended independent wire rope core. A fibre core rope will lose more strength at a given bend ratio than an IWRC rope. This is clearly shown in Fig. 6.2.

As a general rule, a minimum bend ratio of 12 is recommended. Where this would create problems with the size of the fitting, a ratio of 10 is an acceptable compromise for items such as universal roller fairleads.

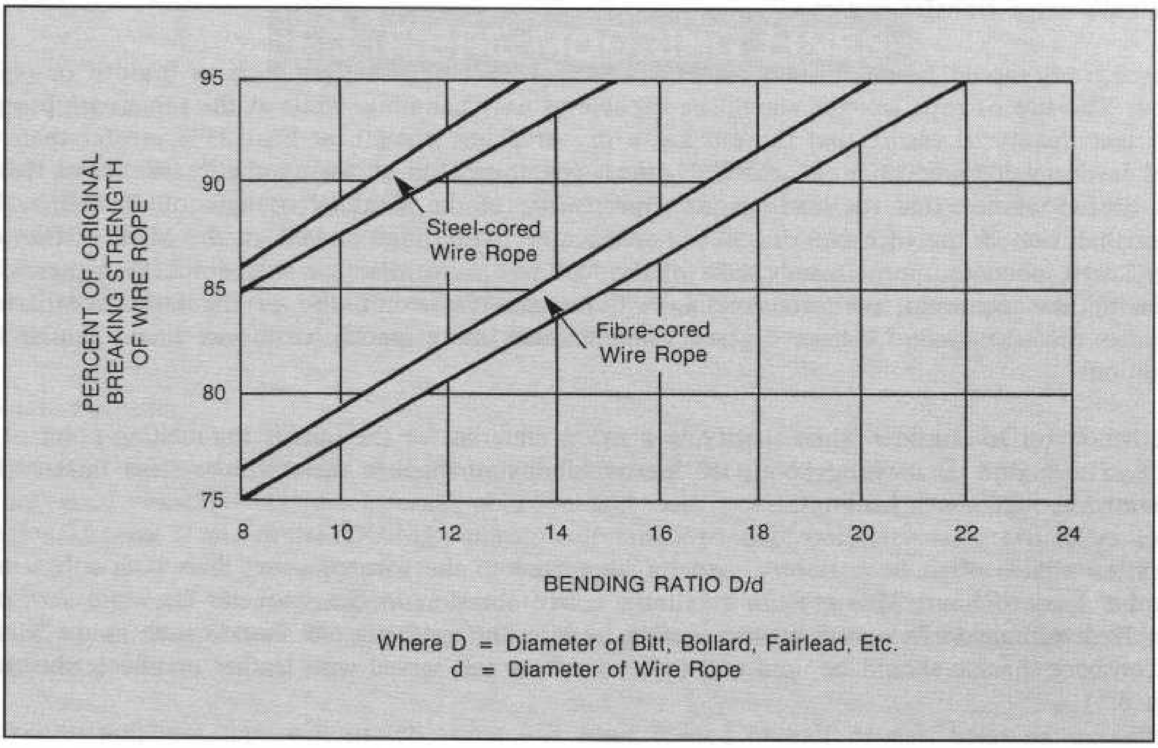


FIGURE 6.2: EFFECTS OF BENDING ON WIRE ROPE STRENGTH

6.2.5 *Maintenance, Handling and Inspection*

These subjects are dealt with in Appendix B.

6.2.6 *Standard Specifications*

Wire ropes are covered by many national standards. The following standard specifications concern wire ropes that comply with the material and construction recommended in paragraphs 6.2.1 and 6.2.2

- American Federal Specification (FS) RR-W-410-A: 6 × 37 with IWRC special improved plow steel.

- DIN 3066-SE-1770; equal lay, right hand, with heavy galvanising per DIN E 1548.

- BS 3021 987.

- ISO 4344.

6.2.7 *Use of Synthetic Tails*

In order to provide additional elasticity, the wire mooring lines of some large tankers are fitted at the shore end with a length of synthetic rope, or tail. This additional elasticity reduces the dynamic loads induced in the wire mooring lines by allowing the ship to respond more closely to various combinations of wind, wave and current, as well as to ships passing nearby at low speeds. Tails also tend to distribute the loadings more evenly among mooring lines in the same service. Finally, the elongation of the mooring line system furnished by the tail serves to reduce the risk potential associated with poor line tending by lessening the frequency and precision of line tending requirements, particularly in berths with large tidal variations and/or high cargo handling rates.

The main disadvantage of tails is that they may introduce a weak link into the moorings which is not readily apparent to the ship operator. Tests conducted on failures have revealed that tails can undergo a substantial reduction in breaking strength in a relatively short period of time. Additionally, if the tail is too elastic the ship movement may be in excess of that which can be tolerated by the terminal's cargo transfer system.

If used, tails should be made of a material with high breaking strength such as braided or plaited nylon. The size of rope selected should be capable of easy handling, while at the same time being of sufficient quality to ensure that the tail has a dry breaking strength at least 25% greater than that of the wire line to which it is attached. Increasing the MBL of the synthetic tail above that of the wire will ensure that the loadings, as a percentage of the breaking strength of the tail, will be lower than that in the wire mooring line. For example, if the load is 55% of the MBL of the wire, it will correspond to approximately 45% of the MBL of a synthetic tail having a 25% higher MBL. Reducing the maximum per cent loading will increase the useful life of the tail, as experience indicates that the cyclic loadings degrade synthetic lines more quickly than wire under similar load conditions.

Another factor to consider when specifying a nylon material for the tails is the melting point of the grade. The higher the melting point, the less possibility of damage there will be from internal heat generated at high cyclic loadings.

Nylon tails have often been observed to be connected to the wire mooring lines with a cow hitch or other type of knot. This practise results in severe abrasion to the synthetic line, and should be actively discouraged. To guard against chafing, a wire line synthetic tail shackle such as the Mandel or Tonsberg shackle should be used and the eyes of the tail served with leather or plastic sheathing. (Fig. 6.3)

In summary, tails, if used, should preferably be made of nylon line (not three strand construction), be about 11 metres long overall and have a dry breaking strength at least 25% greater than the MBL of the wire to which they are attached. Tails should be replaced at least every 18 months unless experience and/or inspection indicates a longer or shorter period is warranted.

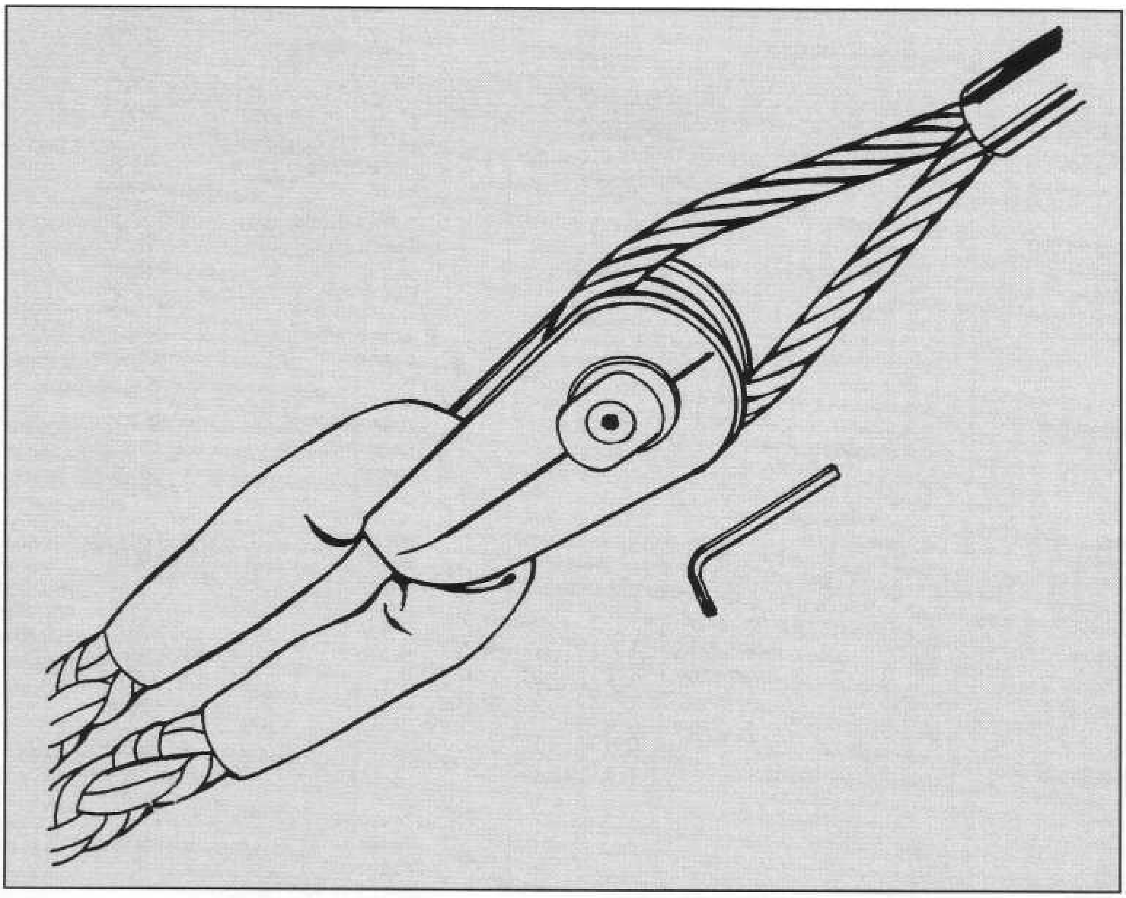


FIGURE 6.3: STAINLESS STEEL SHACKLE FOR LINES WITH TAILS

6.3 FIBRE MOORING LINES

6.3.1 *Material*

Today practically all fibre mooring lines are made from synthetics, the most common materials being polyester, nylon, polypropylene and polyethylene. Some ropes are made of combinations of these materials. Table 6.1 details materials used in making synthetic ropes, giving the construction in which these materials are normally used and the general characteristics of each. Table 6.2 states typical strengths for ropes of different materials. Table 6.3 summarises the recommended application for various ropes. Figure 6.4 shows typical elongation values for various rope materials.

6.3.1.1 *Polyester*

Polyester is the most durable of the common materials. It has high strength, both wet and dry. It has good resistance against external abrasion and does not lose strength rapidly due to cyclic loading. Recent Tests under OCIMF sponsorship indicated that polyester lasts 190 times longer than nylon and 570 times longer than polypropylene when subjected to cyclic loads. Although polyester ropes are the most expensive (about three times that of polypropylene and about 1.2 times that of nylon on an equal strength basis), the initial outlay is more than compensated by their much longer service life.

Broken-in polyester rope will stretch about 9% when loaded to half its new breaking strength. In comparison, wire rope stretches about 1% at half its breaking strength.

Polyester's low coefficient of friction allows it to slide easily around bitts. Its relatively high melting point (230°C/450°F as against 165°C/330°F for polypropylene) reduces the chances of fusion. Polyester is therefore preferred for large and small rope material where strength and durability are important and where high elasticity is not required.

Material	Construction	Features	Deficiencies	Comments
Polyester ("Dacron") ("Terylene")	3 strand 8 strand Double braid	High dry & wet strength Moderate cost Moderate Stretch		
Nylon (Polyamide)	3 strand 8 strand Double braid	High dry strength Moderate cost High stretch	Low wet strength Low fatigue life	Wet strength is about 80% of dry strength.
Polypropylene	3 strand 8 strand	Light weight Low cost Moderate stretch	Low strength Low Melt Point Creep	Special medium strength polypropylene ropes are available.
Polyethelene	3 strand 8 strand	Light weight Low cost Moderate stretch	Low Strength Low Melt Point Creep	Special medium strength polyethylene ropes are available.
Aramid ("Kevlar") ("Twarlon")	Varied	Very Low Stretch High strength to weight ratio	High Cost Low abrasion resistance	D/d bend ratio similar to that of wire. Tight bend radii do not allow migration of load and can cause damage to the fibres.
HMPE ("Dyneema")	Varied	High strength/weight ratio		Better flexibility characteristics than Kevlar, sometimes used as a replacement for wire rope.
Nylon/Polyester/ Polypropylene ("Jetkore")	6 strand	High dry strength Good fatigue life Good abrasion resistance Moderate cost High stretch	Low wet strength Hockles	
Nylon Mono- & Multi-Filament Fibre Mixture ("Atlas Perlen")	6 strand 8 strand	High dry strength Good abrasion resistance Moderate stretch	Low wet strength Low fatigue life	
Polyester/ Polypropylene Fibre Mixture ("Deltaflex")	6 strand 8 strand	Moderate to high dry & wet strength Moderate cost Moderate stretch		
Polyester/ Polypropylene Melt Mixture ("Karat")	3 strand 8 strand	Moderate to high dry & wet strength Moderate cost Moderate stretch		

TABLE 6.1: SYNTHETIC ROPE CHARACTERISTICS

6.3.1.2 *Nylon*

Dry nylon rope is slightly stronger than polyester rope. However, moisture reduces the strength of nylon fibre, and wet nylon rope has about the same strength as wet polyester. Wet nylon rope loses strength much faster under cyclic loading than polyester. Thus a heavily used nylon rope will become weaker than a heavily used polyester rope of the same size. For this reason, if nylon is used for tails, it should have at least 37% more strength than the associated wire rope. This rule is derived from the general 25% recommendation for tails plus an additional 10% allowance for reduction in wet strength.

Nylon is more elastic than any other material. When broken in it will stretch 12% or more at 50% MBL. This characteristic makes it especially suitable for use as tails with wire hawsers where elasticity is required, such as for STS operations and at berths subject to wave motion or surge. It is less suited to operations where rigid moorings are required, such as a large ship in a loading arm envelope at a poor weather berth.

Rope Size No. (See Note 2)	Circ mm. (See Note 2)	Diameter (See Note 2) mm.	in.	Polypropylene tonnes	Nylon (Polyamide) (See Note 3) tonnes	Polyester tonnes
1	24	8	0.33	0.9	1.2	0.9
1.25	30	10	0.4	1.3	1.9	1.4
1.5	36	12	0.5	1.8	2.7	2.0
1.75	42	14	0.6	2.5	3.7	2.9
2	48	16	0.67	3.1	4.8	3.7
2.25	54	18	0.7	4.0	6.1	4.6
2.5	60	20	0.8	4.8	7.4	5.7
2.75	66	22	0.9	5.8	9.0	6.9
3	72	24	1.0	6.7	11	8.2
3.5	84	28	1.1	9.1	14	11
4	96	32	1.3	11	18	14
4.5	108	36	1.5	15	22	17
5	120	40	1.7	17	27	22
5.5	132	44	1.8	21	32	26
6	144	48	1.9	25	38	30
6.5	156	52	2.0	28	44	35
7	168	56	2.2	32	50	40
7.5	180	60	2.5	37	57	45
8	192	64	2.7	42	65	52
9	216	72	3.0	53	81	65
10	240	80	3.3	65	99	80
11	264	88	3.7	78	118	95
12	288	96	4.0	92	139	113

Notes:

1) Spliced strength is approximately 90% of unspliced strength. Based on British Standard breaking strengths.

2) Rope size number is the nominal circumference of the rope in inches. All circumferences and diameters are approximate, as determined by common rope industry practices.

3) Wet nylon strength is about 80% of dry strength. Accordingly, the strengths of nylon provided in this table should be reduced for direct comparison with polypropylene and polyester.

TABLE 6.2: TYPICAL STRENGTHS OF SYNTHETIC 3 STRAND AND 8 STRAND ROPES (New, Dry Rope With Splices)

6.3.1.3 *Polypropylene*

Polypropylene rope has approximately the same elasticity as polyester rope, but is significantly weaker than either polyester or nylon. Polypropylene has a low melting point and tends to fuse under high friction. It also has poor cyclic loading characteristics. Lastly, prolonged exposure to the sun's ultraviolet rays can cause polypropylene fibres to disintegrate due to actinic degradation.

Polypropylene is lighter than water and can be used for floating messenger lines. Otherwise, the use of polypropylene for moorings is not recommended.

6.3.1.4 *Polyethylene*

Polyethylene is similar to polypropylene in appearance, but is generally weaker and is less resistant to abrasion. It has about the same elasticity as polypropylene. Polyethylene rope is acceptable for floating messenger lines provided they are constructed from top quality fibres.

Service	Approximate Load Range in tonnes (See Note 1)	Other Requirements	Recommended Rope	
Winch-Mounted Mooring Lines	20-70	Moderate stretch Abrasion resistant	Polyester Combination	8 Strand 6 Strand
Auxiliary Mooring Lines	20-50	Moderate stretch Abrasion resistant	Polyester	8 Strand
Tails (on wire rope)	20-70	High stretch	Nylon Combination	Double Braid 6 Strand
Heaving Lines	0.05	Light weight Very easy to handle	Polyester Aramid	8 Strand Various
Messenger Lines	2	Light weight Easy to handle (Float on water)	Polyester Polypropylene	8 Strand 8 Strand
Pick-Up Lines	2	Light weight Low stretch Abrasion resistant	Polyester	8 Strand
Stoppers and Snubbers	5	Easy to handle Abrasion resistant	Polyester Staple-fibre	8 Strand
Lifting Lines	5	Low stretch	Polyester	8 Strand

Note:
1) Loads are approximate working or applied loads and appropriate safety factors must be applied to these when purchasing ropes.

TABLE 6.3: SYNTHETIC LINE SERVICE REQUIREMENTS AND RECOMMENDATIONS

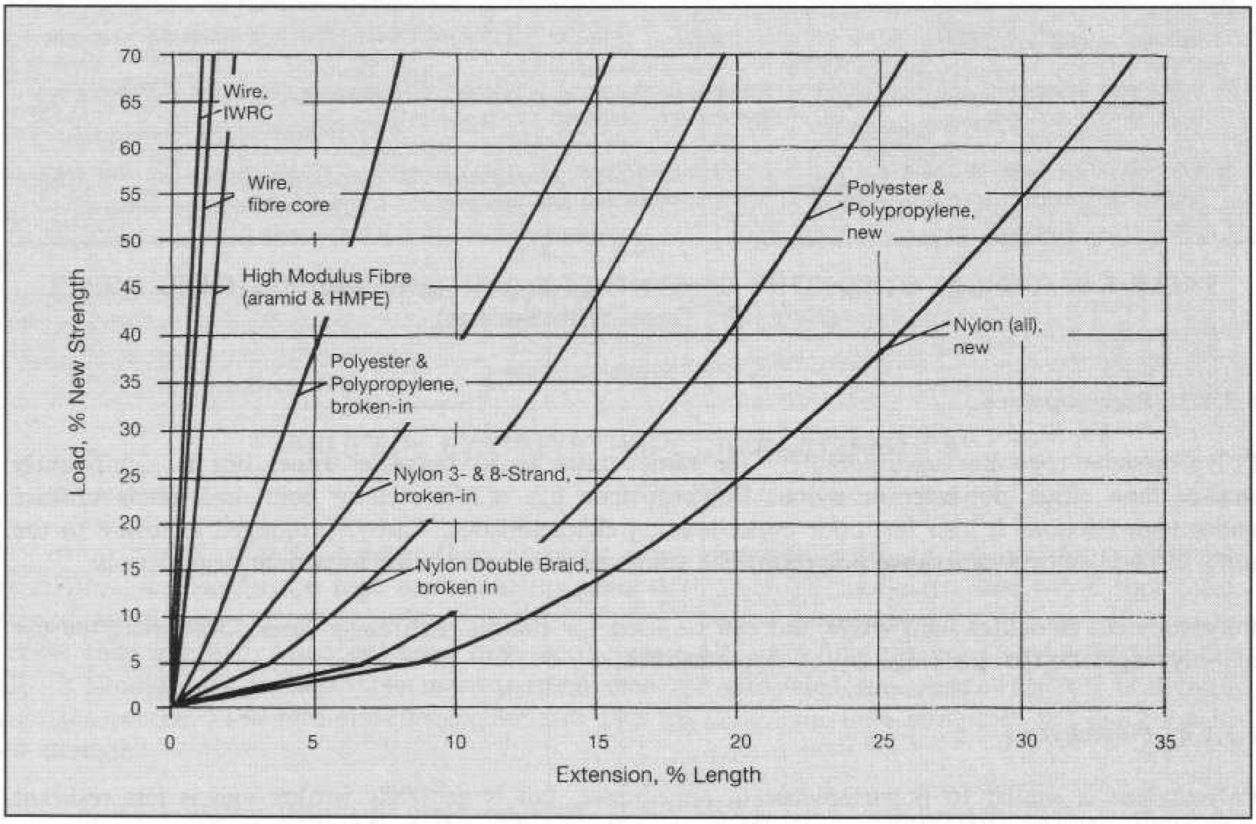


FIGURE 6.4: LOAD-EXTENSION CHARACTERISTICS
Wire and Fibre Ropes, New and Broken-In
Courtesy of Tension Technology International
(Reference 9)

6.3.1.5 *Aramid and Other Materials*

A number of the newer synthetic materials for ropes, such as aramid fibre offers high strength, low weight, low stretch, flexibility and corrosion resistance. Weight for weight aramid fibres are about five times as strong as steel, while elasticity is only about 3% at 50% breaking load.

Despite the relative cost of aramid fibres their use as substitute for other synthetics has already been established, and high modulus polyethylene such as Dyneema has also been installed as replacement for mooring wires.

With regard to aramid fibres the main benefits are in reduction of diameter and therefore weight of line required. The reduction in line size is advantageous if winch drum size is limited, whereas the reduction in weight makes handling substantially easier.

Besides the cost, there are other drawbacks to the use of synthetic fibres. In the case of aramid fibres there are restrictions with regard to bend radius and in the majority of cases there is a need to sheath the fibres to guard against ultraviolet rays and mechanical wear. Damage to the outer sheath can result in rapid degradation of the core fibres, and may result in difficulty to retrofit using existing fittings.

6.3.1.6 *Combinations of Materials*

The following are examples of proprietary ropes manufactured by combining different materials:

Jetkore rope is a six-strand construction comprised of nylon, polyester and polypropylene. Since nylon is the principal strength-carrying element in Jetkore rope, it has essentially the same strength and elasticity characteristics as conventional nylon rope. The unique combination of polyester and polypropylene on the strand surface provides very good resistance to abrasion. Jetkore rope does not suffer significant strength reduction from cyclic loading. It is therefore a suitable alternative for tails on wires where high elasticity is required. It is also used successfully for winch-mounted lines on some ships where motions can be tolerated.

Atlas Perlon rope is a six-strand construction made of both monofilament and multifilament nylon yarns. The monofilaments are almost as large in diameter as a pencil and alternate with multifilament yarns as the outer yarns on the strands. Abrasion resistance is good, but cyclic load performance may be poor. It is used for winch-mounted lines on some smaller ships.

Polyester/Polypropylene Several manufacturers make rope comprised of mixtures of polyester and polypropylene fibres. These ropes are less expensive than pure polyester ropes. Their strength lies somewhere between corresponding ropes made of only polyester and only polypropylene. Depending on how the fibres are arranged in the yarns, abrasion resistance and cyclic load performance can be almost as good as for pure polyester. The better quality polyester/polypropylene mixed fibre ropes are a possible lower cost alternative to polyester ropes where slightly lower strength for a given size rope is acceptable.

Karat rope is made of fibres which are a melt mixture of polyester and polypropylene. It will float on salt water. Karat rope is significantly stronger than polypropylene rope, though some of this extra strength is achieved because the strands are not as tightly twisted. For this reason, Karat rope may not be as resistant to some forms of abrasion and other handling abuses.

6.3.2 *Construction*

Figure 6.5 shows the common structures used in synthetic ropes; three-strand. six-strand, eight-strand and double braid.

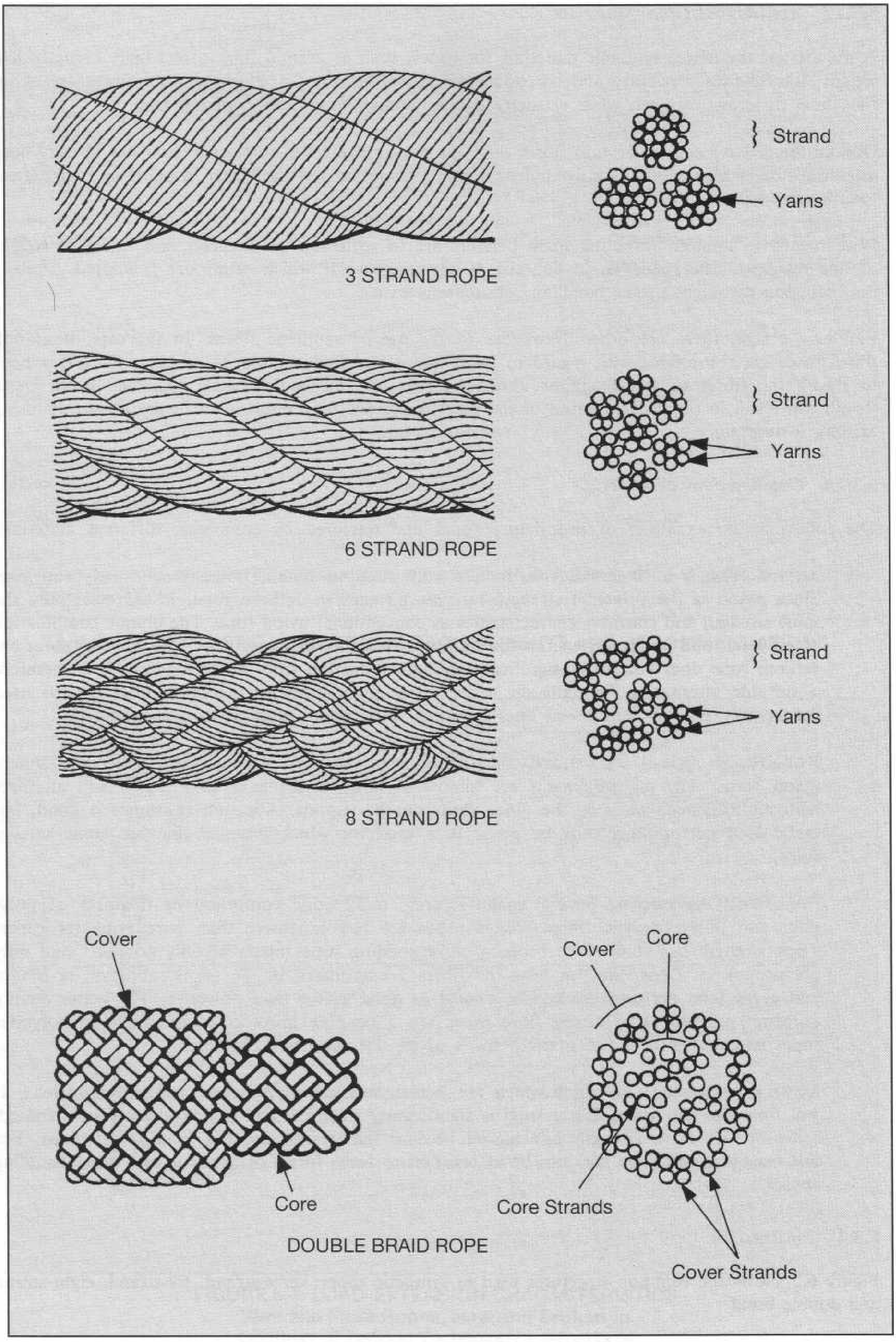


FIGURE 6.5: SYNTHETIC ROPE STRUCTURES

The three-strand rope is the most common form of twisted rope. It is adequate for some tasks, but is prone to hockling (see figure 6.7). This significantly reduces strength, making it a bad choice for use as a mooring rope.

The six-strand with core structure, used in Jetkore and Atlas ropes, is a twisted rope similar to conventional wire rope. It is not as prone to hockling as three-strand rope and is sometimes used for mooring lines.

The eight-strand rope, sometimes called square braid or plaited, is constructed of braided pairs of strands. It has essentially the same strength as a three-strand rope of the same size. It does not hockle and is more durable than twisted rope. It is a good rope structure for mooring lines and for most other purposes on ships.

Double braid rope, sometimes called braid-on-braid, is constructed of a core braided of many small strands and surrounded by a cover which is also braided of many small strands. It is generally stronger than other ropes of the same diameter because the structure is denser. It is commonly used for mooring hawsers at single point moorings (SPMs) and for tails on wire ropes.

6.3.3 *Bend Radius*

The strength loss due to bending is apparently not as critical for conventional fibre lines as for wire given their ability to allow migration of load. Newer types of synthetic fibres such as the aramid types are more bend radius sensitive. As for all fibres the strength loss and durability factors may depend upon the specific material and construction. The rope manufacturers guidelines should be consulted for each specific application.

6.3.4 *Identification*

If the composition of a mooring rope is not known, certain characteristics of appearance, burn properties and density may help to identify the material.

Fibre appearance
Polypropylene fibres are usually coarse and thicker than a human hair and are normally flat instead of round. Polypropylene is often dyed black to provide resistance to sunlight. Sometimes other dye colours, such as yellow or orange, are used.

Thus a coloured or black rope with coarse flat fibres is probably polypropylene. The only exceptions to this are polyethylene and Karat, which can have a similar appearance.

Since polyethylene ropes may not be quite as strong as polypropylene, a burn test may be necessary to distinguish between the two for critical applications where there is a possibility that polyethylene ropes have been furnished.

Karat rope fibres are similar in appearance to polypropylene. They are straw coloured, as opposed to the distinctive yellow colour of some dyed polypropylene fibres.

Nylon and polyester fibres are almost always very thin and fine, distinguishing them from polypropylene. However, it is virtually impossible to distinguish between nylon and polyester by appearance. New nylon fibres have a bluish tinge, but even experts cannot always tell them apart, especially if the rope has been used. The only reliable way is by a burn test.

Aramid ropes are normally covered by a braid strand or extruded plastic jacket. The aramid fibres are very fine and straw coloured.

Density
Polypropylene and polyethylene are lighter than water and can readily be distinguished from polyester and nylon by their density. Place several fibres or yarns in a glass or dish of water and stir, if necessary, to remove any air bubbles adhering to the sample.

If the sample floats, it is polypropylene or polyethylene. If it sinks, it is polyester or nylon.

<table>
<tr><th></th><th>Nylon 6 and 6.6</th><th>Polyester</th><th>Polypropylene</th><th>Polyethelene</th></tr>
<tr><td></td><td colspan="2">Melts and burns</td><td></td><td></td></tr>
<tr><td>In Flame</td><td>Blue flame with yellow tip</td><td>Yellow flame with blue edges</td><td>Shrinks and melts</td><td>Shrinks, curls, melts and drips</td></tr>
<tr><td></td><td>White smoke</td><td>Blackish smoke</td><td colspan="2">Blue flame with yellow tip</td></tr>
<tr><td></td><td>Yellowish melted drops falling down</td><td colspan="3">Melted drops falling down</td></tr>
<tr><td>After Removed From Flame</td><td colspan="2">Stops burning</td><td>Continues to burn rapidly</td><td>Continues to burn slowly</td></tr>
<tr><td></td><td>Small bead on end</td><td>Small Black bead on end</td><td></td><td></td></tr>
<tr><td></td><td colspan="2">Hot melting bead may be stretched into fine thread</td><td>Hot melted substance may be stretched into fine thread</td><td>Hot melted substance cannot be stretched</td></tr>
<tr><td>Residue</td><td>Hard round Yellow bead</td><td>Hard round Blackish bead</td><td>Hard round brown to blackish</td><td>No melting bead Like paraffin</td></tr>
<tr><td></td><td colspan="3">Not crushable</td><td></td></tr>
<tr><td>Smell of Smoke*</td><td>Celery-like fish odour, burnt wool, or hair</td><td>Oily sooty odour Faintly sweet Similar to sealing wax or burning rubber</td><td>Similar to diesel fumes, burning asphalt, wax or paraffin</td><td>Similar to burning paraffin</td></tr>
</table>

* The sense of smell is subjective and should be used only with reservation. The smell may be altered by agents in or on the fibre.

Adopted from Table 3, Netting Materials for Fishing Gear by Gerhard Klust, published by Fishing News Books, Surrey, England, 1973, for United Nations Food and Agriculture Organization (FAO).

TABLE 6.4: BURN TEST CHARACTERISTICS OF SYNTHETIC FIBRES

Burn test
Because this test requires the use of an open flame, it should only be carried out in designated safe areas.

Take a specimen of fibre from the rope. A short length of small yarn is sufficient. On a used rope, the sample should be taken from a portion which is not soiled or oily. Take care not to damage the rope.

Table 6.4 summarises the burning characteristics to be observed.

In a designated safe area, hold the fibre sample over a clean flame, such as that from a gas cigarette lighter. Take care not to allow hot residue from the burning or burned fibres to fall on your skin, clothing or other vulnerable objects.

The sample should then be removed from the flame. Observe the reaction of the specimen and the nature of the smoke given off. While the smell of the smoke may be an indicator, this property alone should not be used for identification because the sense of smell varies between individuals. Taking care not to inhale more fumes than necessary, extinguish any flame on the sample by blowing. After allowing to cool, the residue can be examined.

6.3.5 *Handling, Maintenance and Inspection*

A summary of recommendations is provided in Appendix C. Since synthetic ropes are common on smaller ships, a more detailed discussion concerning them follows below:

Safety hazards
Synthetic lines can pose a great danger to personnel if not properly used. Handling of mooring lines has a higher potential accident risk than most other shipboard activities.

The most serious danger is snap-back, the sudden release of the static energy stored in the stretched synthetic line when it breaks.

When a line is loaded, it stretches. Energy is stored in the line in proportion to the load and the stretch. When the line breaks, this energy is suddenly released. The ends of the line snap back, striking anything in their path with tremendous force.

Snap-back is common to all lines. Even long wire lines under tension can stretch enough to snap back with considerable energy. Synthetic lines are much more elastic, increasing the danger of snap-back.

Synthetic lines normally break suddenly and without warning. Unlike wires, they do not give audible signals of pending failure; nor do they exhibit a few visible broken elements before completely parting.

Line handlers must stand well clear of the potential path of snap-back, which extends to the sides of and far beyond the ends of the tensioned line. Figure 6.6 illustrates potential snap-back danger zones.

