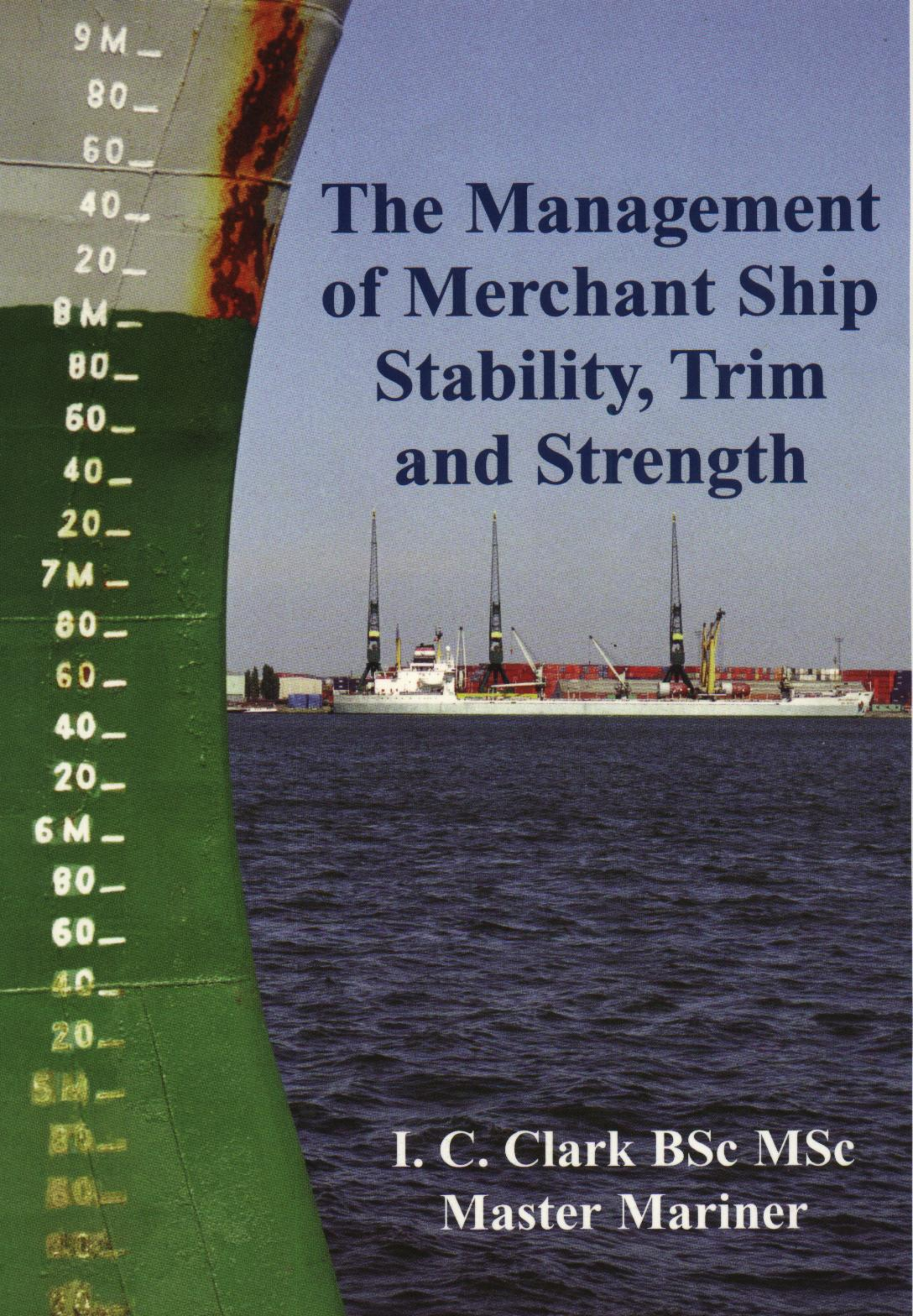

The Management of Merchant Ship Stability, Trim and Strength

I. C. Clark BSc MSc
Master Mariner

THE MANAGEMENT OF MERCHANT SHIP STABILITY, TRIM & STRENGTH

By Ian. C. Clark, BSc, MSc
and Master Mariner

A guide to the theory, rules and calculations carried out to ensure that a vessel maintains seaworthy stability and trim whilst remaining within its limits of strength

THE MANAGEMENT OF MERCHANT SHIP STABILITY TRIM AND STRENGTH

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This book has been prepared to address the subject of merchant ship stability, trim and strength. This should not, however, be taken to mean that this document deals comprehensively with all of the concerns which will need to be addressed or even, where a particular matter is addressed, that this document sets out the only definitive view for all situations.

The opinions expressed are those of the author only and are not necessarily to be taken as the policies or views of any organisation with which he has any connection.

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Cover picture courtesy of Joachim Affeldt

MV New Zealand Pacific at Blohm & Voss ag floating dock Hamburg. During the docking process, careful attention is needed to ship stability. (See chapter 6)

There has been a need to present seafarers with a comprehensive text book on stability which does not go too deeply into naval architecture. As an examiner of masters and mates I have always felt this need particularly as stability is a statutory requirement and ships have to carry approved stability booklets.

The book is well illustrated and its figures are explained with text which is easy to understand. Besides helping the student to get over the complexities of the subject, the student should be in a very good position to face the ordeal of an examination also.

It might be of interest to know that Mr Clark has accomplished this task when actually serving at sea. The philosophy of teaching while practicing seems to have worked well.

Captain P.S. Barve ExC MNI Coordinator
TEF BITS Distance Learning Centre Mumbai

FOREWORD

Captain R.B. Middleton FNI

Seafarers must be responsible for the stability and strength of their vessels. These two aspects linked to navigation are essential disciplines ensuring the safety of our ships when at sea.

The course we steer also influences the behaviour of our ships in a seaway which again is linked to navigation and the dynamic characteristics of the vessel as determined by its stability and loading.

We have needed a mariners guide to ship stability and strength for a long time. This does not detract from the standard texts for certificates of competency but they are generally inadequate to gain a complete picture.

What Mr Clark has succeeded in doing in this well researched and imaginatively illustrated book is present the subject from the seafarer's perspective. The theory is well illustrated but he goes further, he goes on to explain why and how the forces interact both in a static and dynamic way.

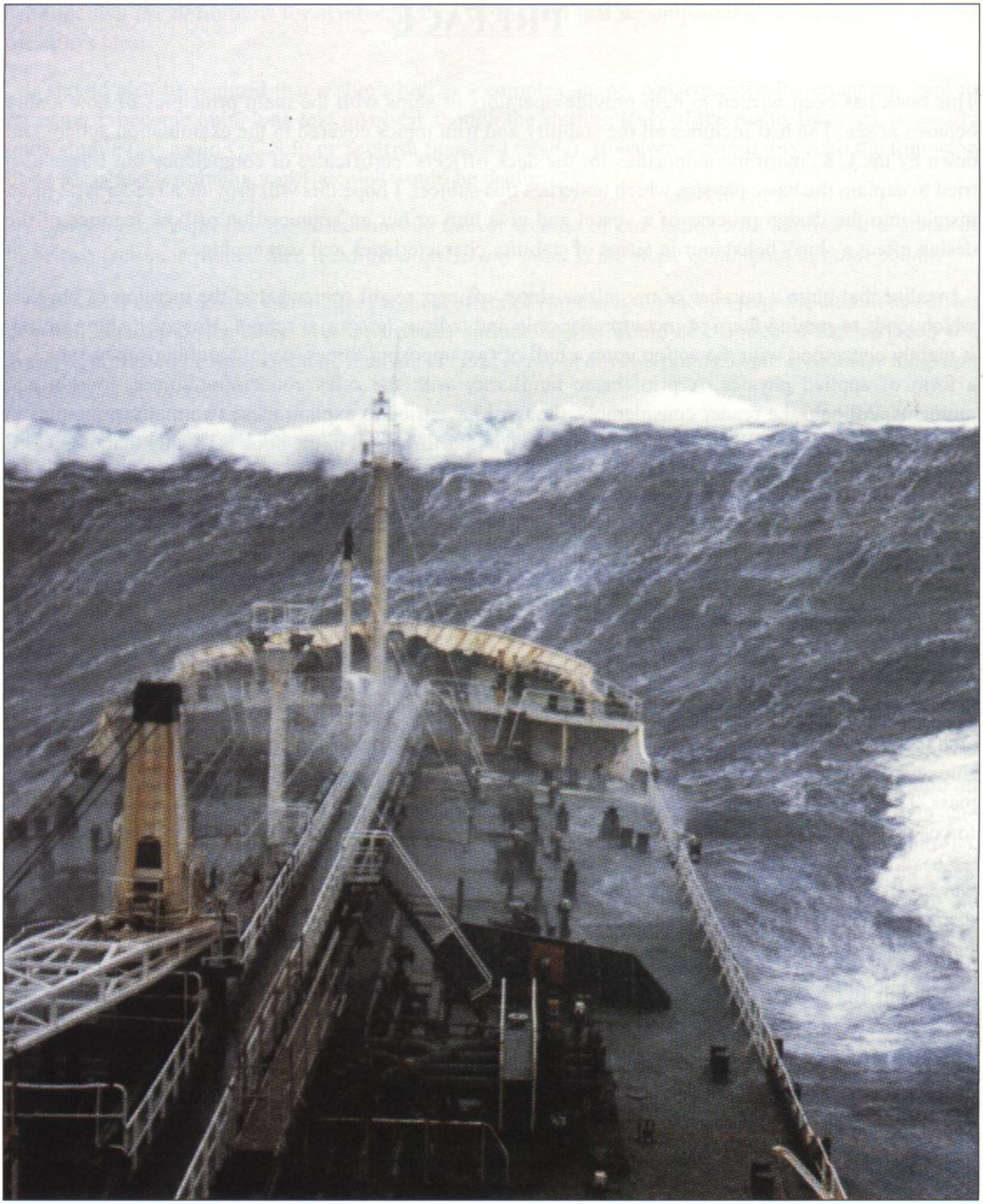
Seafarers are not usually introduced to sub-division rules and Load Line regulations, but their impact on design and stability is predominant and we should have a good understanding of this subject if we are to discuss, for example, modifications and ship improvements at dry docking time.

Mr Clark's practical guide is the missing link between the naval architect and the mariner and as such it needs to be studied by the naval architect whether at university, in industry, classification societies or in government departments.

On board it will be invaluable to the chief mate faced with unusual circumstances and it will enable company superintendents to be more articulate about specifying the way ships are to be loaded in charter parties.

Ultimately managing ship stability and strength is part of the nautical profession's seamanship and this book by Ian Clark enhances this fundamental discipline and takes it to a new higher level of awareness.

Captain R.B. Middleton FNI, President,
The Nautical Institute



Heavy weather is a test of the ship's state of loading in terms of both stability and strength.

PREFACE

This book has been written to help provide operators of ships with the main principles of how a ship behaves at sea. The text includes all the stability and trim topics covered in the examination syllabi laid down by the U.K. maritime authorities for the deck officers' certificates of competency but I have also tried to explain the basic physics which underlies this subject. I hope this will provide a reader with more insight into the design process of a vessel and give him or her an appreciation of how features of the design effect a ship's behaviour in terms of stability characteristics and seakeeping.

I realise that quite a number of my fellow ships' officers recoil somewhat at the mention of physics, which tends to remind them of incomprehensible and tedious lessons at school. However, ship stability is mainly concerned with the action upon a hull of two opposing forces, weight and buoyancy, which is a form of applied physics. A prior basic familiarity with the rules concerning forces, motion and moments will help the reader considerably, though I have tried to explain these two major principles in the early parts of the text. The level of physics used as starting point in the book is usually that of G.CS.E or Scottish Standard Grade but there are some topics, such as circular motion, where a slightly higher level of understanding is required.

The difference between an object's weight and its mass is frequently misunderstood. Mass is the amount of matter in an object (measured in kilograms and tonnes), whereas weight is the gravitational force acting upon that mass (measured in Newtons and KiloNewtons). All forces act to accelerate a mass (where Force = Mass x Acceleration) so a 1 Tonne mass weighs 9.81 KiloNewtons on earth, where the acceleration due to gravity is 9.81 metres per $(\text{second})^2$.

The definitions of force and mass are fundamental to the principles of physics. ***However,*** naval architecture, like many engineering disciplines, usually fails to make such a distinction. The weight of a ship and the water it displaces are usually expressed in Tonnes, which strictly speaking, are measures of mass. The omission of gravity from the equation, however, does not alter the balance between the two forces of weight and buoyancy, as gravity affects both these masses equally. The problem comes when we have to consider the ship's motion and power requirements to overcome forces such as friction and wave resistance. We then have to take account of accelerations other than that of gravity and do have to distinguish between the ship's mass and its weight. The only such example in this book is the effect of turning forces causing a ship to list over. Weight is expressed in tonnes for the rest of the text.

The unit of 'KiloNewtons' is being used more frequently now in nautical circles, particularly in expressing towing forces or 'bollard pull'.

As a one time secondary school physics teacher, I should welcome this belated attempt to properly distinguish between 'mass' and 'weight' but it is generally not appreciated that it is purely co-incidental that the numerical value of gravitational acceleration on earth is very nearly equal to 'ten'. This has unfortunately resulted in a widespread mistaken belief that the 'KiloNewton' is simply a new decimal unit equal to $1/10^{th}$ of a Tonne, which only adds to the confusion between the meanings of 'mass' and 'weight'.

The principles of stability like all forms of applied physics and engineering goes hand in hand with mathematics. Equations and graphs are the language for expressing physical relationships most clearly and concisely. Equations use symbols, rather than words, and there are insufficient letters in the English alphabet to stand for all the different factors used in the book. Even borrowing letters from the Greek alphabet does not entirely solve the problem and can be confusing until the reader is familiar with these. A single letter or symbol may stand for different things in different equations (e.g. 'd' may mean 'draft' in one equation and 'depth' in another).

I hope that the definitions for symbols, given by the text that accompanies each equation, make the meanings clear.

It should also be realised that a ship's hull is a complex shape, consequently, the equations used to measure it, become quite long and involved, though the starting level of the maths involved, is usually quite simple (i.e. again G.C.S.E. or Scottish Standard Grade). However, a familiarity with the following basic ideas and terminology of Calculus would be useful:-

Differentiation expresses the instantaneous rate of change of one factor with another. If a graph of these two factors is plotted then the *differential at any point is the slope of the graph at that point.*

Integration expresses the cumulative physical effect of two changing factors (e.g. the cumulative effect of width measurements, taken over a particular surface length, would be the area of that surface.) If a graph of these two factors is plotted, then *the integral between any two points, is the area under the graph between those points.*

Methods of approximate integration are used extensively in naval architecture to determine areas and volumes and moments. There are two main methods of such approximations, Simpson's rules and the Trapezium method. I have explained both these methods and how they are used, though I have given a slight preference to the 'Trapezium' method, as this is simpler to understand.

In writing the text, I have used line drawings extensively and avoided carrying an particular argument or explanation over from one page to another, except in the case of a few double page spreads, where the two pages appear opposite each other. In the main, each page has a separate title and should stand alone. I hope that this makes it more convenient for a reader to follow the particular points of the text.

Finally, I should like to say few words about the law and legislation as one third of the content is concerned with explaining relevant internationally recognised regulations.

Generally, laws are determined and policed by the governments of individual national states and all commercial ships must comply with the laws of their country of registry (this is known as 'flag state control') and the countries whose waters they trade in (which is known as 'port state control'). International regulation is achieved by countries agreeing to incorporate a common set of rules into their own state legislation so, for example, the 'Load Line' regulations are part of the national laws of all the separate states that signed the International Convention. (The list, understandably, will not include a land locked country such as Nepal, not because it has no regard for international law, but being miles from the sea and perched amidst the world's highest mountains, its government is understandably not greatly concerned with maritime affairs).

This is not a legal textbook and so my explanations should be taken only as an outline of the legal requirements that directly affect a ship's stability in the intact and damaged condition. The laws themselves are often expressed in language that can be extremely torturous to read and to give a full, yet comprehensible, explanation of how they relate to every possible type of ship would require a book in itself. I have therefore chosen to highlight what I feel to be the main points of the regulations and, in doing so, will almost certainly have omitted some of the details.

SUMMARY OF CHAPTER CONTENTS

MV Rickmers Tianjin at berth No. 2 Taichuns discharging a 267 T generator unit with ship's gear for a power station project. Special care is needed when handling heavy lifts. See chapter 5.

Coastal vessels are equally sensitive to static and dynamic stability.

ACKNOWLEDGEMENTS

I would like to thank the following people for their help in producing this book.

Mr J. Parker as Secretary of the Nautical Institute who agreed to publish the book and has given support for the two years that it has taken me to write it.

Captain Barve, former Chief Examiner of Masters and Mates and former Nautical Advisor to the Indian Government who has patiently read through all the various re-writes of the chapters and provided useful advice in the presentation.

Mr Frank Poulloin of Lloyd's Register, London, who has kindly given me assistance in writing Chapter 11 on the Load Line regulations.

Messers Graham Greensmith and Andy Williamson at Lloyd's for their assistance in writing Chapter 10 on the MARPOL subdivision regulations for tankers.

Mr Simon Milne and the staff of the Passenger Ship Department of the Maritime and Coastguard Agency, Southampton, who have provided help with Chapter 10, regarding the subdivision rules.

My good friend Angus Richardson who, as a man more versed in computers than myself and my one time partner in designing and building a 28 foot yacht, has given advice in both the chapter content and how to grapple with some of the more infuriating foibles of the computer.

My wife Hilary, who along with both my children, has been incredibly patient and understanding of most of my lapses into bad temper whilst proof reading the entire text to correct my own peculiar notions of spelling and punctuation.

Finally, I would like to thank all the lecturers, ship's officers and crew who, over the years, have taught me my trade. In particular, I would like to extend my appreciation to Dr Barrass who lectured me in naval architecture at Liverpool Polytechnic all those years ago.

All these people contributed to the completion of this book but any mistakes are entirely my own.

Ian Clark, North Wales, October 2001

CHAPTER 1

AN INTRODUCTION TO THE SHAPE OF A SHIP'S HULLFORM AND THE PRINCIPLES OF HYDROSTATICS THAT ACT UPON IT

SUMMARY

THIS CHAPTER GIVES AN OVERVIEW OF A HOW THE PRINCIPLES OF HYDRODYNAMICS APPLY TO A SHIP'S HULL, BY INTRODUCING THE FOLLOWING TOPICS:-

1) **THE STANDARD TERMINOLOGY AND MEASUREMENTS USED TO DESCRIBE THE SHAPE AND FEATURES OF A SHIP'S HULLFORM.**
2) **THE PRINCIPLES OF BUOYANCY AND FLOATATION WITH REGARD TO SHIP'S DRAFT.**
3) **THE STANDARD DRAWINGS FOR DEFINING THE SHIP'S HULLFORM AND HOW MEASUREMENTS ARE TAKEN TO CONVERT THESE DRAWINGS INTO TABLES OF DATA.**
4) **HOW METHODS OF APPROXIMATE INTEGRATION ARE APPLIED TO THESE DATA TABLES TO CALCULATE HULL AREAS AND VOLUMES.**
5) **THE PRINCIPLES OF MOMENTS WITH REGARD TO LOCATING THE CENTRES OF GRAVITY & BUOYANCY AND HOW THE DISTRIBUTION OF WEIGHT & BUOYANCY ACT UPON A HULL TO PRODUCE HEELING, TRIMMING AND BENDING MOMENTS**
6) **THE MOTION OF A SHIP IN SEAWAY AND THE NATURE OF SEA WAVES**

CONTENTS

THE EVOLUTION OF THE SHAPE OF A SHIP'S HULLFORM

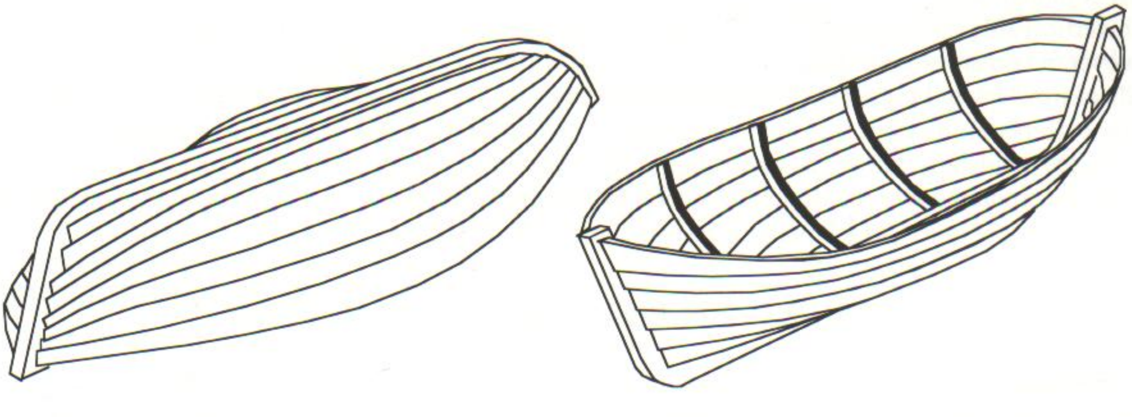
The boat or ship, in the broadest sense of the words, is almost certainly mankind's oldest form of transport other than walking on his own feet. The enormous carrying capacity of water borne craft has long been appreciated. Even now in the late twentieth century where public imagination has been diverted to motor cars and aircraft, the vast bulk of goods in the ever-increasing world trade are carried by ocean going ships. At first glance, a modern large container vessel appears to have little in common with a Viking longship or even a nineteenth century tea clipper, but the basic hull shape of all these vessels is essentially the same. The ship's hull has evolved into a surprisingly subtle shape to meet the following requirements, which are often in conflict with each other:-

1) A good carrying capacity for the overall size of the vessel.
2) Good sea-keeping qualities.
3) The ability to be easily driven through the water.
4) The possession of the ability to remain basically upright in a seaway.
5) The strength to withstand the stresses and strains due to the motions of the sea

As with most of man's engineering ventures, design techniques have progressed considerably through trial and error within the limitations of the material and tools available to build the ship at the time. True understanding of the principles involved has often lagged behind 'rule of thumb' practices and there have been some spectacular and infamous examples of getting things wrong. The 'Wasa' was a sixteenth century warship, which capsized on launching due to its excessive top weight.
Despite these setbacks, the basic general hull shape has survived the test of time and has essentially remained unchanged over the last thousand years, although today we can build ships much larger than ever previously envisaged. This is because, within a broad set of parameters, the evolved hull shape is the best one for the job of moving across a frequently turbulent fluid surface at any kind of reasonable speed and comfort.

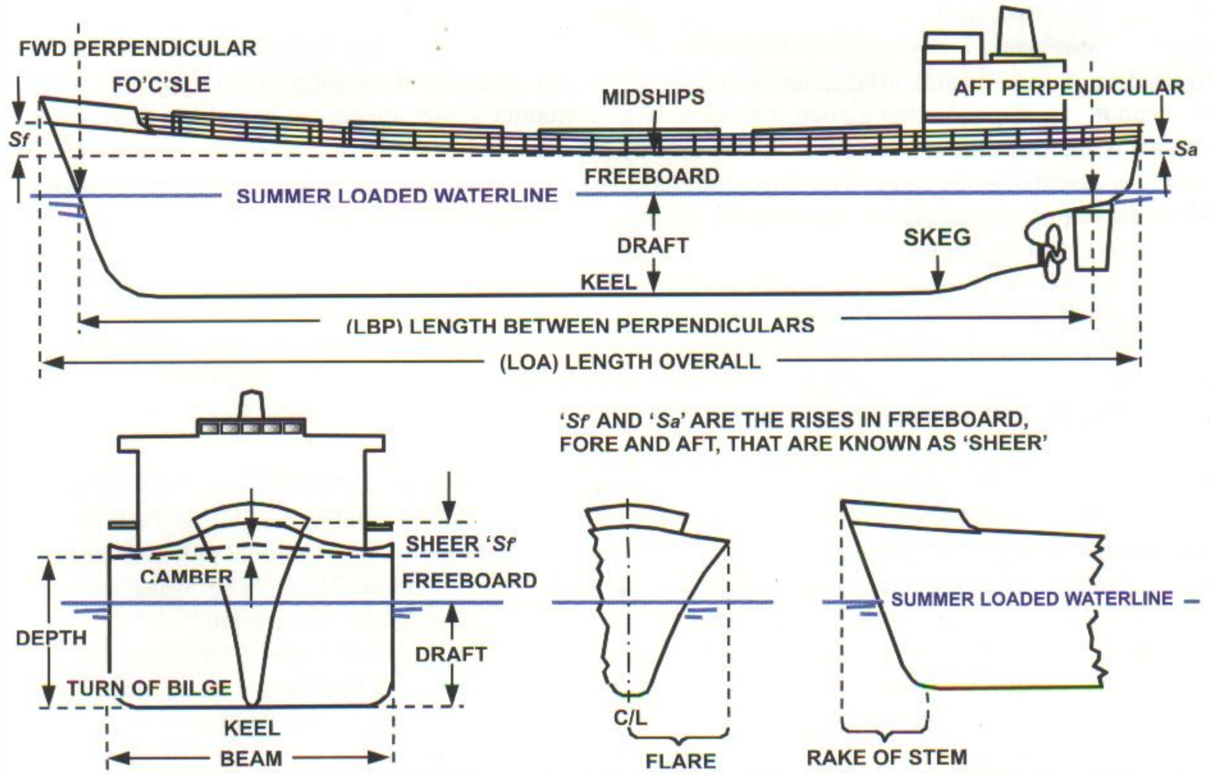
We can see the classical hull shape if we look at a traditional building technique, that of the clinker built boat. In this method, the boat is built by laying overlapping planks from stem to stern working outwards and starting from a strong central keel, which is shaped to form the hull's profile. The fore and aft ends of the keel are fashioned into stem and stern posts to which the planks are nailed, as shown below

THE TRADITIONAL 'CLINKER' BUILT BOAT



The shape of the boat is judged by eye and controlled by the degree of overlap at the fore and aft ends. It may be double ended, as shown above, or the stern may be 'chopped short ' by a flat transverse bulkhead, called the 'transom'. The depth of hull rises naturally near the bow and stern where the planks are pinched together. To achieve this, the planks must be gradually twisted as they are worked away from the midships region of the hull. The hull in the midships region (or the waist) is correspondingly low and wide.

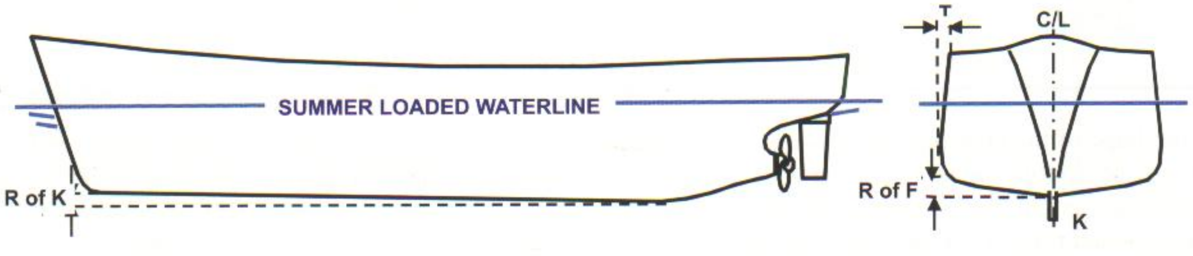
COMMON FEATURES AND TERMINOLOGY OF A SHIP'S HULL



Both sheer of the deck line and flare of the hull at the bow and stern improve the hull's sea-keeping ability by increasing its resistance to being submerged at the fore and aft ends by wave action. The upper deck is given a transverse, or athwartships, curvature, known as camber, to assist water drainage when seas actually break on board. The stem rake is a natural consequence of the concave flare of the bow region and, again, it helps the hull ride over the waves as it moves forward. Many ships have a raised watertight enclosed compartment at the bow, known as the fo'c'sle which provides additional protection from seas breaking over the fore end. A similar raised structure, called the poop, may be built into the stern, but this is becoming less common.

Most commercial cargo carrying hulls are flat bottomed without any external protruding centreline keel with vertical side plating amidships (i.e. the hull amidships is wall sided). Some smaller vessels, however, retain an external keel with a 'vee' shaped bottom (i.e. the hull features a rise of floor). Hulls of trawlers and tugs, which require deep immersion of the propeller for towing, frequently become deeper at the stern (i.e. the hull features a rake of keel). A few vessels have inward sloping ships sides amidships, known as tumblehome. This has no particular hydrodynamic advantage and was usually incorporated into the design to reduce overall enclosed space, which, in the past, gave the vessel a lower tonnage figure, which harbour dues are generally based upon.

RISE OF FLOOR, RISE OF KEEL AND TUMBLEHOME



R of K = RISE OF KEEL, R of F = RISE OF FLOOR, T = TUMBLEHOME, K = KEEL

HULL MEASUREMENTS

DIMENSIONS

Length Overall (LOA) is the extreme length of the ship, from the foremost point on the bow to the aft point on the stern. Its primary importance is in determining the amount of space that a ship requires when tying up alongside jetties and turning in confined rivers or channels.

Length between Perpendiculars (LBP) is measured from the rudder post aft to the point where the stem cuts the waterline in the normal fully loaded conditions at even keel (i.e. no trim by the head or stern). It is an approximation of the submerged length and is used in hydrostatic calculations concerning trim and stability. *(The positions of the 'perpendiculars' changes slightly when 'length' is defined for the purpose of determining a ship's maximum allowable draft.(See Chapter 11-The Load Line Regulations)*

Depth of Hull and Beam are vertical and transverse measurements taken in the midships region, so the depth will be the minimum vertical distance between the uppermost continuous (i.e. full-length) deck whilst the beam will be the maximum width from one side to the other. Quite often **moulded** values are quoted. These are internal measurements and do not include the thickness of the hull plating.

Freeboard is the height above the waterline of the uppermost watertight continuous deck. It generally increases at the bow and the stern due to sheer, so for any particular loaded state, its minimum value occurs at the midships region. The law requires that every commercial vessel's seaworthiness must be assessed and, on the basis of this, each vessel is assigned a minimal legally allowable freeboard which limits the maximum load the ship can carry. The basic calculations produce the **Summer Freeboard**, This maximum allowable waterline must be marked on the ship's sides. A range of additional seasonal and regional adjustments, based upon the Summer Freeboard, is allowed, depending upon the area that a ship is trading in at a particular time of the year. The Summer Load Line and the allowed seasonal adjustments are marked on the ship's port and starboard sides in the midships region. It is a serious criminal offence to leave a port for the open sea with the vessel in an overloaded condition.

Draft is the depth of the hull beneath the waterline. If it remains constant along the ship's length, then the ship is said to be on even keel or level trim. The mean draft indicates the amount by which the ship is loaded and is used in hydrostatic calculations. The maximum draft is important for ensuring that the vessel is not run aground by entering water that is too shallow. The draft produced by the Summer Freeboard is known as the Summer draft and is generally quoted as the designed maximum loaded draft, though it is subject to both seasonal and regional adjustment.

Trim is the difference between the forward and aft drafts. A ship is frequently loaded so that the draft is slightly grater at the stern than the bow, to ensure that the propeller remains well immersed and to minimise taking seas over the bow. This is known as a **Stern Trim or trimmed by the stern**.

Air Draft is the maximum height of any part of the vessel above the waterline for a particular loaded state. It is important for ensuring that the vessel has adequate clearance when passing under bridges or navigating in close proximity of airport runways.

DISPLACEMENT

A ship's **Displacement** is the actual mass of the vessel's structure and all its contents, i.e. the cargo, fuel and stores, so it can be 'broken down' as follows:-

Loaded Displacement	=	**Lightship displacement**	+	**Deadweight (or Burden)**
(Ship's fully laden weight)		(Ship's structural weight)		(Weight of cargo, fuel, water etc)

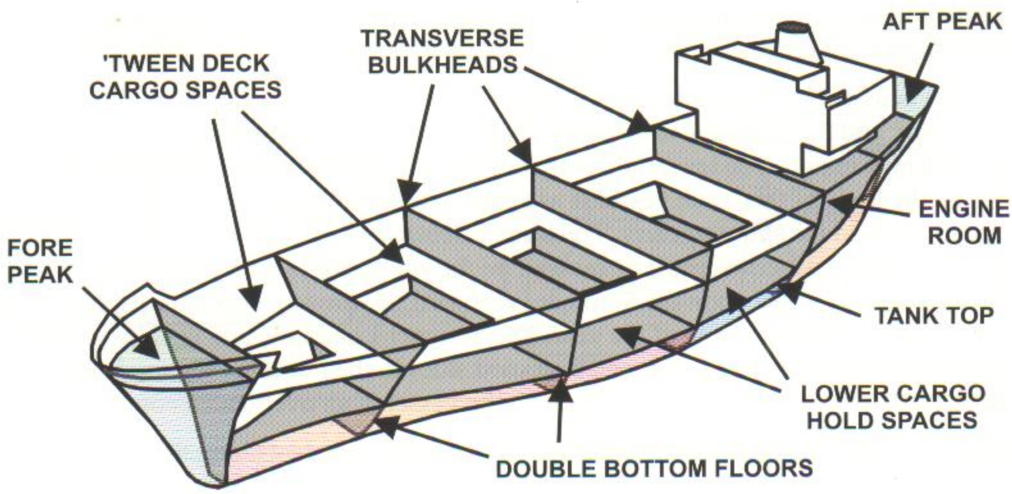
Displacement values are measured, quite correctly, in Tonnes and Kilograms but often wrongly expressed as ships' 'weights'. Many people fail to distinguish between 'mass', which is the amount of matter within an object, and 'weight', which is the downwards force acting on that mass due to the earth's gravity.
Weight, as a force, should be measured in KiloNewtons rather than Tonnes where **1 Tonne weighs 9.81 KiloNewtons on Earth as the acceleration due to gravity is 9.81 m/s^2**. However, the practice of considering 'Tonnes' as a measure of weight is so widespread in ship's data and cargo figures that this book only makes the distinction between weight and mass when it is necessary.

INTERNAL DIVISION WITHIN THE SHIP'S HULL

Internal subdivision of a ship's hull by structural partitioning is important for the following reasons:-

1) The internal decks and bulkheads provide essential stiffness and strength to the hull structure.
2) The resulting subdivision creates suitably sized segregated spaces for fuel, water, ballast, cargo and machinery rooms. A number of convenient sized cargo compartments with separate hatches allows for flexibility in the distribution of mixed commodities loaded for different destinations.
3) Internal subdivision restricts the possible movement of individual cargo stows, particularly liquids that will flow back and forth with the ship's motion.
4) Watertight internal bulkheads and decks limit the extent of flooding that can occur if a ship is accidentally 'holed' under the waterline and so provide it with some chance of remaining afloat.

THE MAIN INTERNAL SUBDIVISION OF A TYPICAL TWO DECK DRY CARGO SHIP



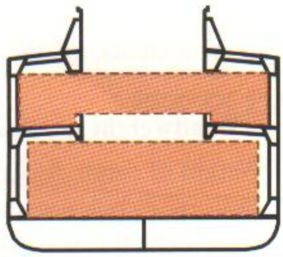
KEY

= MAIN WATERTIGHT SUBDIVISIONS, = FORE AND AFT PEAK BALLAST TANKS,

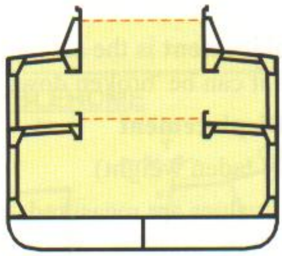
= DOUBLE BOTTOM FUEL TANKS, = DOUBLE BOTTOM FRESH WATER TANKS

THE SHIP'S HULL IS INTERNALLY DIVIDED INTO CARGO HOLDS, FORE AND AFT PEAK TANKS AND THE ENGINE ROOM BY VERTICAL TRANSVERSE BULKHEADS. A DOUBLE BOTTOM SPACE IS CREATED BETWEEN THE HULL BOTTOM AND THE HORIZONTAL TANK TOP. THIS IS DIVIDED INTO TANK SPACES BY CONTINUING THE TRANSVERSE BULKHEADS DOWNWARDS TO THE SHIP'S BOTTOM.(THESE TRANSVERSE VERTICAL DIVISIONS WITHIN THE DOUBLE BOTTOM ARE KNOWN AS 'FLOORS'.) THE CARGO SPACES ARE HORIZONTALLY SPLIT INTO THE LOWER HOLDS AND 'TWEEN DECKS BY A NON-WATERTIGHT DECK WITH HATCHES

CARGO COMPARTMENT CAPACITIES



MIDSHIPS TRANSVERSE SECTION THROUGH THE LOWER HOLD & 'TWEEN DECK' COMPARTMENTS



BALE CAPACITY

THIS VOLUME IS DERIVED BY MEASURING BETWEEN THE INNER EDGES OF THE SHIP'S FRAMES AND UNDERDECK GIRDERS. VOID SPACES BETWEEN THE FRAMING IS 'LOST'

GRAIN CAPACITY

THIS IS THE MAXIMUM VOLUME, DERIVED BY MEASURING BETWEEN THE SHIP'S SIDES, TANK TOP AND UNDERSIDES OF THE DECKS. (I.E. THE MOULDED DIMENSIONS).

A SHIP'S REGISTERED TONNAGE

Merchant ships are 'registered' under a national government 'flag' in order to provide the commercial parties concerned with the vessel with a legal code for settling disputes and for the ship to have a recognised regulatory authority. Part of the registration procedure involves the government authorities assessing the ship's size to record its 'registered tonnage'. This is actually a measure of the **volume of enclosed spaces** on a ship and **not** the vessel's **mass or weight**, though it is expressed in 'Tons', where **1 registered ton is the equivalent to 100 cubic feet or 2.78 cubic metres**. Registered Tonnage is an internationally recognised basis for raising taxes from the shipping industry and setting levels of fees for port dues, towage, pilotage and canal passages. It is also used as a size determiner in both national and international shipping regulations. For example, by international agreement, ships of 1,600 GRT (Gross Registered Tons) or more must have more comprehensive radio equipment than smaller vessels whilst Japanese regulation requires that vessels in excess of 10,000 GRT must be under pilotage in many of their more restricted coastal areas, such as the 'Inland Sea'

The particular term 'tonnage' appears to have originated in thirteenth century England when the King started to levy taxes on the growing wine trade between the South of England and France. The wine was shipped in large wooden barrels, known as 'tunneaux' or 'tuns' so the King's revenue collectors would tax a vessel on the number of such barrels that it could carry in its holds. The word 'tonnage' originally meant the tax paid to the crown but it gradually changed to mean the measurement of the vessel, so although it is confusing to consider 'tons' as a measurement of volume rather than mass, (or even, less correctly, as weight) it is, in fact, a much older meaning of the word.

The long-standing idea of measuring a ship by the number of regular sized units of cargo continues today as the size of modern container ships is often expressed in terms of 'TEU' or 'twenty foot equivalent units' that it can carry. *(Twenty feet being the length of a standard container.)*

Service providers to the shipping industry (e.g. port authorities, pilotage companies etc.) are not obliged to base their fees on registered tonnage and many include other parameters, such as overall length, draft or type of cargo in their charging policies. These can reflect more accurately the costs of dredging, berthing etc. However, registered tonnage remains the most common single charging element.

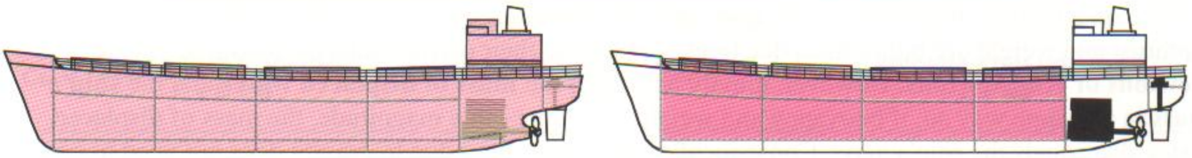
It is not this book's intention to give a detailed description of the tonnage rules, as they are not directly relevant to the hydrostatic principles that determine a ship's behaviour at sea. However, an awareness of tonnage measurements is useful as they affect a ship's operating costs and have greatly influenced merchant ship design over the years resulting, at times, in some very odd ships' features.

Broadly speaking **Gross Registered Tonnage (GRT)** includes all enclosed spaces whilst the **Net Registered Tonnage (NRT)** measures just the enclosed cargo spaces. The rules, however, have a complex history of continual change as shipowners attempt to minimise their costs by requiring ships to be designed with the smallest possible registered tonnage for their size. Much of this revolves around the rules' definition of an 'enclosed volume', as a cargo space may be exempt from measurement if it is technically 'open' to the weather even though it is effectively 'sheltered' from the elements.

The authorities also changed the rules to avoid penalising desirable developments so, in the past, spaces essential for the ship's safe navigation and the well being of crew were exempted from measurement but this is no longer allowed in the present rules.

The current 1969 Tonnage Regulations apply the **'Universal Tonnage Measurement System'** or **'UMS'** for deriving Gross and Net UMS 'tonnage' values, to be expressed simply as numbers without the term 'tons'. The UMS gross is based upon total enclosed volume whilst UMS net is calculated from the cargo space volume and numbers of passengers. UMS net cannot be less than 30% of UMS gross.

A SIMPLIFIED ILLUSTRATION OF GROSS AND NET UMS TONNAGE



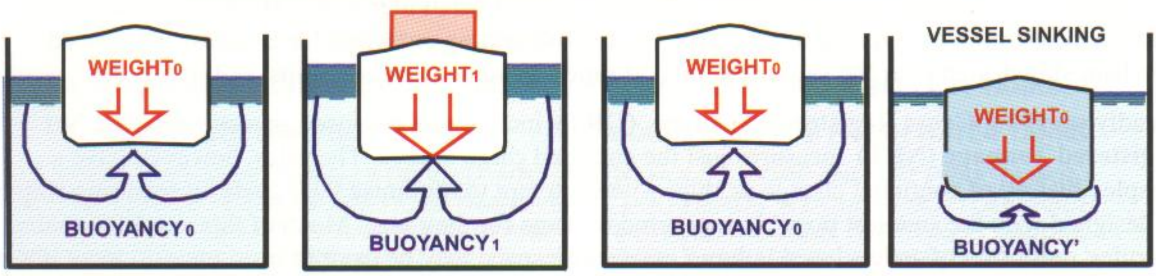
= GROSS TONNAGE VOLUME. INCLUDES ALL ENCLOSED CARGO SPACES, FUEL TANKS AND ENGINE ROOM

= NET CARGO SPACE VOLUME. THE UMS NET VALUE ALSO TAKES PASSENGER NUMBERS INTO ACCOUNT

FLOATATION AND BUOYANCY

A ship floats by pushing its own weight of water up out of the way. This displaced water exerts a supporting force on the ship's hull as gravity tries to restore the original undisturbed level.
The resulting upwards force is called the **Upthrust** or **Buoyancy** and is an example of Newton's third law of Force and Motion, which states that:-
*'A single force must act between two masses and its effect upon one mass, (the **Action**) is equal and opposite to its effect on the other (the **Reaction**)'*
The ship pushes the water upwards, so the water pushing back against the ship with an equal force. Whether this amount is enough to support the ship or not, depends upon the **Volume** of water displaced. If the ship's hull encloses a considerable amount of space containing low density material, including air, then its overall weight will be sufficiently low enough to allow the displaced water to support it completely, so it floats. If the hull is then progressively filled with higher density cargo, the weight increases which requires an ever increasing volume of displaced water to support it, hence the ship floats lower and lower in the water. Eventually, the enclosed spaces of the ship are completely immersed and further increases in cargo weight will not produce any further increases in displaced water, which is no longer sufficient to fully support the ship. The upthrust is still acting upon the ship but is now less than the increased weight of the vessel so it sinks.
Alternatively, a ship may sink if some of its enclosed hull spaces are holed and flooded. The ship's weight remains the same but the flooded compartment no longer contributes to the displacement of water hence the buoyancy is now reduced. This ship must sink lower in the water and, if there is sufficient remaining enclosed space to compensate for the flooding, the vessel will remain afloat at a new deeper draft. If, however, the enclosed spaces become fully submerged without fully compensating the lost buoyancy, the ship will sink.
The principles involved are best illustrated if we consider the ship floating in an enclosed space or, like Archimedes, we imagine a model boat floating in a bath and consider changes in the water level when we increase the boat's load or put a hole in it.