As a general rule, any point within about a 10 degree cone around the line from any point at which the line may break is in danger. A broken line will snap back beyond the point at which it is secured, possibly to a distance almost as far as its own length. If the line passes around a fairlead, then its snap-back path may not follow the original path of the line. When it breaks behind the fairlead, the end of the line will fly around and beyond the fairlead.

If an activity in a danger zone cannot be avoided, the exposure time can at least be reduced by observing some simple rules. When it is necessary to pass near a line under tension, do so as quickly as possible. If it is a mooring line and the ship is moving about, time your passage for the period during which the line is under little or no tension. If possible, do not stand or pass near the line while the line is being tensioned or while the ship is being moved along

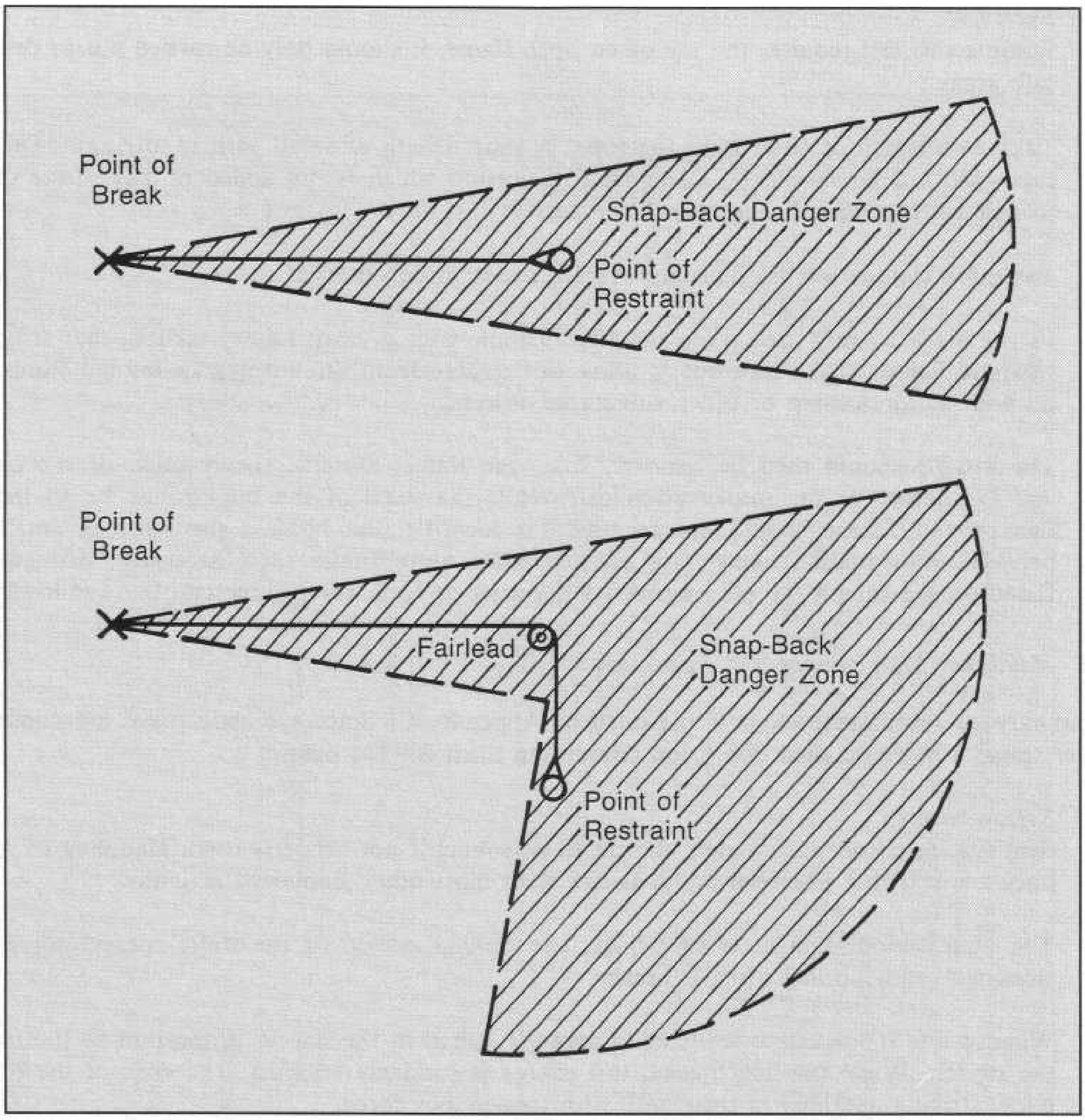


FIGURE 6.6: EXAMPLES OF SNAP-BACK DANGER ZONES

the pier. If you must work near a line under tension, do so quickly and leave the danger zone as soon as possible. Plan your activity before you approach the line. Never have more people than necessary near the line. If the activity involves line handling, make certain that there are enough personnel to perform it in an expedient and safe manner. Instruct observers to stand well clear.

Handling

Crews handling synthetic lines which must be stopped off and made fast to bitts need good training in accident prevention. Surging of lines on winch drum heads, which used to be common practice with natural fibre rope, is not recommended for synthetic lines, polypropylene in particular. The nature of the fibres, combined with the high loads, make it necessary to walk back the winches rather than surge in order to provide slack when stopping off and making fast. Stoppers made of polyester are recommended. They should be used in the double line configurations, where a half hitch is placed over the bitts and the two ends of the stopper are crossed over and under the line being stopped off. Training should include action to be taken during a break-out incident, namely, clearing the area to prevent injuries.

When holding and tensioning the line on the warping drum end, capstan or bitt, the line handler must not stand too close. When the line surges, he could be drawn into the drum or bitt before he can safely take another hold or let go. He should stand back and grasp the line about 1 m (3 ft) from the drum or bitt.

Synthetic lines are not very resistant to cuts and abrasion, and should not be exposed to conditions which might damage them. If they are used in fairleads previously used with wires, make certain the fairleads have not become grooved or roughened by the wires. It may be necessary to grind the fairleads smooth.

Care should be taken when dragging synthetic lines along a deck. Avoid sharp edges and rough surfaces. Small lines should be carried instead of dragged when possible.

When dirt, grit or rust particles are allowed to cling to and penetrate into synthetic ropes, internal abrasion will result. The rope should be brushed or cleaned before storing.

Twisted ropes can be harmed by kinking, which may form into hockles if not properly removed. When a kink forms, the load must be removed and the kink gently worked out.

Twisted rope must be coiled in the proper direction. Most lines are right-hand lay and should be coiled clockwise. When removing new rope from a coil, suspend the coil on a shaft and rotate it.

Winch-mounted synthetic lines should be end-for-ended after about two years to distribute wear, unless inspection dictates a shorter schedule.

Storage

Synthetic lines should be stored in clean, cool, dry surroundings. Excessive heat can damage synthetic fibres, especially polypropylene and polyethylene. Do not store synthetic ropes near steam pipes or against bulkheads which may reach high temperatures.

Ultraviolet rays from sunshine can damage fibres. Polypropylene and polyethylene are especially vulnerable. The potential degree of damage increases as rope size decreases. Never store small polypropylene or polyethylene ropes in direct sunlight.

Synthetic fibres are also subject to chemical damage. Their susceptibility depends on the chemical and the fibre. Nylon is attacked by acids and bleaching agents. Polyester is attacked by some alkalis. Industrial solvents, including paint thinners, will damage most synthetic lines if they are stored in paint lockers or near paints and paint fumes.

Oil and petroleum products will not normally damage synthetic fibres. Nonetheless, care should be taken to avoid contact with them. If a rope becomes oily, it is more difficult to handle. Dirt and grit will adhere to the oil and cause internal abrasion of the rope. If the line becomes oily or greasy, it should be scrubbed with fresh water and a paste-like mixture of granulated soap. For heavy accumulations of oil and grease scrub the line with a solvent such as mineral spirits; then rinse it with a solution of soap and fresh water.

Inspection and replacement

Synthetic lines should be examined frequently while in service. They should be checked for obvious signs of deterioration before each use and undergo a thorough inspection at least once each year.

Some signs of damage such as hockling, cuts, surface abrasion and fusion are readily visible. Others are not as evident. While it is not possible to prescribe definitive retirement criteria, the following sections discuss the types of damage and wear experienced by ropes and provide general guidelines.

Cuts

The degree of damage caused by a cut depends on the depth and extent of the cut and on the rope construction. Each strand of a three-strand, six-strand or eight-strand rope carries a substantial portion of the load. If any one strand is significantly weakened by a cut, then the strength of the entire rope is significantly decreased. In general, any cut which penetrates through 25% of the area of one or more strands critically weakens the rope. The rope should be cut and spliced or retired.

Double braid ropes have many more strands. In conventional synthetic fibre double braid ropes the cover and the core each carry about 50% of the load. Thus, one or several cut strands in the cover normally do not significantly reduce the strength. If more than about 10% of the entire cover strands are cut, then the double braid rope should be retired.

In the case of the newer types of synthetic line such as the aramid fibres, almost the entire load is carried by the inner core. Therefore, should the external sheath be damaged the internal load bearing fibres may rapidly degrade through exposure to ultraviolet rays or through mechanical wear. It is consequently advisable to inspect these lines on a regular basis with a view to pre-emptive repair as necessary.

External abrasion and fusion

A moderate amount of external abrasion is normal and can be tolerated in most synthetic ropes. The abrasion is evident as a general fuzzy appearance. If abrasion reduces the solid diameter by more than about 5%, then the rope should be retired. If the abrasion is localised and the remainder of the rope is in good condition, then the rope may be respliced.

Severe localised abrasion may be of concern. Severe abrasion of even one strand in three-strand, six-strand or eight-strand rope can significantly reduce the strength of the strand and upset the rope structure. The abrasion affects a number of yarns as it extends along the strand, so the degree of damage is not necessarily proportional to the depth of abrasion. If the abrasion on any one strand penetrates more than about 15% of the strand area, the rope should be cut and spliced.

Internal abrasion

Internal abrasion is caused by the strands and yarns rubbing against each other as the rope undergoes cyclic loading. It is a form of fatigue entirely different from the type of fatigue experienced in metals.

The rope should be examined for signs of inter-strand abrasion. Carefully open the structure of three-strand, six-strand or eight-strand rope to examine the surfaces of the strands at points where they contact each other. A general fuzzy appearance at the points where strands rub against each other is an indication of moderate internal abrasion. If the abrasion has progressed to the extent that some yarns are worn through, the rope should be retired.

Internal abrasion in double braid rope is harder to detect because it may appear to be normal external abrasion. Closely examine the broken yarns which appear on the strands at the surface. If they have broken in the valleys between the strands, then it is internal abrasion. This internal abrasion probably extends throughout the entire rope structure. If it is severe, it has significantly decreased the rope strength and the rope should be retired.

Hockling

Hockling normally occurs only in twisted ropes. A hockle resembles a knot in the rope, as shown in Fig. 6.7. Hockles greatly reduce the strength of the rope. When a hockle appears in a rope which is otherwise in good condition, it should be cut out and the rope spliced.

Hockles occasionally occur in the individual strands of three-strand, six-strand and eight-strand ropes. Such hockles upset the balance of load carried by the strands. The rope should be cut and spliced.

Broken Core

The core of a double braid rope may break under high load without resulting in immediate rope failure. Under load, the rope will have a smaller diameter at the point of core break. Under no load, the rope may bend more freely at this point. If the core is broken, the double braid rope should be retired.

Ultraviolet Damage

Ultraviolet rays from the sun destroy the strength of polypropylene and polyethylene fibres. The weakened fibres can easily be rubbed off the surface of the rope. The significance of the

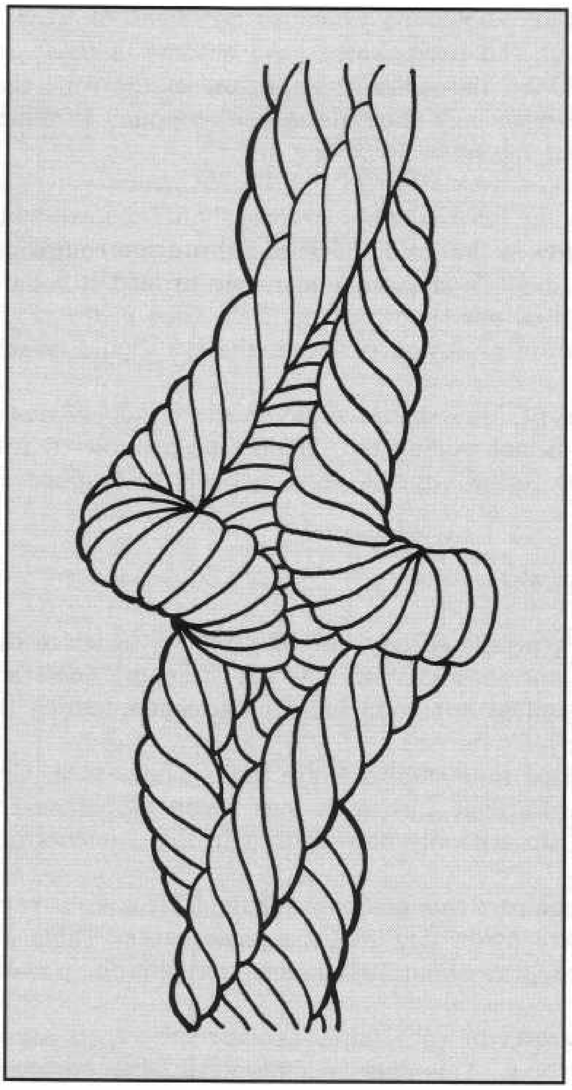


FIGURE 6.7: EXAMPLE OF HOCKLE IN 3 STRAND ROPE

damage depends on the size of the rope. Small ropes, less than about size 3 (24 mm diameter), should be retired when there is evidence of significant ultraviolet damage. Larger ropes are normally not as seriously damaged because only the yarns on the surface of the rope are affected.

Aramid fibres are also vulnerable to ultraviolet rays. They are normally covered by jackets of another material for protection. If the jacket is damaged and the aramid fibres exposed to sunlight, deterioration may follow. If the jacketing of an aramid rope is damaged, it should be repaired in accordance with the manufacturer's instructions.

Chemical damage
Some synthetic rope materials can be damaged by chemical attack. Nylon is affected by acids. The products of rust can be acidic and weaken nylon. If a nylon rope has been significantly discoloured by rust beyond the surface yarns, the affected portion should be cut out or the entire rope should be discarded.

Paints, paint thinners and even the fumes of paints and thinners can damage some synthetic fibres. Ropes should not be stored in the same room as paints and thinners. If a rope has become soaked in paint or thinners, it should be discarded.

Splices
The proper techniques for splicing common types of synthetic rope are described in seamanship manuals and manufacturers' literature and are not covered here. Ropes should be spliced by experienced personnel with reference to the applicable splicing instructions.

Splices in used ropes should be examined for signs of wear. Look for strands which have slipped in the splice and tucks which have become undone, as these upset the load balance. The transition between the splice and the rest of the rope should be examined for signs of internal abrasion which may concentrate at this point. In double braid rope splices, look for any indications that the splice is pulling apart.

Eyes
Abrasion and fusion at the inside back of the eye are common problems. Wear occurs at this point as the rope angle is changed under load around a bollard or hook. On eyes which are protected by thimbles, one should examine the rope in the mouth of the thimble for abrasion. If significant abrasion or fusion is found, the eye should be respliced.

In double braid rope, one should examine the crotch of the eye for broken strands. Make certain the splice is not pulling out. It may be possible to resplice small double braid rope. Large used double braid rope is very difficult to resplice and the rope may have to be replaced.

6.3.6 *Standard Specifications*

The available standards generally specify only the manner in which the rope is made, the manner in which it is to be tested and its minimum new dry strength. There are significant differences in the test methods and the specified minimum breaking strengths among the standards.

The most widely recognized rope standards are those published in Great Britain (British Standards), the United States (U.S. Cordage Institute), and Japan (Japanese Industrial Standards). The U.S. and Japanese standards are generally only applied in the countries of publication.

Tables 6.5, 6.6 and 6.7 compare the minimum required breaking strengths specified by these standards for eight-strand polyester, nylon and polypropylene ropes. Table 6.5 also lists the German DIN Standard 83 331 which requires about 10% higher strengths for polyester ropes.

As indicated in the footnotes of each table, the strength values are not directly comparable due to differences in testing methods. This must be considered when comparing ropes produced to different standards.

Synthetic rope manufacturing is subject to continuous development and improvement. Some manufacturers are reluctant to produce higher strength ropes because they are more expensive: they find it difficult to market stronger ropes at increased prices when there are no standards which recognize the higher specification. Nevertheless, some companies do now produce and market ropes, especially those made from polyester, that are significantly stronger than the 'standard' product. These ropes should be considered for mooring applications since they will be lighter than a standard rope at generally the same cost and strength.

Rope Size No. (See Note 1)	Circ. (See Note 1) mm	Diameter (See Note 1) mm	Diameter in.	U.S. Cordage Ins. (1985) tonnes	British Standard BS 4928 (See Note 2) tonnes	German Standard DIN 83 331 (See Note 2) tonnes	Japanese Standard JIS 2704 (See Note 2) tonnes
1	24	8	0.33	1.0	1.0	1.0	1.0
1.25	30	10	0.4	2.0	1.6	1.7	1.6
1.5	36	12	0.5	2.6	2.3	2.4	2.2
1.75	42	14	0.6	3.3	3.2	3.4	3.0
2	48	16	0.67	4.1	4.1	4.4	3.8
2.25	54	18	0.7	5.1	5.1	5.6	4.8
2.5	60	20	0.8	6.4	6.3	6.9	5.8
2.75	66	22	0.9	7.3	7.6	8.3	6.9
3	72	24	1.0	9.0	9.5	10.1	8.1
3.5	84	28	1.1	12.1	12.2	13.5	10.8
4	96	32	1.3	13.5	15.7	15.1	13.9
4.5	108	36	1.5	15.3	19.4	21.7	17.3
5	120	40	1.7	23.4	24.0	26.3	20.9
5.5	132	44	1.8	27.7	28.6	31.3	
6	144	48	1.9	32.7	33.6	36.7	
6.5	156	52	2.0	37.6	39.0	43.1	
7	168	56	2.2	43.8	44.9	49.4	
7.5	180	60	2.5	49.9	49.9	54.9	44.5
8	192	64	2.7	55.8	58.1	64.0	
9	216	72	3.0	71.2	72.1	79.4	
10	240	80	3.3	85.7	88.5	97.1	76.2
11	264	88	3.7	103.4	106.1	116.6	
12	288	96	4.0	122.5	125.2	137.9	

Notes:

1) Rope size number is the nominal circumference of the rope in inches. All circumferences and diameters are approximate, as determined by common rope industry practices.

2) British, German and Japanese Standards are for rope tested without splices. Spliced strength is approximately 90% of unspliced strength. New ISO Standard strengths are not yet determined.

TABLE 6.5: "STANDARD" STRENGTHS OF POLYESTER 8 STRAND ROPES

Rope Size No. (See Note 1)	Circ. (See Note 1) mm	Diameter (See Note 1) mm	in.	U.S. Cordage Ins. (1985) tonnes	British Standard BS 4928 (See Note 2) tonnes	Japanese Standard JIS 2704 (See Note 2) tonnes
1	24	8	0.33	1.0	0.5	1.2
1.25	30	10	0.4	2.0	2.1	1.8
1.5	36	12	0.5	2.6	3.0	2.8
1.75	42	14	0.5	3.3	4.1	3.7
2	48	16	0.67	4.3	5.3	4.8
2.25	54	18	0.7	5.8	6.7	5.9
2.5	60	20	0.8	6.9	8.3	7.2
2.75	66	22	0.9	8.2	10.0	8.6
3	72	24	1.0	10.3	12.0	10.2
3.5	84	28	1.1	13.5	15.8	13.5
4	96	32	1.3	17.6	20.0	17.2
4.5	108	36	1.5	21.7	24.9	21.6
5	120	40	1.7	26.5	29.9	26.3
5.5	132	44	1.8	31.8	35.8	
6	144	48	1.9	37.6	43.5	
6.5	156	52	2.0	43.3	49.0	
7	168	56	2.2	51.3	55.8	
7.5	180	60	2.5	57.2	64.0	55.8
8	192	64	2.7	66.2	72.1	
9	216	72	3.0	81.5	89.8	
10	240	80	3.3	102.5	109.8	95.3
11	264	88	3.7	122.5	131.1	
12	288	96	4.0	147.0	153.8	

Notes:

1) Rope size number is the nominal circumference of the rope in inches. All circumferences and diameters are approximate, as determined by common rope industry practices.

2) British and Japanese Standards are for rope tested without splices. Spliced strength is approximately 90% of unspliced strength. New ISO and DIN Standards are essentially equivalent to British Standard.

TABLE 6.6: "STANDARD" STRENGTHS OF NYLON 8 STRAND ROPES

Rope Size No. (See Note 1)	Circ. (See Note 1) mm	Diameter (See Note 1) mm	in.	U.S. Cordage Ins. (1985) tonnes	British Standard BS 4928 (See Note 2) tonnes	Japanese Standard JIS 2704 (See Note 2) tonnes
1	24	8	0.33	0.8	1.0	0.8
1.25	30	10	0.4	1.4	1.4	1.1
1.5	36	12	0.5	1.7	2.0	1.7
1.75	42	14	0.6	2.1	2.8	2.2
2	48	16	0.67	2.5	3.5	2.8
2.25	54	18	0.7	3.4	4.5	3.5
2.5	60	20	0.8	4.0	5.3	4.2
2.75	66	22	0.9	4.7	6.5	5.1
3	72	24	1.0	5.7	7.6	6.0
3.5	84	28	1.1	7.5	10.1	8.0
4	96	32	1.3	9.6	12.8	10.3
4.5	108	36	1.5	12.2	16.1	12.8
5	120	40	1.7	14.7	19.5	15.6
5.5	132	44	1.8	17.6	23.1	
6	144	48	1.9	21.2	27.2	
6.5	156	52	2.0	24.9	31.3	
7	168	56	2.2	28.1	35.8	
7.5	180	60	2.5	32.7	41.3	33.1
8	192	64	2.7	36.7	46.7	
9	216	72	3.0	46.7	58.5	
10	240	80	3.3	55.8	72.1	56.7
11	264	88	3.7	66.2	86.2	
12	288	96	4.0	77.6	102.1	

Notes:

1) Rope size number is the nominal circumference of the rope in inches. All circumferences and diameters are approximate, as determined by common rope industry practices.

2) British and Japanese Standards are for rope tested without splices. Spliced strength is approximately 90% of unspliced strength. New ISO and DIN Standards are essentially equivalent to British Standard.

TABLE 6.7: "STANDARD" STRENGTHS OF POLYPROPYLENE 8 STRAND ROPES

Section 7.0

Winch Performance, Brake Holding Capacity and Strength Requirements

7.1 FUNCTION AND TYPE OF MOORING WINCHES

Mooring winches perform a multitude of functions. They secure the shipboard end of mooring lines, provide for adjustment of the mooring line length to suit the mooring pattern in each port and compensate for changes in draft and tide. They serve to store the mooring line when not in use and to haul the ship into position against environmental or inertia forces. They also act as a safety device that releases the line load in a controlled manner once the force in the line increases to the point of near-breakage. General requirements for shipboard mooring winches are dealt with in ISO Standards 3730 and 7825.

Winches can be categorised by their control type (automatic or manual tensioning), drive type (steam, hydraulic or electric), by the number of drums associated with each drive (single drum, double drum, triple drum), by the type of drums (split, undivided) and by their brake type and brake application (band, disc, mechanical screw, spring applied). Each of these features influences the mooring winch function and will be briefly discussed below.

7.2 AUTOMATIC TENSION WINCHES VERSUS MANUAL WINCHES

Automatic tension winches are designed to automatically heave-in whenever the line tension falls below a certain pre-set value. Likewise, they will pay out if the line tension exceeds a pre-set value. Manual winches always require a person to handle the controls for heaving or rendering.

In practice, the use of the self-tension winch is not recommended except for moorings deployed at 90° to the ship's axis. Automatic tension winches should not be used for spring lines, for example, since it has been known for the winches to cause the ship to 'walk' along the pier. This is because spring line forces oppose each other, leaving only the difference between the heaving and rendering values of the winches available to resist longitudinal environmental forces.

In theory, automatic tension winches would be ideal for breast lines forming a 90° angle to the ship axis. If the breast lines cannot be arranged at the optimum 90° angle due to the location of mooring fittings at the pier and on the ship, the mooring restraint capability may be reduced because the lines' fore-and-aft components oppose each other. For this reason, most terminals do not allow the use of the automatic feature and require that the winch be placed on the manual brake while the ship is moored. See also Reference 6, Section 3.5.4.

7.3 WINCH DRUMS

Winch drums may be either split or undivided. The split drum is composed of a tension section and a line storage section. It has the advantage that it can maintain a constant brake holding capacity and heaving force, due to the fact that the mooring line is always run off the first layer of the tension drum. For this reason, split drum winches are preferred by some operators, especially for large ships. The disadvantage of the split drum is the more difficult operation, a factor which can be overcome with proper instructions and operator experience. Another reason for their use is that ISO Standard 3730, Annex A, recommends that synthetic ropes under tension should not be wound on a drum in more than one layer or short life will result, and this can normally only be avoided by using split drums.

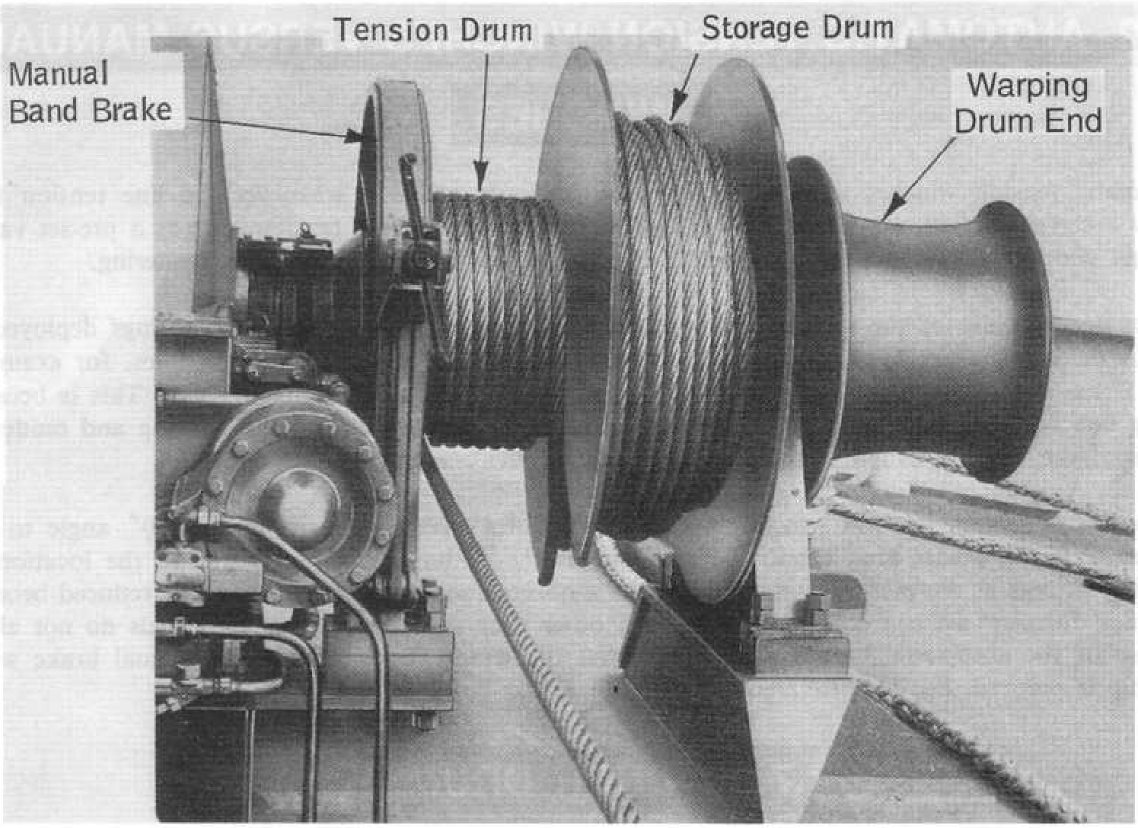
For either type of drum, the minimum drum diameter should be 16 times the wire rope diameter. Split drums should allow 10 turns of rope on the tension section.

While many VLCC owners and operators prefer the split drum type, the undivided drum has its proponents also. The following discusses the pros and cons for each type and also discusses the effects that the number of layers of wire on the tension or working drum has on each.

7.3.1 *Split Drums*

As shown in Figure 7.1 the split drum winch is a common drum divided by a notched flange into a wire storage section and a tension section. It is operated with only one layer of wire on the tension section and theoretically can maintain a constant, high brake holding power.

The split drum winch was designed as a solution to the spooling problem encountered with undivided drum winches. When wires are handled directly off drums, the final turns of the outer layer when under tension tended to bite into the lower layers. This could result in possible wire damage and difficulties when releasing the line. Also the mechanical spooling devices which were used on undivided drums were found to be susceptible to damage.



Courtesy of Pusnes

FIGURE 7.1: THE SPLIT DRUM WINCH

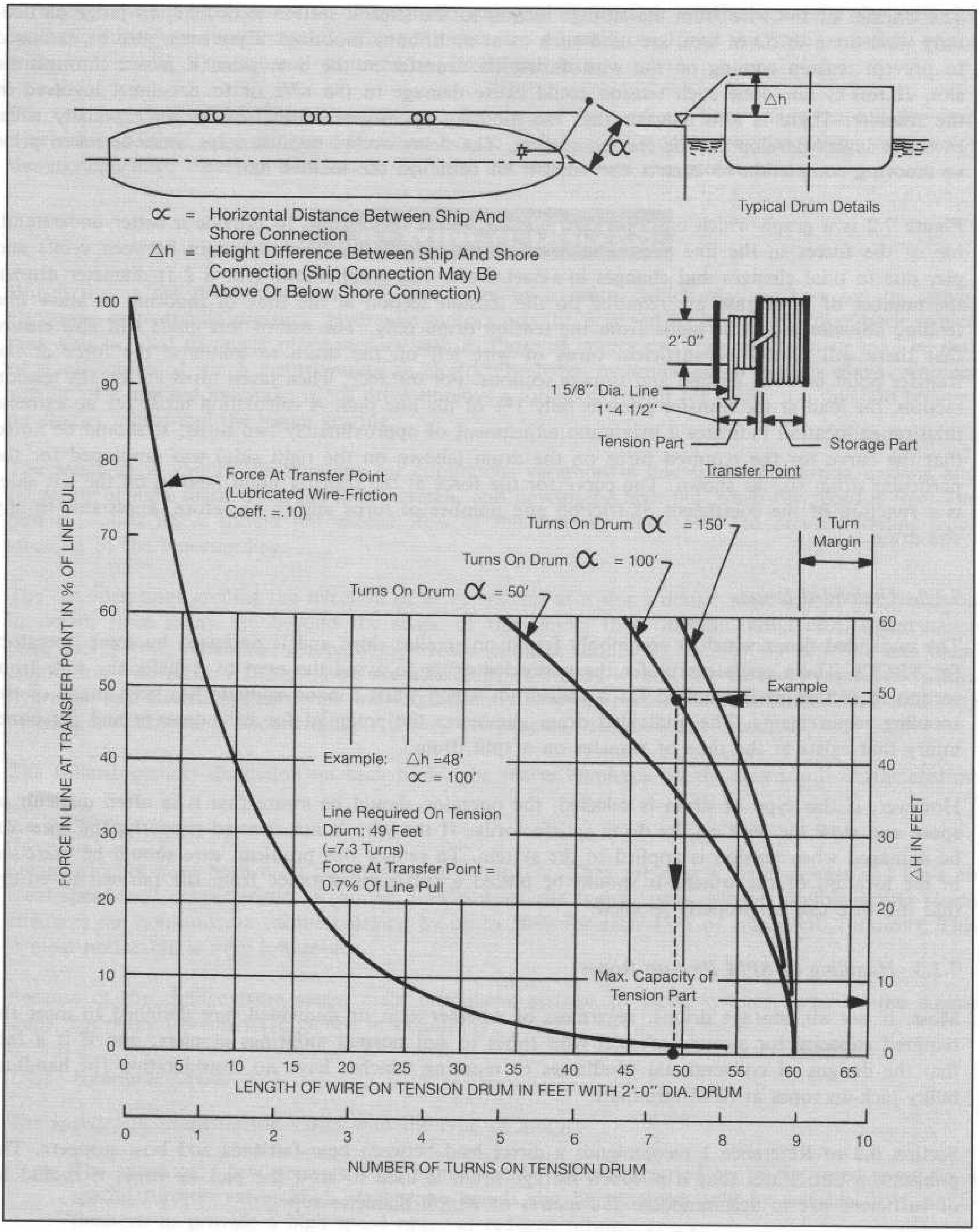


FIGURE 7.2: SPLIT DRUM MOORING WINCHES: TURNS REQUIRED ON TENSION PART OF DRUM AND FORCE IN LINE AT TRANSFER POINT

In operation, the wire from the split drum winch is sent ashore first from the working half and then directly from the storage half. As the wire is recovered, it is wound directly on the stowage half until that time when only sufficient slack wire is available to provide a sufficient number of turns on the tension drum to: (1) hold the tension of the wire on the tension drums only and (2) provide extra turns to allow for adjustments of the line throughout cargo transfer. At that time the wire rope is fed through the slot from the storage portion to the tension portion.

The transfer of the wire from the storage section to the tension section is difficult to judge particularly when long drifts of wire are used such as at multi-buoy moorings. Care must also be exercised to prevent tension coming on the wire during the transfer at the time when it passes through the slot. If this is not done such tension could cause damage to the wire or to personnel involved in the transfer. There is also concern that the mooring operations could take longer especially when excessive layers develop on the tension section. The delay occurs because steps must be taken prior to mooring completion to correct the number of turns on the tension half.

Figure 7.2 is a graph which one operator has distributed to its ships to provide a better understanding of the forces in the line and to show the effect of relative height changes between vessel and pier due to tidal changes and changes in vessel draft. The graph shows (for a 2 ft diameter drum), the number of turns that are required on the tension section at the time of mooring to allow line tending adjustments to be made from the tension drum only. The use of this guide will also ensure that there will always be sufficient turns of wire left on the drum to minimise the force at the transfer point between tension and storage sections. For instance, when seven turns are on the tension section, the load at the transfer point is only 1% of the line pull. A calculation made for an extreme tidal range location indicates a maximum adjustment of approximately two turns. It should be noted that the curve for the required turns on the drum (shown on the right side) was developed for the particular drum size as shown. The curve for the force at the transfer point (shown on the left side) is a function of the coefficient of friction and number of turns and is, therefore, applicable to any size drum.

7.3.2 *Undivided Drums*

The undivided drum winch is commonly found on smaller ships and is preferred by some operators for VLCCs. These operators prefer the undivided drum to avoid the need to transfer the wire from section to section as is required for a split-drum winch when a poor estimate has been made of the spooling requirements. The undivided drum eliminates the potential for wire damage and personnel injury that exists at the time of transfer on a split drum.

However, if this type of drum is selected, the operator should be aware that it is often difficult to spool and stow the wire on the drum satisfactorily. If the wire is not spooled properly, the wire can be damaged when tension is applied to the system. To reduce this problem, care should be exercised in the location of the winch. It should be placed a sufficient distance from the fairlead to ensure that the wire can be properly spooled.

7.3.3 *Handling of SPM Pick-up Ropes*

Most, if not all, storage drums, regardless of whether split or undivided, are designed to meet the required capacity for storage of steel wire ropes to suit normal moorings at piers, and it is a fact that the designs of conventional windlasses or mooring winches have no consideration for handling bulky pick-up ropes at SPM terminals.

Section 6.3 of Reference 1 recommends a direct lead between bow fairleads and bow stoppers. The publication also states that if a winch storage drum is used to stow the pick-up rope, it should be of sufficient size to accommodate 150 metres of 80mm diameter rope.

Whilst it is recognised that either a warping end or a wire drum may be used to heave in the SPM pick-up rope prior to connection of the chafing chain to the stopper, opinions as to which should be used are equally divided.

Following reports of accidents and "near misses" during unmooring at SPMs when the pick-up rope has been released and allowed to fall un-controlled from the ship, it is recommended that each windlass be provided with a special drum of sufficient capacity to accommodate 150 metres of 80mm diameter rope. If fitted at the inboard end of windlasses they will minimise rope turns and angles. These drums may also be used for normal wire moorings.

If main mooring lines comprise of synthetic ropes, then traction winches specially designed for synthetic ropes are recommended. These winches are normally equipped with twin traction drums with grooves and a storage reel mounted on, or below, deck.

For warping ends, which are ordinarily used for natural or synthetic fibre ropes but not for steel wire ropes because of handling difficulties due to stiffness, ISO 6482 provides design criteria for the recommended diameter and length, nominal dimensions and the profile types.

7.4 WINCH DRIVES

Winch drums are driven through reduction gears by a motor which may be powered by steam, electricity or hydraulic pressure. Hydraulic motors may be powered by a central power pack (which may also be used to power other motors such as those of cranes or cargo winches), or may be part of a self-contained winch incorporating an electrically driven hydraulic pump in each winch. Another version of a unitised hydraulic winch drive employs a torque converter for power transmission between an electric motor and the winch gearing.

The winch drive should ideally allow for continuous variation in line speed, permitting heaving and rendering at high speed when the load is small, and developing high pull when the speed is low. The pull at stalled heave should not exceed 50% of the mooring line's MBL to prevent routine over-stressing of the mooring line.

The considerations guiding the selection of drive systems at a ship's design stage will not be discussed in detail, since many are beyond the scope of this guide: these include initial cost, maintenance cost, energy cost, reliability, company experience, climatic conditions, availability of steam and inter-connection with other systems. There are also many variations within each category (especially within hydraulic and electric systems) that may significantly alter the performance, reliability and cost. Any of these are acceptable aboard tankers if they are properly designed and executed.

The following short discussion on each basic type places emphasis on the speed/pull characteristics:

7.4.1 *Steam*

Steam winches have been most popular on large tankers. They are simple, safe, robust and have an ideal speed/pull characteristic. A minor drawback of the typical twin cylinder double acting steam engine is the non-uniform torque (varying by up to 50% for each 45% of crankshaft rotation). This is most noticeable at very low speeds.

Because of the change from steam main propulsion systems to diesel systems, even on the largest ships, hydraulic systems have gained in popularity.

7.4.2 *Hydraulic Drives*

The speed/pull characteristic varies with the type of motor:

- The speed of a low pressure vane motor can be infinitely varied up to the rated speed by a special throttle valve. High slack rope speeds can be obtained only by suitable drive modifications to provide a high speed range at reduced torque.
- A high pressure variable displacement hydraulic motor has an ideal speed/pull characteristic within certain limits. To achieve a high slack line speed, a speed change provision is required.
- A torque converter drive has a similar characteristic as a variable displacement high pressure motor and it also requires a second gear step to provide for a high slack line speed.

7.4.3 *Electric Drives*

Electric drives are the least popular types on tankers due to certain inherent disadvantages. All electric equipment in the cargo tank region must be of the certified safe type. Electric controls are sensitive to moisture and corrosion. The relatively simple AC pole changing motor is inferior to other drives in pull/speed characteristic (speed variable in steps only, high starting torque, on/off control only). The AC/DC drive has excellent drive characteristics, but is expensive and more sensitive to moisture, corrosion and overheating than any other drive type.

7.5 WINCH BRAKES

The brake is the heart of the mooring system, since the brake secures the drum and thus the mooring line at the shipboard end. A further important function of the brake is to act as a safety device in case the line load becomes excessive, by rendering and allowing the line to shed its load before it breaks.

Ideally, a brake should hold and render within a very small range, and once it renders, should shed only enough load to bring the line tension back to a safe level. Unfortunately, the widely used band brake with screw application is only marginally satisfactory in fulfilling these requirements and its operation requires special care.

7.5.1 *Layers of Mooring Line on Drum*

Split-drum Winch

The rated brake holding capacity is only achieved with one layer of wire on the tension drum. Operation with additional layers will decrease the brake holding capacity.

As an example, the theoretical reduction in holding load for more than one layer on a 24″ diameter drum is as follows and assumes a rated brake holding capacity of 55 tonnes:

Layer of wires	*Theoretical holding capacity (tonnes)*	(kN)	*% Rated holding capacity*
1	55	539	100
2	49	481	89
3	45	441	82
4	41	402	75
5	38	373	69

Undivided Drum Winch

Brake holding capacity of the undivided drum winch, as with the split-drum winch, is affected by the number of layers on the drum. It is therefore, essential that the operator of an undivided drum winch knows the number of layers of wires on the drum that the manufacturer states will develop the design brake capacity. More layers on the drum will increase the moment applied to the drum because of the increased moment arm. This means that a smaller mooring line load will provide the moment which will cause the brake to slip. Conversely with less layers on the drum, mooring line loads must be larger to cause the brake to slip. It is very possible that mooring line loads could become excessive thereby increasing the possibility of mooring line breakage. Thus, the operators, as in the case of split-drums, should be advised and trained in the use of the undivided drum winch to ensure integrity of mooring and safety of vessel.

When applying the requirement for winch brakes to hold to a minimum holding load of 60% of MBL to undivided winch drums, due consideration should be given to the number of layers of wire which will be found on a drum during normal operations. For this purpose, it should be assumed that 30–50 m of wire minimum will be outboard of the fairlead.

7.5.2 *Band Brakes*

Band brakes follow the same principle as wrapping a rope around a bitt or warping head to hold a line's force. Relatively little force is required to hold a high load. This principle provides for easy brake application, but has disadvantages such as sensitivity to changes in friction, dependency of setting force on rope force and sensitivity to reeling direction.

The main factors which affect the actual holding capacity of the band brake are discussed below:

7.5.2.1 *Torque applied*

Figure 7.3 shows the results of a series of brake holding load tests on a VLCC when the torque applied is varied. It can be seen that the holding load will drop appreciably when the torque applied is lower than the recommended value. It is, therefore, essential that the operator apply the brake properly.

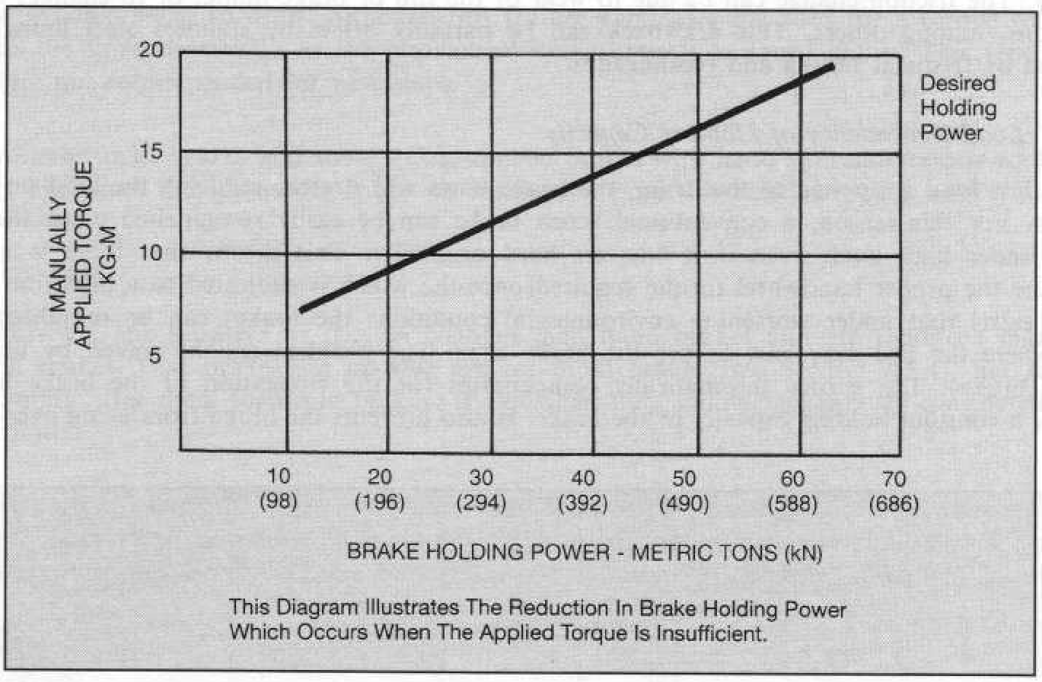


FIGURE 7.3: EFFECT OF APPLIED TORQUE ON BRAKE HOLDING POWER

Shipboard inspections have shown that winch brakes are very often not torqued to their design level; and, in some tests two men were required to apply the required torque. Hence, several VLCC operators are now installing hydraulically actuated brakes on new VLCCs and are retrofitting them on older VLCCs to ensure proper brake holding load. Obviously, it is essential that the winch brakes be operated properly to achieve desired holding loads.

Also, it has been found that some winch actuators and controls are poorly positioned. As a result, the operator is unable to see what is happening on deck. Proper design is essential to ensure proper operation.

7.5.2.2 *Condition of the Winch*

The physical condition of the winch gearing and brake shoe linings have a significant effect on brake holding load capacity. Oil, moisture or heavy rust on the brake linings or brake drum can reduce brake holding load capacity by up to 75%. Many operators run the winch with the brake set slightly to burn off or wear off the oil or moisture. (Care, however, must be taken to ensure that excess wear is not caused by this practice when using composite brake linings.) Excessive winch speed can also reduce brake holding capacity by the build-up of heat in the composite brake lining.

7.5.2.3 *Winch in Gear*

The holding load capacity of a brake can be increased by leaving the winch in gear with the steam on. However, resort to this practice should be regarded as only an emergency measure with limited application since it can only be wholly effective on winches with a single drum, which are not common on VLCCs. On double or triple drum winches, it is not normally possible to engage all drum shafts and at the same time maintain equal tension on the mooring lines. If only one drum is engaged, this would result in the lines with lower brake holding capacity slipping and transferring loads to the drum whose brake is augmented by steam. This could result in the parting of the line. Thus, the particular circumstances should be fully assessed before using this emergency procedure.

7.5.2.4 *Friction Coefficient*

A small change in the friction coefficient will cause a large change in the holding capacity. As a rule of thumb for typical mooring winch brakes, the change in holding capacity is twice the change in the friction coefficient. For example, a 10% change in friction will cause a 20% change in holding capacity. The friction change can be due to wear of the rim or brake lining, or to changes in weather conditions, among others. This drawback can be partially offset by stainless steel lining of brake rims and by frequent testing and recalibration.

7.5.2.5 *Load Dependency of Holding Capacity*

Once a line load is applied to the drum, the brake band will stretch, reducing the load on the brake controls. For this reason, a conventional screw brake can be easily re-tightened when the mooring line is under high load, even if it was set hard originally. This means that there is no way to determine the proper handwheel torque required once the winch is subjected to a high line load. The danger exists that under worsening environmental conditions the brakes can be re-tightened to the point where the line may part before the brake slips. The problem can be solved by using spring applied brakes. The spring automatically compensates for the elongation of the brake band, thus assuring a constant holding capacity of the brake. It also prevents the brake from being over-tightened.

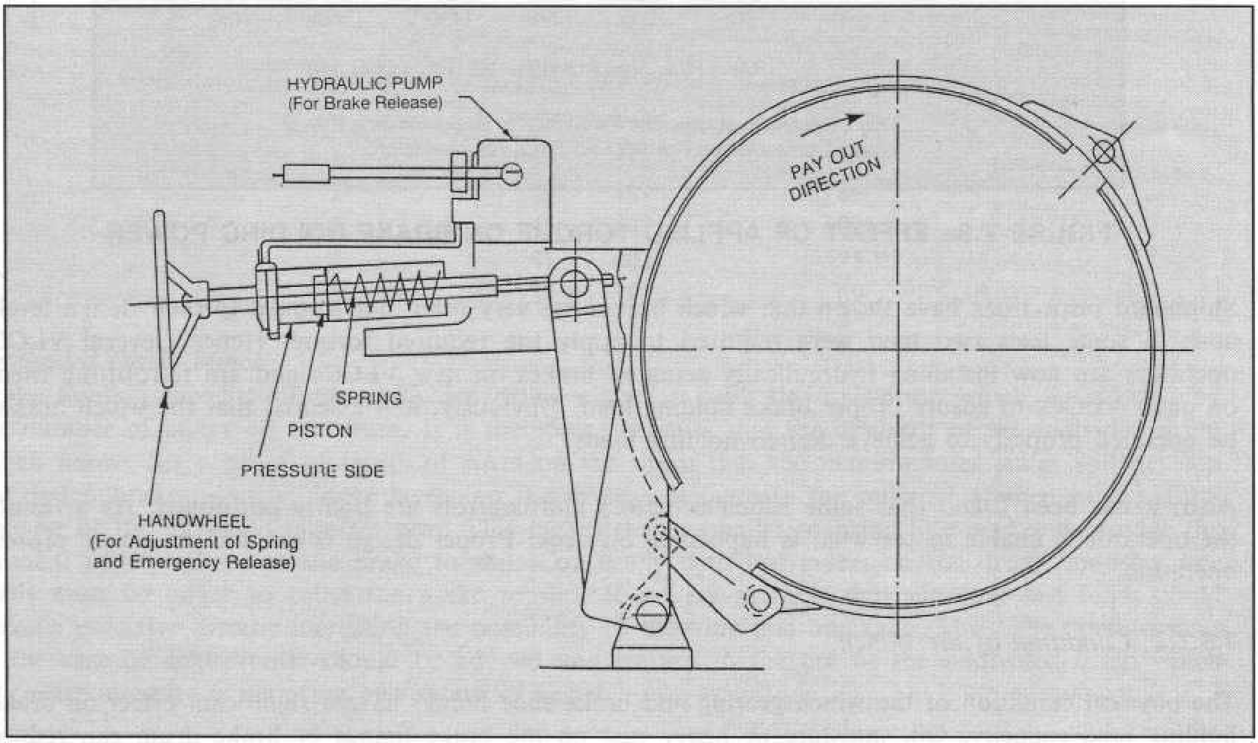


FIGURE 7.4: SPRING-APPLIED BRAKE WITH HYDRAULIC RELEASE

A schematic of a spring-applied band brake is shown in Fig. 7.4. In the case shown, the brake is released by a hydraulic hand pump. To apply the brake, a valve is opened to release the hydraulic pressure. Winches with hydraulic drives can utilise the main hydraulic pressure to release the brake. The handwheel is not used for routine operation: it serves to adjust the spring compression during calibration, and to release the brake in case of a hydraulic malfunction. The handwheel should be secured with a seal after each calibration to prevent tampering. Another advantage of the spring applied brake is its fail-safe feature.

7.5.2.6 *Sensitivity in Reeling Direction*

A band brake is designed to work in one direction only. Therefore the line must always be reeled correctly onto the drum. The line is properly reeled if it is pulling against the fixed end of the brake strap.

Disc brakes work equally well in either direction.

Some winches are equipped with so-called hydraulic-assist brakes. With this type, a hydraulic cylinder sets the brake. A pressure gauge connected to the cylinder allows the brake to be applied with the proper force, predetermined during brake testing. Once the brake is applied with the hydraulic assist, the load is transferred to the mechanical linkage by tightening the handwheel until the pressure in the assist cylinder starts to drop. Although this feature eliminates the need for a torque wrench, the brake has the same drawbacks as the mechanical screw brake mentioned above. This type of brake is therefore not recommended for new ships.

Since hydraulic-assist brakes and some spring-applied brakes with hand hydraulic release look similar, it is very important that operators study the instruction book for their particular ship's installation.

7.5.3 *Disc Brakes*

Disc brakes are less sensitive to friction changes than band brakes and are therefore better suited for winch brakes. Disc brakes are in wide use for input brakes (see Section 7.5.4), but few winch manufacturers offer them for drum brakes. If available, they should be seriously considered for new construction.

Disc brakes do not have the other drawbacks mentioned earlier, but this is of secondary importance, since band brakes can be improved by spring application and proper operation. Spring application is also recommended for disc brakes because of the foolproof simple operation that it permits.

7.5.4 *Input Brakes*

Most hydraulic and electric winches are provided with spring-applied brakes at the drive motor. They are automatically applied by springs when the control lever is in neutral and automatically released when the control lever is in the heave or rendering position (when the motor is powered). ISO Standard 3730 requires automatic brakes with a holding capacity of 1.5 times the rated load for all electric winches.

For single drum winches (one drum per drive), input brakes could serve as the primary brake and would eliminate the need for separate drum brakes. However, if the winch is also provided with a warping head, a drum brake and dog clutch would still be required to allow use of the warping head while the drum is held stationary.

Multiple-drum winches always require a brake for each drum.

Most input brakes are not rated to serve as a primary brake due to strength limitations of the gears. If this is the case, once the ship is moored the drum must be set on the drum brake and disengaged from the drive by means of the dog clutch. If the drum is left engaged and the drum brake is set, both brakes will work in unison (rated holding capacity is directly additive). The combined holding capacity will exceed 100% of the line's MBL, an undesirable result for reasons explained earlier.

7.5.5 *Winch Brake Testing*

Regardless of the brake type, periodic testing is essential to assure a safe mooring. The following provides a guide for testing of mooring winch brakes:

7.5.5.1 *Test Program*

7.5.5.1.1 Frequency

Each winch brake is to be tested individually and tests are to be carried out prior to the vessel's delivery and every year thereafter in line with recommendations in the International Safety Guide for Oil Tankers and Terminals (ISGOTT) Chapter 3. In addition, individual winches are to be tested after completion of any modification or repair involving the winch brakes, or upon any evidence of premature brake slippage or related malfunctions.

7.5.5.1.2 Test specification

For each vessel a winch test specification is prepared incorporating specific instructions for setting up the test gear, preparation of the winch for testing, setting of the winch brakes, application of the test load, revision of torque wrench or hydraulic pressure readings if required, and recording of test results. The coefficient of friction of the brake lining is considerably affected by moisture. To assure constant results the winch is to be operated for a short period with the brake set slightly on to dry the brake surface.

7.5.5.1.3 Supervision of testing

All winch testing is to be carried out under the supervision or in the presence of a senior officer designated by the Master or Chief Engineer or repair superintendent familiar with the test procedure and the operation of the winches.

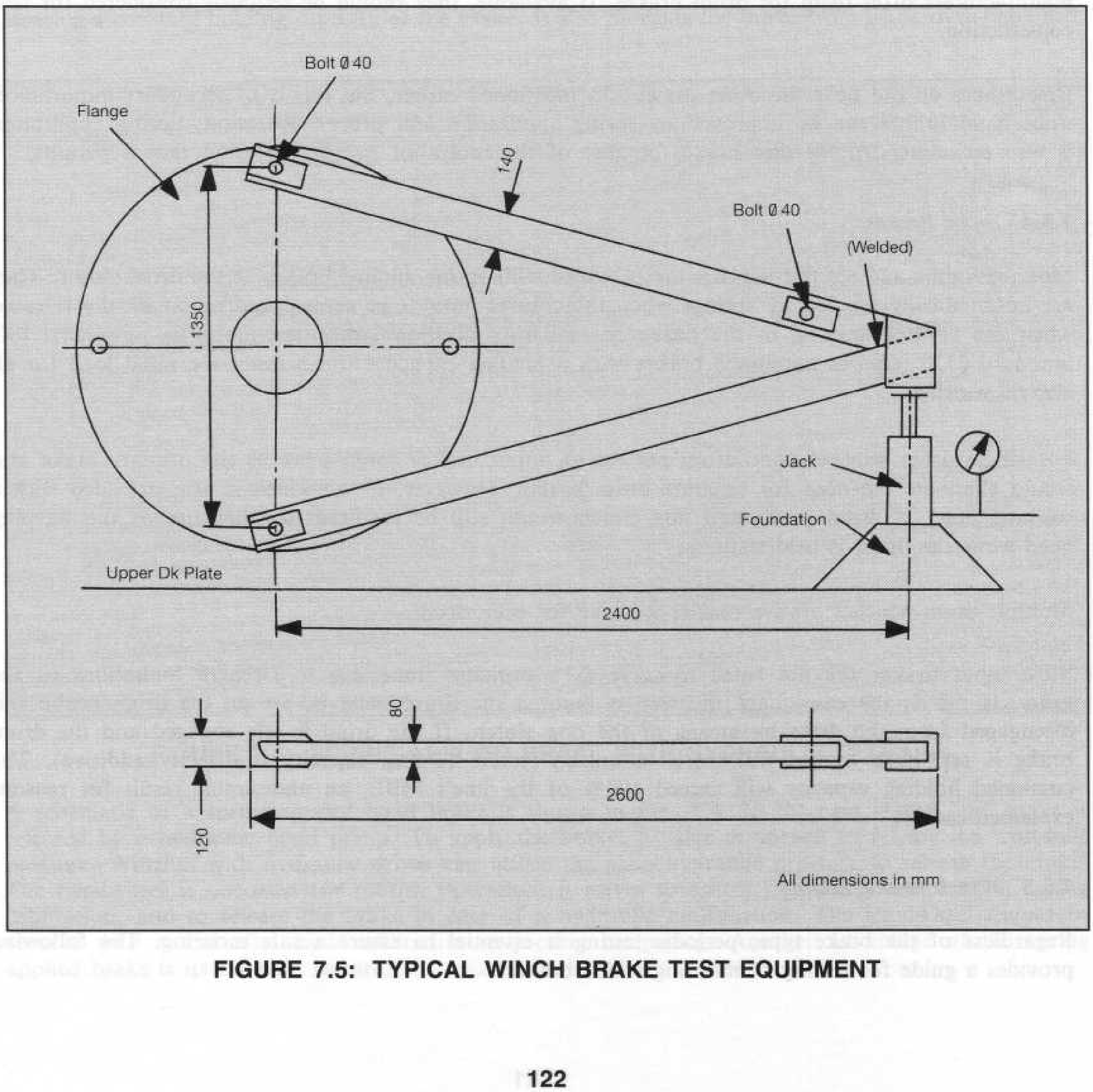


FIGURE 7.5: TYPICAL WINCH BRAKE TEST EQUIPMENT

7.5.5.1.4 Testing equipment

Typical equipment for testing the brakes is shown in Figure 7.5, and includes the following items:

- Lever consisting usually of two pieces of bar as shown on the sketch. The lever is secured to the drum of the winch by means of bolts furnished with the test kit and fitted through holes provided in the drum flange;
- Hydraulic jack with pressure gauge; and
- Foundation to be placed under the hydraulic jack for the purpose of distributing the load into the deck structure.

It is recommended that complete test equipment be placed on board each vessel, properly stowed in an appropriate location. Alternatively, the Owner may elect to procure one or two sets of testing equipment for each type and size of winch and retain this equipment in a convenient central location for shipment to repair facilities as required.

7.5.5.2 *Method of testing*

The testing arms are bolted to the flange of the winch drum with the hydraulic jack pressed under the end of the arms at the designated location and resting on supports. The flange brake is set as recommended in the test specification. If the winches are set normally, a torque wrench should be used. If they are set hydraulically, the pressure gauge should first be calibrated.

Before testing, the detailed instructions for testing included in the test specification should be reviewed and the equipment prepared accordingly. The instructions will include:

- The values for torque wrench or pressure gauge fitted for setting up the brakes;
- A curve or table relating hydraulic jack test pressure to line pull; and
- Hydraulic jack pressure at which the brake is designed to render.

With the winch prepared for testing, the testing gear securely in place and winch brakes set in accordance with the recommendations, pressure is applied to the hydraulic jack. The winch drum is to be carefully observed. At the first sign of movement, the hydraulic pressure applied to the jack is recorded and the following action taken:

- If slippage occurs at a pressure less than designed, the brake should be tightened or repaired and jack pressure reapplied;
- If the recorded pressure corresponds to the design pressure the jack should be released and the test gear removed; or
- If slippage does not occur at the design pressure, the brake setting should be adjusted so the brake can render at the design load.

The lever should be lightweight for easy handling. Testing can be further simplified by reducing the lever to slightly more than the drum flange radius and placing the jack directly on the winch foundation. A schematic of such a test arrangement is shown in Fig. 7.6. In place of the heavy lever, a simple fitting attached to the drum flange suffices. The higher jack loads may pose a problem for existing equipment, but provision for this method can easily be incorporated into new equipment.

Once the brakes are tested and calibrated, the proper setting should be recorded. In case of conventional screw brakes, a tag should be attached stating the proper torque. For spring-applied brakes, the spring compression distance should be recorded and the spring adjustment mechanism secured with a seal.

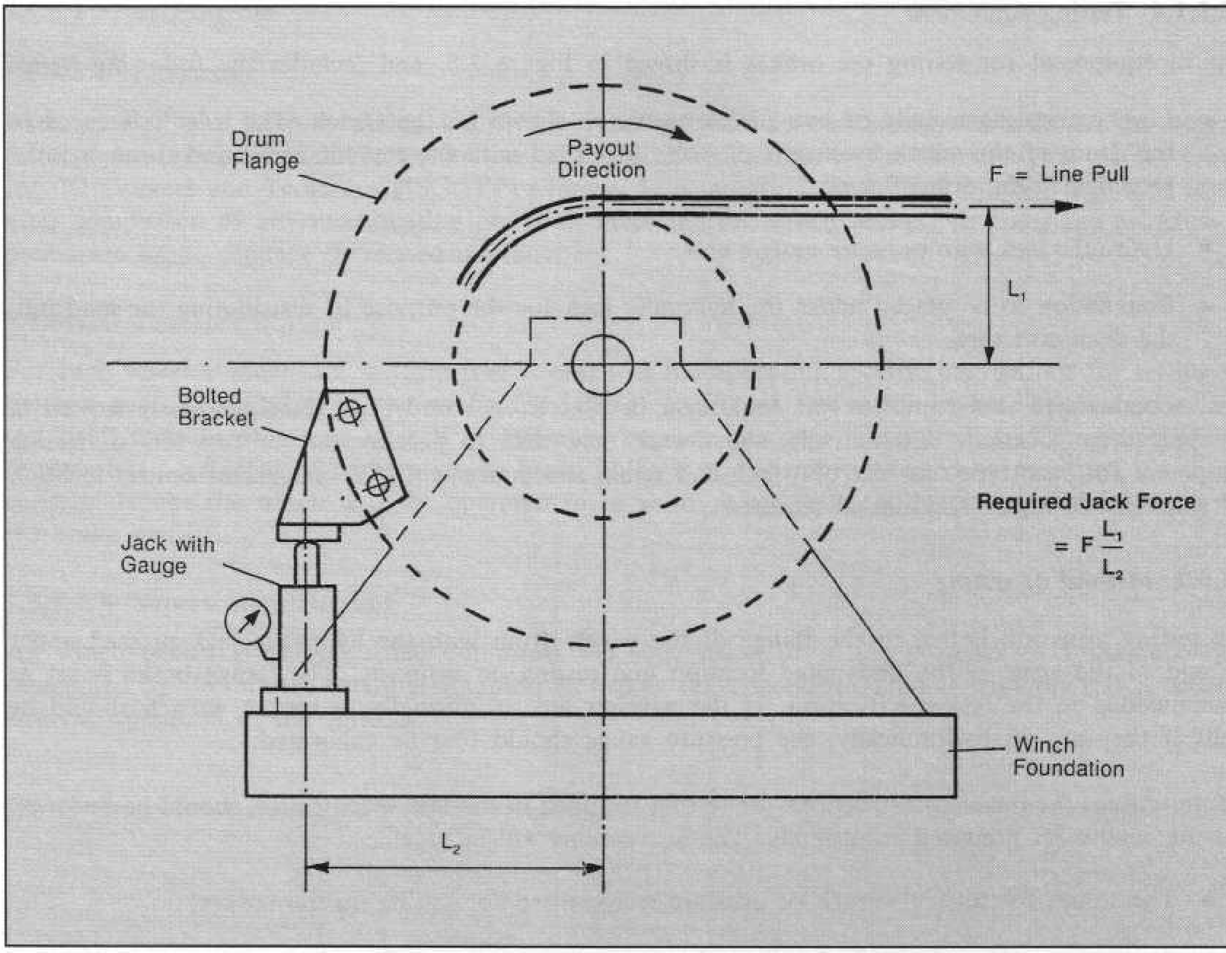


FIGURE 7.6: SIMPLIFIED BRAKE TEST KIT

7.5.6 *Brake Holding Capacity*

The primary brake should be set to hold 60% of the mooring line's MBL. Since brakes may deteriorate in service, it is recommended that new equipment be designed to hold 80% of the line's MBL, but have the capability to be adjusted down to 60% of the line's MBL. A drum brake holding capacity of 80% MBL is also required by Lloyds, DNV and ISO Standard 3730 with the rope on the first layer. If a brake of an undivided drum is set to hold 80% MBL on the first layer, it will hold approximately 65% MBL on the third layer.

7.6 WINCH PERFORMANCE

The principle performance particulars for different size mooring winches should correspond to values listed in Table 7.1. These are in general agreement with ISO Standard 3730, except for minimum light line speed and in-service setting of brake.

7.6.1 *Rated pull (also called drum load or hauling load)*

This is the pull that the mooring line can develop at the rated speed on the first layer. The listed values correspond to approximately 30% of the line's minimum breaking strength. This value assures adequate force to heave in against environmental forces. On the other hand, it is low enough to prevent line overstressing in the stalled condition, considering that the stall pull is generally higher than the rated pull for most drive types.

RATED SIZE	RATED PULL*	RATED SPEED MIN.*	LIGHT-LINE SPEED MIN.**	DESIGN ROPE DIA. (WIRE ROPE)	MINIMUM BREAKING STRENGTH OF ROPE	STALL PULL MAX.	BRAKE HOLDING FORCE* DESIGN VALUE NEW	BRAKE HOLDING FORCE* IN-SERVICE SETTING	MINIMUM DIA. OF DRUM	DRUM CAPACITY*** Normal	DRUM CAPACITY*** High	WIDTH OF TENSION SECTION****
	t	m/min	m/min	mm	t	t	t	t	mm	m	m	mm
12	12	23	45	26	43	21	35	26	416	200	400	260
16	16	23	45	32	66	33	53	40	512	250	500	320
20	20	18	45	36	83	41	67	50	576	250	500	360
25	25	18	45	40	103	61	83	62	640	250	500	400
32	32	15	45	44	124	62	100	75	704	250	500	440
40	40	15	45	48	148	74	119	89	768	250	500	480

* With rope on first layer.

** This is a desirable light line speed in the first layer and is in excess of the 0.5 m/s requirement of ISO Standard 3730. However, achievable light line speed depends on the winch drive as mentioned in 7.4 and 7.6.3, and should be clarified with the winch manufacturer.

*** For split-drums, this applies to the storage part of the drum.

**** Applies to split drums only.

Note: Performance data also applies to winches designed for fibre ropes, but drum dimensions must be increased to suit the larger diameter of the fibre lines.

TABLE 7.1: PERFORMANCE SPECIFICATION FOR MOORING WINCHES

7.6.2 *Rated speed (also called nominal speed or design speed)*

The rated speed is the speed that can be maintained with the rated load applied to the mooring line. The rated speed in combination with the rated pull determines the power requirement for the winch drive. The recommended rated speed is higher for the smaller size winches because the smaller sizes are intended for smaller ships and these warp and moor more quickly than large ships.

7.6.3 *Slack line speed (also called no-load speed or light-line speed)*

This is the speed that can be achieved by the winch with negligible load on the line. High speed is essential to pass a line quickly to a shore mooring point or to bring the line quickly back on board. The listed value recognises that during this phase the line is normally not on the first layer, even on split-drum winches, and the actual line speed could be up to 50% higher. The slack line speeds achievable are different for different types of drives and this must also be considered when specifying performance and drive type for new equipment. For example, a low pressure two-speed vane motor can achieve only twice the rated speed.