ARCHIMEDE'S PRINCIPLES OF BUOYANCY AND FLOATATION



AN INCREASE OF WEIGHT ($W_0 \rightarrow W_1$) REQUIRES GREATER DISPLACEMENT OF WATER TO PRODUCE THE INCREASE IN BUOYANCY REQUIRED TO REMAIN AFLOAT

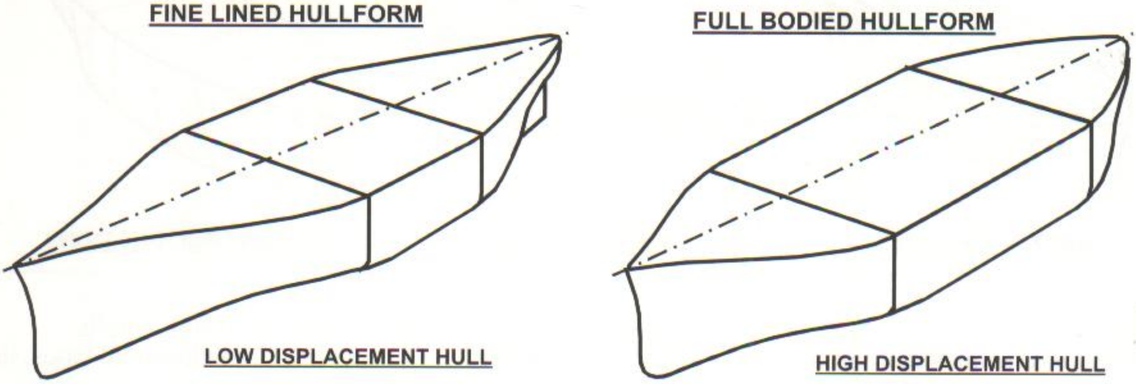
FLOODING REDUCES THE BUOYANCY FOR A GIVEN DRAFT, SO THE HULL SINKS DEEPER. TOTAL FLOODING LEAVES ONLY THE VOLUME OF THE HULL PLATING TO PROVIDE BUOYANCY

A ship remains afloat if the weight of water displaced equals its own weight. To achieve this, its average density, including enclosed void spaces, must be less than that of the water it floats in.
The density of fresh water is slightly less than that of seawater, so a ship will float at slightly deeper draft when it passes from seawater to freshwater.

A force on a body is the product of its **mass** and **acceleration.** I.e. Force = Mass x Acceleration
Upthrust and weight are both forces due to gravity and so their correct units of measurement are **Newtons or KiloNewtons.** However, gravity acts upon the **mass of displaced water** and the **ship's mass** in the same way so, if buoyancy and weight are of the same magnitude, the mass of displaced water also equals the ship's mass. Consequently, the acceleration due to gravity does not need to enter the hydrostatic equation and it is common to loosely express weight and buoyancy in Tonnes. (I.e. the unit of 'mass') There are, however, dynamic situations, such as the ***horizontal*** turning forces acting on a ship's rudder, where the distinction between weight and mass must be made.

THE UNDERWATER HULLFORM AND COEFFICIENTS

We can see from the previous page that, for a given length, draft and beam, a box shaped hull will have the greatest possible displacement and, hence, carrying capacity. Such a craft, however, would have very poor seakeeping qualities and would produce a high level of resistance to being driven through the water. Such a shape is restricted to floating platforms for use in sheltered waters (such as used for floating heavy lift cranes and floating dry docks)
For any ship operating in open waters, carrying capacity must be traded off, to some extent, against seakeeping performance. Such a hull can be considered as three separate sections merging into each other along the ship's length. The forward and aft sections are finely tapered towards the bow and stern where as the midships parallel body section is box shaped. The proportion of this centre region to the total underwater length varies from one design to another. Ships with fine lines have a relatively short length of parallel body where as bluff, or full formed, hulls may be box shaped for over half of their length.

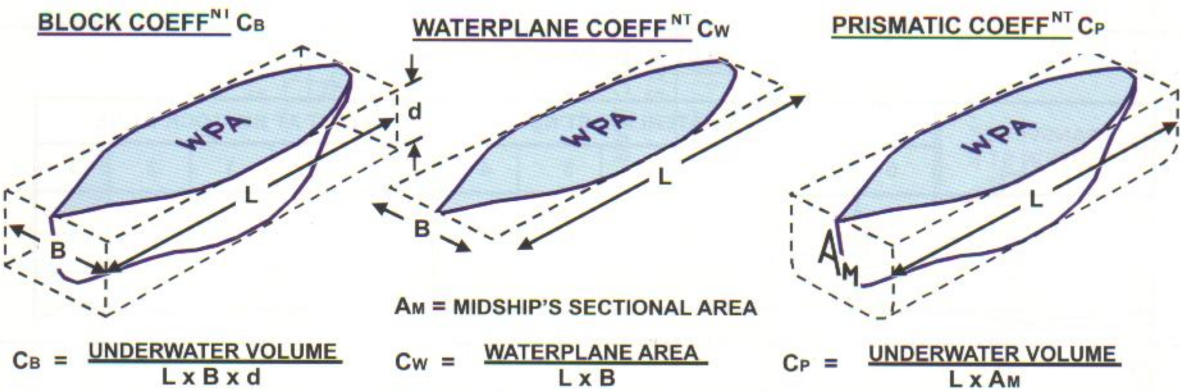


The fine lined hull is more easily driven through the water than the full bodied form but it has a reduced carrying capacity. The hull form also affects other seakeeping qualities, such as the vessel's rolling and pitching characteristics.

UNDERWATER HULL COEFFICIENTS

The degree of hull fineness can be expressed in terms of measured ratios, known as hull coefficients. These coefficients compare the actual immersed hull shape to that of rectangular shapes of the same overall dimensions. The three most commonly used coefficients are as follows:-

HULL FORM COEFFICIENTS



A_M = MIDSHIP'S SECTIONAL AREA

$C_B = \dfrac{\text{UNDERWATER VOLUME}}{L \times B \times d}$ $\quad$ $C_W = \dfrac{\text{WATERPLANE AREA}}{L \times B}$ $\quad$ $C_P = \dfrac{\text{UNDERWATER VOLUME}}{L \times A_M}$

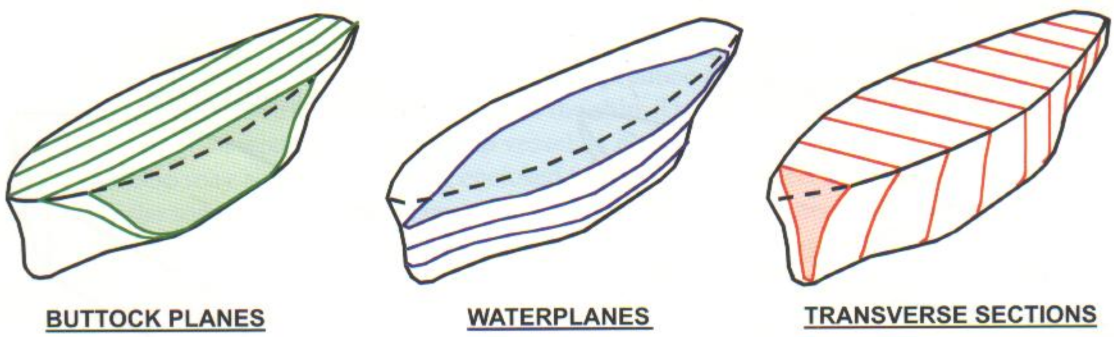
The block coefficient, C_b, is the principal measure of hull form and can vary from about 0.65 for the fine lined hull of a fruit carrier to near 0.9 for a large oil tanker. C_b increases with draft as the hull becomes a fuller shape further away from the keel.

DEFINING THE SHAPE OF A SHIP'S HULLFORM, THE LINES PLAN

The first stage of building a ship is the drawing out of scale plans which accurately define its three dimensional shape on a flat piece of paper. The standard technical drawing approach of three mutually perpendicular views is used to produce plan, side and end elevations but the method is modified to show the changing curvature of the hull in all three dimensions. This is achieved by illustrating slices of the hull at regular intervals in each dimension, as shown below.:-

1) **Buttockplanes:-** These are vertical fore and aft slices taken at regular beam intervals.
2) **Waterplanes:-** These are horizontal fore and aft slices taken at regular draft intervals.
3) **Transverse sections:-** These are vertical athwartships slices taken at regular length intervals.

DEFINING A VESSEL'S HULL FORM



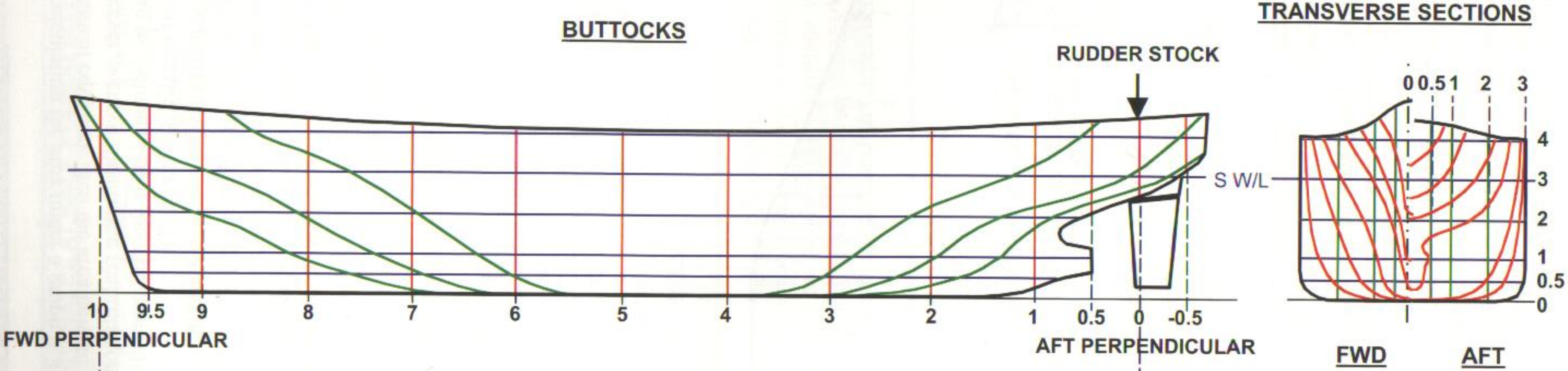
Typically, a ship will be designed around basic specifications, which will include overall length, maximum beam amidships and draft for a given Summer Load Line deadweight. In almost all cases, the hull form of a new vessel will be closely based upon records of a previously built ship of similar size, which allows a shipyard to simply adjust an existing set of plans. If, however, we were starting completely afresh, we would commence by producing a centreline profile and an amidships section that met the specified dimensions. We could then draw waterplanes based upon an estimate of the degree of fineness needed to meet the ship's carrying capacity at the specified draft.
The intersection of these waterlines at buttock and transverse stations on the plan view can then be measured and the values used to construct buttockplanes and transverse sections. This will reveal any unwanted hollows or bumps in these two dimensions, which can then be smoothed out and the original waterplane estimates re-adjusted accordingly. The procedure is a re-iterative one by which the three views and sets of curves are progressively built up and modified by cross referencing to ensure that each set of slices is fair and actually representing the same hull form. When this point is reached, a table of measurements, known as the **Offsets**, is made up to define each transverse section and so define the entire hull shape.

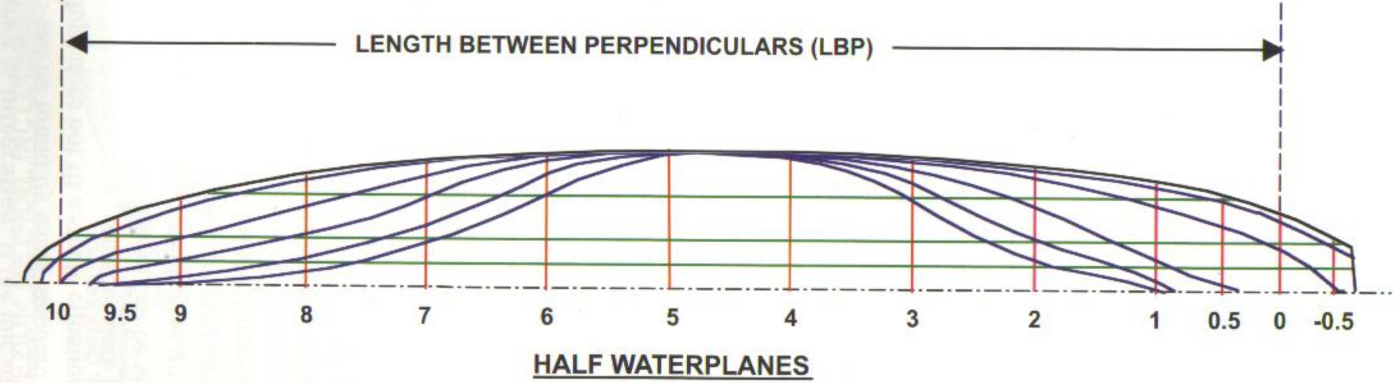
TABLE OF OFFSETS

TRANSVERSE SECTION ST'N	$\frac{1}{2}$ WIDTHS AT WATERLINE ST'NS						HEIGHT AT BUTTOCK ST'NS				
	0	$\frac{1}{2}$	1	2	3	4	0	$\frac{1}{2}$	1	2	3
$\frac{1}{2}$											
1											
2											

Transverse station intervals are, by convention, taken at every one tenth of the length between perpendiculars (LBP), though half or even quarter station intervals are used close to the bow and stern in order to more accurately define the hull in these regions where curvature changes more rapidly with length. At one time, full size drawings used to be made and the process was called lofting the lines because it was carried out in the shipyard's rigging loft.

SKETCH ILLUSTRATING THE LINES PLAN OF A HULL, SHOWING THE BUTTOCKS, WATERPLANES AND TRANSVERSE SECTIONS





HEIGHTS ARE MEASURED FROM THE KEEL

HALF WIDTHS ARE MEASURED FROM THE CENTRELINE

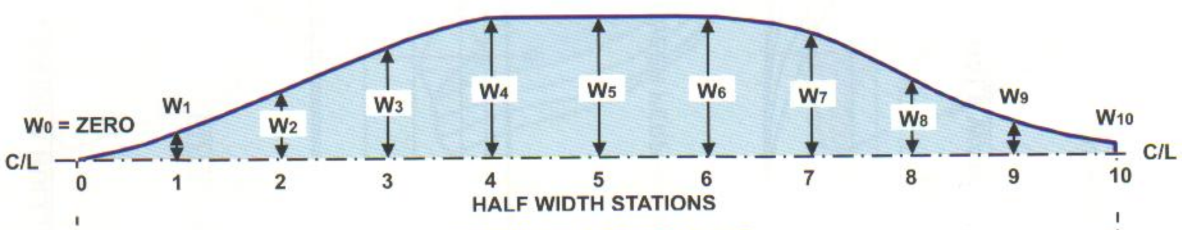
THE SCALE OF A LINES PLAN IS TYPICALLY 1:100

HALF STATION INTERVALS ARE USED IN THE REGIONS WHERE HULL FORM IS CHANGING RAPIDLY WITH LENGTH. I.E. AT THE BOW AND STERN.

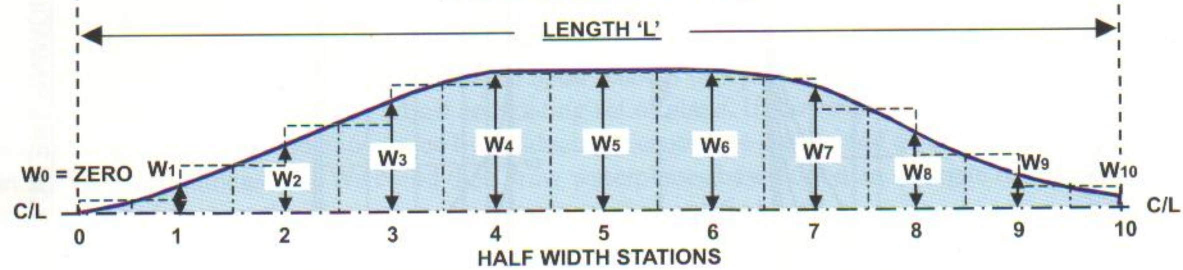
CALCULATING AREAS FROM THE LINES PLAN

The process of calculating the area under a graph is known as **integration**. If the line of a graph obeys a single mathematical equation, then the area beneath it can be found by using standard mathematical formulae. If however, a curve illustrating real relationships does not fit a single equation, then the area enclosed by it can be estimated by a form of approximate integration, which considers the area to be made up of a lot of equally thin strips. The piece of curve bordering each strip can be simplified so that the area of each strip can be calculated and then the individual area added up to give an approximation of the total area. The accuracy is improved by taking more strip measurements at shorter regular intervals. The measuring interval is usually called the **Common Interval, C.I.**
The simplest such process is called the **Trapezium Method**, which assumes that the curve can be made up of a lot of short sections bounded by **straight lines** so each strip can be assumed to be a trapezium. The process is best understood if we see how it is applied to estimating the area of a waterplane area, as shown below.

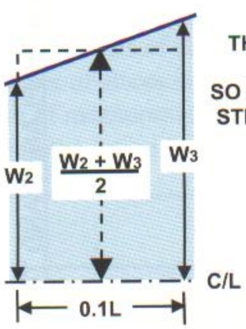
CALCULATING WATERPLANE AREA BY THE TRAPEZIUM RULE



ACTUAL HALF WATERPLANE



ESTIMATED HALF WATERPLANE



THE AREA OF A TRAPEZIUM = AVERAGE HEIGHT x HORIZONTAL WIDTH

SO AREA OF 1/2 WATERPLANE STRIP BETWEEN ST'NS 2 & 3 $= 0.1L\left(\frac{W_2 + W_3}{2}\right)$ IF IT IS A TRAPEZIUM

$$\text{So TOTAL } \tfrac{1}{2}\text{WPA} = 0.1L\left(\frac{W_0 + W_1}{2} + \frac{W_1 + W_2}{2} + \text{---} + \frac{W_8 + W_9}{2} + \frac{W_9 + W_{10}}{2}\right)$$

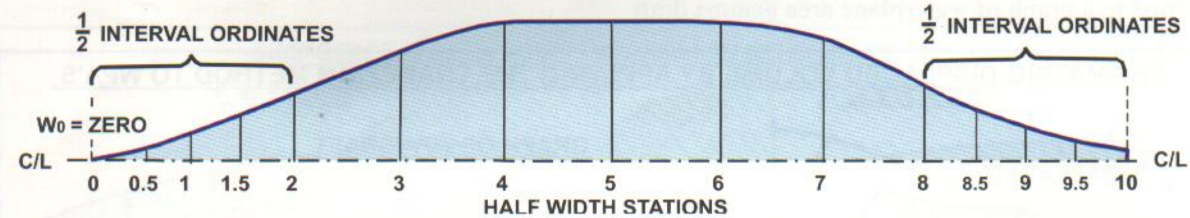
$$\text{So TOTAL } \tfrac{1}{2}\text{WPA} = 0.1L\left(\frac{W_0}{2} + W_1 + W_2 + W_3 + \text{------} + W_8 + W_9 + \frac{W_{10}}{2}\right)$$

$$\text{So TOTAL WATERPLANE AREA} \approx 2 \times 0.1L\left(\frac{W_0}{2} + W_1 + W_2 + W_3 + W_4 + W_5 + W_6 + W_7 + W_8 + W_9 + \frac{W_{10}}{2}\right)$$

WHERE 0.1L IS THE COMMON INTERVAL BETWEEN HALF WIDTH MEASURING STATIONS

The simplicity of the above approximation relies upon the ordinate measurements (in this case, the half width) being taken at regular intervals so the base width of each trapezium remains constant. Accuracy tends to be lost at the fore and aft ends where the line curvature tends to be most pronounced and changes most rapidly with length. This loss of accuracy can be reduced by including additional width measurements, taken at half station intervals. A slightly more complex formula is produced, as shown on the following page.

THE TRAPEZIUM METHOD, INCLUDING ORDINATES AT HALF STATION INTERVALS



IF WE USE ADDITIONAL HALF STATION WIDTH ORDINATES AT THE BOW AND THE STERN, THEN THE WATER -PLANE AREA CAN BE APPROXIMATED AS FOLLOWS;-

$$\text{TOTAL WPA} = \frac{2L}{10}\left[\left(\frac{W_0}{4}+\frac{W_{0.5}}{2}+\frac{W_1}{2}+\frac{W_{1.5}}{2}+\frac{W_2}{4}\right)+\left(\frac{W_2}{2}+W_3+W_4+W_5+W_6+W_7+\frac{W_8}{2}\right)+\ldots\right.$$

$$\left.\ldots+\left(\frac{W_8}{4}+\frac{W_{8.5}}{2}+\frac{W_9}{2}+\frac{W_{9.5}}{2}+\frac{W_{10}}{4}\right)\right]$$

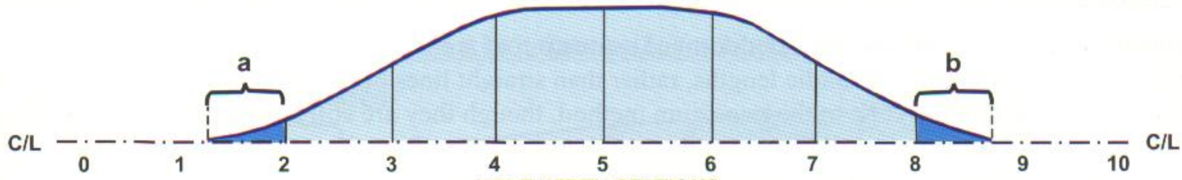
$$\boxed{\text{TOTAL WPA} = \frac{2L}{10}\left[\frac{W_0}{4}+\frac{W_{0.5}}{2}+\frac{W_1}{2}+\frac{W_{1.5}}{2}+\frac{3W_2}{4}+W_3+W_4+W_5+W_6+W_7+\frac{3W_8}{4}+\frac{W_{8.5}}{2}+\frac{W_9}{2}+\frac{W_{9.5}}{2}+\frac{W_{10}}{4}\right]}$$

Notice that the end ordinates, w_0 and w_{10} must be included in the equation, even though that their value is frequently zero.