7.6.4 *Stall heaving capacity (also called stalling load)*

This is the line pull the winch will exert when the control is in heave and the line is held stationary. A high stall heave capacity is desirable to winch a ship onto the pier against high environmental loads. On the other hand, the stall pull should not be so high that there is any danger of line breakage, and should never exceed 50% of the mooring lines MBL. Achievable stall pull depends on the drive type and control. As a rule of thumb the following ratios of stall pull to rated pull may be assumed:

Steam-driven	1.50
Hydraulic	1.05
Electric	1.25

7.6.5 *Drum capacity*

The drum should be capable of stowing the total line length. Table 7.1 lists the drum capacities as specified in ISO Standard 3730. Two capacities, 'normal' and high' are indicated. Winches with undivided drums could be more suitable for the 'normal' capacity since this would reduce the number of layers required. Nevertheless, many operators prefer longer hawsers and 365 metres would be a representative length. ISO Standard 3730 also specifics that for undivided drums the total number of layers shall not exceed five for normal capacity drums and eight for high-capacity drums when the total hawser length is stowed on the drum. For split drums, the number of layers on the storage section may be higher. The ISO standard notes the possibility of line damage if large loads are applied while more than four layers of rope are reeled on the drum. Presumably this applies to wire rope, since the standard also warns of a short rope life if synthetic rope under tension is wound in more than one layer.

On split-drum winches, it should be possible to store the total length of the mooring line on the stowage part of the drum. This would facilitate the rapid pay out of a line during mooring without the need to transfer the line through the dividing flange slot. An allowance for 1.5 extra turns should be made when determining the flange diameter of 'normal' capacity drums. For 'high' capacity drums, no allowance need be made if wound with the layers superimposed directly upon each other (i.e. without a half-rope diameter offset between adjacent layers). On split drums, the tension section should be wide enough for 10 turns of line.

7.7 STRENGTH REQUIREMENTS

The strength of the mooring winch structure should be based on the breaking strength of the line and the criteria given in Section 4. The winch should therefore be clearly marked with the range of rope strengths for which it is designed.

Although brakes should slip before the line breaks, under extreme conditions such as overtightening of the brake or overriding rope turns, the winch may be subjected to the full MBL of the line. For this reason, it is recommended that all structural components of the winch including brakes, foundations and supporting deck structure should be designed in accordance with Table 4.3, Sheet 4.

Where a drive input brake is intended to be the primary brake, all components, including the reduction gears, must be based on the full MBL of the mooring line. For winches with undivided drums, the line must be assumed to be at the layer that produces the highest stress in each individual component. For split-drum winches the line shall be assumed to be on the first layer.

Ship designers should pay special attention to the support structure in way of the fixed end of band brakes, as this is the major load transfer point. (See Section 5 for further details.)

7.8 WINCH TESTING

In addition to brake testing per 7.5.4, the following acceptance tests are recommended. The tests are in general agreement with ISO Standard 3730.

7.8.1 *Rules concerning testing at manufacturer's works for the acceptance of the manufacturer and purchaser:*

- *Type testing*

One winch of each batch must be tested. This test may be replaced by a prototype test certificate if agreed by the manufacturer and purchaser.

The test shall be carried out as follows:

(1) operation under load: alternately hauling and rendering at the rated load of the winch for 30 minutes continuously.

(2) holding test: to be tested by applying the holding load to a rope led off the drum, with no rotation of the drum; this may be carried out on board ship if agreed between purchaser and manufacturer.

(3) automatic brake system test: this test may be carried out on board ship if agreed between purchaser and manufacturer.

(4) throughout testing, the following should be checked:

(a) tightness against oil leakages
(b) temperature of bearings
(c) presence of abnormal noise
(d) power consumption
(e) speed of rotation of the drum.

Where tests are required in excess of the type test, they should be agreed between the purchaser and manufacturer at the time of the contract.

- *Individual tests*

The following tests should be carried out:

(1) operation under no load: running for 30 minutes, 15 minutes continuously in each direction, at light line speed.

(2) correct operation of brake system.

(3) throughout testing, the following should be checked:

(a) tightness against oil leakages
(b) temperature of bearings
(c) presence of abnormal noise
(d) power consumption
(e) speed of rotation of the drum.

7.8.2 *On-board acceptance tests and inspections*

It is recommended that the following inspections and tests be carried out on board the ship using ship's power.

- *Running tests*
  The winch is to be run for ten minutes at light-line speed, five minutes continuously in each direction. Bearing temperature rises must be checked.

7.9 SUMMARY OF RECOMMENDATIONS

The following is a summary of the recommendations contained in Section 7.

7.9.1 *Recommendations for ship designers*

- All winches should be controlled in the manual mode, including self-tensioning winches as presently designed.
- In selecting drive systems, the speed-pull relationship for various types must be considered.
- Minimum rated winch pull should be about 30% of the line's MBL.
- Minimum rated speed should be selected on the basis of ship size, from 0.13m/sec for large ships to 0.20m/sec for small ones.
- Minimum light line speed should be 0.75m/sec., if practicable for the chosen drive system.
- Maximum stall pull should be 50% of the mooring line's MBL.
- New winch brakes should be capable of holding 80% of the line's MBL, with provision for adjustment to 60% of the line's MBL.
- Winch drums should have a minimum diameter of 16 times the rope diameter.
- The tension section of a split drum should be wide enough for ten turns of line.
- Winch brakes should be of the fail-safe spring applied type.
- Disc brakes are a good alternative for winch drums.
- For optimum ease, security of operation and manpower reduction, single drum winches with full capacity 'fail-safe' input brakes should be considered.
- Winch brake testing provisions should be incorporated into the winch design.
- The strength of all structural winch components, including drum brakes and deck support, should be based on the line MBL. Special attention should be given to the connection of the brake band anchor point to the ship's structure.
- Winches should be clearly marked with the rope MBL for which they are designed.

7.9.3 *Recommendations for ship operators*

- The instructions for the particular mooring winches should be followed carefully. Items of equipment which appear similar may require very different operation.
- Automatic tension mooring winches should only be used in the manual mode.
- Winch brakes should be tested annually and adjusted to hold 60% of the MBL of the mooring lines. After adjustment, a tag with proper setting values should be attached to the winch. On spring-applied brakes, the spring adjustment mechanism should be secured with a seal to prevent inadvertent tampering.
- Mooring lines must be spooled onto the drum in the correct direction, since band brakes are designed to work in one direction only.
- Winches that are provided with both a drum brake and an input brake should be operated with one brake only while the ship is moored. On multiple drum winches, this will always be the drum brake. On single drum winches, the instruction book should be consulted to determine which brake is the primary brake designed to hold 60% of the line's MBL. Where the drum brake is the primary brake, the clutch between drum and shaft should be disengaged while the ship is moored. This applies also to winches without input brakes.

Section 8.0
Deck Mooring Fittings

8.1 INTRODUCTION

Many national, shipyard and vendor standards for mooring fittings exist. In almost all cases these standards lack complete information on the true strength of the fittings. Even where back-up calculations are provided, questions still arise. For complete evaluation of existing standards, extensive stress analysis would be required for each fitting and component. This has not been undertaken here. Instead, existing standards have been reviewed and recommendations made based upon successful or unsuccessful experience with them. In some cases even superficial analysis is sufficient to expose deficiencies. Any fitting of undocumented or incomplete strength characteristics should be verified for compliance with the recommended strength criteria by detailed calculations and prototype testing.

Generally, reference is made to the widely-used International Standards (ISO), British Standards (BS) and Japanese Industrial Standards (JIS). A complete listing of ISO, BS and JIS standards for mooring fittings and equipment may be found in Section 4.

One difficulty in establishing the general acceptability of a particular fitting is the inconsistency in load and design parameters between various standards. This can be partially compensated for by the continuing adoption of ISO Standards as the basis of national or vendor standards (e.g. the BS and Japanese standards for bollards now incorporate the appropriate ISO Standard requirements). Unfortunately, the ISO Standards do not yet cover all mooring fittings, nor do they always provide the detail included in some national standards. Moreover, some ISO and national standards have not been updated to keep pace with industry progress, adding further to the difficulty in knowing which strength criteria are relevant to a fitting.

Reproduction of the relevant ISO and national standards, which would have assisted in understanding the complexities of load and design criteria for mooring fittings and improved the effectiveness of this report, has been precluded owing to copyright constraints. The individual standards institutions should be contacted for detailed applications.

The following notes itemise some of the critical elements in selecting and installing acceptable deck mooring fittings.

8.2 VERTICAL BOLLARDS

ISO 3913 and BS MA 12, Parts 1 and 2, show standards for bollards with diameters from 100mm to 800mm. Although the ISO Standard is recommended for the design of bollards, where detail is insufficient, e.g. in construction and weld details, reference should be made to the British Standard dealing with these points. The bitts should penetrate the baseplate rather than just be welded to the top of the baseplate, and strengthening rib plates should be fitted in the base. The JIS and other national standards, which closely follow the ISO Standard, are acceptable provided construction details follow the above principle. The range of sizes covered by other standards may vary, such as the JIS which also includes 355, 450 and 560mm sizes.

The tabulated 'single rope maximum loading' quoted in the ISO Standard is the SWL when the rope is belayed in a figure-of-eight fashion (the ISO safety factor is the same as that recommended in Section 4, Table 4.3). According to the ISO Standard (and some national standards), two ropes of this value may be applied in figure-of-eight fashion near the base, or alternatively a single rope of twice the load may be applied, as a loop, at heights up to 1.2 times the bollard nominal diameter.

Nominal Size of Bollard (D) in mm	**Total Maximum** Rope Loading in Tonnes if Load is Applied at 1.2 D above Base, or Lower	
	Figure-of-Eight Belayed	Attached with Eye
100	3	6
125	4	8
160	5	10
200	8	16
250	12	24
315	20	40
400	32	64
500	46	92
630	70	140
710	82	164
800	100	200

Notes:

1) Bollard Scantlings per ISO 3913
2) The "figure-of-eight" values correspond to the "single rope maximum loading" in ISO 3913 and are the values recommended to be marked on the fitting as the SWL.

TABLE 8.1: MAXIMUM PERMISSIBLE ROPE LOADING OF BOLLARDS

The British Standard refers to the methods of belaying the rope as 'mooring' for figure-of-eight belaying, and 'towing' for loop belaying. The 'towing' SWL is twice the 'mooring' SWL. Table 8.1 shows the permissible rope loadings. The reason that the SWL depends on the method of rope belaying is that certain belaying methods tend to pull the two posts together and thus induce a higher stress in each barrel than that produced by an eye laid around a single post. With figure-of-eight belaying, the loading in each post corresponds to the sum of all forces in the successive rope layers, which can be higher than the maximum rope load. Experienced mariners are aware of this phenomenon and have devised methods that effectively distribute the external load over the two posts (for instance, by taking one or two turns around the first post before starting to belay in figure-of-eight fashion). Nevertheless, ISO takes a conservative approach by assuming that some mariners may lack this knowledge. Fig. 8.1 illustrates the two methods of belaying a bollard. Care must be taken in interpreting tabulated maximum rope loadings for other standards, since design criteria in terms of belaying methods, and number and height of ropes applied, are not consistent with ISO definitions. See also Table 8.1.

8.3 CRUCIFORM BOLLARDS

ISO 3913, Addendum 1, covers bollards with a barrel diameter from 70mm to 400mm. JIS F2804 covers bollards (called cross bitts by JIS) from 150mm to 350mm.

Similarly to vertical bollards, the tabulated 'single rope maximum loading' value quoted in ISO for double cruciform bollards is the SWL when the rope is applied in figure-of-eight fashion. Since a single cruciform bollard cannot be overstressed if certain belaying methods are used, their SWL is twice that of a double bollard. No information on height of rope application is given in the standard; since the scantlings are the same, it is assumed to be the same as for vertical bollards. The design criteria recommended in Section 4, Table 4.3 should therefore be used.

The BS Standard, MA 12, Part 2 (which is similar to ISO 3913) includes OCIMF recommended sizes for single cruciform bollards for use at tanker manifolds.

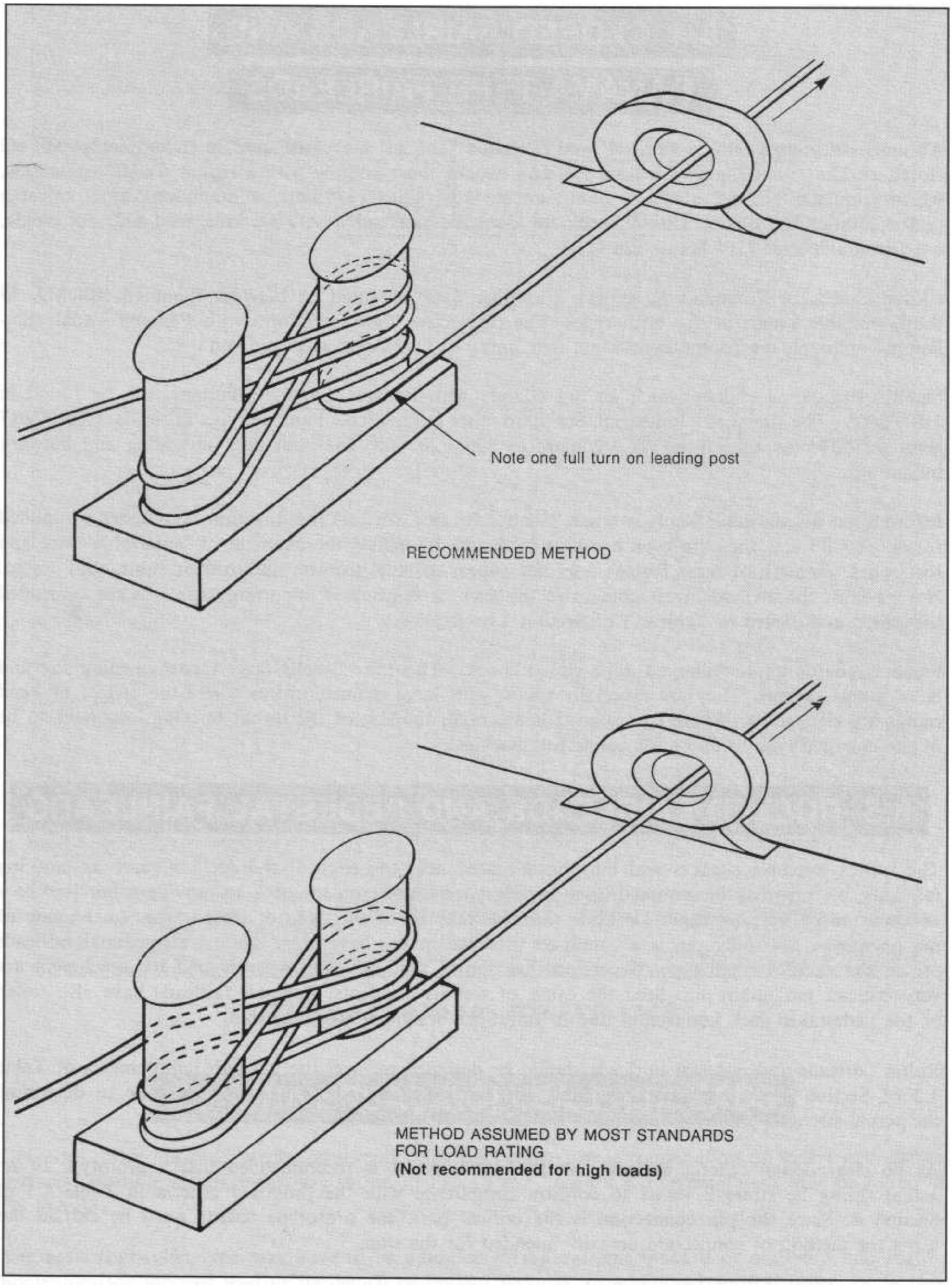


FIGURE 8.1: METHODS OF BELAYING A BOLLARD

8.4 CLOSED FAIRLEADS AND PANAMA-TYPE FAIRLEADS

Although the terms 'closed fairlead' and 'Panama fairlead' are often used interchangeably, not all closed chocks are 'Panama chocks'. Panama chocks must comply with Panama Canal regulations, which stipulate, among others, a minimum surface radius (178mm), a minimum throat opening (300 × 250mm for single, 350 × 250mm for double type) and a safe working load (32t for single, 64t for double type (314 kN or 628 kN)).

Closed chocks or Panama-type chocks are either deck mounted or bulwark mounted. BS MA 19 shows suitable standards for both types. The three sizes shown conform with Panama Canal regulations, although the footnotes indicate that only the two larger sizes are suitable.

Smaller size closed chocks which do not comply with Panama Canal regulations, can be found in JIS F2005. The Japanese Industrial Standard also covers true Panama-type fairleads (JIS F2017 gives six different sizes from 310 × 260mm to 500 × 260mm, for both deck mounting and bulwark mounting).

Information on allowable loads is scant. Neither BS nor JIS lists the direction or number of applied ropes. The BS lists the wire rope breaking load and JIS F2005 the diameter of 'applicable' wire and soft ropes. Strength of these fittings does not appear to be a problem due to their substantial design. Nevertheless, the method of attachment to the deck or bulwark is important, and it is recommended that the criteria listed in Table 4.3 of Section 4 be applied.

Some standards quote 'enlarged' type closed chocks. These are usually large throat opening size and large radius fittings. They are especially useful with large diameter wires where the effects of bend radius are significant. Where soft rope tails are used, the size of the throat opening may have to be of the enlarged type to allow for connector shackles.

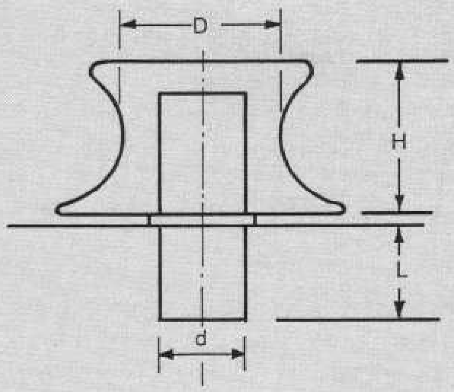
8.5 ROLLER FAIRLEADS AND PEDESTAL FAIRLEADS

The rollers resemble sheaves and may be mounted near the edge of the deck to serve as mooring fairleads; or they may be mounted upon a pedestal elsewhere on the deck to provide a fair lead to a winch drum or warping head. Deckside fairleads may be of the open or closed type. In the case of the open type the roller pin is a cantilever attached at the base only. Almost all pedestal fairleads are of the cantilever pin type. Experience has shown that the cantilever pin and its attachment are very critical: pin failure has been the cause of serious accidents. Pedestal fairleads have also failed at the pedestal-to-deck connection due to improper design or workmanship.

Roller fairleads and pedestal fairleads should be designed to meet the strength requirements of Table 4.3 of Section 4. Strength data available for existing standard fittings is inadequate to determine the actual strength. Table 8.2 compares the BS and JIS standards.

As no clear design criteria are given in the standards, it is recommended that a prototype of an actual fitting be strength tested to confirm compliance with the proposed criteria in Table 4.3 of Section 4. Since the pin connection is the critical part, the prototype testing must be carried out using the method of connection actually intended for the ship.

Figure 5.5 in Section 5 shows an example of a pedestal fairlead designed for a wire breaking strength of 85t (834 kN) with wire wrapped 180° (170t total load (1667 kN)) and a safety factor as listed in Table 4.3 of Section 4.



D [mm]	BS MA 22 H [mm]	BS MA 22 L [mm]	BS MA 22 d [mm]	BS MA 22 "Load"* [t]	JIS F 2014 H [mm]	JIS F 2014 L [mm]	JIS F 2014 d [mm]	JIS F 2014 "Load"* [t]
130	85	NOT LISTED	70	8	—	—	—	—
150	—		—	—	150	40	60	15
160	106		90	12	—	—	—	—
200	132		110	20	185	49	80	23
250	160		140	30	220	58	90	29
300	—		—	—	240	61	110	45
320	180		165	48	—	—	—	—
350	—		—	—	260	65	125	65
				66	—	—	—	—
400	—		—	—	280	67	140	81
450	—		—	—	292	90	155	104

* "Load" Definition: BS MA 22 : Safety factor and wrap unknown
JIS F 2014: Breaking strength of listed rope. Wrap unknown

TABLE 8.2: ROLLERS FOR FAIRLEADS

8.6 UNIVERSAL ROLLER FAIRLEADS

Universal roller fairleads consist of several cylindrical rollers, or a combination of rollers and curved surfaces. Possible arrangements are shown in Figs. 8.2, 8.3 and 8.4. Further details may be found in BS MA 23.

The basic four-roller type may have to be modified to suit extreme inboard or outboard line angles. Sometimes the inboard lead to the winch or bollard requires an additional vertical roller, or in rare cases, two additional vertical rollers as shown in Fig. 8.4. Extreme outboard angles can be accommodated with chafe plates as shown in Fig. 8.3.

Some line angles are impractical and seldom occur. For example, the inboard lead seldom runs in an upward direction and the outboard angle is upward only at terminals with a large difference in tide or when moored in canal locks. In this case the Type C shown in Fig. 8.2 would be adequate and result in a lower overall height of the fairlead.

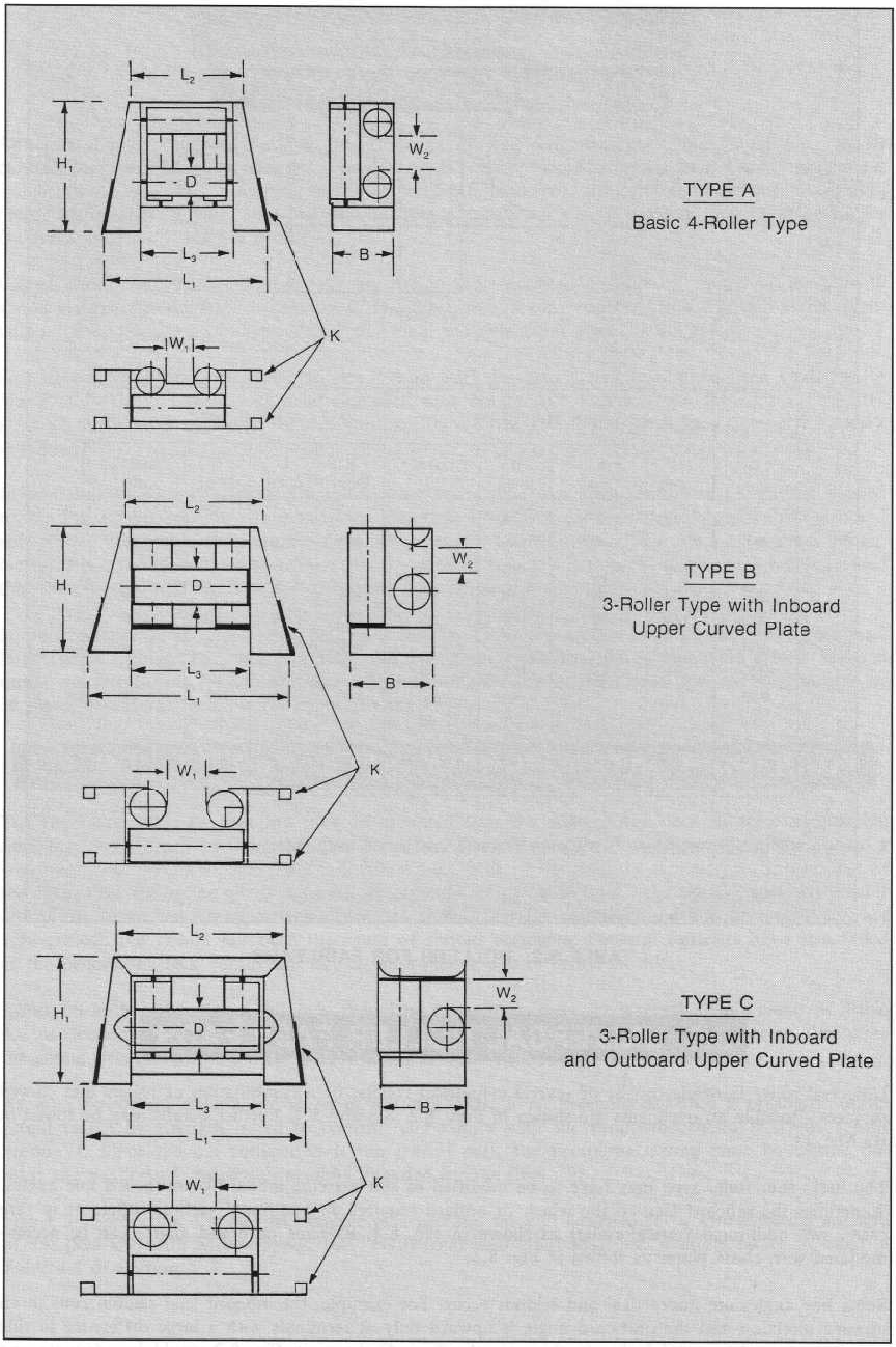


FIGURE 8.2: TYPES OF UNIVERSAL ROLLER FAIRLEADS

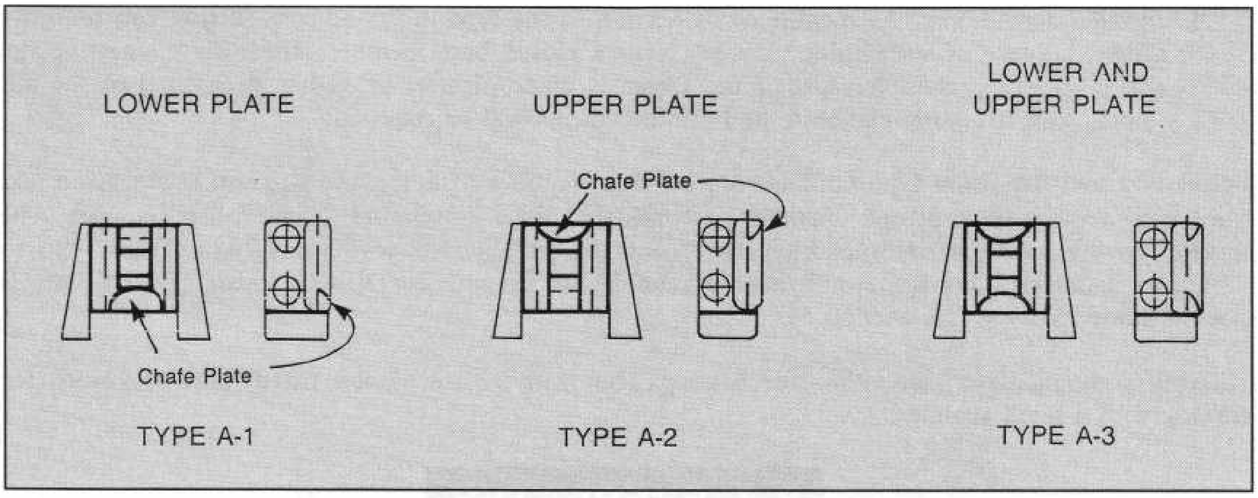


FIGURE 8.3: ADDITIONAL CHAFE PLATES FOR TYPE A FAIRLEADS

Care must be taken when installing fairleads on sloping bulwarks, such as those in the bow area, to avoid line chafing at the upper outboard edge of the frame. Type A-2 or A-3 of Fig. 8.3 would be suitable for this case.

Apart from line leads, the following considerations apply when selecting universal fairleads:

- *Roller diameter.* A small diameter will reduce the strength of the line (refer to Section 6.2.4). For use with wire rope with independent wire rope core, the roller diameter should be about 12 times the rope diameter.

- *Opening size* (dimensions Wl and W2 in Fig. 8.2). The minimum size is determined by the space required to pass the eye of the line or end fittings (such as those required for tails) through the fairlead. The following minimum dimensions are recommended, provided that the roller diameter is 12 times the diameter of the wire rope and four times the diameter of a synthetic rope:

  Width = 1.00 × roller diameter
  Depth = 0.65 × roller diameter

- *Strength.* Strength should be the main criterion. The recommended strength criteria is shown in Table 4.3 of Section 4. Many existing designs do not comply with these criteria. Often the frame is not strong enough to resist longitudinal forces such as those applied to spring lines. Proper frame strength can be provided by fitting flat bar reinforcements to the outer edges of the frame, as indicated by the letter 'K' in Fig. 8.2.

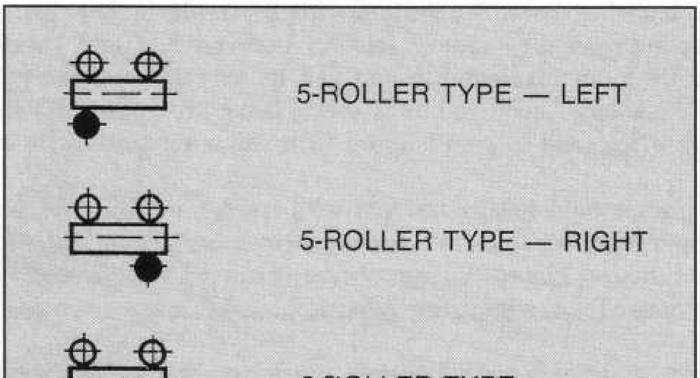


FIGURE 8.4: UNIVERSAL FAIRLEADS WITH ADDITIONAL INBOARD ROLLERS

- *Installation/material.* As mentioned in Section 5, the type of frame construction can seriously affect the ease of installation. Designs with a closed base member effectively connecting the end posts are preferable. Also if the frame is made of steel of higher strength than the adjoining hull structure, elaborate hull reinforcements will be required.

Four-roller and five-roller type fairleads are covered by JIS F2026, but the strength is not given and the listed 'applicable wire rope diameter' is considered large in relation to the roller diameter. The largest, having rollers of 315mm diameter, would be suitable for wire ropes (with IWRC) up to 32mm in diameter. This size would not be suitable for instance for VLCCs. Also, the JIS base is not the preferred type (see Section 5).

All rollers should have lubrication-free bearings (but with grease nipples fitted) and provisions for turning with a hook spanner.

8.7 EYEPLATES

Eyeplates are used in the manifold area for securing hoses (see Section 9) and have various other applications aboard ships. The British Standard BS MA 10 shows suitable eyeplates and includes safe working loads at any angle of pull.

8.8 STOPPERS

The most common loose fitting is a stopper, which is required to temporarily hold the load of a line while the line is in the process of being belayed on a bollard.

Common methods of stopping-off wire or synthetic and natural fibre ropes are shown in Fig. 8.5.

Stopping off a line is a dangerous operation. High loads in the line must be avoided, since the stoppers have less strength than the mooring line. Only a properly sized 'carpenter's' stopper can approach the line's strength.

Standard fibre stoppers for fibre ropes, as well as chain stoppers for wire ropes, are covered by the BSRA Shipbuilding Standard No. SIS 23. Three different sizes are listed for each type, including all details and the SWL. The fibre stoppers are of three-strand polyester rope and may be used on any synthetic or natural fibre mooring rope.

8.9 SELECTION OF FITTING TYPE

At the ship design stage, a decision must be taken regarding the types of fairleads employed at the shipside for use with mooring lines. Roller-type chocks result in less line wear and improve the winch hauling capacity because they reduce friction between line and chock. On the other hand, roller-type fittings require more maintenance and can be very large if the rollers are of proper size for the intended service. A large bend radius is much easier to realise in Panama-type fairleads and for this reason they are often used in combination with winch-mounted wire ropes.

Winch-mounted fibre lines should ideally be used with roller-type fairleads, since the friction created by the fixed fairleads can lead to rapid line damage. However, the current practice of operating with reduced crewing has resulted in closed fairleads being preferred for all rope types, mainly due to the higher maintenance associated with roller fairleads.

Secondary mooring lines associated with bollards (lines not required to meet the mooring restraint requirement listed in Section 2) and tug lines are usually served by fixed fairleads.

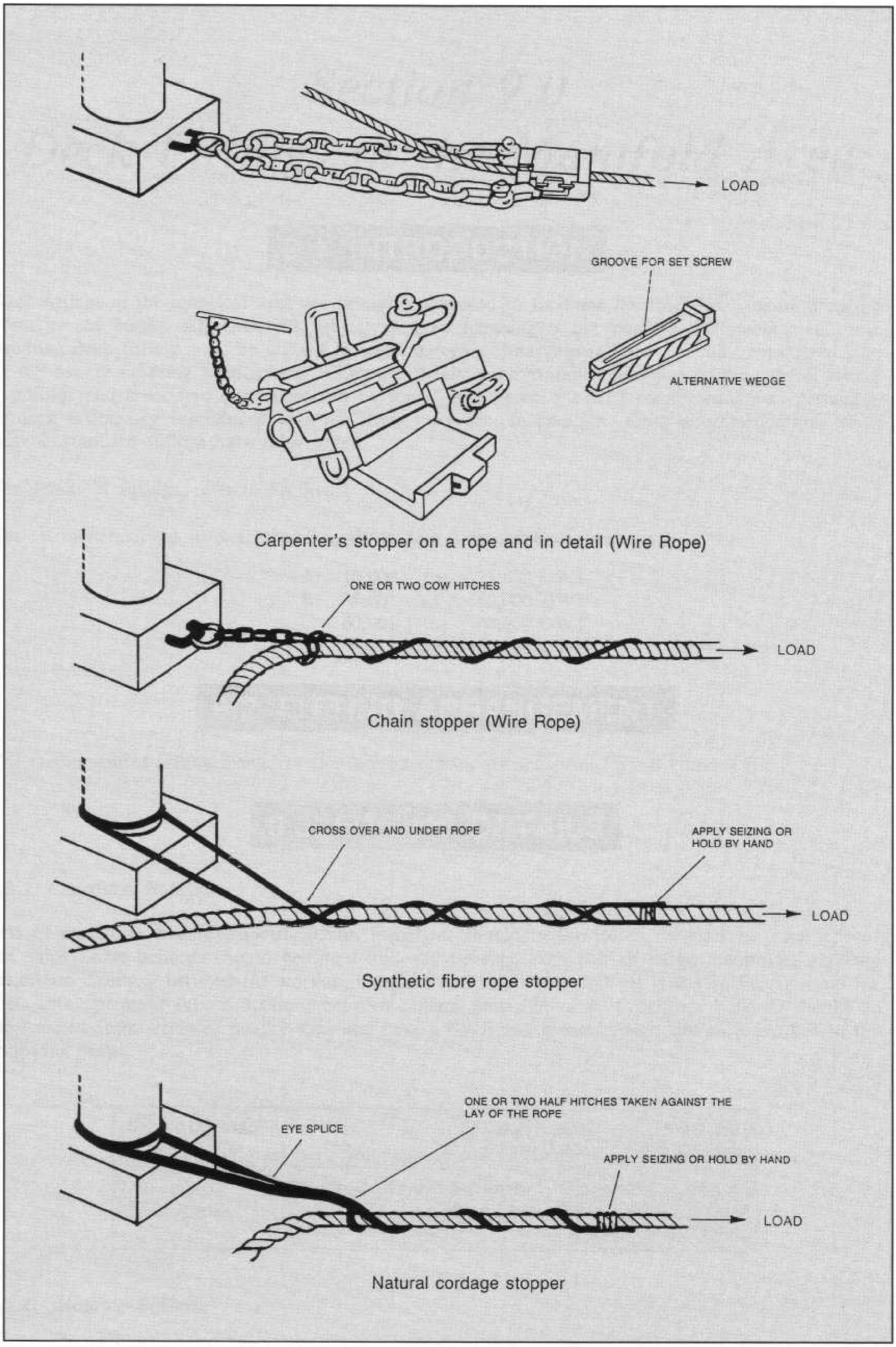


FIGURE 8.5: STOPPERS

Section 9.0
Deck Fittings in the Manifold Area

9.1 INTRODUCTION

Deck fittings in the manifold area are primarily designed to facilitate hoisting and hanging of cargo hoses at sea berths. Although the subject of hose handling is not part of this publication, hose handling deck fittings may be utilised in some mooring situations and are generally considered part of the overall mooring arrangement. Complete details for recommendations pertaining to oil tanker manifolds and associated equipment can be found in Reference 2. The recommendations pertaining to deck fittings are reproduced here for ready reference. In addition, some recommendations as to suitable standard fittings have been added.

For details of fittings, refer to Section 4.

The recommendations in Reference 2 apply to four ship tonnage categories:

A	16,000	to	25,000	DWT
B	25,001	to	60,000	DWT
C	60,001	to	160,000	DWT
D		Over	160,000	DWT

9.2 FITTING ARRANGEMENT

The recommended arrangements for the four categories are shown in Figs. 9.1 and 9.2.

9.3 FITTING DETAILS

9.3.1 *Cruciform Bollards*

Sets of cruciform bollards, about 600mm in height, should be welded to the deck on each side of the ship. These bollards should be fitted mid-way between each pair of cargo manifolds, allowing maximum clearway between the working platform and ship's side and yet giving sufficient room for their safe operation (about 300mm) between bollard and spill tank. Cruciform bollards should be constructed from weldable quality steel and have a SWL and recommended size as prescribed in the following table:

SHIP TONNAGE CATEGORY	SWL	RECOMMENDED STANDARD	NOMINAL SIZE mm
A B C	250 kN (25 t)	ISO 3913 or BS MA 12 Part 2	250
D	400 KN (40 T)	ISO 3913 or BS MA 12 Part 2	315

9.3.2. *Mooring Bollards*

Two pairs of bollards, should be provided on each side of the ship, with one pair located forward and one pair aft of the manifold. The bollards should be conveniently placed near to, but outside

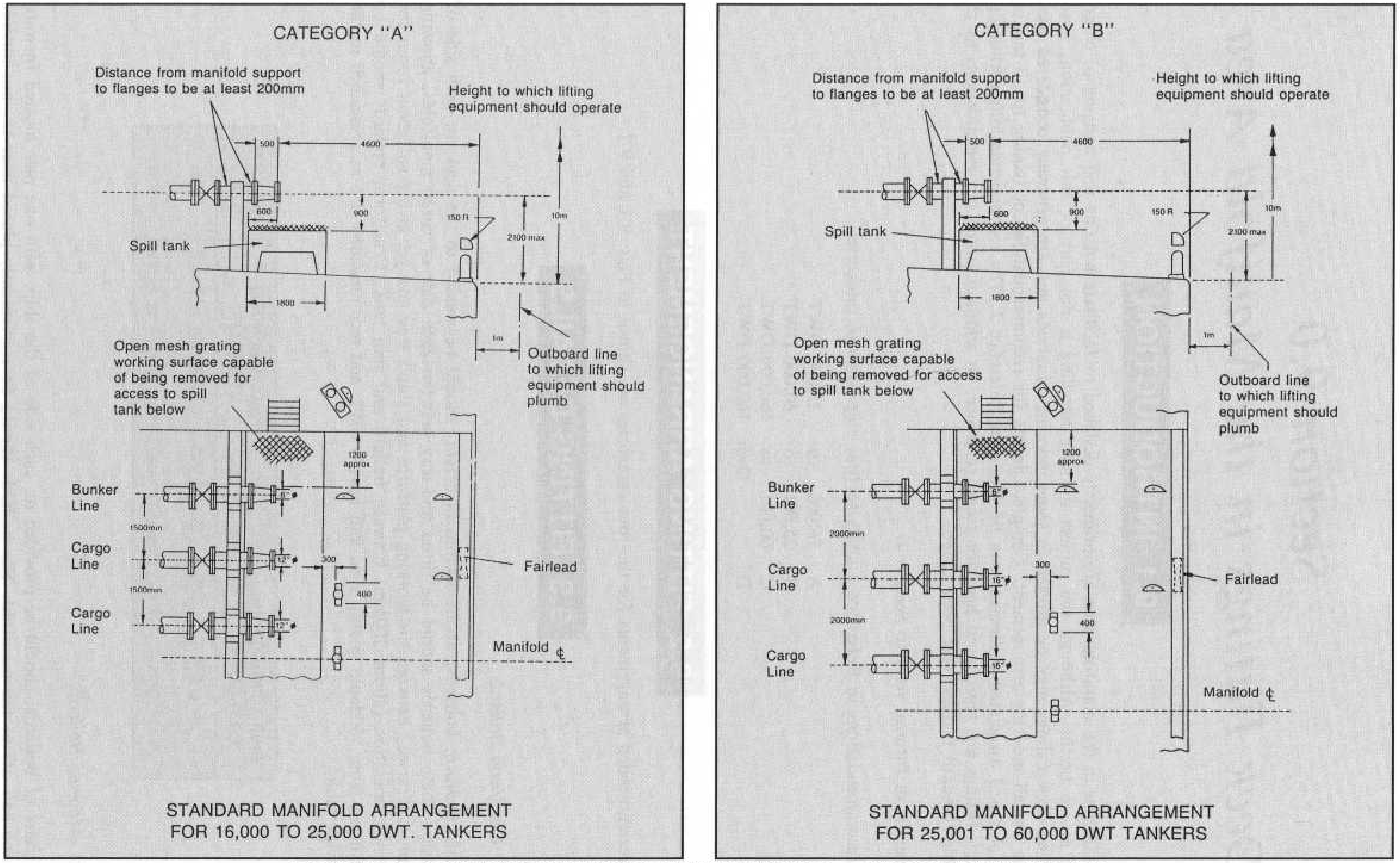


FIGURE 9.1: STANDARD MANIFOLD ARRANGEMENTS — CATEGORY "A" AND "B"
(for manifold incorporating vapour recovery system; see Annex to Reference 2)

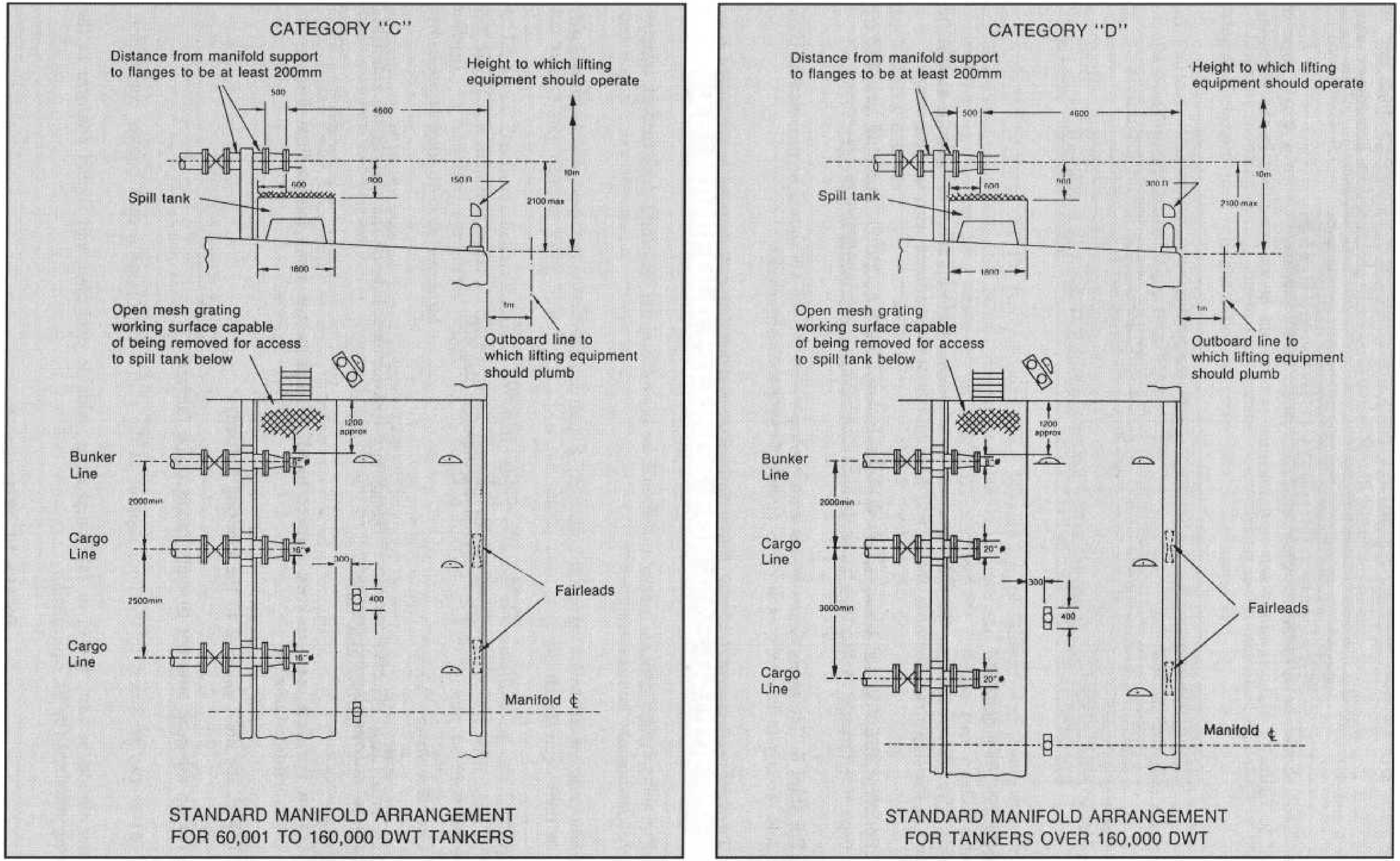


FIGURE 9.2: STANDARD MANIFOLD ARRANGEMENTS — CATEGORY "C" AND "D"
(for manifold incorporating vapour recovery system; see Annex to Reference 2)

of, the deck area between the manifold and the ship's side and in the same longitudinal line as the cruciform bollards. The sets of bollards should be aligned and positioned in accordance with Figs. 9.1 and 9.2.

The bollards should be constructed from weldable quality steel; the SWL (when a single rope is belayed in figure of eight fashion) and recommended size should be as prescribed in the following table.

SHIP TONNAGE CATEGORY	SWL	RECOMMENDED STANDARD	NOMINAL SIZE mm
A B C	250 kN (25 t)	ISO 3913 or BS MA 12 Part 2	300
D	400 kN (40 T)	ISO 3913 or BS MA 12 Part 2	350

9.3.3 *Panama-type Fairleads*

Fairleads with clear openings 400 × 250mm and a minimum surface radius of 180mm should be provided at the ship's side for hose chains and hose hoisting in accordance with the following:

For ships in Categories A and B, two fairleads having a SWL of 250 kN should be provided on each side directly in line with forward and aftermost cargo presentation flanges.

For ships in Categories C and D, fairleads having a SWL of 400 kN should be provided on each side directly in line with each cargo manifold presentation flanges.

The outboard face of fairleads should be in line with the side shell to prevent chafing of hose chains on the ship's structure.

For ships in Category D, a clear run should be provided along the ship's deck to deck winches for leading the hose hoisting wires.

Fairleads should be of cast steel construction and conform to one of the following Standards or their equivalents:

BS MA 19, nominal size 40

JIS F 2017, 1982, nominal size 400

9.3.4 *Oval Eyeplates*

Eyeplates of at least 15t (147 kN) SWL should be fitted on each side of the ship in the following positions (also see Figs. 9.1 and 9.2):

(a) on the deck next to each bunker manifold and in the same longitudinal line as the cruciform bollards

(b) on the deck adjacent to the mooring bollards

(c) on the deck adjacent to each Panama-type fairlead

(d) on the deck next to the hose support at the ship's side opposite each bunker connection.

Eyeplates should be cast, forged or fabricated from weldable steel plate and should conform to the following standard (or its equivalent):

BS MA 10, 1971, size 16

9.4 LIGHTENING ARRANGEMENTS

(Over 160,000 Tonnes DWT)

To facilitate mooring during ship-to-ship transfer operations, it is recommended that ships in tonnage category D be provided with the following (refer also to Reference 4):

(a) Fairleads
Two additional sets of heavy duty closed fairleads with clear openings of 500 × 400 mm should be positioned on the starboard side of the ship. These fairleads should be located 35 m forward and aft of the mid-length of the ship or as close to this position as possible. All fairleads associated with ship-to-ship transfers should be smoothly finished both inboard and outboard to prevent chafing of ropes.

It should be noted that the recommended 500 × 400 mm size of fairlead is not covered by British or Japanese standards (the largest BS fitting is 600 × 320 mm and the largest JIS fitting is 500 × 260 mm). Section 3.9.2 discusses an alternative arrangement utilising two smaller fairleads instead of one 500 × 400 mm fairlead.

(b) Mooring bollards
550 mm diameter twin sets of bollards should be located inboard of the fairleads for handling the moorings used during transfer operations. Suitable standard bollards would be:

ISO 3919 or BS MA 12, Part 1, nominal size 630
JIS F 2001, nominal size 560

9.5 MANIFOLD FITTINGS FOR GAS CARRIERS

Since gas carriers usually moor at berths with articulated chicksan loading arms, the practice has been not to carry any special fittings in way of manifolds to assist in handling hoses.

It is recommended, however, that a limited number of such fittings be included for use in the event that ship-to-ship transfer using hoses handled by the ship's crane is required.

At least two cruciform bollards should be fitted, either between the manifold or one forward and one aft of the manifold area. A closed chock fairlead should be fitted for use with the cruciform bollards. In addition, eye plates similar to those used for oil tankers should be installed in the immediate vicinity of each manifold.

Cruciform bollards should be approximately 300mm diameter and constructed to ISO, BS or equivalent standard (see Section 9.3.1). Closed chocks should be the same as those detailed in Section 9.3.3 for oil tankers.

Appendix A
Design Standard for Tanker Mounted SPM Fittings

The recommendations apply to the number of mooring connections used, the safe working loads of fittings, the dimension of chafe chains and attendant fittings and the type and location of securing devices and fairleads on board. The recommendations only deal with those features that are necessary to ensure the correct matching of equipment used for mooring at SPMs.

A.1 BOW CHAIN STOPPERS

Ships likely to trade to SPMs should be equipped with bow chain stoppers designed to accept 76 mm chafe chain in accordance with the following table:

Ship Size	Number of Bow Stoppers	S.W.L. (tonnes)	S.W.L. (kN)
150,000 tonnes DWT or less (approx. 175,000 displt.)	1	200	1961
Over 150,000 but not greater than 350,000 tonnes DWT (approx. 175,000–400,000 displt.)	2	200	1961
Over 350,000 tonnes DWT (approx. 400,000 displt.)	2	250	2452

Notes:

a) "Tonnes DWT" refers to the deadweight at maximum summer draft.

b) The safety factor on yield of bow stoppers should be 1.50 SWL. The test load should be equivalent to the SWL.

c) The recommendations above are based strictly on ship size as being the only general criteria that can be used. Although terminal operators may increase the SWL of the chafe chain and equipment to take account of local environmental conditions, it is essential that the dimensions of the material remain as detailed in Section 5 of Reference 1, in order that it matches the onboard equipment.

In practice, bow chain stoppers have been found to be safe and easy to use and maintain. The bow chain stopper should be designed to secure standard 76 mm diameter stud-link chain when the chain engaging pawl or bar is in the closed position. The chain stopper should be designed to freely pass a standard 76 mm diameter stud-link chain and associated fittings when the chain engaging pawl or bar is in the open position (See Figures A1 and A2).

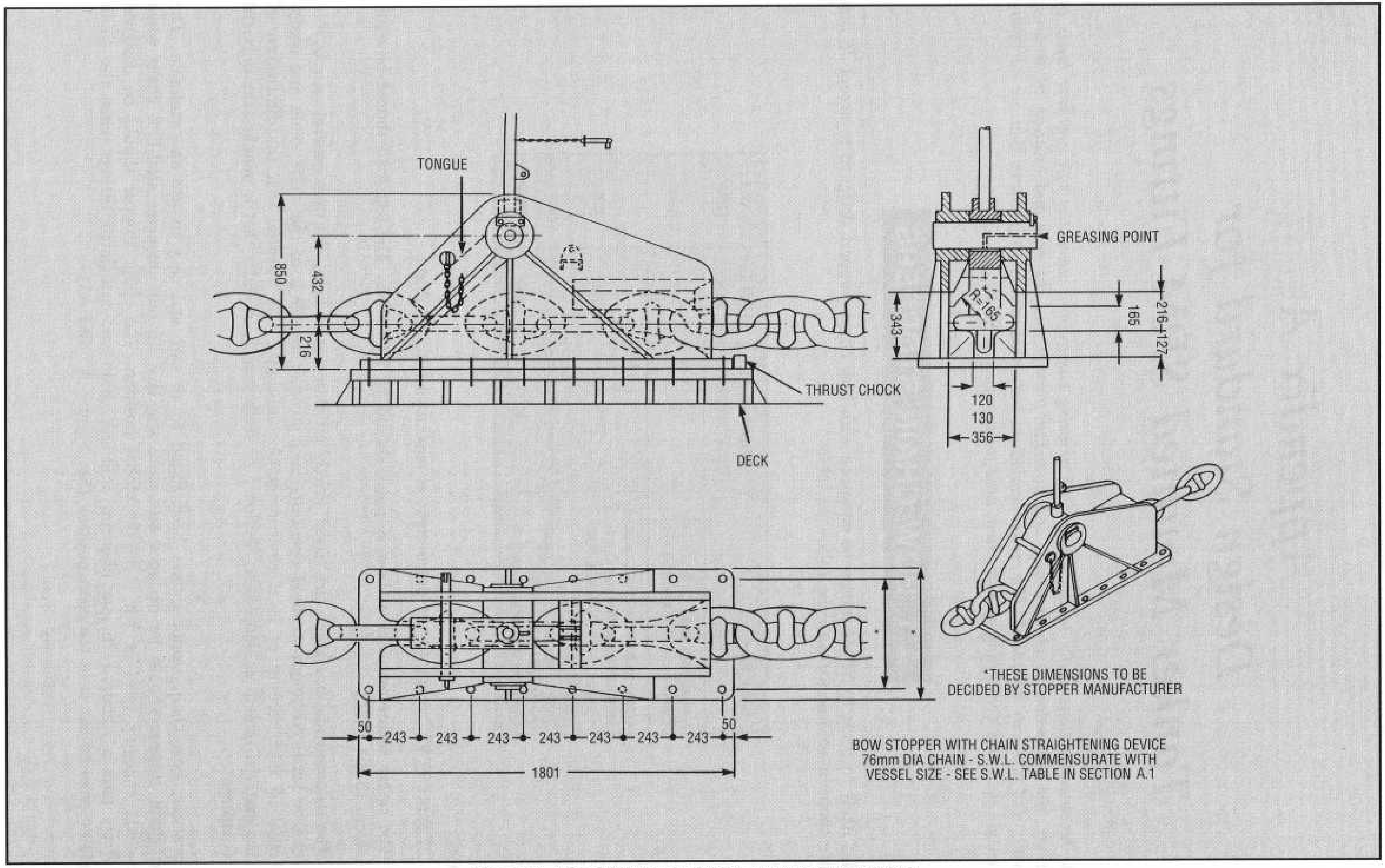


FIGURE A1: TONGUE TYPE CHAIN STOPPER

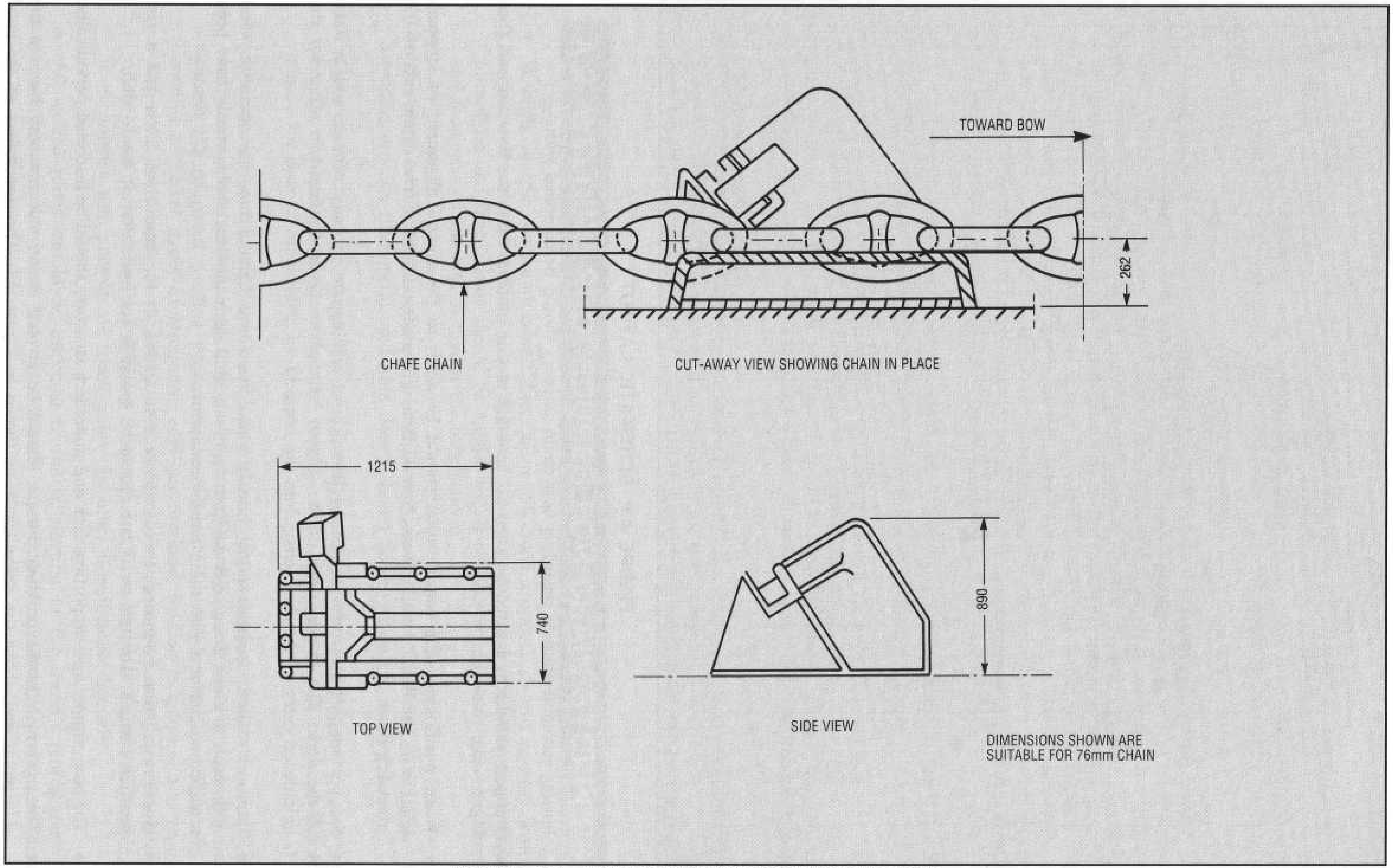


FIGURE A2: HINGED BAR TYPE CHAIN STOPPER

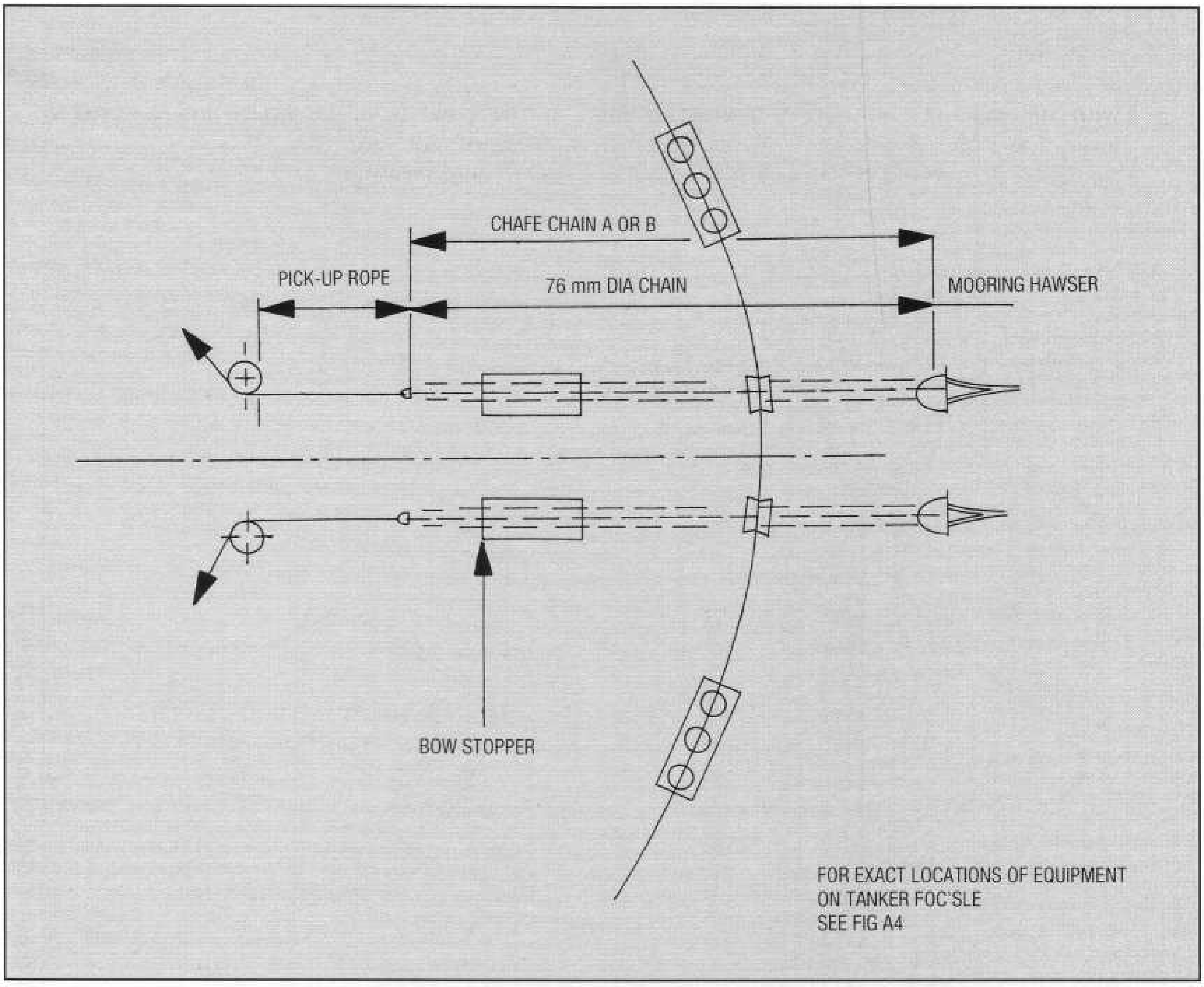


FIGURE A3: SCHEMATIC LAYOUT
(SHIP OF MORE THAN 150,000 TONNES DEADWEIGHT FITTED WITH TWO BOW STOPPERS)

The following recommendations regarding bow chain stoppers should be complied with: –

- Stoppers should be located between 2.7 and 3.7 metres inboard from the bow fairlead (See Figure A4).
- When positioning, due consideration should be given to the correct alignment of stoppers relative to the direct lead between bow fairlead and pedestal lead or the drum end of the winch/capstan.
- Stopper support structures should be trimmed to compensate for any camber and/or rake of the deck. The leading edge of the stopper base plate should be faired to allow for the unimpeded entry of the chafe chain into the stopper.
- Improperly sited stoppers and/or ancillary equipment will hamper mooring operations and may result in ships being rejected from terminals until such time the arrangements have been modified to conform with these recommendations.
- Upon installation bow stoppers should be test loaded to the equivalent SWL and a test certificate issued. The test certificate should be available for inspection on board ship.
- The bow stopper and supporting deck and underdeck structure should be inspected periodically.
- The pawl in a pawl-type chain stopper should be pivoted on an axle mounted between the side plates and above the chain tracks as shown in Figure A1. The dimensions of the pawl

and the position of the axle should be such that the pawl, when resting on the horizontal chain link, forms an angle of approximately 40° with the centreline of the chain. The surface of the pawl which rests against the chain should be concave to essentially conform with the external contour of the chain link. If the pawl is to be lifted and lowered by a lever, the lever should be easily operated by a force of not more than 245 N (55 lb) exerted on its end. The pawl should be provided with a counterweight if necessary to meet this criteria.

- The bar in a bar-type chain stopper should be pivoted on brackets off one of the side plates such that it falls into slots cut in the side plates, as shown in Figure A2. The arrangement and orientation of the slots and the position of the pivot should be such that the side of the bar which rests against the upright chain link makes an angle of approximately 50° with the chain centreline and that the longitudinal axis of the bar is horizontal in the closed position. The surface of the bar which rests against the chain should be concaved to essentially conform with the external surface of the chain link. The bar should be lifted and lowered by a man standing on the deck and be capable of being lifted by a force of not more than 245 N (55 lb) exerted on the handle. The bar should be provided with a counterweight having a striking surface for raising the bar with the blow of a hammer.

- The pawl axle bearing surfaces, the crank mechanism, and the bar pivot surfaces should be designed to resist corrosion and be provided with suitable lubrication points.

- A latching pin(s) or mechanism(s) should be provided to hold the pawl or bar in the closed position such that the chain cannot release and to hold the pawl or bar in the open position such that the chain can run freely. The latching pin or mechanism should be attached to the frame of the stopper by a short chain if it might otherwise be separated from the stopper.

- The chain stopper should be constructed of rolled, cast, or forged steel. Cast iron is not acceptable. If the chain stopper is to be welded to the deck, the steel used in the chain stopper base should be either ordinary-strength or high-strength steel as defined by the Classification Societies.

A.2 WELDING PROCEDURES FOR CHAIN STOPPERS

- Welding in the construction and fabrication of chain stoppers should comply with the requirements of this section, unless approved otherwise by owner. In all instances welding procedures and filler metals should be applied which produce sound welds that have strength and toughness comparable to that of the base material.

- The plans submitted for construction and installation should indicate clearly all welds. The welding process, filler metal, and joint design should be shown on the detail drawings.

- Only qualified welders should be used for the construction and installation. On owner's request, the manufacturer and installer should submit up-to-date records of the qualification papers for each of the welders to be used.

- Procedures for the welding of all joints should be established before construction or installation of the chain stopper. Upon owner's request the manufacturer and installer should submit the proposed welding procedures to be used. The procedures should include the welding processes, types of electrodes, edge preparations, welding techniques, and positions proposed. The procedures should be in accordance with the appropriate parts of Section 30 of the *Rules for Building and Classing Steel Vessels* by the American Bureau of Shipping or in accordance with appropriate specifications by other Classification Societies.

- All welded joints should be inspected to the satisfaction of the owner's representative by established non-destructive test methods such as radiographic, ultrasonic, magnetic-particle, or dye-penetrant inspection.