APPENDAGE AREAS

Although it would be possible to divide each waterplane length into ten equal slices this is not usually done in practice because we want to measure width ordinates at station intervals common to all the waterplanes. As these increase in length progressively with the height of the waterplane above the keel, there will usually be residual areas protruding beyond the fixed measuring stations. Consider, for example, a waterplane close to the keel that does not extend to the fore and aft perpendiculars.

ESTIMATING THE AREAS OF APPENDAGES



'a' & 'b' ARE THE LENGTHS OF THE BOW AND STERN APPENDAGES

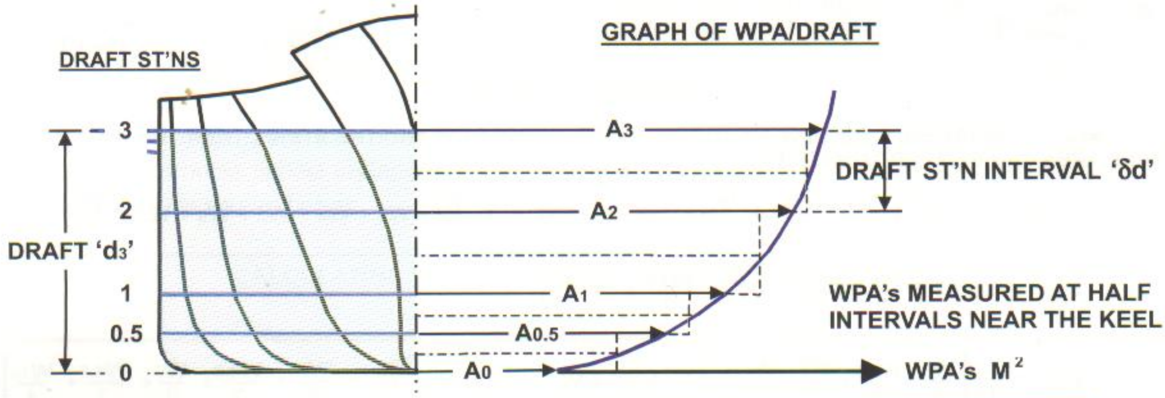
$$\boxed{\text{TOTAL WPA} = 2\left[\frac{a \times W_2}{2}+\frac{L}{10}\left(\frac{W_2}{2}+W_3+W_4+W_5+W_6+W_7+\frac{W_8}{2}\right)+\frac{b \times W_8}{2}\right]}$$

The darker shaded appendage areas are assumed to be triangular with base lengths 'a' and 'b' & perpendicular heights 'w_2' and 'w_8' respectively. For the sake of simplicity, in this case we have only taken half ordinates at full station intervals, though the principle of calculating the appendage areas, could still be applied if we had included half station measurements at station 1.5 and 8.5. This would reduce the areas of appendages and increase the accuracy of the approximation. Notice that the common interval for the calculation remains as 0.1 of the length between perpendiculars, even though this particular waterplane does not actually extend over the entire length. At deeper drafts, the waterplanes extend beyond the perpendiculars and may include widths measured at station 10.5 aft of the rudder stock or station -0.5 forward of the fwd perpendicular.

INTEGRATING WATERPLANE AREAS WITH DRAFT TO CALCULATE VOLUME

The immersed volume for the hull at a particular draft can be calculated by applying the trapezium method to a graph of waterplane area against draft.

ESTIMATING DISPLACED VOLUME BY APPLYING THE TRAPEZIUM METHOD TO WPA'S



$$\text{SUBMERGED VOLUME AT DRAFT 'd3'} = \delta d \left(\frac{A_0}{4} + \frac{A_{0.5}}{2} + \frac{3A_1}{4} + A2 + \frac{A_3}{2} \right) M^3$$

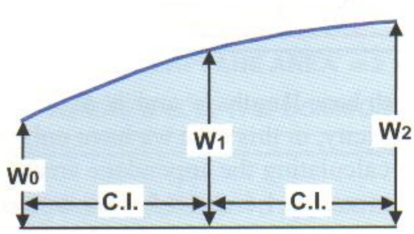
Notice that, close to the keel, half draft station intervals have been used to increase accuracy.
The immersed volume can also be calculated by applying the trapezium method to transverse sectional areas or buttock-plane areas taken at regular station intervals along the ship's length. Comparison of the values of immersed volume, calculated by these three different sets of data, will indicate the degree of error involved in the approximate integration method. If this appears to be unacceptably large, then the calculations can be repeated with more ordinates taken at half, or even quarter, station intervals. An average of the three results can then be used.
Although the extensiveness of these calculations may seem quite daunting, taken step by step, the procedures are relatively simple and suitable for computerisation.

ALTERNATIVE METHODS OF APPROXIMATE INTEGRATION - SIMPSONS RULES

Simpson's rules are a more sophisticated form of estimation of areas, which assumes that curves of the ship's lines consist of short parabolic lengths, rather than straight lines between the ordinate stations. These are preferred in the U.K. to the trapezium method, though they are still an approximation of the actual hull lines. There are two different versions of the integration formula, depending upon how many station intervals are included in the data and these are shown below:-

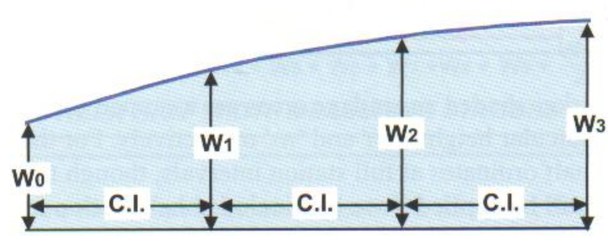
SIMPSON'S RULES

THE 1-4-1 RULE



$$\text{AREA} = \frac{C.I.}{3} \left(W_0 + 4\,W_1 + W_2 \right)$$

THE1-3-3-1 RULE



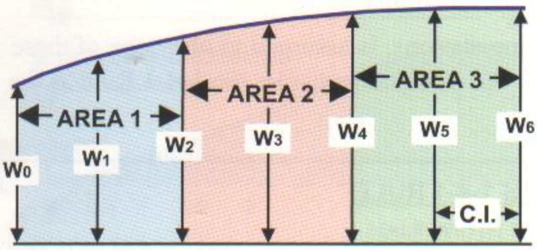
$$\text{AREA} = \frac{3 \times C.I.}{8} \left(W_0 + 3\,W_1 + 3\,W_2 + W_3 \right)$$

SIMPSON'S R ULES (Cont.)

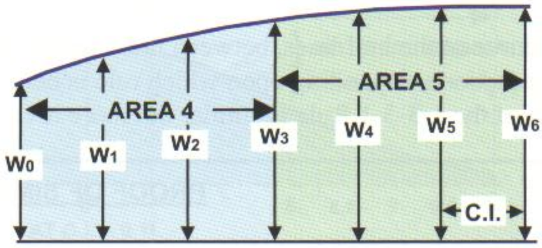
The total area under a curve can be estimated by combining the separate areas, defined either by groups of three (using Simpson's 1-4-1 Rule) or four (using 1-3-3-1 Rule) ordinates, as shown below.

SIMPSON'S RULES OF APPROXIMATE INTEGRATION

1- 4- 1 RULE



1- 3- 3 -1 RULE



1- 4- 1 RULE

$$\text{AREA 1} = \frac{C.I.}{3}(W_0 + 4W_1 + W_2)$$

$$+\ \text{AREA 2} = \frac{C.I.}{3}(W_2 + 4W_3 + W_4)$$

$$+\ \text{AREA 3} = \frac{C.I.}{3}(W_4 + 4W_5 + W_6)$$

$$\text{TOTAL AREA} = \frac{C.I.}{3}(W_0 + 4W_1 + 2W_2 + 4W_3 + 2W_4 + 4W_5 + W_6)$$

1- 3- 3 -1 RULE

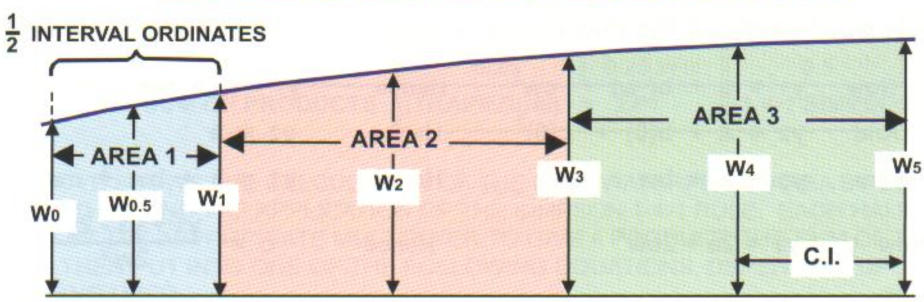
$$\text{AREA 4} = \frac{3xC.I.}{8}(W_0 + 3W_1 + 3W_2 + W_3)$$

$$+\ \text{AREA 5} = \frac{3xC.I.}{8}(W_3 + 3W_4 + 3W_5 + W_6)$$

$$\text{TOTAL AREA} = \frac{3xC.I.}{8}(W_0 + 3W_1 + 3W_2 + 2W_3 + 3W_4 + 3W_5 + W_6)$$

WHERE HALF ORDINATES ARE USED, THE MULTIPLYING FACTORS (1-4-1 OR 1-3-3-1) ARE HALVED

THE 1- 4- 1 RULE INCLUDING HALF INTERVAL ORDINATES



1- 4- 1 RULE

$$\text{AREA 1} = \frac{C.I.}{3}(0.5W_0 + 2W_{0.5} + 0.5W_1)$$

$$+\ \text{AREA 2} = \frac{C.I.}{3}(W_1 + 4W_2 + W_3)$$

$$+\ \text{AREA 3} = \frac{C.I.}{3}(W_3 + 4W_4 + W_5)$$

$$\text{TOTAL AREA} = \frac{C.I.}{3}(0.5W_0 + 2W_{0.5} + 1.5W_1 + 4W_2 + 2W_3 + 4W_4 + W_5)$$

PROOF OF SIMPSON'S RULES

It is not necessary to understand the proof of Simpson's rules in order to use them so the following explanation is just for the benefit of those readers curious enough to want to know. The proof does require an understanding of basic calculus to follow and involves two types of parabolic equations.

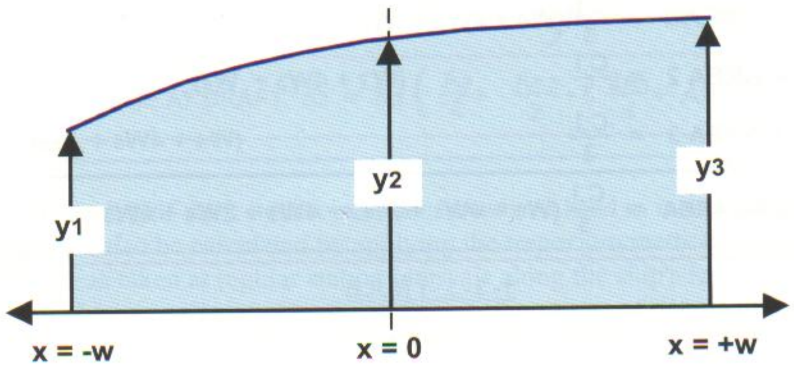
A 2nd order parabolic equation has the form of '$y = a + bx + cx^2$' (Simpson's 1-4-1 Rule)
A 3rd order parabolic equation has the form of '$y = a + bx + cx^2 + dx^3$' (The 1-3-3-1 Rule)

Where a, b, c and d are constants

The curve of the hull lines, between the selected measuring stations, is assumed to obey one of these two equations, depending upon which rule is being used. We shall just consider the 1-4-1 Rule as the proof of the 1-3-3-1 Rule is similar but more involved.

PROOF OF SIMPSON'S 1-4-1 RULE

A 2ND ORDER PARABOLIC CURVE IS DEFINED BY THREE y ORDINATES, 'y' 1, y2 AND y3, SPACED AT THE COMMON INTERVAL OF 'w'. THE CURVE CAN BE EITHER CONCAVE, AS SHOWN, OR CONVEX



IN GENERAL, $y = a + bx + cx^2$

And AREA UNDER CURVE $= \int_{-w}^{w} a + bx + cx^2$ So AREA $= \left[ax + \frac{bx^2}{2} + \frac{cx^3}{3} \right]_{-w}^{w}$

So THE AREA UNDER THE CURVE $= \frac{1}{3} 2cw^3 + 2aw$

WE NOW MUST EXPRESS THE CONSTANTS 'a' AND 'c' IN TERMS OF THE 'y' ORDINATES

Now $y_1 = a + b(-w) + cw^2$ Hence $y_1 = a - bw + cw^2$

And $y_2 = a + b(0) + c(0)^2$ Hence $y_2 = a$

Also $y_3 = a + bw + cw^2$ Hence $y_3 = a + bw + cw^2$

Now $y_1 + y_3 = 2a + 2cw^2$ Hence $c = \frac{y_1 + y_3 - 2y_2}{2w^2}$

WE CAN NOW RETURN TO THE EQUATION FOR THE AREA, SUBSITUTING 'a' AND 'c'

THE AREA UNDER THE CURVE $= \frac{1}{3}2w^3 \; \frac{y_1 + y_3 - 2y_2}{2w^2} + 2wy_2$

So THE AREA UNDER THE CURVE $= w\left(\frac{1}{3}y_1 + \frac{1}{3}y_3 - \frac{1}{3}2y_2 + 2y_2\right)$

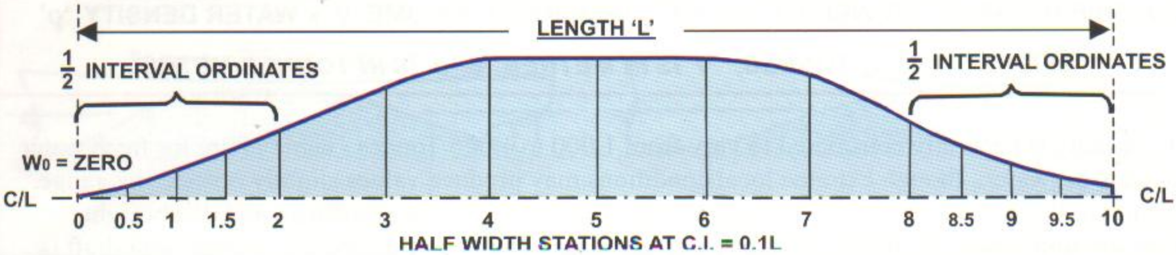
Hence THE AREA UNDER THE CURVE $= w\left(\frac{1}{3}y_1 + \frac{4}{3}y_2 + \frac{1}{3}y_3\right)$

WE NOW HAVE THE MULTIPLIERS '1', '4' & '1' WITH THE COMMON INTERVAL FACTOR OF '1/3'

THE PROCEDURE TO ESTIMATE THE AREA UNDER A CURVE

Whichever method of approximate integration is used to calculate any area under a curve, the key to conducting such a calculation, is to lay out the figures in a tabular form and follow a simple methodical procedure, as shown in the following example

THE PROCEDURE FOR CALCULATING A WATERPLANE AREA



HALF WIDTH MEASURING STATION	HALF WIDTH ORDINATE	TRAPEZIUM RULE		SIMPSON 1-4-1 RULE	
		MULTIPLIER	PRODUCT	MULTIPLIER	PRODUCT
0	W_0	0.25	0.25 W_0	0.5	0.5 W_0
0.5	$W_{0.5}$	0.5	0.50 $W_{0.5}$	2	2 $W_{0.5}$
1	W_1	0.5	0.50 W_1	1	W_1
1.5	$W_{1.5}$	0.5	0.50 $W_{1.5}$	2	2 $W_{1.5}$
2	W_2	0.75	0.75 W_2	1.5	1.5 W_2
3	W_3	1	W_3	4	4 W_3
4	W_4	1	W_4	2	2 W_4
5	W_5	1	W_5	4	4 W_5
6	W_6	1	W_6	2	2 W_6
7	W_7	1	W_7	4	4 W_7
8	W_8	0.75	0.75 W_8	1.5	1.5 W_8
8.5	$W_{8.5}$	0.5	0.50 $W_{8.5}$	2	2 $W_{8.5}$
9	W_9	0.5	0.50 W_9	1	W_9
9.5	$W_{9.5}$	0.5	0.50 $W_{9.5}$	2	2 $W_{9.5}$
10	W_{10}	0.25	0.25 W_{10}	0.5	0.5 W_{10}
	SUM OF PRODUCTS	(TRAPEZIUM) = ΣP_T		(1-4-1 SIMPSONS) = ΣP_S	

THE TABLE ABOVE SHOWS THE CALCULATION OF THE HALF WATERPLANE AREA BY BOTH THE TRAPEZIUM METHOD AND THE APPLICATION OF THE SIMPSON 1-4-1 RULE. EACH HALF WIDTH IS MULTIPLIED BY THE APPROPRIATE MULTIPLYER TO GIVE A PRODUCT. THE SUM OF THESE PRODUCTS IS THEN PUT INTO ONE OF THE FOLLOWING EQUATIONS, DEPENDING UPON WHICH METHOD IS BEING USED.

½ WATERPLANE AREA BY TRAPEZIUM RULE = $0.1\,L\Sigma P_T$

TOTAL WATERPLANE AREA BY TRAPEZIUM RULE = $0.2\,L\Sigma P_T$

So ½ WATERPLANE AREA BY SIMPSON'S RULE = $\frac{0.1L}{3}\Sigma P_S$

So TOTAL WATERPLANE AREA BY SIMPSON'S RULE = $\frac{0.2L}{3}\Sigma P_S$

IF SIMPSONS METHOD IS TO BE USED, THEN, IN THIS EXAMPLE, THE 1-4-1 RULE MUST BE APPLIED TO FIT EXACTLY THE NUMBER OF MEASURING STATIONS USED IN THE CALCULATION

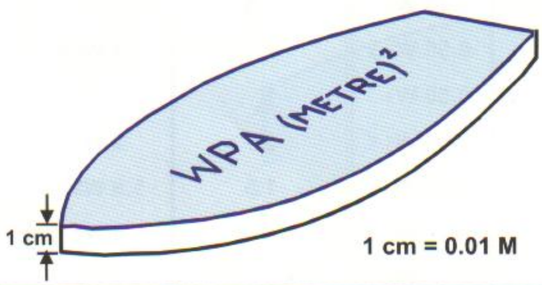
DISPLACEMENT CALCULATIONS TONNES PER CENTIMETRE AND FRESH WATER ALLOWANCE

If we know a ship's underwater volume at given drafts and we know the density of the water it floats in, then the Law of Floatation states that:-

The weight of a floating body = The weight of the volume of fluid it displaces

> **I.E. A SHIP'S DISPLACED WEIGHT 'ΔT' = ITS IMMERSED VOLUME 'V' x WATER DENSITY 'ρ'**
>
> ***WHERE 'ΔT' IS IN TONNES, 'V' IS IN METRES^3 & 'ρ' IS IN TONNES/METRE^3***

Water density is generally considered to vary from 1.000 to 1.025 Tonnes / cubic metre for fresh water through to seawater , though extreme local conditions may produce values slightly outside this range. It is normal, however, to use these two figures to produce scales of displacement and deadweight against the ship's mean draft for both fresh and salt water. The rate of increase in weight with draft is expressed in **'Tonnes required to increase the draft by 1 cm'** or **'TPC'.** For a ship shaped hull, the TPC increases as the vessel is loaded deeper and its submerged hull form becomes fuller.



THE TPC IS THE WEIGHT OF WATER VOLUME DISPLACED BY A 1 cm WATERPLANE SLICE

$$\text{i.e. TPC} = \frac{\text{WPA}}{100} \times \text{WATER DENSITY}$$

So

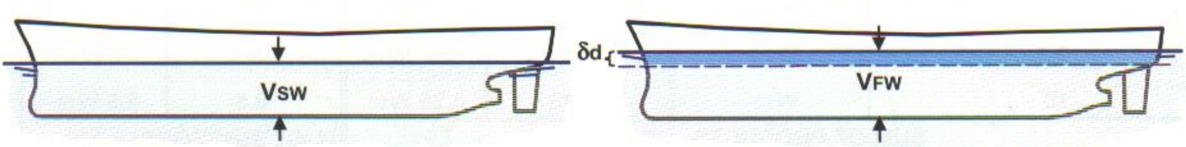
$$TPC_{SW} = 0.01025\ \text{WPA}\quad \text{T/cm}$$

$$TPC_{FW} = 0.01000\ \text{WPA}\quad \text{T/cm}$$

If a ship moves from salt to fresh water, there will be a bodily sinkage to compensate for the decrease in the water's density. This change in draft can be related to the TPC provided that the waterplane area does not significantly alter in the change.

FRESH WATER ALLOWANCE 'FWA'

SALT WATER | **FRESH WATER**



$$'\Delta T' = 1.025 \times V_{SW}\ \text{T} \qquad\qquad '\Delta T' = (V_{SW} + \text{WPA} \times \delta d) \times 1.000$$

WHERE 'δd' IS THE CHANGE IN DRAFT BETWEEN SW AND FW

So $1.025\ V_{SW} = V_{SW} + \text{WPA} \times \delta d$ THEREFORE $.025\ V_{SW} = \text{WPA} \times \delta d$

WHERE $= \dfrac{TPC_{SW}}{0.01025}$ *AND* $V_{SW} = \dfrac{\Delta T}{1.025}$

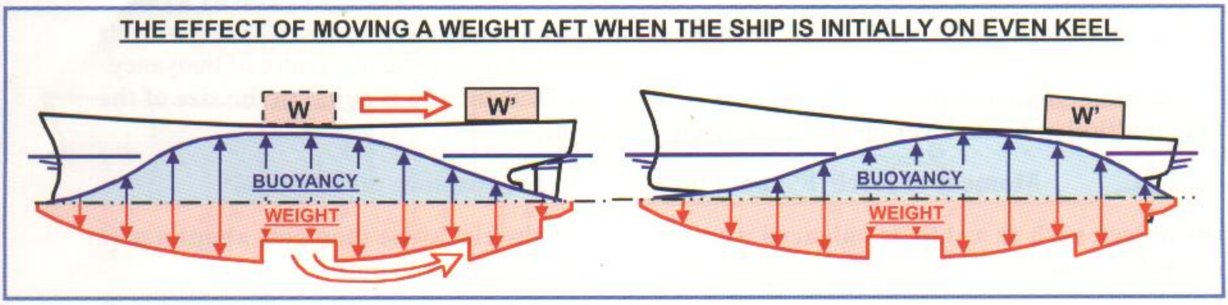
So if $\delta d = \dfrac{0.025\ V_{SW}}{\text{WPA}}$ THEN $\delta d = \dfrac{0.025\Delta T \times 0.01025}{TPC\ sw + 1.025}$

> **So THE FRESH WATER ALLOWANCE 'δd' =** $\dfrac{\Delta T}{4000\ TPC_{SW}}$ **M or** $\dfrac{\Delta T}{40\ TPC_{SW}}$ **cm**

The Freshwater Allowance, or **FWA**, for the summer loaded draft is marked on the ship's side amidships, in addition to the seasonal and regional allowances to the load limits.

THE SHIP'S ANGLE OF TRIM AND HEEL

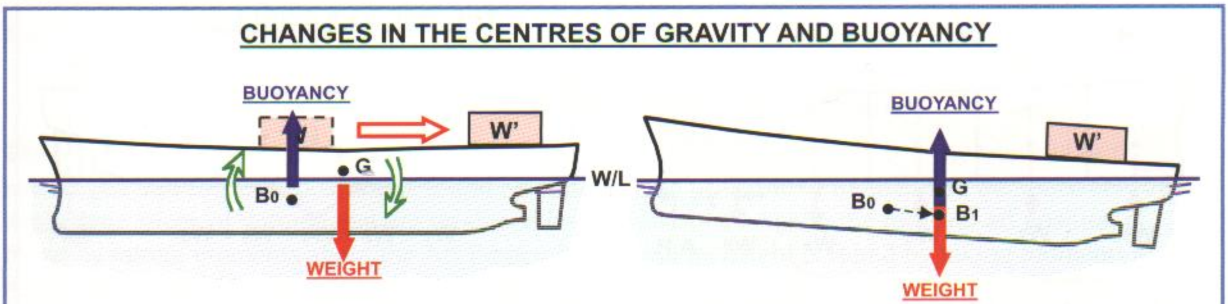
A ship floats if buoyancy due to the displaced water equals the ship's weight. However, it may lean over to one side, (i.e. have an angle of heel) and/or sit lower in the water at the stern or the bow (i.e. have an angle of trim). The attitude of the ship in the water will adjust until the overall distributions of weight and buoyancy balance each other out across its beam and length.



On even keel, the buoyancy distribution is approximately symmetrical about midships with the greatest concentration being found in the parallel body parts of the hull where the immersed volume is the largest. In the loaded condition, the weight distribution tends to follow a similar pattern as the midships region also has the greatest carrying capacity but it is more variable if there are empty spaces to provide alternative stowage arrangements. The sketches above illustrate this. Initially the areas enclosed by the buoyancy and weight distribution curves are both equal and centred around the same point in the midships region of the ship's length. Moving the weight *w'* from the midships region to the stern causes the concentration of weight distribution to move aft. The ship consequently sinks deeper at the stern and rises at the bow. In doing so, aft buoyancy is increased whilst there is corresponding decrease in buoyancy forward. Eventually the ship settles with a stern trim as the two curves of weight and buoyancy reach a new balanced distribution. A similar sequence of events will occur if we move a weight outboard from the centreline and cause the ship to heel over to the heavier side.
Changes in weight distribution cause a change of a ship's underwater shape as the immersed hull adjusts to reach a new balance between the distributions of weight and buoyancy.

THE CENTRES OF BUOYANCY AND GRAVITY

Although buoyancy acts along the entire length of the hull by varying degrees, there is a single point about which all the separate turning effects, or **moments**, of buoyancy cancel each other out. This point is called **the Centre of Buoyancy, or C of B. It is the geometric centre of volume for the underwater hull and we can consider that the entire force of buoyancy acts through this single point.** Similarly, the entire weight of the vessel can be considered to act through **the centre of gravity, or the 'C of G'. The weight and buoyancy distributions are balanced when these two points are in vertical alignment.** When the two points are out of alignment, a turning moment is produced which causes the hull to adjust its attitude in the water and consequently re-align the C of B vertically with the C of G.



In the left-hand sketch, there is a trimming moment by the stern. The hull responds by adjusting its attitude in the water, which changes buoyancy distribution so that the C of B moves from B_0 to B_1 to produce the new state of equilibrium shown by the right hand sketch.

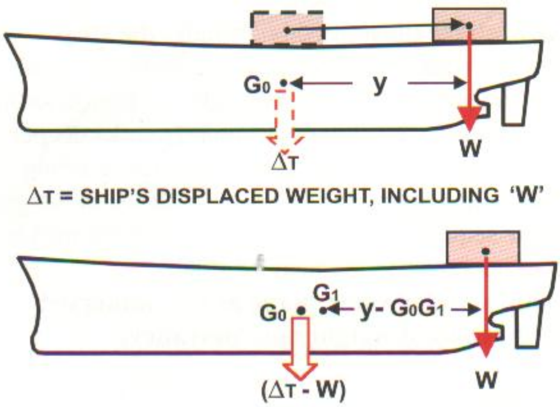
APPLYING THE PRINCIPLE OF MOMENTS TO LOCATING THE CENTRES OF GRAVITY AND BUOYANCY

It is very convenient to consider that all the separate forces of weight and buoyancy act only through two points, namely the C of G and C of B respectively. However, in order to do this, we must locate the positions of these two points. The key to achieving this lies in the fact that all the individual weights balance about the centre of gravity. A ship would balance on its centre of gravity as the turning effects of all the weights trying to tip it one way are cancelled out by the effect of weights trying to tip it the other. Similarly, all the buoyancy forces balance about the centre of buoyancy. **The turning effect of a force about a point is called its *Moment* and is equal to the size of the force multiplied by its leverage distance from that point.**

i.e. **Moment of force 'F' about point 'P' = F x distance of 'F' from P**

Any object, such as a ship, is balanced when the sum of the clockwise moments about a point equals the sum of the anti-clockwise moments taken about the same point.
If we look at the weight that was moved aft in the previous page, then we can use the principle of moments to determine the shift of C of G (G_0 to G_1) caused by the movement of that weight.

THE TRIMMING MOMENT DUE TO A SHIFT OF A SINGLE WEIGHT



THE SHIP BALANCES ABOUT G_0 BEFORE MOVING W AFT BUT NOW THE C of G MUST SHIFT TO A NEW POINT OF BALANCE G_1, WHERE

$$(\Delta T - W)G_0G_1 = W(y - G_0G_1)$$

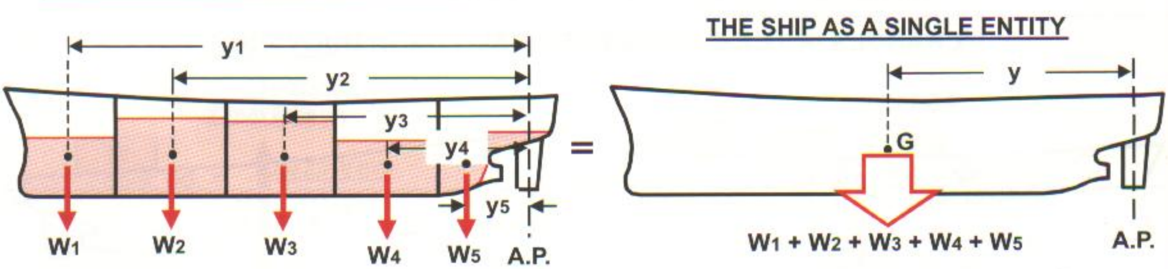
So $\Delta T\, G_0G_1 - W\, G_0G_1 = W y - W\, G_0G_1$

Therefore $\Delta T\, G_0G_1 = W y$

So $$\boxed{G_0G_1 = \frac{W y}{\Delta T}}$$

We can apply the Principle of Moments to locate the Centre of Gravity of the entire ship, providing that we have a detailed knowledge of the ship's weight distribution. We chose a convenient turning axis on the ship, (the aft perpendicular is used when locating the fore and aft position of the C of G) and calculate the moments of all the individual weights about this axis. The sum of these individual moments must equal the moment of the ship's total weight about the same point. As we know the value of the total weight, we can work out the distance of the C of G from this axis as follows:-

LOCATING THE LONGITUDINAL POSITION OF THE CENTRE OF GRAVITY (THE LCG)



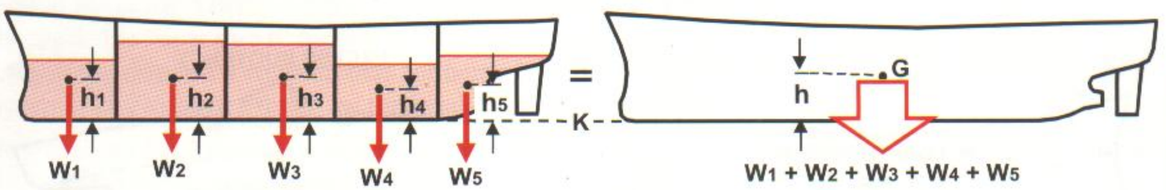
$$W_1y_1 + W_2y_2 + W_3y_3 + W_4y_4 + W_5y_5 = (W_1 + W_2 + W_3 + W_4 + W_5)\,y$$

So DISTANCE 'y' OF 'G' FWD OF THE A.P. $= \dfrac{W_1y_1 + W_2y_2 + W_3y_3 + W_4y_4 + W_5y_5}{(W_1 + W_2 + W_3 + W_4 + W_5)}$ M

APPLYING THE PRINCIPLE OF MOMENTS TO LOCATING THE CENTRES OF GRAVITY AND BUOYANCY (cont.)

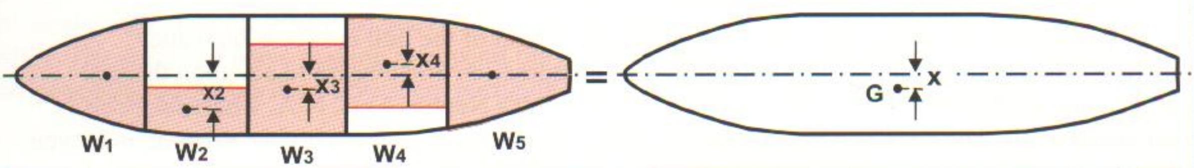
If we want to locate the vertical position of the Centre of Gravity then we apply the method of the previous page to moments taken about the keel, whilst its athwartships position is determined by taking moments about the centreline.

DETERMINING THE VCG (THE VERTICAL HEIGHT OF THE C of G)



$$\text{HEIGHT 'h' OF G ABOVE THE KEEL} = \frac{W_1h_1 + W_2h_2 + W_3h_3 + W_4h_4 + W_5h_5}{(W_1 + W_2 + W_3 + W_4 + W_5)} \text{ M}$$

THE ATHWARTSHIPS OFFSET OF THE C of G FROM THE CENTRELINE



$$\text{OFFSET 'x' OF G FROM THE CENTRELINE} = \frac{W_1 \times \text{ZERO} + W_2X_2 + W_3X_3 + W_4X_4 + W_5 \times \text{ZERO}}{(W_1 + W_2 + W_3 + W_4 + W_5)} \text{ M}$$

It must be appreciated that these sketches are a very simplified representation of the entire weight distribution of a real ship and its cargo. Naval architects who design the ship will estimate the position of the Lightship Centre of Gravity from their knowledge of the distribution of the ship's structural weight. The actual co-ordinates of the C of G, however, must be confirmed by experiments carried out on the actual ship on completion of its building. The Inclining experiment measures the height of the C of G above the keel and its position relative to the centreline whereas its longitudinal position is determined by the trim of the ship in lightship condition and knowledge of its Centre of Buoyancy at that draft and trim..

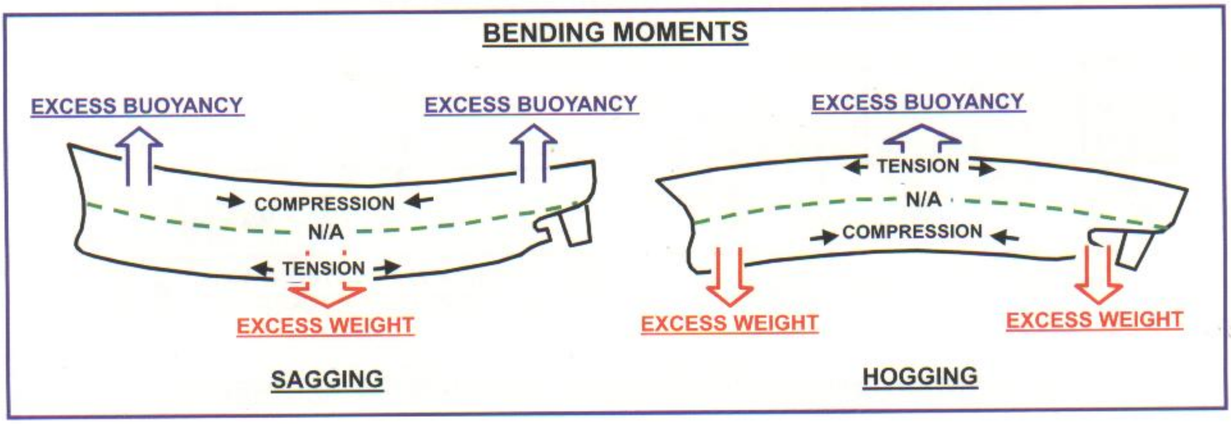
Determining the Centres of Gravity of individual weights of cargo, fuel stores etc. involves a certain degree of estimation and approximation on the part of the ship's officers. The shipbuilders will provide detailed information on the positions of the centres of volume (or Centroid) of every cargo and tank space on the vessel, which can be used as the C of G of any weight of uniform cargo that completely fills a space. Part filled spaces or stows of different commodities in the same compartment do, however, give more scope for error.

LOCATING THE CENTRE OF BUOYANCY

The Centre of Buoyancy is the centre of the displaced volume of water, i.e. it is the centroid of the submerged hull form. Its position is found by applying the principle of moments to the transverse and waterplane slices of volume that are produced by the Linesplan. Again, the fore and aft position is found by taking moments about the Aft Perpendicular and its height is found by taking moments about the keel. This requires applying approximate integration formulae to the offsets taken from the Linesplan. The full analysis of the underwater hullform must determine the shift of the C of B resulting from changes in draft, trim and heel as these are essential factors affecting the ship's hydrostatic behaviour. It is a lengthy procedure, which requires the processing of a lot of data but has been standardised by the shipyards over a long period of time.

THE BENDING OF A SHIP'S HULL BY THE FORCES OF WEIGHT AND BUOYANCY

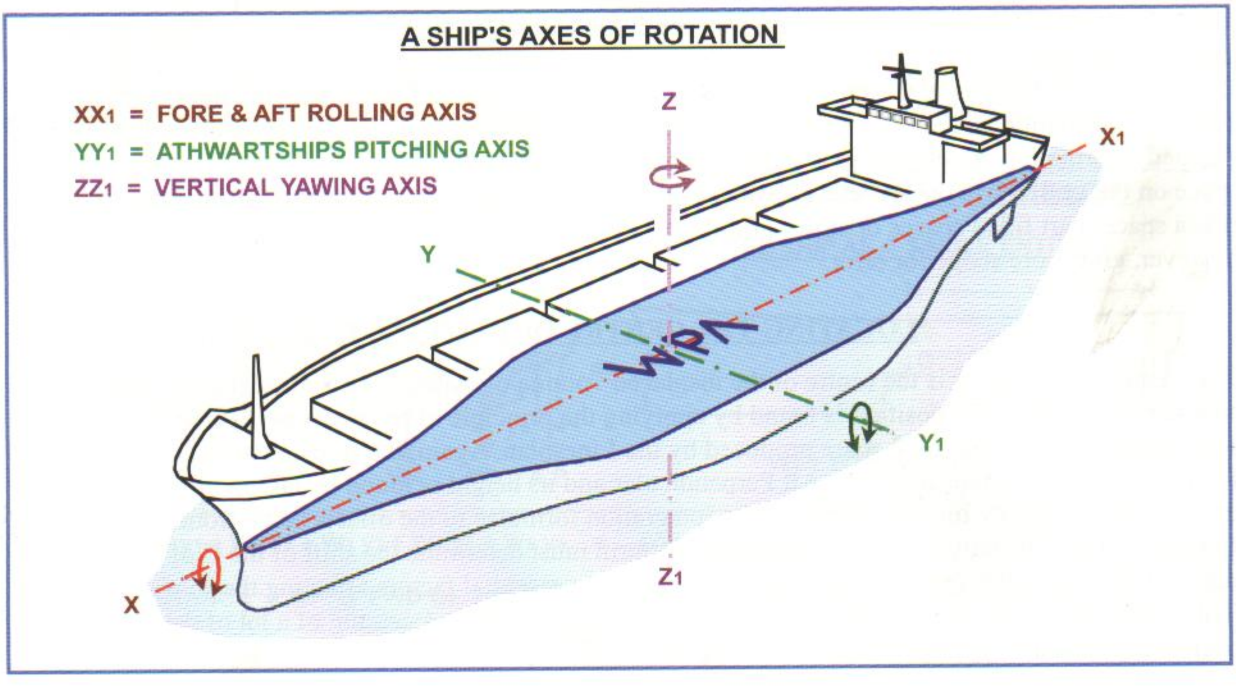
The distribution of weight and buoyancy can still be mis-matched, even though the centres of gravity and buoyancy are in vertical alignment. In this situation, the hull is subjected to bending rather than trimming moments which either cause the middle of the hull to sink lower **(sagging)** or rise higher (**hogging)** than the fore and aft ends.



In both sagging and hogging, the excess weight in one region of the hull is excess buoyancy elsewhere. The bending causes deforming forces of compression and tension to act upon the hull plating, particularly at positions furthest away from the hull's Neutral Axis, N/A, where no stress is experienced at all. The deck and keel plating in the midships region is most highly stressed and, even if the weight and buoyancy distribution are well matched in still water, the action of waves moving along the hull will cause an alternating cycle of hogging and sagging. A ship must be built to withstand a degree of bending stresses so that it can flex in a seaway without breaking up. It is important that the safe limits of these stresses are well defined by the shipbuilder so that the officers responsible for the load distribution of the vessel can ensure that they are not exceeded.

A SHIP'S MOTION IN A SEAWAY

A ship's motion at sea is a complex combination of swinging to and fro about the three mutually perpendicular axes through the waterplane, as shown below:-



The waterplane axes, shown on the previous page, continually shift in position relative to each other and the ship because the buoyancy distribution is always changing as the underwater hullform alters shape with the pitching and rolling. The terminology used for these forms of motion is as follows:-

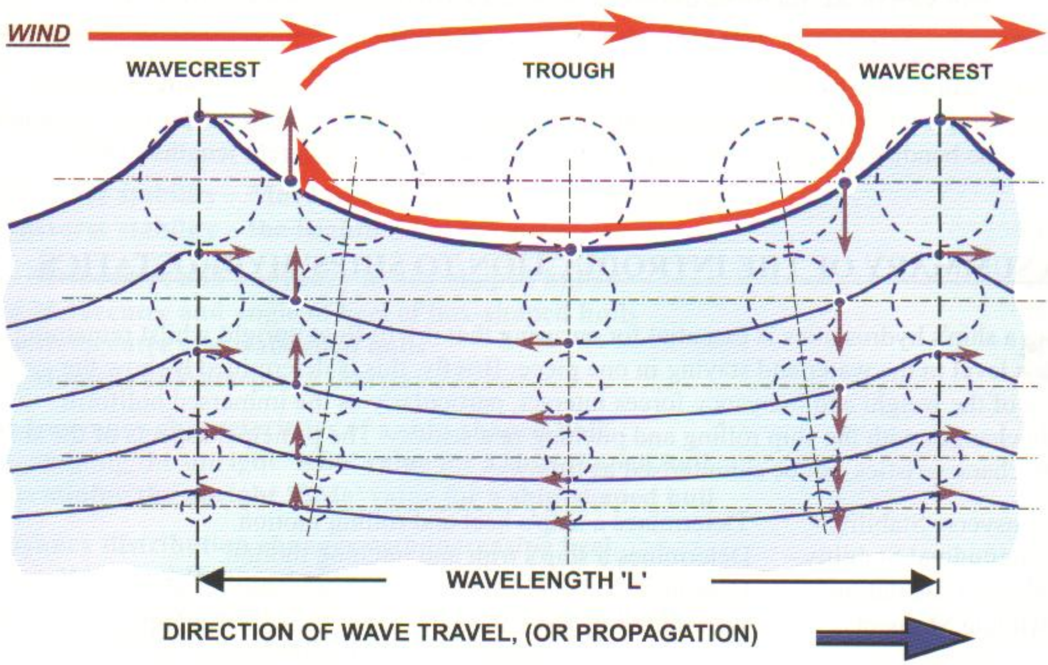
Rotation about the **Fore & Aft** axis is known as ***'Rolling' or 'Heeling'.*** If the ship has a constant bias for heeling to one side (port or starboard), due to the distribution of weight then the ship is said to have an ***'angle of list'***.

Rotation about the **Athwartships** axis is usually known as ***'Pitching',*** (though traditionally, this was only applied to **'bow down'** motion and the term ***'Scending'*** referred to the **'bow up'** half of the cycle)**.** If the ship has a constant bias for being bow or stern down, due to the weight distribution then the ship is said to have a ***'trim'*** by the head or the stern respectively.

Rotation about the **Vertical** axis is known as ***'Yawing'***. This affects the ship's steering stability and is not one of the topics covered by this book.