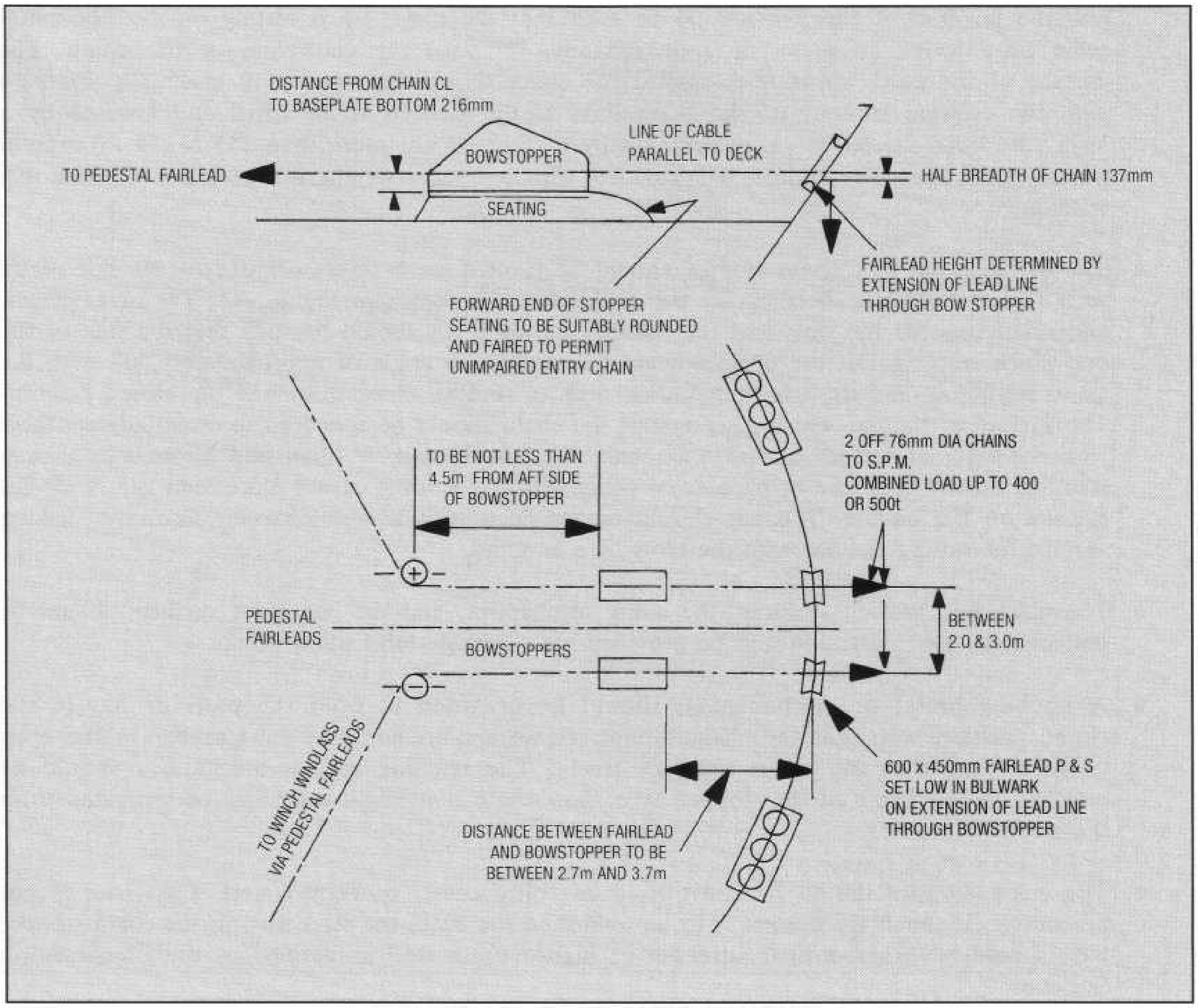


FIGURE A4: POSITIONING OF FORWARD FAIRLEADS, BOW CHAIN STOPPERS AND PEDESTAL ROLLER LEADS

(SHIP OF MORE THAN 150,000 TONNES DEADWEIGHT FITTED WITH TWO BOW STOPPERS)

A.3 BOW FAIRLEADS

In practice, it has been found that the use of a single central fairlead often creates problems when attempting to heave the second chafe chain inboard, as the first chain tends to obstruct the direct line of pull. A certain amount of interaction between mooring hawsers, thimbles and hawser floats also occurs when mooring is effected by a single central lead, which frequently results in chafing and damage to flotation material. As a result, two bow fairleads are recommended for ships of over 150,000 tonnes deadweight at maximum summer draft (equivalent to approx. 175,000 tonnes displacement).

The following recommendations refer to the size, location and type of bow fairleads (See also Figures A4 and A5):

- The chock opening must be large enough to pass the mooring/towing components but not so large that it allows the components excessive lateral or vertical movement.

- All bow fairleads should measure at least 600 x 450 mm. The 600 x 450 mm chock opening of Figure A5 will allow passage of two 76 mm (3 in.) diameter chain links side by side, provided the chain is diagonally oriented. Chain normally comes through the chock with the links diagonal to the chock surface.

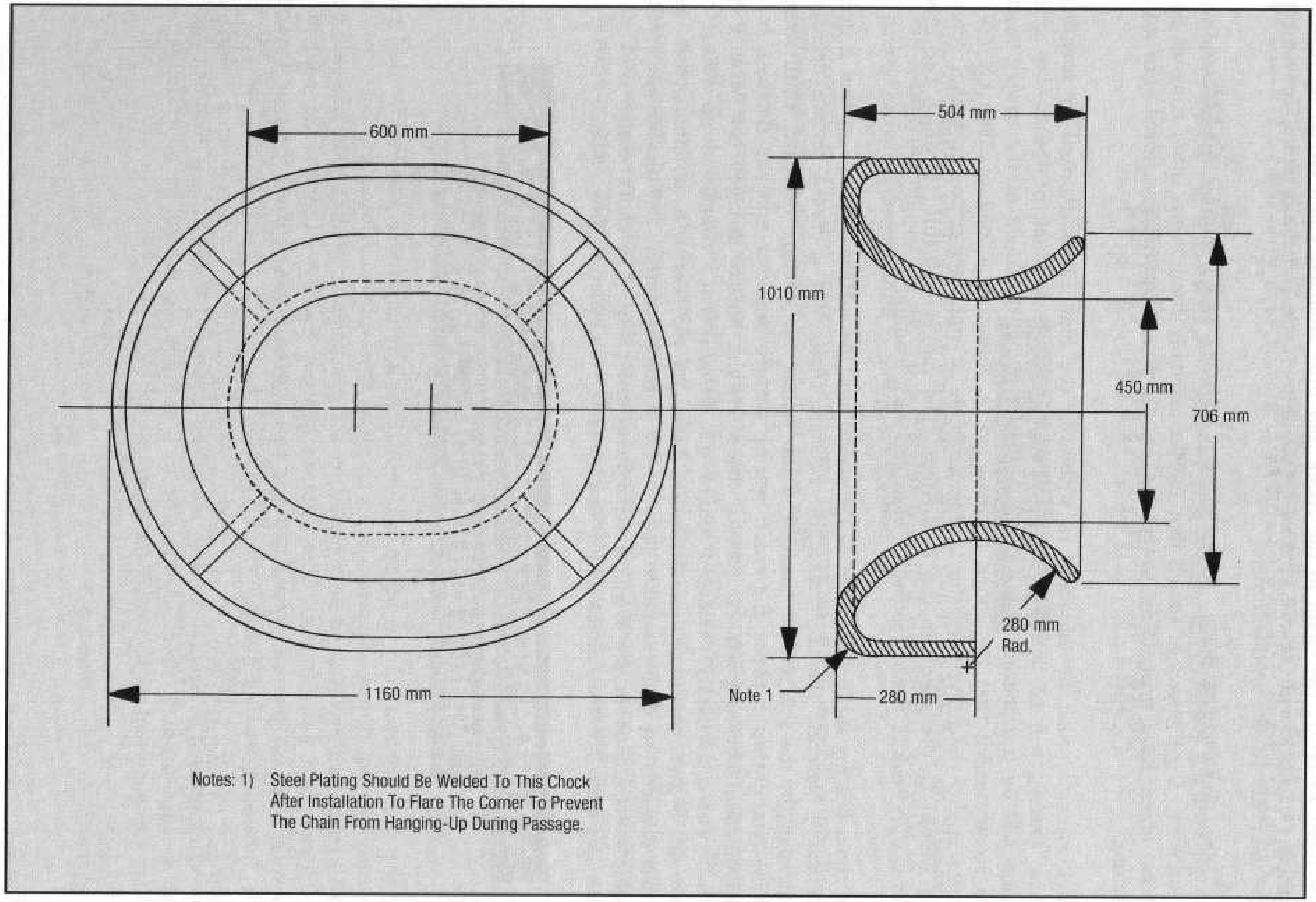


FIGURE A5: TYPICAL CHOCK DESIGN

- Fairleads should be spaced 2.0 metres centre to centre apart, if practicable, and in no case be more than 3.0 metres apart. Ships of 150,000 tonnes deadweight or less at maximum summer draft need only provide one fairlead which should be on the centre line.

- The chock opening should be oval or have rounded corners, as in Figure A5. The curved shape tends to centre the chain in the chock and thus minimize chain lateral movement and abrasion. A chock with square corners would allow excessive chain lateral movement and allow the chain to jam in the corners.

- The outboard lip of the chock should be flush with the tanker hull or smoothly faired into the bulwark. Experience has shown that hawser assembly components often hang-up on bow chocks which have protruding lower lips or which are improperly faired into adjacent structure.

- Chock strength must be adequate to withstand the high loads imposed by towing. The highest loading will be lateral when the tanker yaws. How much of the towing assembly tension will be transmitted to the chock will depend on the angle of the towing assembly. It is recommended that the chock be designed to withstand a load of 3900 kN (397 tons metric) applied laterally, with the towing assembly at right angles to the tanker centreline. This should ensure adequate chock strength. The tanker structure to which the chock is secured may require reinforcement to withstand such loading.

- The chain bearing surface diameter of the chock must be sufficiently large so that it does not significantly decrease the chain strength. Bending chain over a curved surface decreases its axial strength; the sharper the chain bend the greater the strength decrease. Until recently a precise relationship between the chain-bearing surface diameter and the decrease in chain strength could not be given. As part of this study the stresses in chain bent over a curved surface was analytically examined and a test program was conducted on actual chain. Appendix F gives the results of this work. Based on the results, a minimum chain bearing surface diameter to chain diameter ratio of 7:1 is recommended. This should not result in a significant reduction in the breaking strength of the chain. The chock design of Figure A5 meets this ratio for the recommended 76 mm (3 in.) diameter chafing chain.

A.4 POSITION OF PEDESTAL FAIRLEADS, WINCH DRUMS OR CAPSTANS

To enable the mooring operation to be carried out safely, it is imperative that the forward pedestal roller fairleads are positioned correctly. It is essential that roller fairleads/winches/capstans are positioned to enable a direct pull to be achieved on the continuation of the direct lead line between the bow fairleads and bow stoppers. The distance between the bow stoppers and pedestal leads should be considered, so that an unrestricted line pull can be achieved from the bow fairlead and through the bow stopper. (See Figure A4).

Winches or capstans used to handle moorings should be capable of lifting at least 15 tonnes (147 kN). If a winch storage drum is used to stow the pick-up rope, it should be of sufficient size to accommodate 150 metres of 80 mm diameter rope.

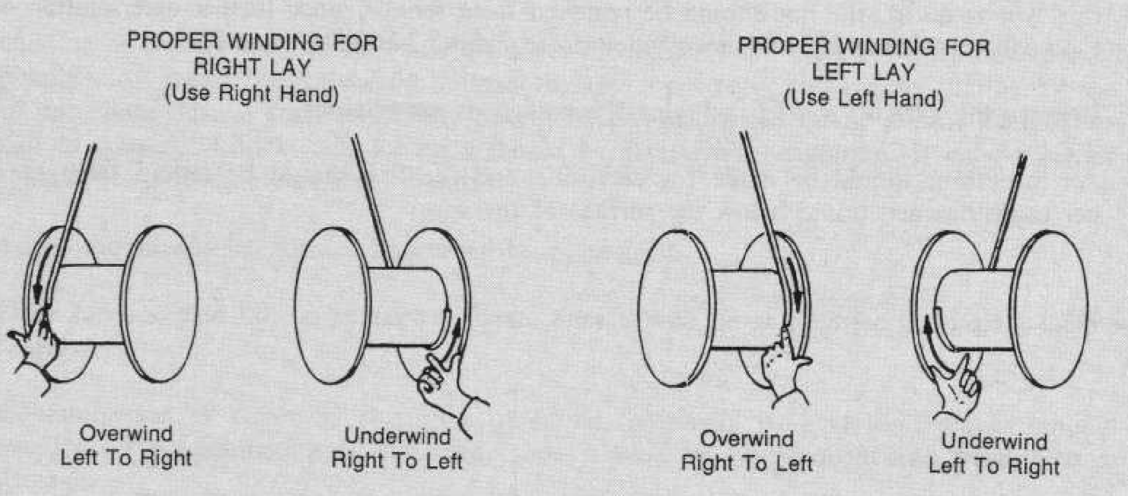
Appendix B
Guidelines for Handling, Inspection and Removal from Service of Wire Mooring Lines

B.1 HANDLING

- Prevent kinking of lines. When unreeling, the reel should be mounted on a spindle and the line pulled directly off the reel, not over the end. If a loop forms, it should be thrown out immediately, before any load is placed on the line.

- The coiling of the first layer of a line on a plain winch drum, depending on the lay of the line, should be in the direction shown in Figure B1.

- Wire lines should be lubricated periodically. Proper lubrication reduces the abrasive effect of individual wires sliding against one another and helps to prevent corrosion. Wire lines are lubricated during manufacture, but this initial treatment is lost during use, particularly in marine applications. Ideally, the line should be lubricated every two or three months. Several patent varieties of wire line oil are available and the lubricant may be brushed on or a box lubricator used. Mooring line manufacturer's recommendations should be followed.



This rule will help to determine the correct lay (left or right) of rope to use on a given drum. The right hand represents right lay rope and the left hand represents left lay rope. The index finger and thumb are extended with the other fingers being closed into a fist. The fist represents the drum, the index finger is the rope coming onto the drum, and the thumb points to the side at which the rope is to be fastened. For overwind the palm is down and for underwind the palm is up.

FIGURE B1: WINDING LEFT AND RIGHT LAY ROPES ON THE WINCH DRUM

- The ends of a wire should be periodically reversed in order to evenly distribute the wear.

- When points of wear develop on wire lines, the wire up to and including the worn sections should be cut off and removed.

B.2 INSPECTION OF WIRE LINES AND GUIDELINES FOR REMOVAL FROM SERVICE

Since wire lines deteriorate in service, regular inspection is necessary to assess damage to the wire and to perform remedial action. Following are some guidelines for inspection:

- The wire diameter should be checked. A marked decrease in wire diameter is a sign that the line should be removed from service. Reasons for the decrease could be core deterioration, internal wear and wire failure, or internal corrosion.

- The wires of the outer layer should be inspected for wear and breaks. If lubrication has been good and operating conditions such that the inside wires are intact, the reserve strength of the inside wires will be maintained. In this case, only a strength reduction corresponding to the broken outer layer wires need be deducted from the rope strength.

- Wires should be checked for abrasion. If the outside wires show a considerable loss of metallic area, the line should be removed from service.

- Individual wires on the strand crowns should be inspected for fatigue breaks. These are characterized by squared-off ends perpendicular to the broken wire. If these wire breaks occur at the strand crowns, the residual wire line strength may be estimated by counting the number of breaks in the length of one lay and subtracting these from the number of wire strands in the line in order to determine the number of remaining unbroken strands. A wire lay is the distance parallel to the longitudinal axis of the rope in which a strand makes a complete turn about the axis of the line. One wire lay has been selected as the length for which to note the breaks because the loss due to a broken wire is effective for approximately this distance.

- Look for fatigue breaks at or near the valley positions of the strands. If one fatigue break of this type is noted, the line should be removed from service, since further deterioration of this type will doubtlessly have taken place where it cannot be easily detected.

- Remove the wire up to and including the kinked or cut sections.

- An inspection should be made for corrosion and the line should be retired from service if corrosion has penetrated below the surface of the wires.

Appendix C
Care, Handling, Inspection and Replacement of Fibre Ropes

C.1 CARE AND HANDLING

Following are some recommendations for the care and handling of fibre ropes:

- New fibre rope of right-hand lay construction should be uncoiled from the centre of the coil in a counter-clockwise direction. When re-coiling it, the rope should be coiled in a clockwise direction. If it is a left-hand lay rope, the opposite would apply. If the rope is on a reel, the reel should be placed on a spindle or a rod to allow the reel to revolve freely. The rope should not be removed from over the end of the reel or while the reel is lying on its side.

- The ropes should be flaked down with as large a flake as possible to avoid kinking the ropes when storing them.

- Excessive build-up of turns in rope or loss of turns should be avoided. Excessive twist should be worked out of the rope by hand before loading.

- A capstan or winch drum rotating clockwise will add turns to a right-hand laid rope and one turning counter-clockwise will remove turns. To avoid this, the rope should frequently be turned end for end on winch drums.

- Ropes should not be dragged over sharp or rough edges, or along the ground, as they could pick up abrasive particles.

- Chafing at chocks and fairleads and on edges of dolphins and piers should be avoided where possible. All metal fittings should be smooth, and ropes protected against chafing by the use of anti-chafe devices such as leather jackets. Winch drums should be as smooth and free of rust as possible. Chocks and fairleads should be in a similar condition. If chocks are of the roller type, they should be free running.

- Ropes should not be exposed unnecessarily to sunlight.

- Fibre ropes should not be exposed to heat. They should never be dried by placing them near heaters.

- Contamination of ropes by chemicals or fumes, especially by acids and alkalis, should be avoided. If contamination is suspected, ropes should be hosed down and washed to avoid damage.

- Sharp bends on a rope should be avoided. Wire lines and synthetic ropes should not be placed on the same bollard or mooring hook.

- Extreme care should be exercised when easing out synthetic ropes from around bitts, cleats or other holding devices, to avoid sudden slipping of the line. Factors contributing to slipping are the low coefficient of friction between synthetic ropes and steel and the large elongation of synthetic ropes under load. Nylon and polypropylene are particularly prone to slipping.

- Due to the high stretch of synthetic ropes, large amounts of energy can be stored in a line under load. Sudden failure of the rope can then result in a potentially dangerous snapping back of the line.

- Mooring ropes should never be knotted. Knots weaken a rope considerably, even after they are removed.

- A left-hand rope should not be coupled to a right-hand rope.

C.2 INSPECTION OF FIBRE ROPES

Fibre ropes lose strength and deteriorate through normal use and must eventually be replaced. Weak points and potential areas of failure can be detected and the line repaired or retired before it parts in service.

For inspection, the rope should be laid out and the inspector should run the rope between his hands, examining about a foot length at a time. As he proceeds, he should rotate the rope and open the strands or spread the yarns to expose the strand interior surfaces and fibres.

C.3 REPLACEMENT OF FIBRE ROPES

The following guidelines will aid in determining when a fibre rope should be replaced:

- *Fibre deterioration*. The rope should be retired if the fibre is breaking up or if powdered fibre is present.

- *Damage due to external wear*. For this purpose, an unused rope sample may be helpful for comparison. If strand crowns are worn down considerably, the rope should be retired. If a significant number of outer yarns are also severed, the rope should no longer be used as a mooring line.

- *Local abrasion*. Heavy chafing or fusion of surface fibres are indications of severe abrasion. If these sections are localised, they can be removed and the rope spliced in accordance with the manufacturer's recommendations.

- *Hockles*. Hockling of fibre ropes indicates a severe reduction in rope breaking strength. The hockle should be cut out, if possible, or the rope removed from mooring service.

- *Chemical attack*. This may be indicated by staining, or by the ease with which filaments or fibres from the yarns can be plucked or rubbed off. If the rope has been chemically damaged, it should be removed from service.

- *Attack by heat*. This may be manifested by glazing of the rope surface. In extreme cases, local fused sections on synthetic rope indicates heat through friction and considerable loss of strength can be expected.

When inspecting mooring lines it is best to be conservative. Cut out damaged places if warranted and splice following manufacturer's recommendations. If damage is not localised, retire the rope.

Appendix D
Example of Force Predictions for Use with Hand Calculation Procedures

Problem:

Determine wind and current force predictions for use with the hand calculation procedure for a 250,000 dwt tanker for the conditions listed below.

Wind load calculations

Step 1: Determine the vessel's characteristics.
Vessel particulars for a 250 kdwt tanker in a fully loaded condition are

A_L (Longitudinal or Broadside Wind Area) = 3225 m^2
A_T(Transverse or Head-on Wind Area) = 818 m^2

Step 2: Obtain wind coefficient at $\theta_w = 150°$

$C_{Xw} = -0.73$ Figure 2.4
$C_{YAw} = +0.19$ Figure 2.3
$C_{YFw} = +0.13$ Figure 2.3

Step 3: Compute wind forces from equations

$$F_{Xw} = C_{Xw}\left(\frac{\rho_w}{7600}\right) V_w^2 A_T$$

$$F_{YAw} = C_{YAw}\left(\frac{\rho_w}{7600}\right) V_w^2 A_L$$

$$F_{YFw} = C_{YFw}\left(\frac{\rho_w}{7600}\right) V_w^2 A_L$$

where:

$\rho_w = 1.223 \dfrac{\text{kg}}{\text{m}^3}$ air at 20°C

V_w = 60 knots

$$F_{Xw} = (-0.73)\left(\frac{1.223}{7600}\right)(60)^2(818) = -346 \text{ kN}$$

$$F_{YAw} = (+0.19)\left(\frac{1.223}{7600}\right)(60)^2(3225) = +346 \text{ kN}$$

$$F_{YFw} = (+0.13)\left(\frac{1.223}{7600}\right)(60)^2(3225) = +243 \text{ kN}$$

Current load calculations

Step 1: Determine vessel characteristics.
Vessel particulars for a 250 kdwt tanker in a fully loaded condition are

$$L_{BP} = 329.3 \text{ m}$$
$$T = 19.2 \text{ m}$$

Step 2: Obtain current coefficient at $\theta_c = 170°$

$$WD/T = 1.1{:}1$$

and that the tanker has a "conventional" bow

$C_{Xc} = 0$		Figure 2.6
$C_{YAc} = -0.175$		Figure 2.5
$C_{YFc} = +0.13$		Figure 2.5

Step 3: Compute the current forces

$$F_{Xc} = C_{Xc} \frac{\rho_c}{7600} V_c^2 T L_{BP}$$

$$F_{YAc} = C_{YAc} \frac{\rho_c}{7600} V_c^2 T L_{BP}$$

$$F_{YFc} = C \frac{\rho_c}{7600} V_c^2 T L_{BP}$$

where:

$$\rho_c = 1025 \frac{\text{kg}}{\text{m}^3} \text{ salt water at } 20°\text{C}$$

$$V_c = 2 \text{ knots}$$

$$F_{Xc} = 0$$

$$F_{YAc} = (+0.175)\left(\frac{1025}{7600}\right)(2)^2(19.2)(329.3) = 597 \text{ kN}$$

$$F_{YFc} = (+0.45)\left(\frac{1025}{7600}\right)(2)^2(19.2)(329.3) = 1534 \text{ kN}$$

Ballasted 250 kdwt tanker

For a ballasted 250 kdwt tanker, the following tanker characteristics are used:

A_T (Transverse or Head-On Wind Area)	$= 1380 \text{ m}^2$
A_L (Longitudinal or Broadside Wind Area)	$= 7622 \text{ m}^2$
L_{BP} (Length between perpendiculars)	$= 329.3 \text{ m}^2$
T (Average Draft)*	$= 6.1 \text{ m}$
Trim (Down by the stern)	$= 1°$

The same equations used for the loaded case apply to the ballasted case also. Only the pertinent characteristics and the wind and current coefficients vary.

*This draft is less than the IMO draft (abt. 8.6 m) assumed for formulae listed in Section 2.4 since the vessel investigated is a pre-MARPOL type.

Summary of wind and current loads for Standard Environmental Criteria

Loaded 250 kdwt tanker

Table showing the wind loads on a fully loaded 250 kdwt tanker for a wind speed of 60 knots at a 10 m elevation.

Force (kN)	*Angle of wind on the stern*								
	0°	30°	60°	65°	90°	100°	120°	150°	180°
F_{Xw}	+353	+304	+137	—	+20	—	−157	−346	−451
F_{YAw}	—	+677	+922	+930	+863	—	+706	+355	—
F_{YFw}	—	+147	+324	—	+481	+500	+451	+243	—

Table showing current loads on a fully loaded 250 kdwt tanker for the current speeds shown and a draft of 19.2 m.

Force (kN)	*Angle and speed of current on the stern average velocity*					
	0°/3 knots	10°/2 knots	80°/3/4 knots	100°/3/4 knots	170°/2 knots	180°/3 knots
F_{Xc}	+276	0	18	−22	0	−268
F_{YAc}	—	+1569	+758	+652	+597	—
F_{YFc}	—	+358	+600	+681	+1467	—

Maximum combination of forces – loaded case

$$F_X = -451 - 268 = -719 \text{ kN from ahead}$$
$$F_{YA} = +930 + 1569 = 2499 \text{ kN}$$
$$F_{YF} = +500 + 1467 = 1967 \text{ kN}$$

Ballasted 250 kdwt tanker (*Wind speed = 60 knots at 10 m height*)

Force (kN)	*Angle of wind on the stern*								
	0°	30°	60°	70°	90°	115°	120°	150°	180°
F_{Xw}	+490	+441	+216	—	—	—	−108	−363	−697
F_{YAw}	—	+1550	+2384	+2420	+2295	—	+1678	+775	—
F_{YFw}	—	+334	+1128	—	+1903	+2190	+2168	+1285	—

Table showing current loads on a ballasted 250 kdwt tanker for the current speeds shown and a draft of 6.1 m.

Force (kN)	*Angle and speed of current on the stern average velocity**					
	0°/3 knots	10°/2 knots	75°/3/4 knots	110°/3/4 knots	170°/2 knots	180°/3 knots
F_{Xc}	+132	+63	+2	0	−47	−93
F_{YAc}	—	+54	+64	+49	+60	—
F_{YFc}	—	+22	+50	+63	+49	—

*based on coefficient for water depth to draft ratio of 1.5:1 since data is not available for the standard ratio of 3.0:1

Maximum combination of forces – ballast case

$$F_X = -697 - 93 = -790 \text{ kN from ahead}$$
$$F_{YA} = +2420 + 64 = +2484 \text{ kN}$$
$$F_{YF} = +2190 + 63 = +2253 \text{ kN}$$

As can be seen the maximum longitudinal force and maximum transverse force at the forward perpendicular are developed in the ballast condition. The maximum transverse force at the aft perpendicular is practically equal in ballast and full load condition. These values compare with those of formulae (10), (9) and (11) in Section 2.4.1 as follows:

Formula (10) $F_X \text{ max} = 0.498 \times 1380 + 0.04613 \times 6.1 \times 329.3 = 780 \text{ kN}$

Formula (9) $F_{YA} \text{ max} = 0.2839 \times 3225 + 0.2482 \times 19.2 \times 329.3 = 2485 \text{ kN}$

Formula (11) $F_{YF} \text{ max} = 0.284 \times 7622 + 0.02595 \times 6.1 \times 329.3 = 2217 \text{ kN}$

As can be seen, the values derived by formulae (10), (9) and (11) are in good agreement with the maximum forces obtained in this example.

The maximum forces can be compared with the restraint capacities developed for a 250 kdwt tanker in Appendix E. This comparison will provide a rough check as to whether the ship, when moored at an idealized mooring arrangement, can stay moored under the recommended design mooring conditions.

Appendix E
Calculation of Mooring Restraint and Loads
Details of Hand Calculation Method Example
Comparison with Computer Method Results

E.1 INTRODUCTION

Reference should be made to Figure E1 which illustrates the general mooring layout and nomenclature.

E.2 BASIC PRINCIPLES OF ANALYSIS

- Mooring line behaviour under load is elastic, i.e., tension is proportional to extension and vice versa. In the absence of pretensioning, the imposition of an external load on a vessel means that the vessel must move away from the fixed shore points for the necessary restraining loads to be developed in the mooring lines. The distribution of tensions between the various lines in a mooring layout is determined by the geometry of the whole layout. Movements of the different parts of the vessel must be compatible, as must be the resultant of the line tensions with the external loads. Individual lines cannot be considered in isolation.
- In lateral restraint the tension developed in any line is proportional to $\frac{\cos \alpha}{L}$, where α is defined as the horizontal angle of a line relative to a true breast and L as the total length of line between jetty hook and vessel winch. The line length is relevant as relative extensions and tensions are required in considering multi-line layouts. The component of this tension effective in resisting the lateral load is proportional to $\frac{\cos^2 \alpha}{L}$.
- The maximum restraint capability of a mooring layout is reached when the most heavily loaded line is at its limiting tension, in design cases 55% (MBL). Beyond this point it would be assumed this line had failed. The imposed load would then have to be resisted by fewer lines. In most cases they would not be able to sustain it and a sequential failure would ensue. The maximum restraint capability therefore includes a factor for the tension in the critical line indicated by subscript "c".
- Vertical line inclinations defined by θ the angle of a line above horizontal, affect tensions and restraint capabilities in the same manner as horizontal angles defined by α.
- By corollary the efficiency of any line in resisting an imposed load is related to the geometry of the critical line as well as its own. These concepts may be illustrated by study of an apparently paradoxical example. For a jetty with all main mooring points on a single line parallel to the vessel's heading, the lateral restraint provided by two true breast lines, one angled head and one angled stern line is less than that of two angled head and two angled stern lines.

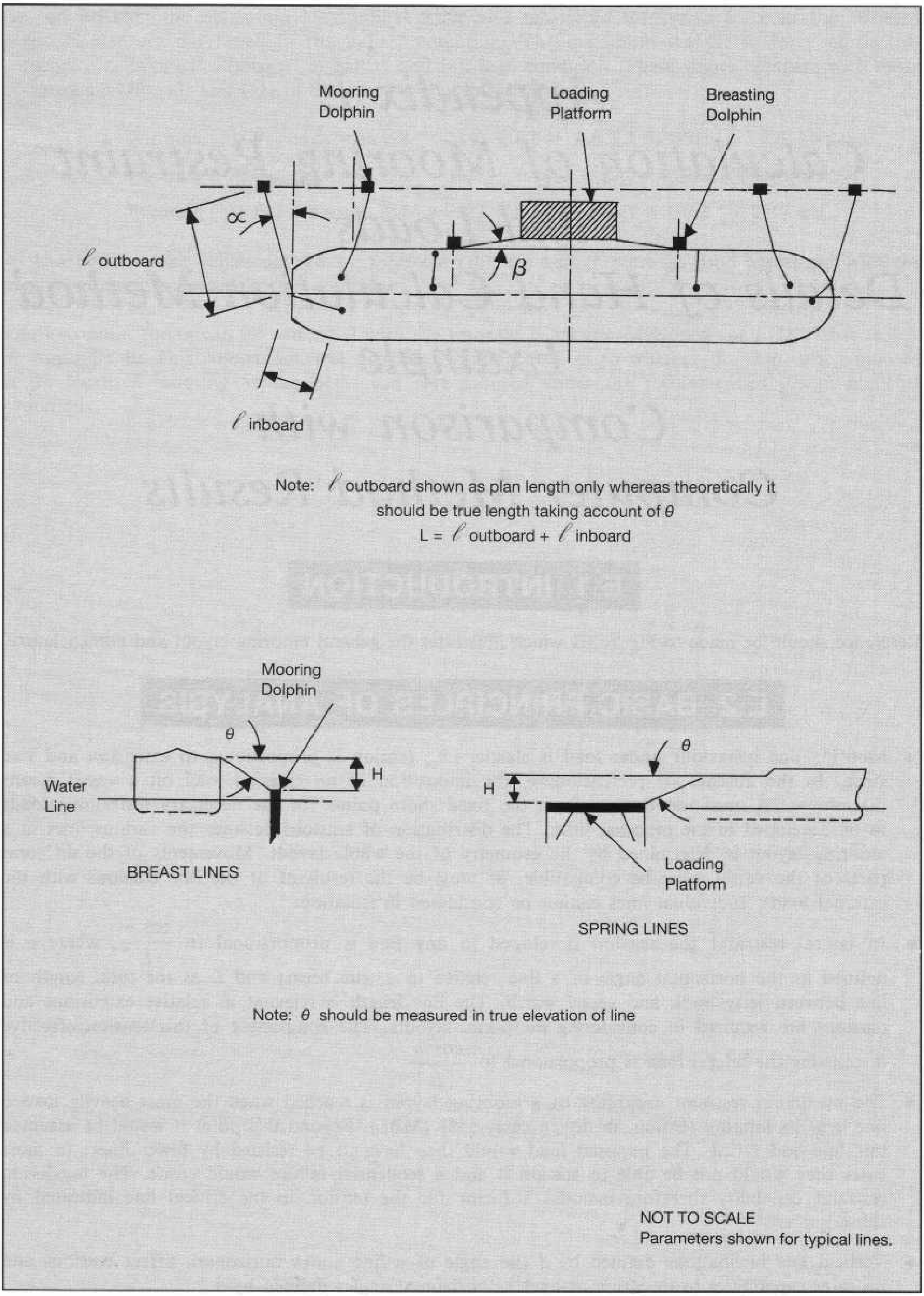


FIGURE E1: VLCC BERTH MOORING LAYOUT NOMENCLATURE FOR HAND AND CALCULATION METHOD

E.3 ASSUMPTIONS

1. The problems may be fairly analysed in two separate parts by resolution of the loadings into lateral and longitudinal components.

   — Generally moorings are arranged in accordance with this concept as discussed in Sections 1.3 and 1.5. Longitudinal restraint should be provided by springs and lateral restraint by breast lines. While angled head and stern lines are not recommended, they can be taken into account in the lateral restraint case, if they are tended like breast lines.

   — Angled head and stern lines by virtue of their orientation and length take up lesser tensions than well placed springs and breasts. Provided they are arranged more as lateral restraint lines, i.e., α less than 45°, their effects on the case of longitudinal loading are not generally significant.

   — Provided that the mooring line layout is reasonably symmetrical about the jetty centre-line, in the lateral loading case the longitudinal components of head and stern line tensions cancel out.

2. Mooring line tension is proportional to extension.

   — In fact, load/extension characteristics are not linear at very small or very large loads, but with a small pre-tension applied they do not deviate significantly from linearity in the zone of interest up to 55% (MBL).

   — A small pre-tension, generally less than 10% (MBL), effectively removes the sag from a line and allows this complication to be ignored.

3. When the imposed loads are zero, the tensions in the mooring lines are assumed to be zero, although the lines are just taut. For the purposes of the calculation, no further tending of lines takes place as external loads are imposed.

   — While some pre-tension is required to remove sag and it is good practice to apply a significant pretension to reduce vessel movement under moderate load fluctuations, uneven application or poor line tending do nullify the benefits as far as ultimate restraint capacity is concerned.

   — In practice, values of pre-tension are seldom more than 20% (MBL), and measurements indicate that they are usually substantially lower.

   — The assumption is slightly conservative in its results but is probably reasonably representative of the majority of situations.

4. The problem of lateral restraint calculation may be further split into two parts by considering the bow and stern of the vessel separately.

   — Lateral restraint lines are usually grouped near the bow and stern of the vessel.

   — Generally, imposed loadings at bow and stern are not too dissimilar, so angular displacements are not very significant.

5. In "mixed" mooring layouts synthetic lines are ignored.

   — By comparison with wires they take up insignificant tensions due to their much greater elasticity.

   — Any benefit from pre-tensioning is excluded by assumption 3.