All three modes of motion tend to react with each other so rolling can induce pitching and vice-versa, hence the complex motion of a ship when subjected to moderately rough seas. This book is primarily concerned with how the weight carried within a ship should be distributed to ensure that the vessel meets certain government requirements with regard to the still water buoyancy distribution at any particular draft. The criteria used for making these regulations are based upon ensuring that the ship can safely withstand the effects of heavy seas and, as such, it is worth considering the nature of sea waves. The theory of wave making is beyond the scope of this book, but the wind turbulence in the air that generates sea waves does give them particular characteristics, which are shown below:-

DEEP WATER WAVE PROFILE & MOTION



THE VERTICAL SCALE IS EXAGGERATED, RELATIVE TO THE HORIZONTAL SCALE

THE TURBULENCE OF THE WIND SETS THE WATER PARTICLES MOVING IN ORBITAL MOTION. THIS PRODUCES A DISTINCTIVE PROFILE WITH LONG TROUGHS AND SHORT STEEPER CRESTS THAT IS KNOWN AS A 'TROCHODIAL' WAVEFORM, WHICH HAS THE FOLLOWING RELATIONSHIPS

WAVE SPEED 'C' = 1.56 x WAVE PERIOD 'T' M/S or 'C' = $\sqrt{\textbf{1.56 x WAVELENGTH 'L'}}$ **M/S**

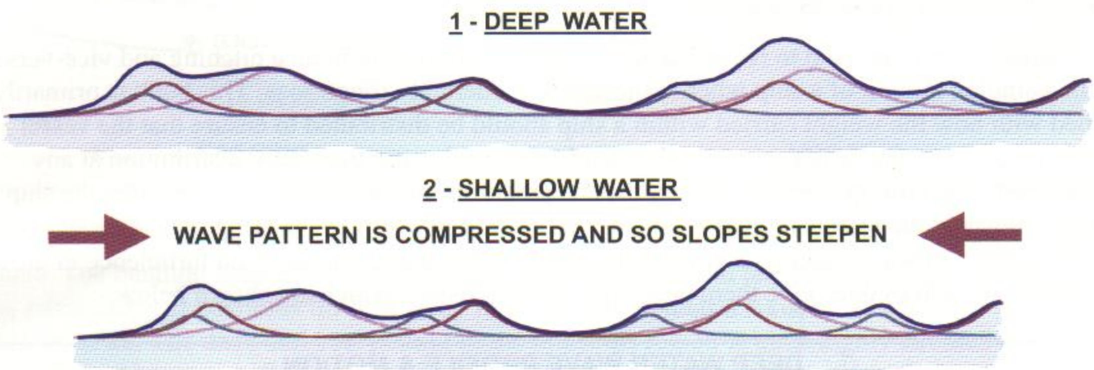
FOR WAVES LESS THAN 150 METRES

THE WAVE HEIGHT FROM CREST TO TROUGH $\approx$ 0.05 x WAVELENGTH 'L' METRES

A SHIP'S MOTION IN A SEAWAY (Cont.)

The deep water wave profile, shown on the previous page, is often modified by the interaction of the wave with other waves or with the seabed, which occurs when waves encounter depths less than their wavelength. When waves of different lengths are superimposed over each other, they sometimes tend to cancel each other out in one place, whilst, elsewhere they will re-inforce each other. This produces the typical rough sea in which a ship will encounter exceptionally large waves, interspersed with calmer patches of water.
The disturbance of sea by the wind becomes progressively less at increasing water depth below the surface and has effectively disappeared at a depth equal to the wavelength. Waves in shallower water that 'feel the bottom' will slow down and so the crests will be drawn closer together, which will increase the steepness of the water slopes encountered by a ship.

SEA PROFILE PRODUCED BY SUPERIMPOSING THREE DIFFERENT WAVELENGTHS



THE VERTICAL TO HORIZONTAL SCALE IS EXAGGERATED BY ABOUT 3 : 1

Wave characteristics, such as wavelength, wave height, steepness and period are fundamental in determining a ship's pitching and rolling behaviour, whilst knowledge of the wave profile is essential for assessing the bending moments that a ship will be subjected to by different lengths of waves.

A SUMMARY OF THE INTRODUCTION TO SHIPS' HYDROSTATICS

Calculating a ship's hydrostatics is essential for ensuring that it will float upright whilst remaining more or less level in the water and staying in one piece. Briefly, this is determined by how the distribution of the weight and buoyancy forces interact, particularly as the immersed hullform continually changes with the ship rolling and pitching in a seaway. The principal aspects of the ship's hydrostatic characteristics can be summarised as follows:-

1) Transverse Stability.- Determines a ship's heel and rolling motion.
2) Longitudinal Stability,- Determines a ship's trim and pitching motion.
3) Bilging Calculations,- Determine a ship's ability to withstand partial flooding.
4) Bending Moments,- Determines a ship's ability to resist being broken up.

It is common practice for students to consider these items as separate topics and there is a certain amount of convenience in doing so. However, it is worth remembering that they are all different effects of the interaction of the same two forces of weight and buoyancy and that the same principles are applied to transverse and longitudinal stability.
Bilging involves the loss of buoyancy as part of the underwater hull is flooded and the ship must retain positive stability, as well as buoyancy, after this change of immersed hullform has occurred.
The study of bending moments is necessary to determine whether a ship's weight distribution is within safe limits to withstand the cycle of sagging and hogging stresses due to the continual changes in the buoyancy distribution as the ship pitches and rolls in a seaway.

CHAPTER 2

AN INTRODUCTION TO THE SHAPE OF A SHIP'S HULLFORM AND THE PRINCIPLES OF HYDROSTATICS THAT ACT UPON IT

SUMMARY

THIS CHAPTER INTRODUCES TRANSVERSE STABILITY AND THE RIGHTING MOMENT. IT CONSIDERS MEANS FOR DETERMINING THE BUOYANCY DISTRIBUTION AND HOW THIS CHANGES WITH THE ANGLE OF HEEL BY EXAMINING THE FOLLOWING: -

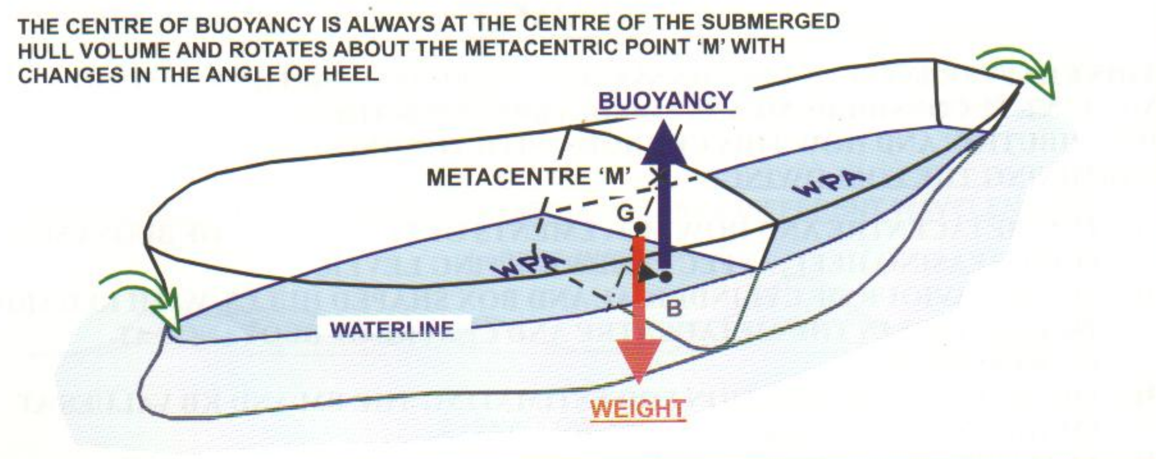
1) **THE METACENTRE AND HOW MOVEMENTS OF THE CENTRE OF BUOYANCY AT INCREASING HEEL, AFFECT THE RIGHTING LEVER.**
2) **THE BEHAVIOUR OF CYLINDRICAL AND BOX SHAPED HULLS, WITH REGARD TO THE SHIFT IN THE METACENTRE AND CENTRE OF BUOYANCY AT INCREASING HEEL.**
3) **THE WALL-SIDED EQUATION FOR ESTIMATING THE BM AND KB VALUES AT SMALL ANGLES OF HEEL**
4) **ANALYSIS OF AN UPRIGHT SHIP-SHAPED HULL, BY APPROXIMATE INTEGRATION TO CALCULATE THE 'KB0' AND 'BM0' VALUES.**
5) **ANALYSIS OF THE SHIP-SHAPED HULL AT INCREASING ANGLES OF HEEL, TO TRACK THE SHIFT IN THE CENTRE OF BUOYANCY AND SO PRODUCE 'KN' CURVES FOR USE IN STABILITY CALCULATIONS.**

CONTENTS

TRANSVERSE STABILITY – MOVEMENTS OF THE CENTRE OF BUOYANCY

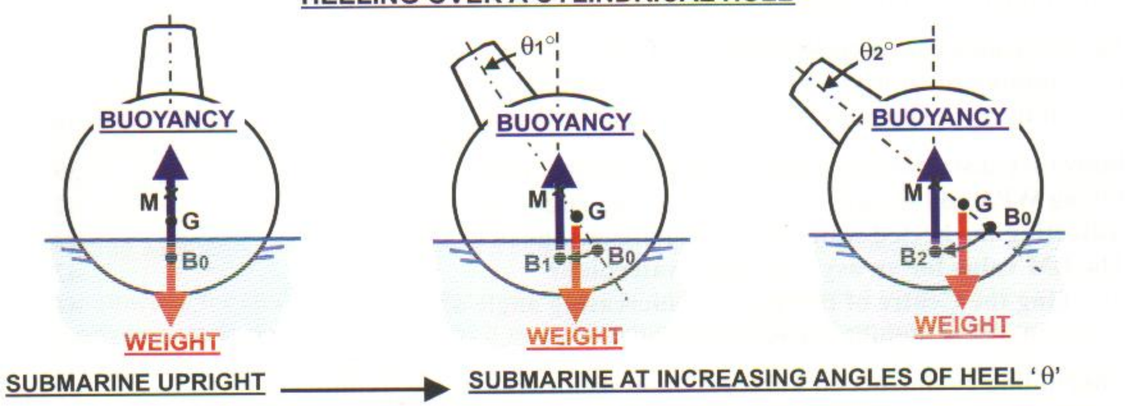
When a ship is heeled over, the underwater shape and hence its distribution of buoyancy changes, which causes a shift in the position of the Centre of Buoyancy, as shown by the following illustration.

THE FORCES PRODUCED BY HEELING A SHIP-SHAPED HULL



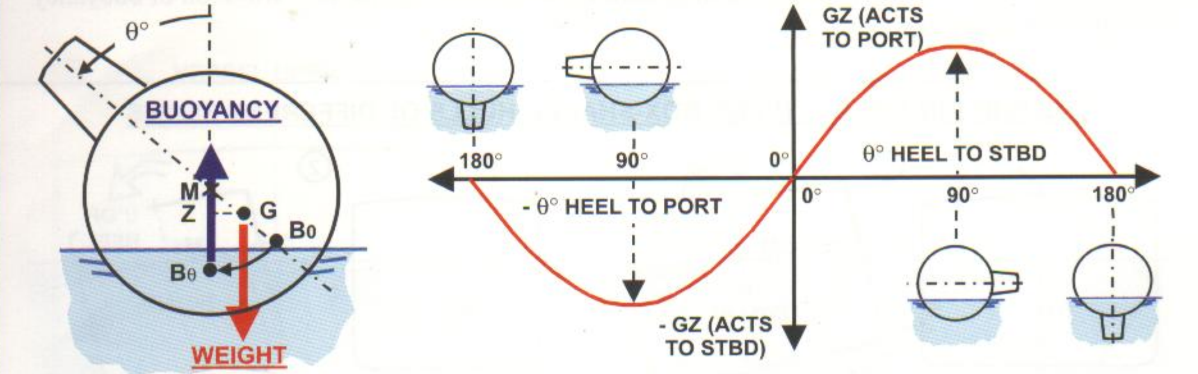
As a ship-shaped hull heels over, water is displaced from the downward side and the underwater shape becomes asymmetrical about the centreline. This causes the Centre of Buoyancy 'B' to swing in an arc of radius BM **about a point called the 'Metacentre 'M'**, towards the down side of the underwater volume so the upward vertical force of Buoyancy also moves outboard to the low side of the vessel. If the buoyancy's line of action is moved further outboard than the swing of the Centre of Gravity, the two equal but opposing vertical forces of buoyancy and weight produce a turning effect or **Righting Moment** directed at pushing the hull back towards the upright. The horizontal separation of these two forces is known as the **Righting Lever**.
The shift of 'B' is entirely dependent upon how the underwater hull shape changes with heel as this determines the value of the **Metacentric Radius 'BM'**.
A ship's ability to resist being rolled over depends upon how the Centre of Buoyancy moves, relative to Centre of Gravity, as the hull is heeled over. The simplest hullform to consider is the cylindrical shape of a submarine.

HEELING OVER A CYLINDRICAL HULL



The immersed shape of a cylindrical hull remains unchanged with increasing angles of heel 'θ', so the Centre of Buoyancy swings around the Metacentre 'M', which, in this case, coincides with the centre of the circular section of submarine. The Centre of Gravity 'G' also swings about the centre of the circular section and, because 'M' is higher up the centreline than 'G', 'B' remains on the down side of 'G' so the forces of Weight and Gravity act to restore the submarine to the upright.

THE RIGHTING LEVER, OR 'GZ' CURVE OF A CYLINDRICAL HULL



AT ANY ANGLE OF HEEL 'θ', THE DISTANCE 'GZ' IS THE HORIZONTAL SEPARATION BETWEEN THE VERTICAL OPPOSING BUT EQUAL FORCES OF WEIGHT AND BUOYANCY. IT IS THE LEVER BETWEEN THESE TWO FORCES, WHICH ACTS TO RIGHT THE HULL, PROVIDING THAT THE METACENTRE 'M' WHICH THE CENTRE OF BUOYANCY 'B' ROTATES ABOUT, IS HIGHER UP THE CENTRELINE THAN THE CENTRE OF GRAVITY 'G'. 'M' COINCIDES WITH THE GEOMETRIC CENTRE OF A CYLINDRICAL HULL AND SO, IN THIS CASE, REMAINS FIXED AT ALL ANGLES OF HEEL

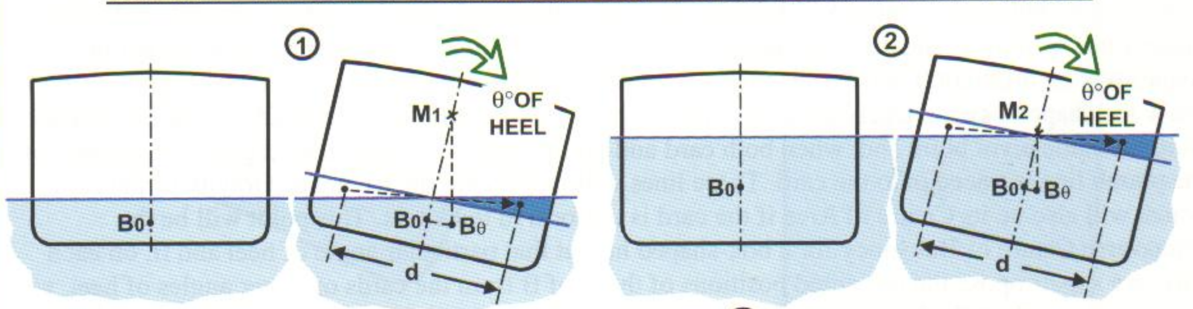
GZ_θ, THE RIGHTING LEVER AT θ° OF HEEL = GM x Sinθ° ***FOR A CYLINDRICAL HULL***
& THE RIGHTING MOMENT AT θ° OF HEEL = GZ_θ x THE SHIP'S DISPLACED WEIGHT

This chapter is concerned with the shift in the Centre of Buoyancy as a ship is heeled over, for this determines a hull's stability characteristics for a given height of the Centre of Gravity above the keel.

THE METACENTRE AND BUOYANCY IN BOX-SHAPED HULLS

If a box-shaped hull is heeled over then, unlike the cylindrical hull, both the water plane area and the submerged (or displaced) volume change shape. At small angles of heel, the change in waterplane is relatively insignificant and the Centre of Buoyancy shifts in response to the change of submerged hull shape, which is due to a wedge of buoyancy being transferred from the high to the low side of the hull. The shift in the C of B, and hence the radius of its swing about the Metacentre 'M', will depend upon the volume of this transferred wedge of buoyancy, relative to the overall displaced volume. This, in turn, will depend upon how deeply laden the hull is for any given angle of heel.

THE SHIFT IN THE C of B FOR A BOX-SHAPED HULL AT DIFFERENT DRAFTS



① **LIGHTSHIP DRAFT** ② **FULLY LADEN DRAFT**

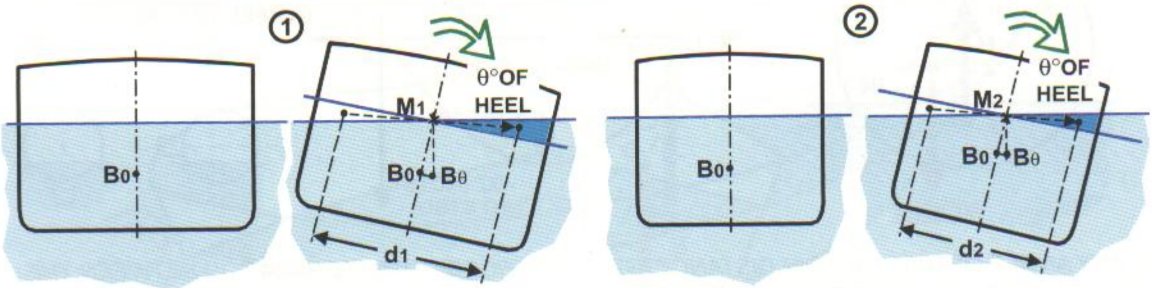
THE SHIFT 'B_0B_θ', DUE TO THE WEDGE OF BUOYANCY 'v' MOVING THROUGH DISTANCE 'd' = $\frac{v \times d}{V}$

UNDERWATER VOLUME 'V' IS MINIMUM SO B_0B_θ AND RADIUS BM ARE MAXIMUM

UNDERWATER VOLUME 'V' IS MAXIMUM SO B_0B_θ AND RADIUS BM ARE MINIMUM

The shift in the Centre of Buoyancy depends upon the volume of the transferred wedge of buoyancy, relative to the total submerged volume and the previous page showed how this is affected by the draft of a box-shaped hull. The beam is also an important factor controlling this redistribution of buoyancy as the following diagram shows;-

THE SHIFT IN THE C of B FOR BOX-SHAPED HULLS OF DIFFERENT BEAMS



① **WIDE BEAM HULL** ② **NARROW BEAM HULL**

THE SHIFT 'B_0B_θ', DUE TO THE WEDGE OF BUOYANCY 'v' MOVING THROUGH DISTANCE 'd' = $\frac{v \times d}{V}$

BOTH THE TRANSFERRED VOLUME 'v' OF BUOYANCY AND THE DISTANCE 'd' THROUGH WHICH IT IS MOVED, ARE GREATER AT WIDER BEAMS, SO THE SHIFT OF THE C of B, 'B_0B_θ' AND ITS RADIUS OF SWING, 'BM' ARE ALSO INCREASED WITH INCREASING BEAM.

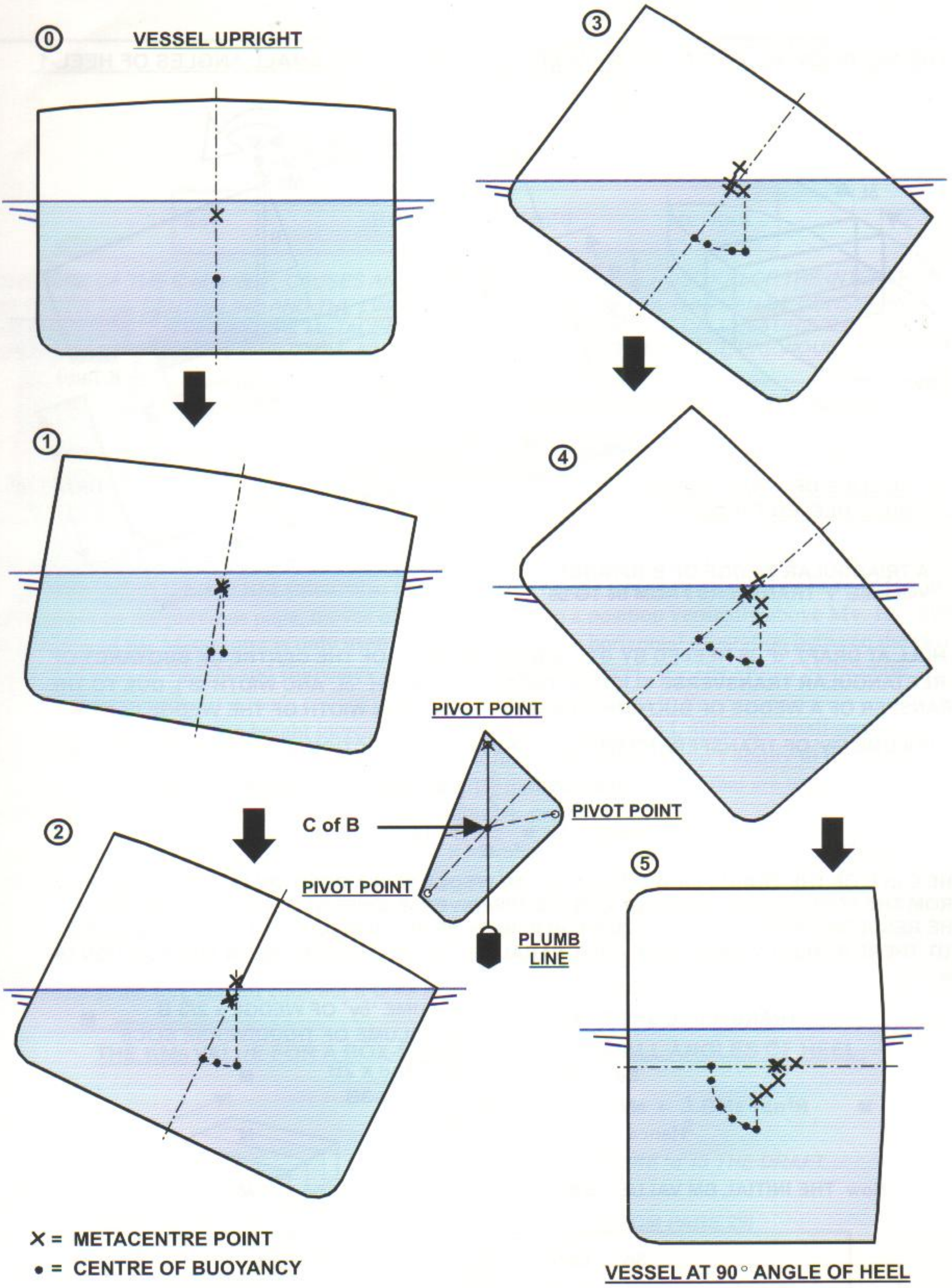
The first pages of this chapter showed that the position of the Metacentre, relative to the Centre of Gravity, is a useful guide to a ship's state of Transverse Stability. The equations for the Righting Lever and, hence, Righting Moment that are true for the cylindrical hull discussed on pages 25 and 26 can also be applied to box-shaped and ship-shaped hulls at small angles of heel where the waterplane shape can also be considered as constant and the Metacentre effectively remains in the same position on the centreline. i.e:-

GZ_θ, THE RIGHTING LEVER AT $\theta°$ OF HEEL = GM x Sin $\theta°$
& RIGHTING MOMENT AT $\theta°$ OF HEEL = GZ_θ x THE SHIP'S DISPLACED WEIGHT
For any vessel at small angles of heel θ

However, if we heel a box or ship shaped hull over progressively larger angles of heel, we will find that the Centre of Buoyancy does not swing through an arc of constant radius about a single fixed metacentric point. Both the BM value and the position of the metacentre continually change as the angle of heel increases. The following page illustrates this by plotting the track of these two points, for a box-shaped hull, over a range of heel from 0° to 90°. We can do this by making shaped cards of the underwater section for the different angles of heel. The area of each card will be the same but the shapes will be different. Each card is then suspended successively on two or three pivot points, which allow the shape to swing freely under gravity. On each such occasion, a plumb line is also suspended from the same pivot point and, when both card and plumb line have stopped swinging, the position of the plumb line is traced onto the card. These lines will intersect at the Centre of Gravity of the card, which will also be its Centre of Area if the card is uniform in thickness. This point will be the Transverse Centre of Buoyancy for a box-shaped hull at that particular angle of heel and if, on each card, we superimpose the measured positions of the C of B from the cards of lesser angles of heel, we can plot the track of the C of B up to the card's own particular angle of heel. We can then draw lines from two adjacent points that are perpendicular to the curved track of the C of B and the approximate position of the Metacentre over that range of heel will be at the intersection of these two lines.
The BM value increases as M rises up and moves off the centreline with increasing angles of heel and waterline beam until the deck edge is immersed. At this point, the trend reverses as the waterplane beam is reduced to its minimum value at 90° of heel.

THE METACENTRE AND BUOYANCY IN BOX-SHAPED HULLS (Cont.)

THE MOVEMENT OF THE CENTRE OF BUOYANCY AND METACENTRE IN A BOX-SHAPED HULL WHEN THE ANGLE OF HEEL IS PROGRESSIVELY INCREASED

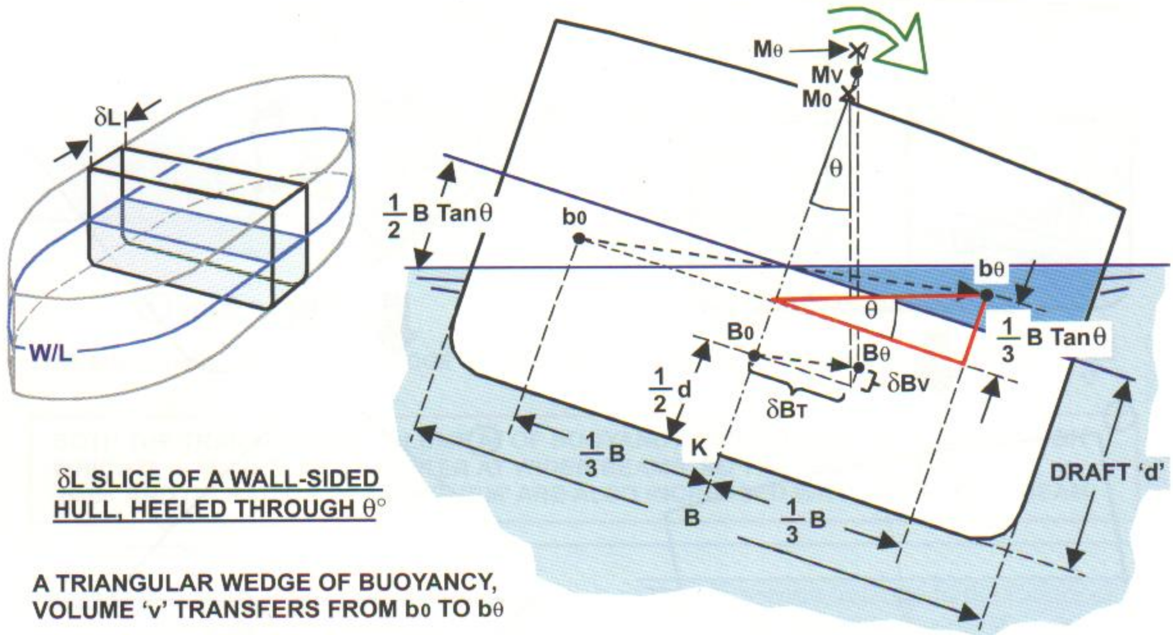


THE C of B MOVES OUTBOARD AT AN INCREASING RATE UNTIL THE DECK EDGE IS SUBMERGED

THE METACENTRE AND ANGLE OF HEEL OF BOX-SHAPED HULLS

The midships section of most merchant ship hulls is rectangular, or box shaped. If we consider heeling over just a rectangular slice of a hull, we can derive an equation for the metacentric radius (otherwise known as the BM value), of this slice in terms of the angle of heel, provided that this is within about 10° and the deck edge is not immersed.

THE METACENTRE FOR A BOX-SHAPED SLICE OF HULL AT SMALL ANGLES OF HEEL-1



δL SLICE OF A WALL-SIDED HULL, HEELED THROUGH θ°

A TRIANGULAR WEDGE OF BUOYANCY, VOLUME 'v' TRANSFERS FROM b0 TO bθ

A HULL AT DRAFT 'd', IS HEELED BY θ°. B0Bθ IS THE SHIFT OF THE CENTRE OF BUOYANCY OF A RECTANGULAR TRANSVERSE SLICE OF THE HULL (LENGTH 'δL' AND WIDTH 'B'), DUE TO THE TRANSFER OF A WEDGE OF BUOYANCY FROM b0 TO bθ. THE WIDTH OF THE WEDGE IS 1/2 B

VOLUME 'δv' OF TRANSFERRED WEDGE = TRIANGULAR SECTIONAL AREA OF WEDGE x δL

$$\text{So} \quad \delta v = \frac{1}{2}\left(\frac{1}{2} B \tan\theta\right) \times \frac{1}{2} \times B \times \delta L \quad M^3$$

$$\text{Hence} \quad \delta v = \frac{\delta L}{8} B^2 \tan\theta \quad M^3$$

THE C of B OF THE TRANSFERRED TRIANGULAR WEDGE IS IN A POSITION 2/3 OF ITS WIDTH FROM THE FORE AND AFT HEELING AXIS, SO TRANSVERSE SHIFT b0 TO bθ = 2 x 2/3 (1/2 B). THE RESULTING SHIFT OF THE SLICE'S C of B 'B0Bθ', IS PREDOMINATELY TRANSVERSE (δBT) BUT THERE IS ALSO A SMALL RISE (δBv), PARALLEL TO THE CENTRELINE, IN THE POSITION OF Bθ.

$$\text{TRANSVERSE SHIFT } '\delta B_T' = \frac{\text{VOLUME } '\delta v' \text{ OF WEDGE} \times 2/3\ B}{\text{TOTAL VOLUME OF TRANSVERSE SLICE}} \quad M$$

$$\text{So} \quad \delta B_T = \frac{\delta L \times B^2 \tan\theta \times 2B}{8 \times \delta L \times d \times 3B} \quad M$$

$$\text{Hence} \quad \delta B_T = \frac{B^2 \tan\theta}{12d} \quad M$$

$$\text{Now THE INITIAL BM VALUE } 'BM_0' = \frac{\delta B_T}{\tan\theta} \quad M$$

$$\text{So} \quad BM_0 = \frac{B^2}{12d} \quad \text{METRES}$$

FOR A SHORT TRANSVERSE WALL-SIDED SLICE , 'B' METRES WIDTH, AT ANGLES OF HEEL LESS THAN DECK EDGE IMMERSION

THE METACENTRE FOR A BOX-SHAPED SLICE OF HULL AT SMALL ANGLES OF HEEL–2

THE RISE OF THE C of B, 'δBv' IN THE WALL-SIDED SLICE AT $\theta°$ OF HEEL, IS GIVEN BY:-

$$\text{THE RISE } \delta Bv = \frac{\text{VOLUME } \delta v \text{ OF WEDGE} \times 1/3\, B \tan\theta}{\text{TOTAL VOLUME OF TRANSVERSE SLICE}} \quad M$$

$$\text{So} \quad \delta Bv = \frac{\delta L \times B^2 \tan\theta \times B\tan\theta}{8 \times \delta L \times d \times 3B} \quad M$$

$$\text{Hence} \quad \delta Bv = \frac{B^2 \tan^2\theta}{24d} \quad M$$

$$\boxed{\text{So} \quad \delta Bv = \frac{1}{2} BM_0 \tan^2\theta \quad \text{METRES}}$$

THIS RISE OF THE C of B 'δBv', CAUSES AN EQUIVALENT RISE IN THE POSITION OF THE INTERSECT BETWEEN THE CENTRELINE AND THE LINE OF ACTION OF BUOYANCY. THIS CAN BE CONSIDERED AS THE EFFECTIVE OR VIRTUAL METACENTRE FOR THE PURPOSES OF CALCULATING THE GM VALUE AND THE SHIP'S STABILITY AT $\theta°$ OF HEEL. IT IS LABELLED 'Mv' IN THE PREVIOUS DIAGRAM.

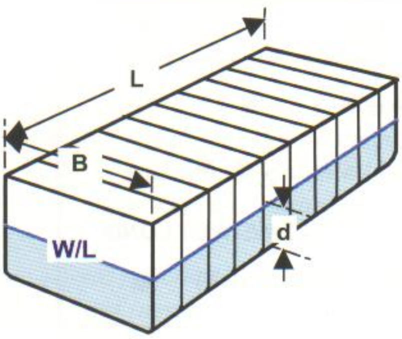
So THE METACENTRIC HEIGHT 'KMθ' = $KB_0 + BM_0 + \frac{1}{2} BM_0 \tan^2\theta$ ***AT $\theta°$ OF HEEL***

FOR A SHORT TRANSVERSE RECTANGULAR SLICE, 'B' METRES WIDTH, AT ANGLES OF HEEL LESS THAN THAT OF DECK EDGE IMMERSION.

The waterplane width of the rectangular slice increases with increasing angles of heel, so the metacentric radius (i.e. the BM value) also increases. This means that the actual Metacentre, 'Mθ', in the diagram on the previous page, moves off the centreline to a position vertically above Mv. However the position of the Metacentre is only useful in determining the horizontal separation between the two opposing forces of Weight and Buoyancy at a particular angle of heel 'θ'. The virtual Metacentre 'Mv', on the centreline is a more convenient reference for relating this to θ.
This rise in the Metacentric height increases at greater angles of heel and, although small compared with the effect of the transverse shift in the C of B, it further enhances a ship's ability to right itself. It is particularly significant at restoring positive stability in the case of vessels that are unstable in the upright condition (see Chapter 3, page 51 and Chapter 4, pages 91 to 95).

These equations can be applied to determine the KM value of a box-shaped hull as it is made up of transverse slices that all have the same value of beam 'B'.
The box shape is the simplest example of a ***wall-sided hull,*** which consists of rectangular transverse sections that can vary in width to produce a hullform with a waterplane area that is constant for any draft. This can be a useful approximation for merchant ship hulls as it allows for tapering the waterplane at the fore and aft ends but it does not account for any flare in the hull.

THE KMθ VALUE FOR A BOX-SHAPED HULL AT SMALL ANGLES OF HEEL



$$KM\theta = KB_0 + BM_0 + \frac{1}{2} BM_0 \tan^2\theta \quad M$$

But $KB_0 = \frac{1}{2}d$ ***WHERE 'd' IS THE DRAFT***

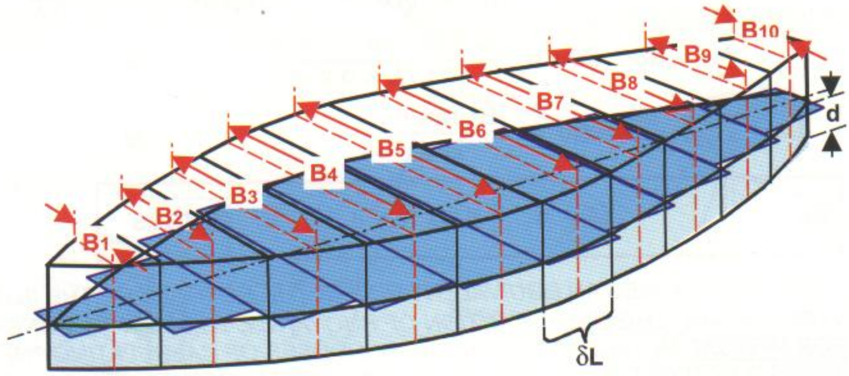
And $BM_0 = \frac{B^2}{12d}$ ***(FROM PAGE 29)***

$$\boxed{\text{So } KM\theta = \frac{1}{2}d + \frac{B^2}{12d}\left(1 + \frac{1}{2}\tan^2\theta\right) \quad M}$$

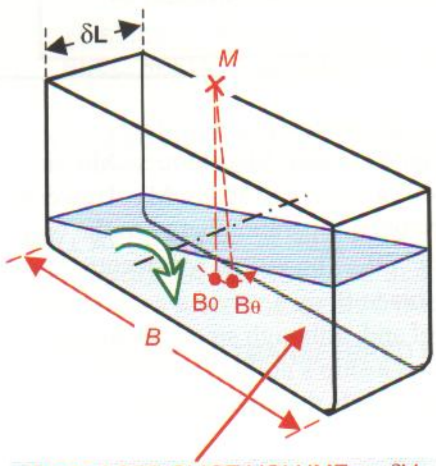
THE METACENTRE AND WALL-SIDED HULLS

The KM value of a wall-sided hull is determined by considering the hull to be made up of individual short rectangular box-shaped slices. The upright 'KB' value will be equal to half the hull's draft

THE KM VALUE FOR A TAPERED WALL-SIDED HULL



THE HULL IS DIVIDED INTO TEN EQUAL LENGTHS 'δL' OF EQUIVALENT RECTANGULAR SLICES. TERMS REFERING TO THESE SLICES, ARE WRITTEN IN *ITALICS*, OR LABELLED IN RED



DISPLACED SLICE VOLUME = δV

WE CAN CONSIDER THE SLICES SEPARATELY AS THE HULL IS HEELED THROUGH θ°. IN EACH TRANSVERSE SECTION, THE C of B OF EACH SLICE WOULD SWING ABOUT ITS OWN BM VALUE, AS DETERMINED BY THE WALL-SIDED EQUATION

FOR EACH SLICE, $BM_0 = \frac{B^2}{12d}$

And SHIFT OF C of B 'B_0B_θ' $= BM_0 \tan\theta$

THE MOMENT PRODUCED BY THIS SHIFT IS GIVEN BY:-

I.e. MOMENT / SLICE $= B_0B_\theta \times$ SLICE BUOYANCY

So MOMENT / SLICE $= BM_0 \tan\theta \times \delta V$ M^4

So MOMENT / SLICE $= \frac{B^2}{12d} \tan\theta \times (\delta L\, B\, d)$ M^4

So MOMENT / SLICE $= \delta L\, \frac{B^3}{12} \tan\theta$ M^4

THE SUM OF THE MOMENTS DUE TO HEELING EACH INDIVIDUAL SLICE, MUST BE EQUAL TO THE TOTAL MOMENT PRODUCED BY THE SHIFT IN THE C of B FOR THE VESSEL AS A WHOLE , SO:-

I.e. MOMENT DUE TO HEELING THE ENTIRE VESSEL $= BM_0 \tan\theta \times V_{\Delta T}$ M^4

WHERE 'BM_0' *RELATES TO THE ENTIRE VESSEL AND* '$V_{\Delta T}$' *IS ITS TOTAL DISPLACED VOLUME*

Then $\frac{\delta L}{12}(B_1^3 + B_2^3 + \text{---} + B_9^3 + B_{10}^3)\tan\theta = BM_0 \times V_{\Delta T} \tan\theta$ M^4

So $\frac{\delta L}{12}(B_1^3 + B_2^3 + \text{---} + B_9^3 + B_{10}^3) = BM_0 \times V_{\Delta T}$ M^4

So, FOR A WALL-SIDED HULL, $BM_0 = \frac{\delta L}{12\, V_{\Delta T}} \sum_{10}^{1} B^3$ M

And AT DRAFT 'd', $KM_0 = \frac{\delta L}{12\, V_{\Delta T}} \sum_{10}^{1} B^3 + \frac{1}{2} d$ M

WHERE 'B' VALUES ARE THE BEAM MEASUREMENTS OF SLICES 'δL' *METRES LONG*

Also AT SMALL ANGLES OF HEEL θ $KM_\theta = KB_0 + BM_0 + \frac{1}{2} BM_0 \tan^2\theta$ M

THE MOMENT OF WATERPLANE INERTIA

The Volumetric Moment of Waterplane Inertia (called simply the Moment of Waterplane Inertia or, more correctly, the Second Moment of Area*) indicates the rate at which the underwater hull shape changes with angle of heel and is an important factor in determining the hullform's resistance to rolling. It is the moment caused by the shift in the Centre of Buoyancy per radian of waterplane area rotation, as defined below:-

DEFINING THE MOMENT OF WATERPLANE INERTIA 'I_{WPA}'

VOLUMETRIC MOMENT FOR CHANGE OF HEEL '$\delta\theta^r$' = $\frac{\delta L}{12}\sum B^3\ \theta^r$ (Metres)^4

THE TERM $\frac{\delta L}{12}\sum B^3$ IS USUALLY KNOWN AS THE '**WATERPLANE MOMENT OF INERTIA 'I_{WPA}'**

> The Rotational Volumetric Moment of Inertia of an area about any axis, is the moment of **'THE SWEPT VOLUME' / RADIAN OF ROTATION, of that area about the axis**

TRANSFERRED BUOYANCY WEDGE OF VOLUME 'v'

TOTAL MOMENTS OF SWEPT VOLUMES ABOUT 'X' IS
$v\,bX + v\,Xb_1 = (bb_1)\,v$

A HULL IS HEELED THROUGH $\delta\theta$ RADIANS

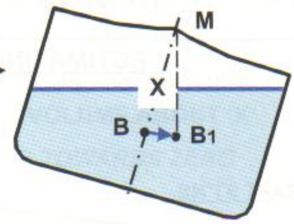
$$(bb_1)\,v / \delta\theta^r = I_{WPA} = (BB_1)\,V_{\Delta T} / \delta\theta^r$$

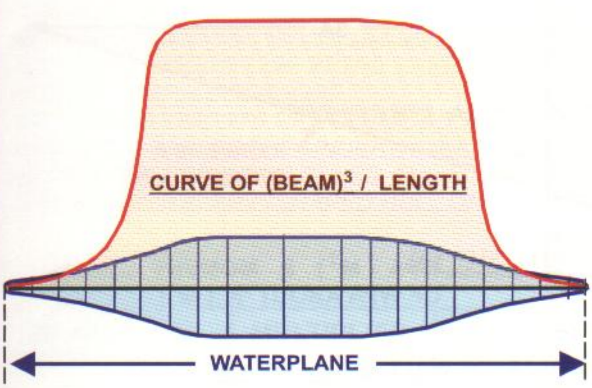
But $BB_1 = BM\ \delta\theta^r$

So $I_{WPA} = BM\ V_{\Delta T}$

Or $$\boxed{BM = \frac{I_{WPA}}{V_{\Delta T}}}$$

UNDERWATER HULL OF DISPLACED VOLUME $V_{\Delta T}$





ESTIMATING 'I_{WPA}'

IF WE PLOT A GRAPH OF **(BEAM)_ / LENGTH** FOR A SYMMETRICAL WATERPLANE, THEN :-

AREA UNDER (BEAM)^3 CURVE = $\delta L \sum B^3$

So $$\frac{\text{AREA UNDER (BEAM)}^3\text{ CURVE}}{12} = I_{WPA}$$

HENCE, WE CAN ESTIMATE THE MOMENT OF WATERPLANE INERTIA 'I_{WPA}' BY METHODS OF APPROXIMATE INTEGRATION

We can see from the above diagrams and definitions that the 'BM' value at any particular angle of heel is obtained by **dividing** the Moment of Waterplane Inertia '**I_{WPA}**' by the volume of displacement **$V_{\Delta T}$'** Obviously, as the waterplane area changes with angle of heel, so will the values of '**I_{WPA}**'and BM. We can, however, plot the shift in the Centre of Buoyancy from zero to 90° of heel by determining the waterplane and its Moment of Inertia at regular intervals of heel angle. This is the basis for the hullform analysis that is explained in the following pages of this chapter.

**Strictly speaking, Moments of Inertia are moments of 'mass' and the Moment of Waterplane Inertia should really be the Second Moment of Waterplane Area multiplied by the density of saltwater. The BM value would then be obtained by dividing the proper Moment of Waterplane Inertia by the Mass of Displacement. This will naturally give the same BM value as the density factor is cancelled out of the equation.*
It is normal practice in naval architecture to refer to 'Second Moments of Area' rather incorrectly as 'Moments of Inertia'. This may confuse people with a physics or engineering background who have encountered Moments Of Inertia relating to spinning flywheels or gyroscopes etc.