E.4 LIMITATIONS OF METHOD

In general, the method is applicable with reasonable accuracy to effective mooring layouts, in particular:

— Lateral and longitudinal restraint functions separated.

- Reasonably symmetrical layouts.
- Vertical line angles (θ) less than 25°.
- Head and stern line deviations from breast (α) less than 45°.
- All lines of similar material, construction, make-up and size.
- Lateral lines effectively grouped at head and stern.

It must be understood that this method only covers static strength requirements for a mooring system. It does not make allowance for dynamic effects, neither does it evaluate vessel movements under imposed loading. These must be considered separately.

E.5 CALCULATION PROCEDURE

(Note: for spring lines substitute β for α)

1. Divide the mooring lines into groups for the different functions, i.e., lateral restraint at bow and at stern and longitudinal restraint forward and aft.

2. Assess the relevant parameters α, θ, L, = 1 (outboard) + 1 (inboard) for each line. Depending on the accuracy required and personal preference, scale drawings or calculations may be used. Secondary parameters such as H, the height difference between jetty hook and vessel fairlead, may be introduced. The results are best tabulated as shown in the examples. As regards accuracy, the procedure already includes a number of approximations which for most layouts have little effect. Further approximations may be made at this stage, e.g.,
   - Take average values for grouped lines.
   - Take only plan lengths of lines, ignoring correction for slope.

3. Evaluate and tabulate $\cos\alpha$, $\cos\theta$ for each line.

4. Evaluate and tabulate $\dfrac{\cos^2\alpha}{L}$ for each line and total (symbol Σ) for each group. In the examples a constant of 100 is introduced for convenience. It is eliminated in the last line of the calculation.

5. Assess for each group the critical line, i.e., the one that will break or slip first. This is the line with the highest value of $\dfrac{\cos\alpha}{L}$. It is the most efficiently placed line, with low values of α and L. Subscript c is used to identify the relevant parameters.

6. Final step for each group:

   lateral restraint

   $$R_Y = 0.55\ (\text{MBL}) \cdot \Sigma \frac{\cos^2\alpha}{L} \cdot \frac{L_c}{\cos\alpha_c} \cdot \cos\theta_c$$

   which has to resist the imposed load.

   The nominal restraint capability of the lines is:

   $$\text{nom } R_Y = 0.55\ (\text{MBL}) \cdot N \text{ where } N = \text{no. of lines}$$

   $$\text{efficiency of layout} = \frac{\text{nom} R_Y}{R_Y}$$

   longitudinal restraint may be calculated by a similar formula, but often the lines have very similar values of L, β, θ and an approximation may be used:

   $$R_X = 0.55\ (\text{MBL}) \cdot \cos\beta_c \cdot \cos\theta_c$$

The number of lines required to hold a vessel in a given situation would have to be assessed by trial and error. Very approximate solutions may be obtained by use of the formulae listed in Section 2.6. These may then be verified by a full calculation.

Sample calculations have been worked out for a 250 kdwt tanker for both the loaded and ballasted conditions. These are shown on pages 167 and 169.

E.6 POSSIBLE REFINEMENTS OF CALCULATION

1. When all the lines are wires fitted with synthetic tails, their elastic properties and consequently the load distribution are altered. Each tail may be considered as an equivalent wire whose length is greater than that of the tail in proportion to the ratio of percentage elongations of the tail and the wire at the same load (this allows for tail MBL greater than wire MBL). In the calculation only L has to be modified. For example, if each line is fitted with an 11 m tail and the ratio of percentage elongations is 10, then $11 \times (10 - 1) = 99$ m is added to the length of each line already assessed.

2. Where a shore pulley is provided to take a bight of a vessel's line, both legs are effective and assume the same tensions. In the calculation for each leg the average values of $\cos\alpha$, $\cos\theta$ and L for the two legs should be used.

   Caution is required if tails are included as the tail on the one line is effectively shared by the two legs.

3. In ill-conditioned layouts with several lateral restraint lines taking up vertical inclinations of 25° or more, the accuracy may be improved by use of a more complicated formula:

   critical line has highest value of $\dfrac{\cos\alpha \cdot \cos\theta}{L}$

$$R_Y = 0.55\ (\text{MBL}) \cdot \Sigma \frac{\cos^2\alpha \cdot \cos^2\theta}{L} \cdot \frac{L_c}{\cos\alpha_c \cdot \cos\theta_c}$$

EXAMPLE – LOADED VESSEL

250 000 dwt tanker on 19.2 m draft
14 no. 42 mm wires of MBL 1110 kN
Layout as shown in Figure E.2.

Line	$\alpha°$	l_{out} *plan*	H	$\theta°$	$l_{out} + l_{in} = L$ *true*	$\cos\theta$	$\cos\alpha$	$100\dfrac{\cos\alpha}{L}$	$100\dfrac{\cos^2}{L}$
1	15.0	47	3.4	4.1	47 + 6 = 53	0.997	0.966	1.82	1.76
2	17.3	46	3.4	4.2	46 + 6 = 52	0.997	0.955	1.84*	1.75
3	17.2	41.5	3.4	4.7	41.5 + 42.5 = 84	0.997	0.955	1.14	1.09
4	14.2	37.5	3.4	5.2	37.5 + 44 = 81.5	0.996	0.969	1.19	1.15
5	12.4	35.5	3.4	5.5	35.5 + 35 = 70.5	0.995	0.977	1.39	1.35
									Σ = 7.10
6	9.0	68	3.4	2.9	68 + 35 = 103	0.999	0.988		
7	8.8	69.5	3.4	2.8	69.5 + 35 = 104.5	0.999	0.988		
8	10.2	60.5	3.4	3.2	60.5 + 35 = 95.5	0.998	0.984		
9	10.4	59	3.4	3.3	59 + 35 = 94	0.998	0.984		
10	11.0	40	3.4	4.9	40 + 32 = 72	0.996	0.982	1.36	1.34
11	12.8	41.5	3.4	4.7	41.5 + 32 = 73.5	0.997	0.975	1.33	1.29
12	16.0	44	3.4	4.4	44 + 7.5 = 51.5	0.997	0.961	1.87*	1.79
13	10.2	58.5	3.4	3.3	58.5 + 6 = 64.5	0.998	0.984	1.53	1.50
14	10.8	65	3.4	3.0	65 + 6 = 71	0.999	0.982	1.38	1.36
									Σ = 7.29

*Indicates critical line of group.

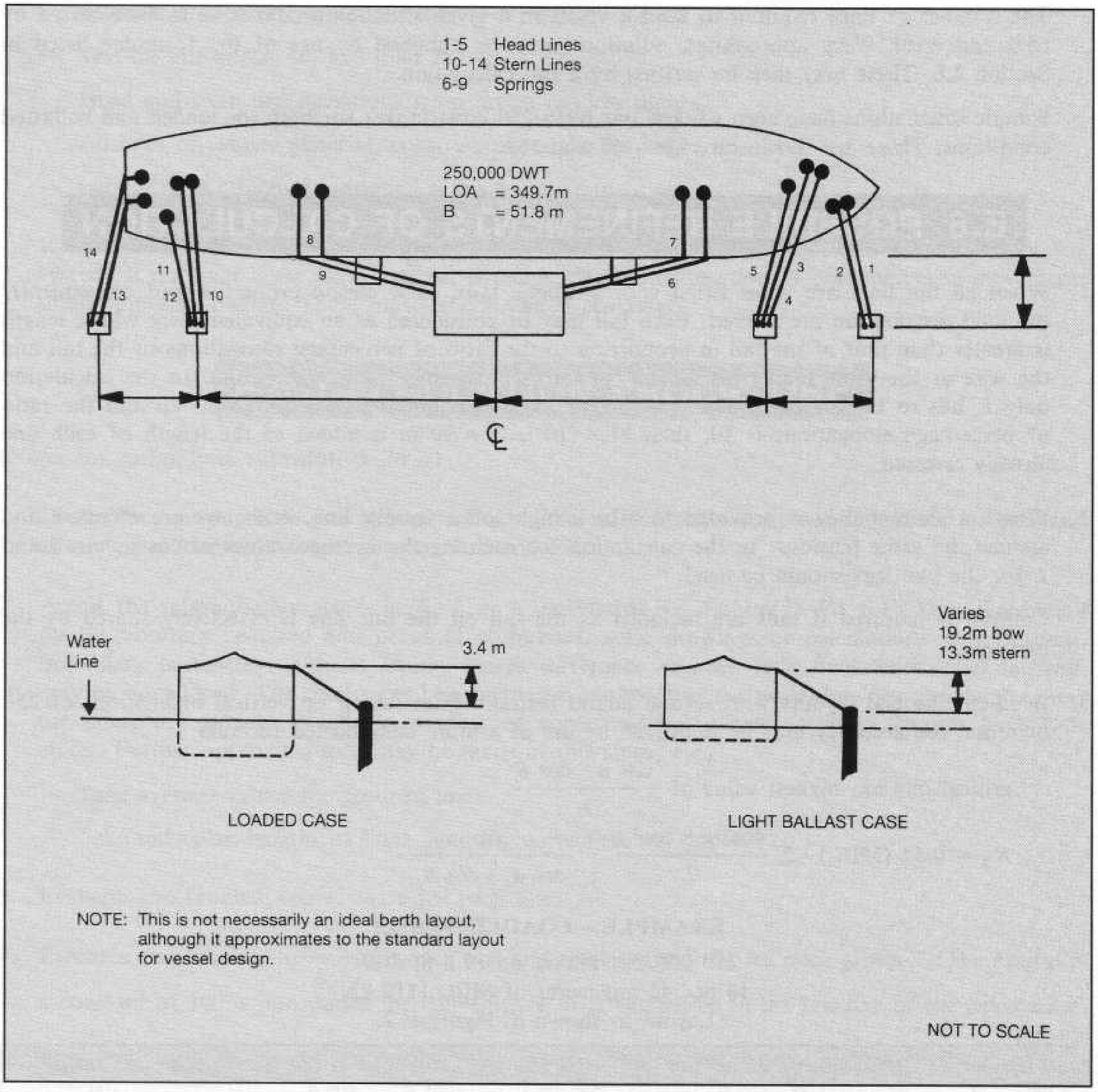


FIGURE E2: MOORING LAYOUT USED FOR COMPUTATIONAL PURPOSES

$$R_Y = 0.55\ (\mathrm{MBL})\left(\Sigma \frac{\cos^2 \alpha}{L}\right)\left(\frac{L_c}{\cos \alpha_c}\right) \cos \theta_c$$

$$R_{YF} = 0.55\ (1110)(7.10)\left(\frac{1}{1.84}\right)(0.997) = 2350\ \mathrm{kN}$$

$$R_{YA} = 0.55\ (1110)(7.29)\left(\frac{1}{1.87}\right)(0.997) = 2373\ \mathrm{kN}$$

$$R_X = N\ (0.55)(\mathrm{MBL}) \cos \beta \cos \theta$$

As variation in L, $\cos \beta$ and $\cos \theta$ is less than 2% the simpler approximate formula may be used giving:

$$R_{XF} = 2\ (0.55)(1110)(0.988)(0.999) = 1205\ \mathrm{kN}$$

$$R_{XA} = 2\ (0.55)(1110)(0.984)(0.998) = 1200\ \mathrm{kN}$$

EXAMPLE – LIGHT BALLASTED VESSEL

250 000 dwt tanker, lightest operational draft 6.1 m, trim 1%

14 no. 42 mm wires of MBL 1110 kN

Layout as shown in Figure E.2.

Note: As values of θ go over 25° and are larger and more variable than those of α, the method uses refinement 3 described in the text.

Line	$\alpha°$	l_{out} *plan*	H	$\theta°$	$l_{out} + l_{in} = L$ *true*	$\cos\theta$	$\cos\alpha$	$100\dfrac{\cos\alpha\cos\theta}{L}$	$100\dfrac{\cos^2\alpha\cos^2\theta}{L}$
1	15.0	47	19.2	22.2	51 + 6 = 57	0.926	0.966	1.57*	1.40
2	17.3	46	19.2	22.6	50 + 6 = 56	0.923	0.955	1.57*	1.39
3	17.2	41.5	18.8	24.4	45.5 + 42.5 = 88	0.911	0.955	0.99	0.86
4	14.2	37.5	18.8	26.6	42 + 44 = 86	0.894	0.969	1.01	0.87
5	12.4	35.5	18.8	27.9	40 + 35 = 75	0.884	0.977	1.15	0.99
									$\Sigma = 5.52$
6	9.0	68	18.0	14.8	70.5 + 35 = 106	0.967	0.988		
7	8.8	69.5	18.0	14.5	72 + 35 = 107	0.968	0.988		
8	10.2	60.5	14.7	13.7	62 + 35 = 97	0.972	0.984		
9	10.4	59	14.7	14.0	61 + 35 = 96	0.970	0.984		
10	11.0	40	13.7	18.9	42.5 + 32 = 74.5	0.946	0.982	1.25	1.16
11	12.8	41.5	13.7	18.3	43.5 + 32 = 75.5	0.950	0.975	1.23	1.14
12	16.0	44	13.7	17.3	46 + 7.5 = 53.5	0.955	0.961	1.72*	1.57
13	10.2	58.5	13.3	12.8	60 + 6 = 66	0.975	0.984	1.45	1.39
14	10.8	65	13.3	11.6	66.5 + 6 = 72.5	0.980	9.82	1.33	1.28
									$\Sigma = 6.54$

*Indicates critical line of group.

$$R_Y = 0.55\ (\text{MBL})\left(\Sigma\frac{\cos^2\alpha\cos^2\theta}{L}\right)\left(\frac{L_c}{\cos\alpha_c\cos\theta_c}\right)$$

$$R_{YF} = 0.55\ (1110)(5.52)\left(\frac{1}{1.57}\right) = 2147\ \text{kN}$$

$$R_{YA} = 0.55\ (1110)(6.54)\left(\frac{1}{1.72}\right) = 2321\ \text{kN}$$

$$R_X = N\ (0.55)(\text{MBL})\cos\beta\cos\theta$$

$$R_{XF} = 2\ (0.55)(1110)(0.988)(0.967) = 1167\ \text{kN}$$

$$R_{XA} = 2\ (0.55)(1110)(0.984)(0.970) = 1165\ \text{kN}$$

Comparison of predicted forces and hand calculated restraint capacity

Appendix D predicts the following maximum forces for the assumed conditions:

Max $F_X = -790$ kN (in ballast conditions)

$F_{YF} = +2253$ kN (in ballast conditions)

$F_{YA} = +2499$ kN (in full load conditions)*

*Practically the same as ballast conditions.

The minimum restraint capabilities calculated in this section are for the ballasted case and are as follows:

R_{XA} = 1165 kN

R_{XF} = 1167 kN

R_{YF} = 2147 kN

R_{YA} = 2321 kN

The longitudinal restraint capabilities fore and aft are clearly adequate. The lateral restraint capabilities fore and aft are about 5 % and 7 % deficient respectively.

Comparison of mooring line requirements with approximate formulae in Section 2.6

Formula (22): S × MBL = 3271 kN

Formula (23): BR × MBL = 12923 kN

Formula (24): S × MBL = 3555 kN

Formula (25): BR × MBL = 12620 kN

This compares with the available S × MBL of 4 × 1110 = 4440 kN, and BR × MBL of 10 × 1110 = 11100 kN.

In all cases the provided spring lines are more than adequate. The breast lines are 14% deficient per formula (23) and 12% deficient per formula (25).

E.7 SAMPLE OF COMPUTER CALCULATIONS

For the 250 kdwt tanker carrying 14 wire lines, computer runs were made for the 60 knot wind from 0° to 180° and for the currents as given in the design mooring condition section. Both the ballasted and loaded cases, with a water depth to draft ratio of 1.1:1 for the loaded case, and 3.0:1 for the ballast case were analysed, and the maximum loadings in the 42 mm IWRC wire lines are as provided in the table below: Details as to wind area, draft etc. are found in Appendix D in which the forces acting on the vessel are calculated. Figure E2 illustrates the mooring line layout used for computational purposes. Vertical and horizontal angles are as illustrated in Figure 2.10.

Wire Number	*Load in Tonnnes*	*Wire Number*	*Load in Tonnes*
1	63.9*	8	48.1
2	63.5*	9	48.5
3	47.6	10	45.4
4	43.1	11	44.4
5	48.5	12	59.4
6	54.9	13	60.8
7	54.4	14	56.2

*Slightly above 55% MBL: but well within order of accuracy of calculations.

- The loads given in the table are the highest loads generated in each line by the varying combinations of wind, current and vessel draught.
- Normal allowable load = 0.55 × 1110 kN = 610 kN
- 1110 kN is minimum breaking load of the 42 mm IWRC wire line which is recommended as the minimum sized wire line for VLCCs.

Thus, based on our design mooring conditions and assumed ballast condition, 14 wire mooring lines as specified are required to safely moor the vessel.

Comparison of hand with computer method results from example

It can be seen that the results from the hand method are slightly worse than those given by the computer method. This measure of agreement should be expected for a good mooring layout. For ill-conditioned layouts the hand method will be less accurate.

The results obtained by the approximate formulae in Section 2.6.1 are slightly conservative since most mooring layout are not as efficient as the layout assumed in this example.

Appendix F
Strength of Chain Tensioned over a Curved Surface

The chafing chain at the end of a tow line or SPM hawser normally is guided through a chock at the edge of the tanker deck. This chock restrains the chain from excessive lateral movement and provides a curved surface for the chain to bear against. The diameter of this curved surface must be large enough to prevent overstressing of the chain when it is subjected to high towing or mooring loads. The problem of defining a criteria for the minimum diameter of chock surfaces was addressed by both analysis and experiment in this study.

F.1 THREE CASES OF CHAIN ON A CURVED SURFACE

Analysis of the stresses in a chain tensioned over a curved surface is more complex than first appears. Three cases of chains tensioned over curved surfaces are illustrated in Figure F1. Case 1 is that of an ungrooved surface, such as a chock. The chain links lie at angles to the surface, alternatively lying to one side and then the other, with all links bearing against the surface. This is the case of primary interest in this study.

The second and third cases apply to grooved surfaces, such as anchor chain windlasses and chain sheaves. On grooved surfaces, every second chain link projects into a groove in the surface and has its transverse axis perpendicular or upright to the surface. Intervening links lie with their transverse axis parallel or flat on the surface. In case the groove is so shallow that the upright link rests on the bottom of the groove and the flat link is lifted tree of the surface. In this case bending stresses are exerted on the upright links only. In case 3 the groove is so deep that the upright links do not touch bottom, and the flat links bear against the curved surface. Only the flat links are subjected to bending stresses in this case.

Grooved surfaces are discussed to distinguish them from the ungrooved case and to allow discussion of other analyses and recommendations. An intermediate case can exist in which both the flat and upright links touch the surface. In the case of chain lying over a very small ungrooved surface with only one or several links in contact, alternating links may lie flat and upright to the surface as in the second and third cases.

F.2 DEFINITION OF ANGLES AND OTHER TERMS

Dimension and angles of chain links lying on a curved surface are shown in Figure F1. The chain diameter, d, is the nominal diameter of the bar from which the common stud link is formed. For common stud links the overall link length L is 6 times d and the link with B is 3.6 times d.

The diameter of the curved surface over which the chain is bent will be defined as D. This must not be confused with the diameter of the opening in the bow chock. For case 2, the shallow-grooved surface with contact at the bottom of the groove. D is analyzed as the diameter to the bottom of the groove.

The ratio of bending surface diameter to chain diameter D/d will be used as a criteria for surface diameter. This diameter ratio should not be confused with the ratio of the surface radius to chain diameter which is sometimes referred to in recommendations made by others.

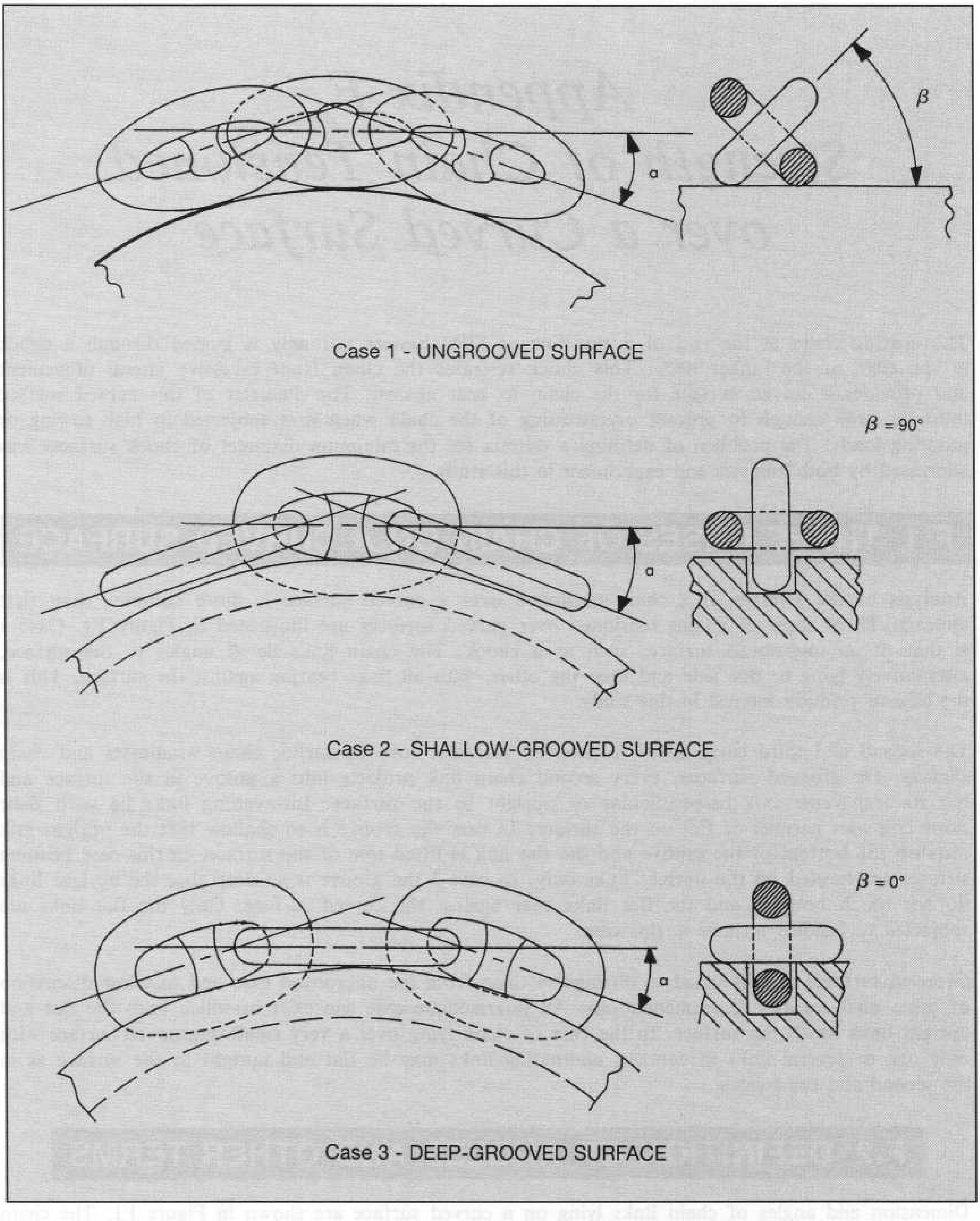


FIGURE F1: THREE CASES OF A CHAIN BENT OVER A CURVED SURFACE

Certain angles, shown in Figure F2, are of interest in analyzing chain tensioned over a curved surface. The angle between the centrelines of two adjacent chain links will be defined as α. This is also the angle subtended by rays from the centre of surface curvature to the centres of adjacent links. The angle between the transverse axis of a chain link and the surface will be defined as β. The angle α can be found as a function of the diameter ratio D/d and the angle β by the following equation:

$$\alpha = 2 \arctan \frac{4}{1 + D/d + 2.6 \sin\beta}$$

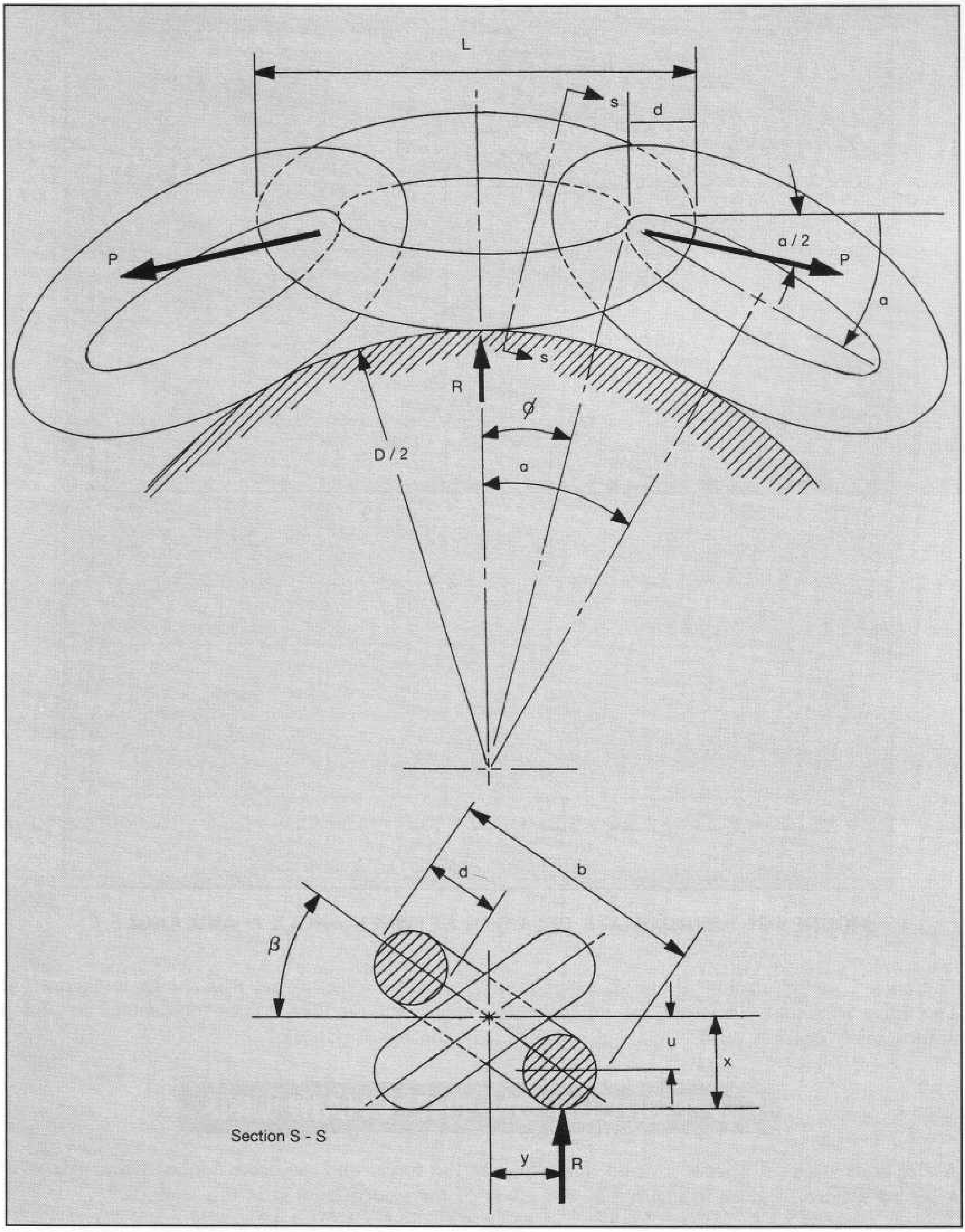


FIGURE F2: GEOMETRY OF A CHAIN BENT OVER A CURVED SURFACE

Unfortunately, the angle β cannot be determined by a closed-form analytical solution. The problem is a very complex solid geometry problem involving the interlocking of two toroids intersecting at an angle which depends on angles α and β. As α increases then the angle β decreases. Figure F3 shows the approximate relationship between the angle β and the angle α. These values were measured with small chain having proportions similar to large stud link chain. The angle β decreases slowly from approximately 40° to approximately 30° as α increases to about 70°, and then it drops rapidly as α increases to 90°.

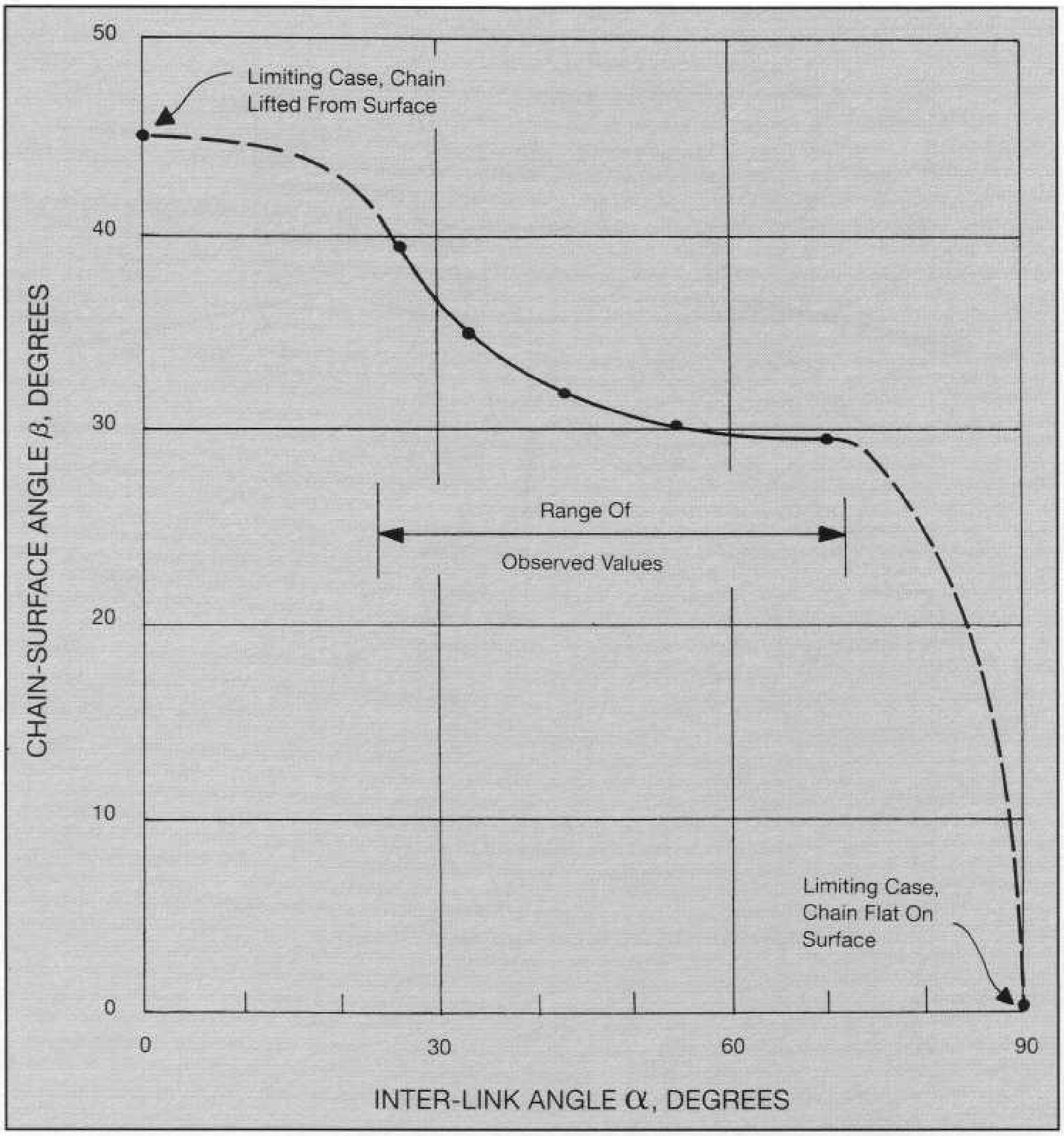


FIGURE F3: APPROXIMATE RELATION BETWEEN ANGLE α AND ANGLE β

The angle α as a function of the diameter ratio D/d and various angles β is shown in Figure F4. The angle α cannot, therefore, be uniquely determined without first having determined β, and β unfortunately depends on α. Thus a closed form solution is not possible.

F.3 FORCES ON THE CHAIN LINK

A free-body diagram of half a chain link, showing the forces and moments applied when bent over a curved surface, is given in Figure F5. The effect of the stud is ignored in this analysis.

The force P is the tension in the chain. For the purpose of analysis, it is assumed the tension force P is applied at the centreline of the chain. In the case of chain on an ungrooved surface, the tension force P between chain links acts at an angle $\alpha/2$ to the chain centreline, because every chain link bears on the surface. This is an important distinction from the case of chain on a grooved surface, in which every other chain link is not in contact with the surface and thus is under tension only. The angle at which the tensile force P acts on the bent chain link in the grooved surface case is α. Only the solution of the ungrooved surface case will be addressed here.

The reaction force of the surface against the chain link is R. For the half chain link free body shown, only half this reaction force applies. Because the chain lies at an angle to the surface, the

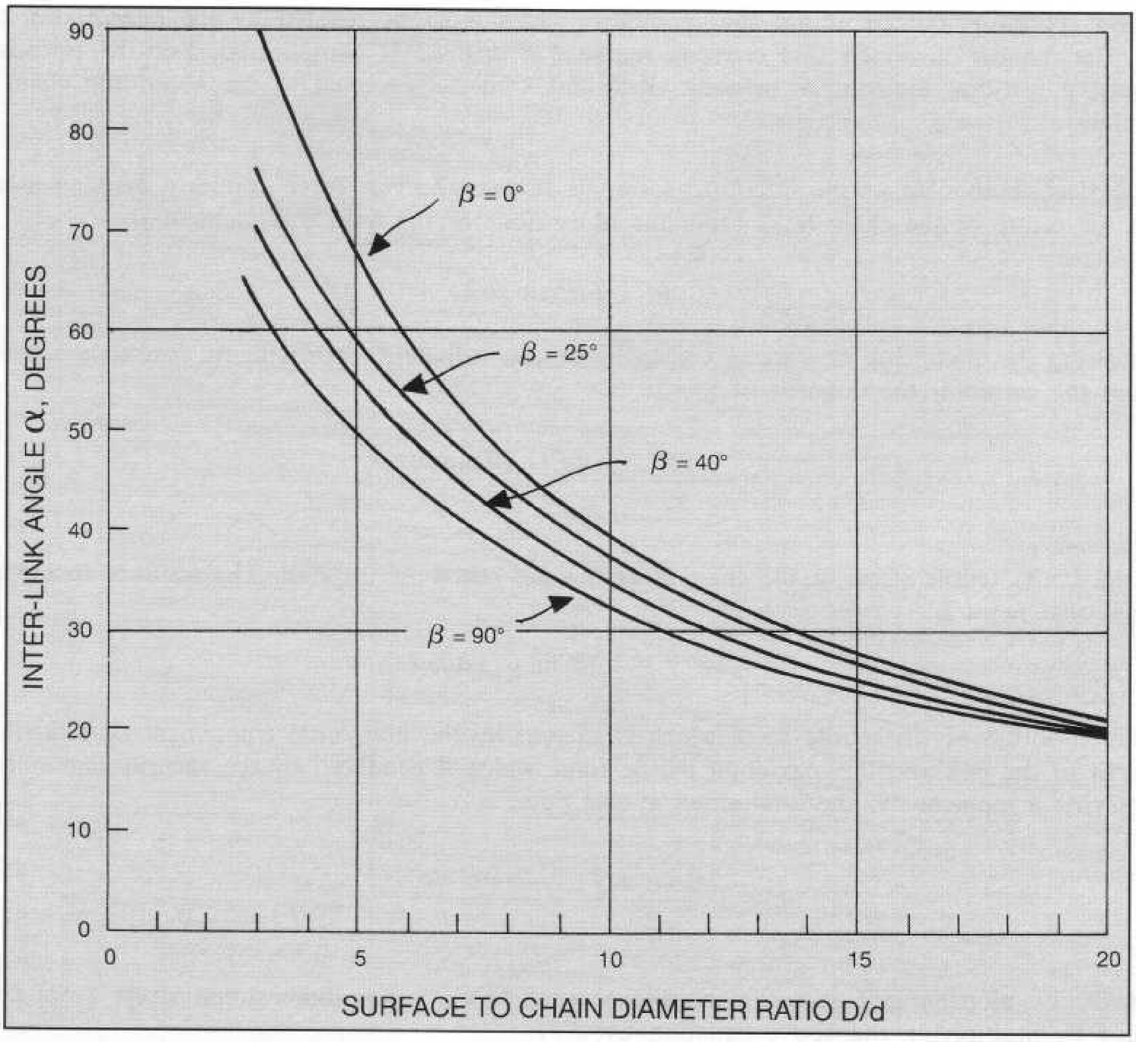


FIGURE F4: ANGLE α AS FUNCTION OF D/d FOR VARIOUS ANGLES β

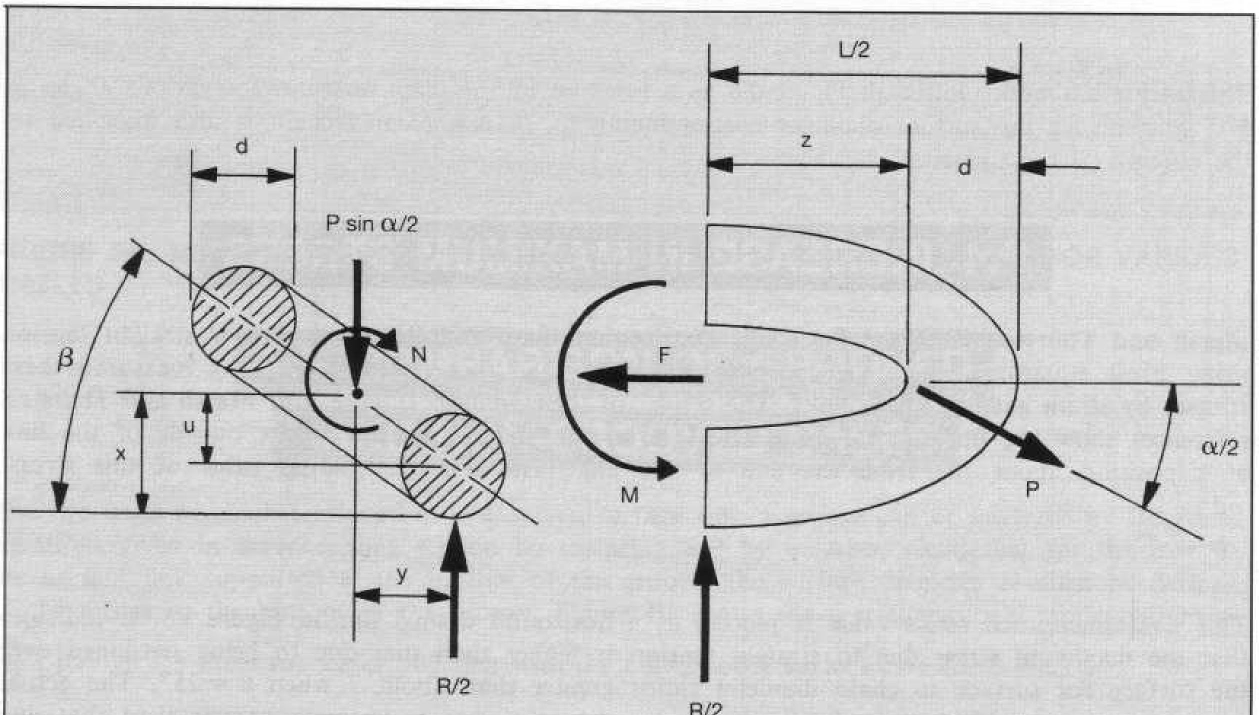


FIGURE F5: FREE BODY ANALYSIS OF HALF CHAIN LINK

reaction applies a moment about the centreline, which must be resisted by the interlocking chain link. The manner in which this resisting moment is applied is complicated. For the purpose of analysis, a resisting moment N between chain links will be assumed to act about the chain link centreline.

The vertical component of the interlink tension is $P \sin \alpha/2$. This force applies a bending moment M at the centre of the chain link. From the dimensions of the link, this moment is

$$M - 2dP \sin \alpha/2$$

Considering the chain link as a beam consisting of two cylinders, each with its centreline a distance u from the centroid, the moment of inertia is

$$I = \frac{\pi d^4}{32} + \frac{\pi d^2 (1.3d \sin \beta)^2}{2}$$

The maximum tensile stress in the chain is at the top centre of the link. The distance to the outer fibre at that point is

$$C = x = 1.3d \sin \beta + 0.5d$$

By superposition of the tensile force which is exerted by the horizontal component of P acting on the area of the link and the maximum tensile force which is produced by the vertical component of P applying a moment M, the total stress at this point is

$$\sigma = \frac{2P \cos \alpha/2}{\pi d^2} + \frac{2PdC \sin \alpha/2}{I}$$

To facilitate plotting and comparison of the stress level, a non-dimensional stress parameter is defined by multiplying the above equation by d^2/P

$$\frac{\sigma d^2}{P} = \frac{2}{\pi} \cos \alpha/2 + \frac{2Cd^3}{I} \sin \alpha/2$$

This factor has been plotted in Figure F6 as a function of the diameter ratio for various angles β. For convenience the surface diameter corresponding to 76 mm (3 in.) chain is also indicated on the abscissa of the figure.

F.4 INCREASE IN MAXIMUM STRESS

Marsh and Thurston analyzed the stress distribution in a stud link chain under straight tension using stress equations which are more sophisticated than those used here. They measured these stresses by strain gauging chain links. The results of their analysis (Figure 3 of Marsh and Thurston reference) show the maximum tension stress in straight tension occurs at the outside of the link at a position about 70° from the end of the link. The non-dimensional value of this stress, $\frac{\sigma d^2}{P}$ is 2.

This non-dimensional stress value is plotted as a horizontal dashed line in Figure F6. It indicates that the maximum stress due to straight tension is higher than that due to being tensioned over the surface for surface to chain diameter ratios greater than about 7 when $\beta = 25°$. The actual diameter ratio at which stress due to tensioning over the surface becomes greater than that due to tensioning over the surface becomes greater than that due to straight tension depends on the angle β, which is undetermined.

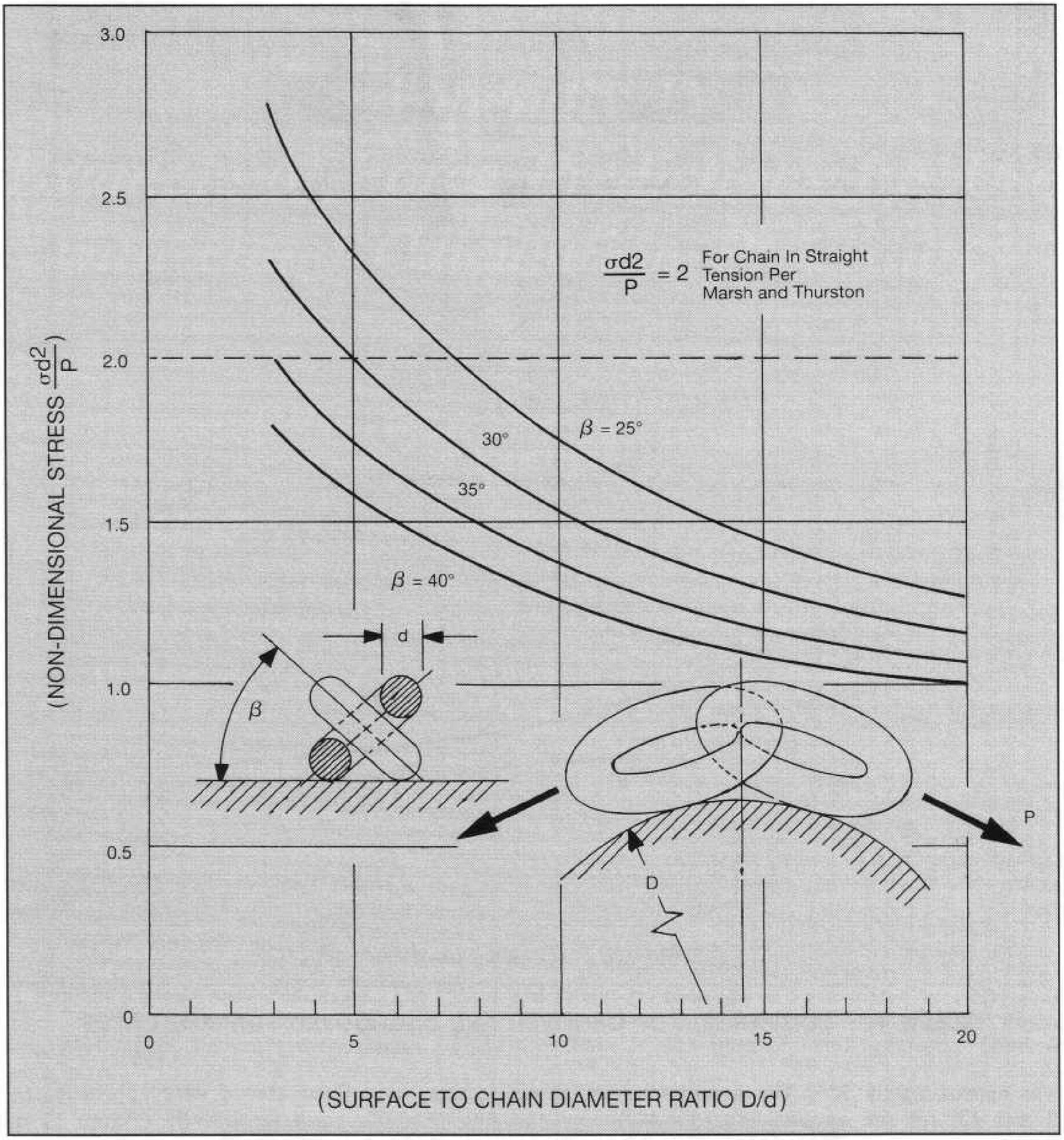


FIGURE F6: NON-DIMENSIONAL STRESS FACTOR AS A FUNCTION OF *D/d* FOR VARIOUS ANGLES β

F.5 COMPARISON OF GROOVED AND NON-GROOVED CASES

The two cases of chain tensioned over a curved surface with a groove can be analyzed by the stress equations given in the preceding section by replacing $\alpha/2$ by α. Also, recall that for the case of the upright link supported at the bottom of the groove, the surface diameter D must be defined as the radius to the bottom of the groove. Figure F7 shows the non-dimensional stress parameter $\frac{\sigma d^2}{P}$ plotted for these two cases, together with the non-grooved surface case for $\beta = 25°$.

As reported by Buckle, the two grooved surface cases were analyzed by Lloyds Register of Shipping. The analysis was done for stud-link chain with a finite-element technique. The inter-link angle α

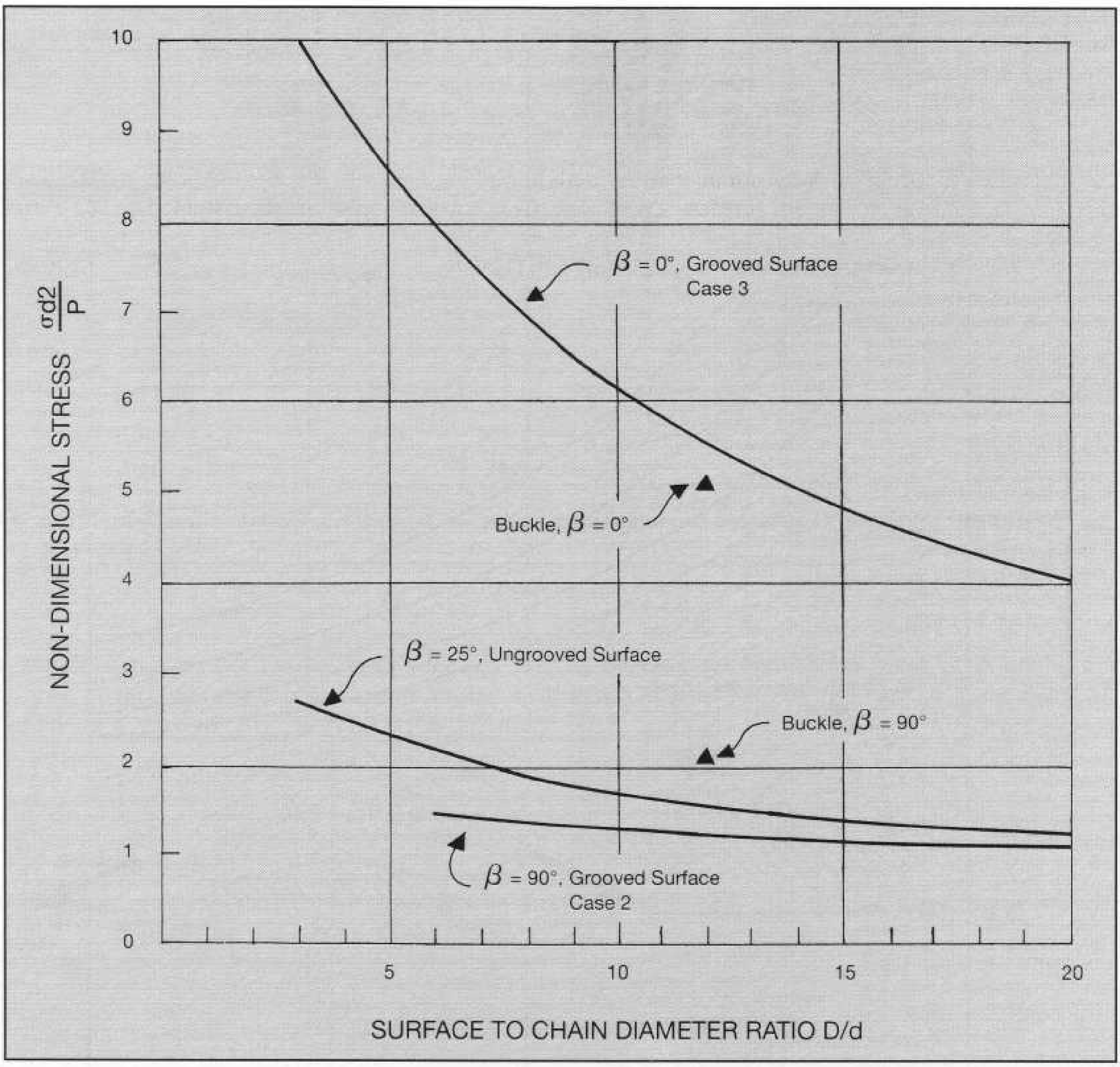


FIGURE F7: COMPARISON OF GROOVED AND UNGROOVED SURFACE CASES

was approximately 30°. The non-dimensional stress factors, as defined above, were approximately 2 and 4.6 for the upright link ($\beta = 90°$) and flat link ($\beta = 0°$) cases respectively (Figure 12 of Buckle reference).

Comparing the Buckle results with those of the present analysis gives some indication of the adequacy of the present calculations. For the 0° case (flat link), Buckle gives a stress only slightly lower than the present analysis. For the 90° case (upright link) Buckle gives a stress approximately 70% higher. This higher value may be due to the action of the stud putting a concentrated load on the middle section of the link.

The comparison indicates the simplified analysis done here may not be as accurate as the finite-element analysis performed by Buckle. Correlation is good for the 0° case, but poor for the 90° case. The intermediate cases of interest, i.e. $\beta = 25°$, may be less inaccurate.

In the present study, a more accurate analysis of the case of a chain tensioned over an ungrooved curved surface could have been obtained using finite-element methods. However, a more exact analysis of the forces between two interlocking links would be necessary. A precise definition of the inter-link forces is a major problem. Further analysis does not appear to be warranted.

F.6 TESTS OF CHAIN TENSIONED OVER CURVED SURFACE

An experimental program was conducted to determine the load at which stud link chain breaks when tensioned over a curved surface. Specimens from two samples of flash-welded grade-2 steel stud-link chain from different manufacturers were tensioned around pins having surface diameters ranging from 2.3 to 14 times the chain diameter. They were loaded until they failed. Properties of the chain specimens are given in Table F.1.

	Sample A		Sample B	
Type of chain	Grade 2 stud-link		Grade 2 stud-link	
Diameter	35 mm	$1\frac{3}{8}$ in.	32 mm	$1\frac{1}{4}$ in.
Certified proof load	490 kN	111,000 lb	410 kN	92,200 lb
Certified breaking load	690 kN	155,000 lb	570 kN	129,000 lb
Actual breaking load	830 kN	186,000 lb	700 kN	156,500 lb
Metallurgical coupon test results:				
Yield strength at 2% offset	415,000 kPa	60,200 psi	373,000 kPa	54,100 psi
Ultimate strength	568,000 kPa	82,400 psi	555,000 kPa	80,500 psi
Elongation, 2 in. gauge length	34%		29%	
Reduction in area	73%		70%	
Hardness	Rockwell B 81		Rockwell B 84	

TABLE F.1: CHAIN TENSIONED OVER CURVED SURFACE, PROPERTIES OF CHAIN SAMPLES

A typical test set up is shown in Figure F8. Load was applied by a large hydraulic cylinder mounted horizontally. One end of the chain specimen was attached to the cylinder head by a detachable link.

The other end was run over the test pin, mounted with its axis horizontal between two plates extending from a stationary frame and connected through a detachable link to a chain swivel bolted to a strong rail in the test floor.

A total of 24 tests were conducted. The set-up and results of each test is summarized in Table F.2. The first 17 tests were conducted with chain A. The results of these tests indicated the chain specimens might be untypically ductile. Therefore chain B was obtained and specimens from it were tested.

F.7 RESULTS OF CHAIN TESTS

When loaded in straight tension, both chain samples broke at loads significantly higher than their rated break test loads. These straight break strengths were used as the basis for defining percent of strength for those chains loaded on the curved surfaces. There was very good agreement between identical tests with chain A. Thus tests were not repeated with chain B.

There was wide scatter of results depending on the geometry of the chain in relation to the surface. When interlocking chain links were arranged to contact the surface at angles near 45°, the chains broke at the interlocking point at loads somewhat lower than the straight break test load. When the interlocking chain links were alternating upright and flat to the surface, as happens with smaller surface diameters, chain strength was not significantly reduced. Sometimes the straight chain links free of the surface broke before the bent chain links did.

Figure F9 shows the tests results. The lowest breaking load obtained for each diameter ratio divided by the straight breaking strength is plotted as a function of the surface diameter to chain diameter ratio. In the case of identical tests, the average of the two loads is plotted.

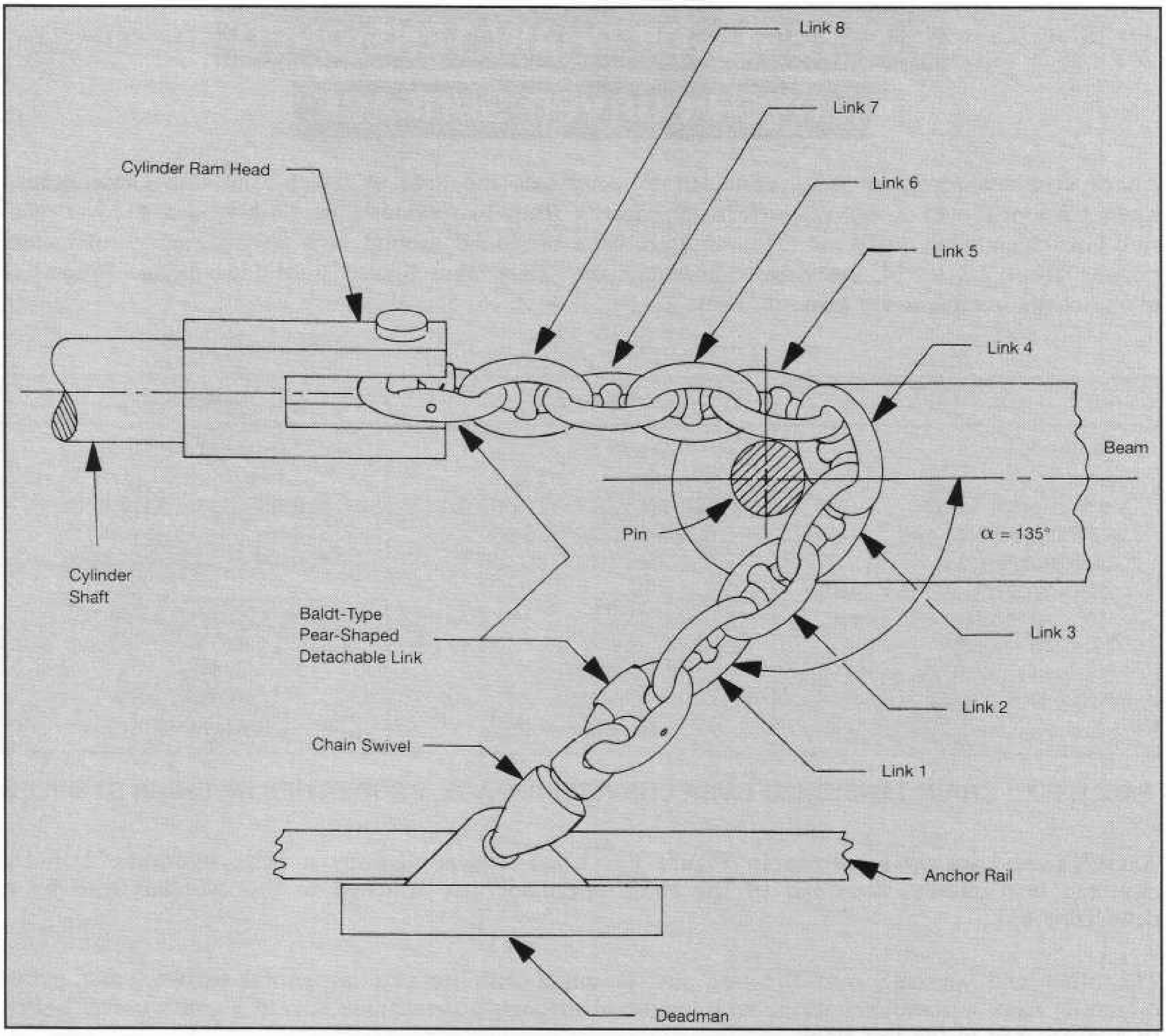


FIGURE F8: TEST SET-UP. TEST 15, $\alpha = 135°$, $D/d = 4$, 8 LINKS

Although chain A was at first thought to break at untypically high loads, because of greater than normal ductility, chain B performed even better and appeared to be even more ductile.

It was thought that links which were deformed around the curved surface would have lower strength when loaded in straight tension. This did not occur. In the tests in which deformed links were loaded in straight tension, the chains broke at points other than the deformed links.

The results of the chain tests should be generally applicable over a wide range of chain sizes, even though only 32 and 35 mm ($1\frac{1}{9}$ and $1\frac{3}{8}$ in.) chains were tested. The diameter ratio results have been treated as non-dimensional quantities. Percent reduction in strength as a function of the ratio of surface diameter to chain diameter chain link proportions is independent of the experiment scale. Also, material properties and manufacturing processes are the same for the chains tested as for larger chains. The findings of the chain tests generally agree with those of the analytical study. Therefore, the results of these studies should be applicable to larger chains.

The curve in Figure F9 is believed to be a reasonable indication of the manner in which surface diameter to chain diameter ratio affects chain strength. The results indicate that strength reduction due to tensioning chain over a curved surface is not significant for surface diameter to chain diameter ratios greater than about 6.

Test No.	Chain sample	Bend angle	D/d ratio	No. of links	Links in contact	Attitude angle	Pre-stress load (lb)	Breaking load (lb)	Percent straight strength	Broken link	Between links	Remarks
1	A	0°	—	8	—	—	—	185,800	100%	5*	4–5	Straight break test
2	A	0°	—	8	—	—	—	186,400	100%	5*	5–6	Straight break test
3	A	90°	14	10	2		—	185,300	99.5%	6*	6–7	Links 2, 3, 4 bent
					3 and 4	−40°						
4	A	90°	14	10	5		—	184,800	99.2%	5*	5–6	Links 2, 3, 4 bent
5	A	90°	9	10	2		—	180,500	96.9%	4	3–4	Links 2, 3 bent
					3	−35°						
6	A	90°	9	10	4	−40°	—	179,500	96.4%	4	3–4	Links 2, 3 bent
7	A	90°	6	8	2		—	174,400	93.8%	2	2–3	Links 2, 3 bent
					3	−20°						
8	A	90°	6	8	4	−70°	—	177,400	95.3%	2	2–3	Links 2, 3 bent
9	A	90°	4	8	2	Almost horizontal	—	153,500	82.4%	2	2–3	
					3	−60°						
					4	Almost horizontal						
10	A	90°	4	8	2	Almost vertical	—	184,800	99.2%	6*	6–7	Broke in free link; links 2 and 3 severely bent
					3	Horizontal						
					4	Almost vertical						
11	A	90°	4	6	— As test 9 —		—	158,800	85.3%	2	2–3	
12	A	90°	2.9	6	2	−45	—	174,900	93.9%	2	1–2	Link 2 severely bent
					3	−10°						
13	A	90°	2.9	6			—	169,600	91.1%	3	2–3	Link 2 severely bent
14	A	135°	4	10	3	−30°	—	177,900	95.6%	3	3–4	
					4	−25°						
15	A	135°	4	8	5	−10°	—	185,800	100%	5	4–5	Links 3, 4 severely
16	A	135°	2.9	7	3	− 5°	—	149,000	80.0%	4	3–4	
					4	−60°						
17	A	135°	2.9	7	3	−90°	—	186,400	100%	5*	—	Link 5 was free of surface under high load; links 3, 4 severely bent
					4	Flat						
18	B	0°	—	7	—	—	—	156,500	100%	3	2–3	Straight break test
19	B	90°	2.3	8	2 and 3	Diagonal	—	141,600	90.5%	3	2–3	Links 2 and 3 severely deformed
20	B	90°	2.3	8	2 and 3	Diagonal	92,000	—	—	—	—	Chain first pre-stressed to rated proof load over pin. Links 2 and 3 deformed. Test chain then straight break tested
		0°	—		—	—	—	155,400	99.3%	5*	5–6	
21	B	90°	4	8	2, 3 and 4	Diagonal	—	151,100	96.5%	2	2–3	Link 2 severely deformed
22	B	90°	4	8	2, 3 and 4	Diagonal	92,000	—	—	—	—	Chain first pre-stressed to rated proof load over pin. Link 2 slightly deformed. Test chain then straight break tested
		0°	—	—	—	—	—	155,400	99.3%	7*	7–8	

TABLE F.2: SUMMARY OF TEST RESULTS, CHAIN TENSIONED OVER CURVED SURFACE

Test No.	Chain sample	Bend angle	D/d ratio	No. of links	Links in contact	Attitude angle	Pre-stress load (lb)	Breaking load (lb)	Percent straight strength	Broken link	Between links	Remarks
23	B	90°	2.3	8	2 3	−90° Flat	130,000	–	–	–	–	Chain first pre-stressed to 130,000 lb to cause significant chain deformation. Test chain then straight break tested
		0°	–	–	–	–	–	154,300	98.6%	8*	7–8	
24	A	0°	–	6	–	–	–	173,600	>93%	Shackle	–	Straight break test of unbroken section of chain from test 10 including severely deformed links. Broke in shackle before chain broke

* Indicates break occurred in straight chain link which was not m contact with curved surface.

TABLE F.2: (*continued*)

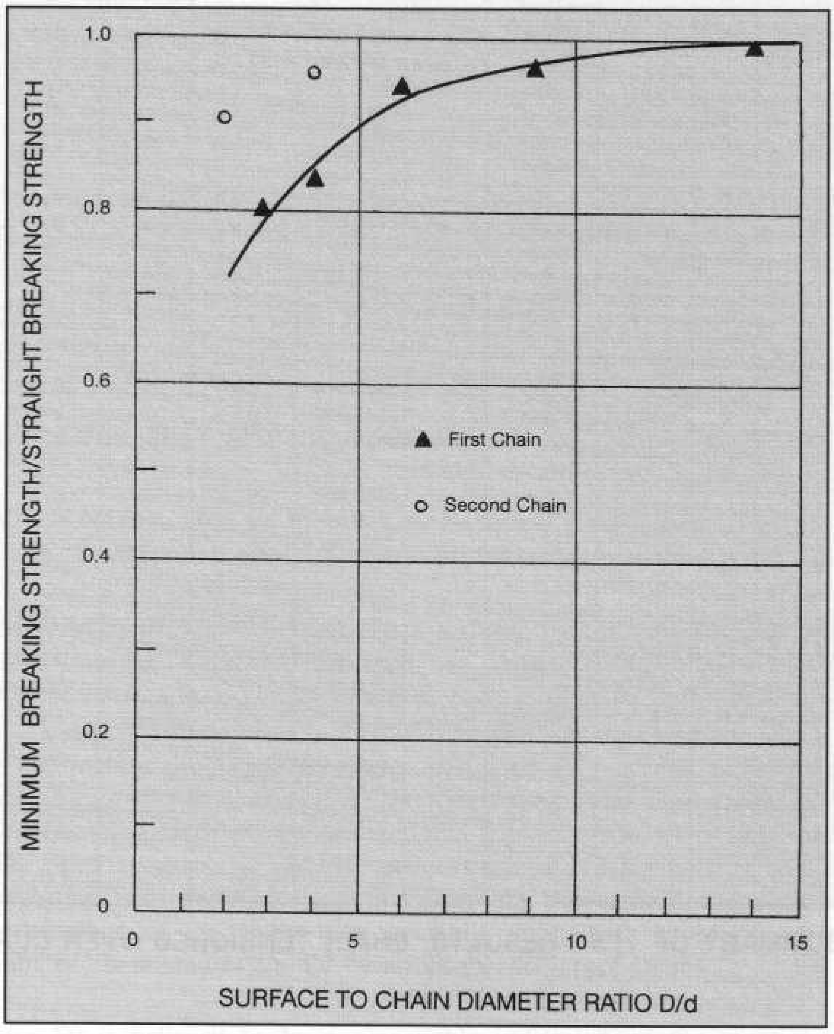


FIGURE F9: RESULTS OF TESTS OF CHAIN TENSIONED OVER CURVED SURFACE

F.8 RECOMMENDATIONS FOR CHOCK SURFACE DIAMETER

Practicality must be considered in specifying a chock diameter for SPM and towing chains. Retro-fitting or modification might be necessary on many tankers. If an impractical or too large diameter is required, many operators and owners may choose to resist or defer making modifications.

Many recommendations have been made in the past regarding the safe diameter for surfaces on which chains bear. Review of these past recommendations indicates that only some of them apply to chain tensioned over an ungrooved surface, the case of interest in SPM and towing chain chocks. The Lloyds investigation reported by Buckle, the only other known analysis, applies only for a grooved surface. No other experimental work of chain tensioned over ungrooved surfaces is known. Thus most past recommendations are apparently only conservative assumptions based on experience in various dissimilar services.

In the present investigation, both analytical and experimental findings indicate that stud link chain will not be significantly weakened provided the diameter of the surface over which it is tensioned is not less than 7 times the diameter of the chain.

Chocks with a 600 by 450 mm (24 by 18 in.) opening and a surface diameter of 560 mm (22 in.) are available. Such chocks are known as Mississippi fairleads, and are manufactured and marketed by several companies. When used with the 76 mm (3 in.) chain, recommended for SPM and towing chafing chain, the 560 mm surface curvature of such chocks provides a surface diameter ratio of 7.4. Thus, the recommended surface diameter to chain diameter ratio of 7 is practical.

Many tankers are already fitted with bow chocks which comply with this recommendation. Some tankers probably have chocks with smaller surface diameters. The most common commercially available 600 by 450 mm (24 by 18 in.) opening chock has a surface diameter of 360 mm (14.1 in.) as specified by Japanese Industrial Standards (JIS). Such chocks provide a surface diameter to chain diameter ratio of only 4.7 with 76 mm chain. However, such chocks can be readily replaced with larger surface diameter chocks.