THE BUOYANCY CHARACTERISTICS OF A SHIP-SHAPED HULL

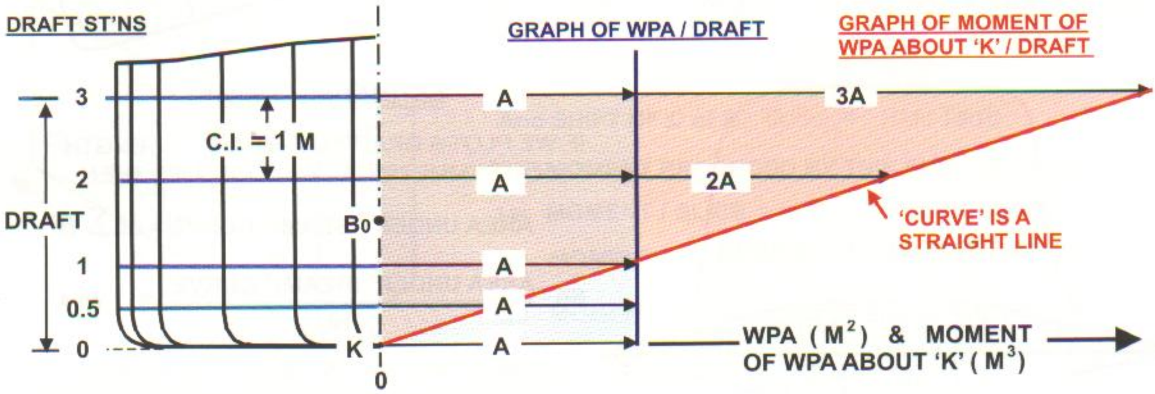
The previous pages considered the change of underwater shape (and, hence, the distribution of buoyancy) when simple hullforms are heeled over. In particular, the reduction of beam at the fore and aft ends of a tapered wall-sided hull, greatly reduces the waterplane transverse moment of inertia which means that the shift in the Centre of Buoyancy and the hull's righting ability are reduced. A real ship-shaped hull incorporates flare, which enhances the waterline beam in the bow and stern regions at increasing angles of heel, in order to overcome this deficiency. It is the job of the shipyard design team to produce an accurate set of data to be used in stability calculations. This requires the ship's hull shape to be fully analysed to track the position of the Centre of Buoyancy for varying angles of heel at different drafts. The first stage of this analysis would be to determine its initial vertical position (the KB_0 value) when the hull is upright at different drafts.

DETERMINING THE UPRIGHT 'KB_0' VALUE OF A SHIP-SHAPED HULL

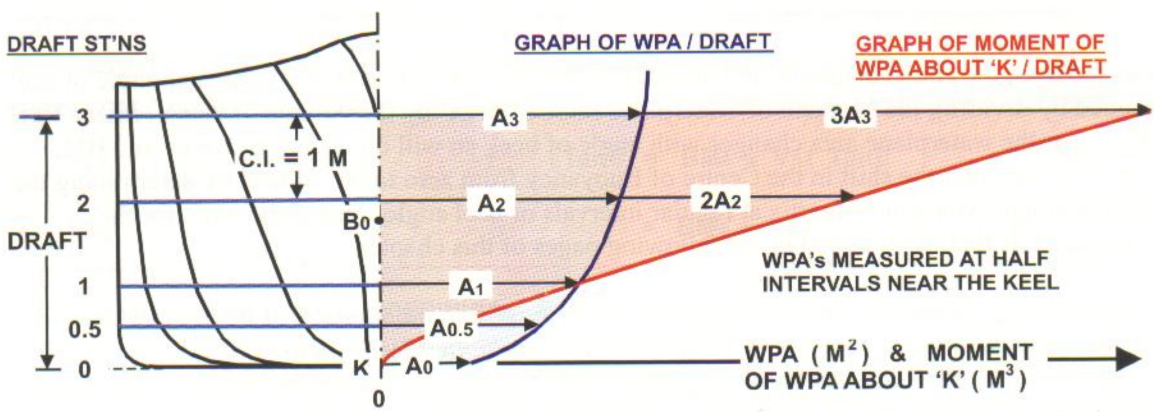
If we have a set of waterplane areas at different drafts for a vessel, then we can plot a graph of waterplane area against draft. The area enclosed by such a graph, between the keel and any particular draft station, represents the hull's underwater volume at that draft (see page 13). We can also plot a graph of each WPA Moment about the keel against the draft and the area enclosed by this curve up to any draft, will represent the total moments of volume about the keel at that draft

ESTIMATING OF THE 'KB_0' VALUE FROM WATERPLANE AREAS

A WALL-SIDED HULL OF CONSTANT WPA 'A'



A SHIP-SHAPED HULL WHERE DRAFT WPA 'A_3' = WALL-SIDED WPA 'A'



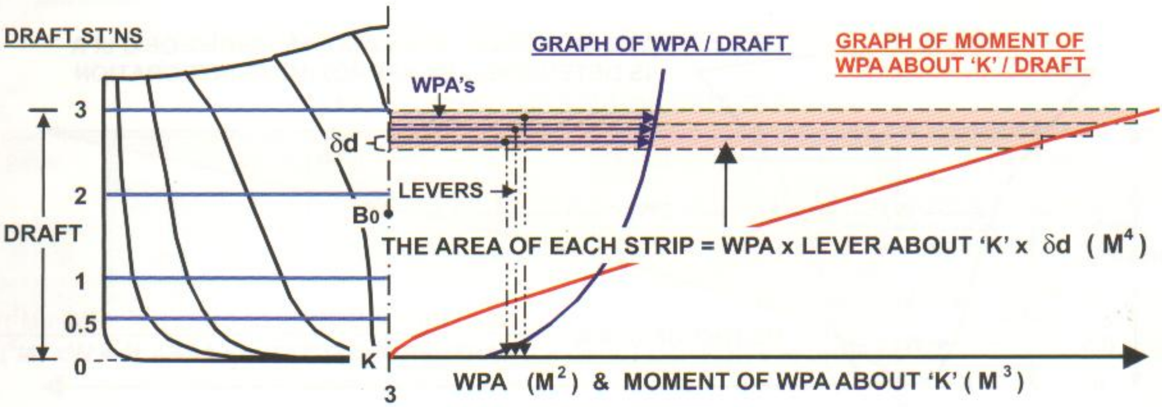
AT A DRAFT OF 3 METRES

$$KB_0 = \frac{\text{TOTAL MOMENTS OF VOLUME ABOUT 'K'}}{\text{TOTAL UNDERWATER VOLUME}} = \frac{\text{AREA UNDER MOMENTS CURVE } K \rightarrow d_3}{\text{AREA UNDER WPA CURVE } K \rightarrow d_3}$$

DETERMINING THE UPRIGHT 'KB0' VALUE OF A SHIP-SHAPED HULL (Cont.)

The previous page shows how the total Moments of Underwater Volume about the Keel is equal to the area enclosed by a 'Moments of **Area** about the Keel' curve, up to the waterline draft. This allows the KB0 to be calculated by the methods of approximate integration.

ESTIMATING OF THE 'KB0' VALUE FROM WATERPLANE AREAS (Cont.)



THE AREA UNDER THE MOMENTS CURVE ≈ THE SUM OF ALL THE STRIPS, 'δd' M THICK

So THE AREA UNDER THE MOMENTS CURVE ≈ THE TOTAL MOMENT OF VOLUME ABOUT 'K'

THE ERROR IN THE APPROXIMATION REDUCES AS 'δd' IS DECREASES TOWARDS ZERO

WHEN WE USE APPROXIMATE INTEGRATION METHODS, WE SIMPLY SAMPLE THE MOMENTS CURVE AT REGULAR DRAFT INTERVALS AND EFFECTIVELY INTERPOLATE THE INTERVENING MOMENT ORDINATES BY ASSUMING THAT THE CURVE FITS A SIMPLE MATHEMATICAL EQUATION. THIS IS A STRAIGHT LINE IN THE CASE OF THE TRAPEZIUM METHOD, OR A PARABOLA IN THE CASE OF SIMPSON'S RULES

THE TOTAL MOMENTS OF VOLUME ABOUT THE KEEL FOR DIFFERENT DRAFTS CAN EASILY BE ESTIMATED WHILST THE UNDERWATER VOLUME IS CALCULATED BY EITHER THE TRAPEZIUM METHOD OR SIMPSON'S RULES. THE FOLLOWING TABLES ILLUSTRATE THE PROCEDURE AS APPLIED TO THE SHIP-SHAPED HULL ON THE PREVIOUS PAGE

BY THE TRAPEZIUM METHOD

DRAFT	WPA	MULTIPLIER	VOLUME PRODUCT	LEVER	MOMENT PRODUCT
0	A_0	0.25	$0.25\ A_0$	0 x C.I	0
0.5	$A_{0.5}$	0.5	$+\ 0.5\ A_{0.5}$	0.5 x C.I	$+\ 0.25\ A_{0.5}$ x C.I
1	A_1	0.75	$+\ 0.75\ A_1$	1 x C.I	$+\ 0.75\ A_1$ x C.I
2	A_2	1	$+\ A_2$	2 x C.I	$+\ 2\ A_2$ x C.I
3	A_3	0.5	$+\ 0.5\ A_3$	3 x C.I	$+\ 1.5\ A_3$ x C.I
			Σ VOLUME PRODUCT		Σ MOMENT PRODUCT

HULL VOLUME TO DRAFT ST'N 3 = C.I. x Σ VOLUME PRODUCT M^3

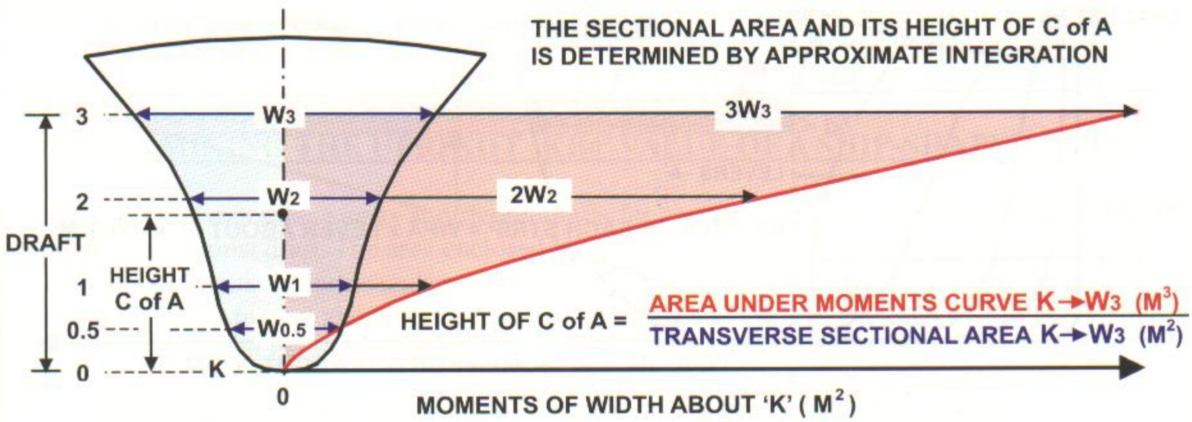
MOMENTS OF VOLUME ABOUT 'K' TO DRAFT ST'N 3 = C.I. x Σ MOMENT PRODUCT M^4

So HEIGHT OF C of B ABOVE 'K' WHEN UPRIGHT = $\dfrac{\Sigma \text{ MOMENT PRODUCT}}{\Sigma \text{ VOLUME PRODUCT}}$ METRES

DETERMINING THE UPRIGHT 'KB$_0$' VALUE OF A SHIP-SHAPED HULL (Cont.)

The height of the Centre of Buoyancy of a hullform can also be calculated from the set of transverse hull sections. We start by determining the submerged transverse sectional areas and the heights of their Centres of Area for different drafts. This is a more involved procedure than the previous calculation based upon the waterplanes.

CALCULATING THE TRANSVERSE SECTIONAL AREA AND THE HEIGHT OF ITS C of A



THE SECTIONAL AREA AND ITS HEIGHT OF 'C of A', ARE CALCULATED FOR EACH DRAFT AT EVERY TRANSVERSE STATION ALONG THE SHIP'S LENGTH. WE CAN USE THE METHOD OF APPROXIMATE INTEGRATION, AS SHOWN PREVIOUSLY, TO DETERMINE THE UNDERWATER VOLUME AND KB$_0$. THIS TIME, WE WILL USE SIMPSON'S RULES, SIMPLY TO ILLUSTRATE AN ALTERNATIVE METHOD.

BY SIMPSON'S 1-4-1 RULE

DRAFT	WIDTH	MULTIPLIER	AREA PRODUCT	LEVER	MOMENT PRODUCT
0	W_0	0.5	$0.5\ W_0$	0 x C.I	0
0.5	$W_{0.5}$	2	+ $2\ W_{0.5}$	0.5 x C.I	+ $W_{0.5}$ x C.I
1	W_1	1.5	+ $1.5\ W_1$	1 x C.I	+ $1.5\ W_1$ x C.I
2	W_2	4	+ $4\ W_2$	2 x C.I	+ $8\ W_2$ x C.I
3	W_3	1	+ W_3	3 x C.I	+ $3\ W_3$ x C.I
			Σ AREA PRODUCT		Σ MOMENT PRODUCT

TRANSVERSE SECTIONAL AREA TO DRAFT ST'N 3 = $\frac{1}{3}$ C.I. x Σ AREA PRODUCT M^2

MOMENTS OF VOLUME ABOUT 'K' TO DRAFT ST'N 3 = $\frac{1}{3}$ C.I. x Σ MOMENT PRODUCT M^3

So HEIGHT OF SECTION C of A ABOVE 'K' = $\dfrac{\Sigma \text{ MOMENT PRODUCT}}{\Sigma \text{ VOLUME PRODUCT}}$ METRES

NOTICE THAT THE PROCEDURE ABOVE IS ESSENTIALLY THE SAME AS THAT PREVIOUSLY USED TO CALCULATE THE UNDERWATER VOLUME AND THE KB$_0$ FROM THE WATERPLANES, *EXCEPT THAT THE MULTIPLIERS AND C.I. FACTOR ARE CHANGED TO THOSE REQUIRED BY THE SIMPSON 1-4-1 RULE. (THE LEVERS REMAIN UNCHANGED)*

A TABLE OF SECTIONAL AREAS AND HEIGHTS OF C of A MUST BE MADE FOR EVERY DRAFT AT EACH TRANSVERSE SECTION STATION ALONG THE LENGTH OF THE VESSEL

DETERMINING THE UPRIGHT 'KB0' VALUE OF A SHIP-SHAPED HULL (Cont.)

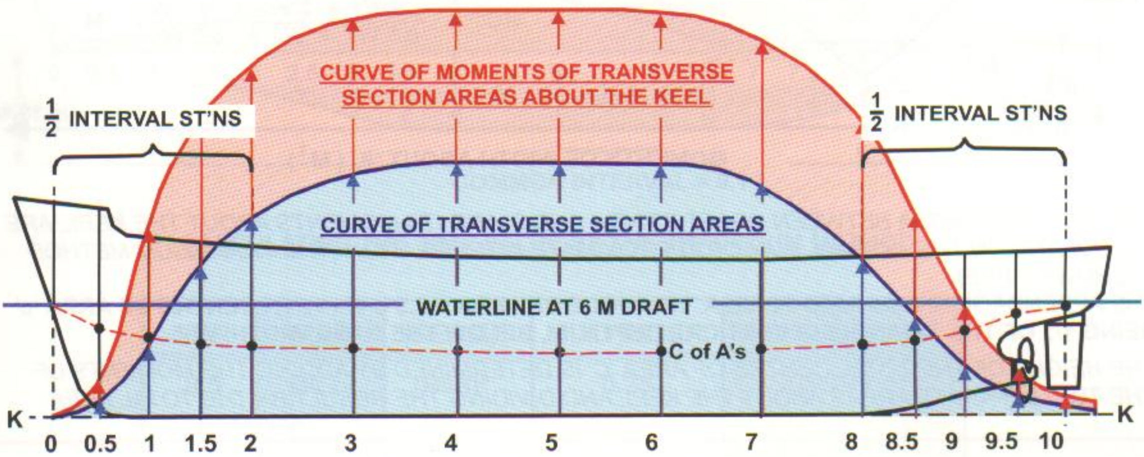
The transverse sectional areas, for each draft, are plotted against the ship's length and the area under the resulting curves will equal the submerged volume at those drafts. The lever about the keel, of each section area, is the height of the section's centre of area, so multiplying the two together will give the ordinates for plotting a curve of Moments about the keel against the ship's length. The area under this curve will be the total Moment of submerged Volume about the keel and dividing this by the volume itself, will produce the KB0 value for the draft relating to that set of data.

CALCULATING THE KB0 FROM THE TRANSVERSE SECTION DATA

TYPICAL LAYOUT OF SECTION DATA TABLE

DRAFT	STATION 0			STATION 0.5L			STATION 1L	
	AREA	LEVER	MOMENT	AREA	LEVER	MOMENT	AREA	LEVER
0.5 M	A_0	H_0	$A_0 \times H_0$	$A_{0.5}$	$H_{0.5}$	$A_{0.5} \times H_{0.5}$	A_1	H_1
1.0 M								

DATA PLOTTED AT INTERVALS OF 0.1 LBP FOR 6 M DRAFT



PROCESSING THE TRANSVERSE SECTION DATA BY THE TRAPEZIUM METHOD

ST'N	X-AREA	MULTIPLIER	VOLUME PRODUCT	LEVER	MOMENT PRODUCT
0	A_0	0.25	$0.25 A_0$	H_0	$H_0 \times 0.25A_0$
0.5	$A_{0.5}$	0.5	$+ 0.5 A_{0.5}$	$H_{0.5}$	$+ H_{0.5} \times 0.5A_{0.5}$
1	A_1	0.5	$+ 0.5 A_1$	H_1	$+ H_1 \times 0.5 A_1$
1.5	$A_{1.5}$	0.5	$+ 0.5 A_2$	$H_{1.5}$	$+ H_{1.5} \times 0.5 A_{1.5}$
10	A_{10}	0.25	$+ 0.25 A_{10}$	H_{10}	$+ H_{10} \times 0.25A_{10}$
			Σ VOLUME PRODUCT		Σ MOMENT PRODUCT

SEE PAGE 13 FOR MULTIPLIERS

HULL VOLUME TO 6 M DRAFT = 0.1 L x Σ VOLUME PRODUCT M^3

MOMENTS OF VOLUME ABOUT 'K' TO 6 M DRAFT = 0.1 L x Σ MOMENT PRODUCT M^4

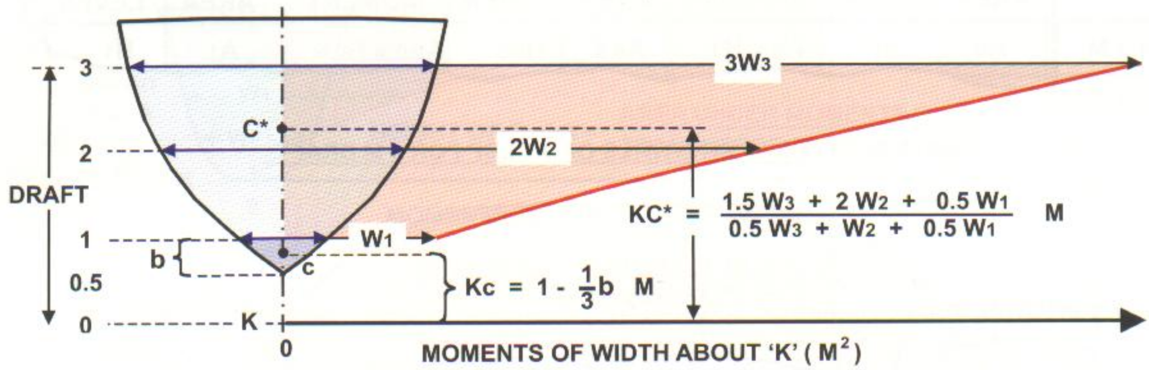
So UPRIGHT KB0 VALUE AT 6 M DRAFT = $\dfrac{\Sigma \text{ MOMENT PRODUCT}}{\Sigma \text{ VOLUME PRODUCT}}$ METRES

DETERMINING THE UPRIGHT 'KB0' VALUE OF A SHIP-SHAPED HULL (Cont.)

The procedure for determining the underwater hull and the position of its Centre of Buoyancy will almost certainly require including **appendage** areas in the calculations of the total underwater volume and the volumetric moment about the keel. These are the parts of the hullform that do not end neatly at one of the fixed interval measuring stations. Near the keel, the fixed waterplane sections will terminate aft at some arbitrary point short of the aft perpendicular, whilst higher up in the hull, the waterplanes extend beyond both perpendiculars. Page 12 dealt with appendages in the calculation of waterplane areas and it is worth looking at the procedure again in relation to the KB0 calculations

CALCULATING THE TOTAL SECTION AREA, INCLUDING APPENDAGE

THE LOWEST POINT OF AN AFT SECTION IS b METRES BELOW DRAFT STN 1, SO A TRIANGLE OF SECTIONAL AREA, WITH C of A c, LIES OUTSIDE OF THE FIXED DRAFT MEASURING STATIONS



THE SECTIONAL AREA BETWEEN DRAFT ST'NS 1 AND 3, AND ITS MOMENTS ABOUT THE KEEL, ARE CALCULATED IN THE NORMAL WAY OF APPROXIMATE INTERGRATION (THE TRAPEZIUM METHOD IS SHOWN HERE)

THE APPENDAGE IS ASSUMED TO BE A **TRIANGLE OF AREA = 0.5 b W1** WITH **CENTRE OF AREA 'c' BEING 1/3 OF THE TRIANGLE'S VERTICAL DEPTH 'b', BELOW THE BASE WIDTH 'W1'**

THE HEIGHT OF THE TOTAL CENTRE OF AREA 'C' IS DETERMINED BY ADDING THE MOMENTS OF THE TWO SEPARATE AREAS ABOUT THE KEEL AND DIVIDING THE RESULT BY THE TOTAL AREA

So $\quad$ **TOTAL AREA** $= (0.5\,W_3 + W_2 + 0.5\,W_1) + 0.5\,b\,W_1 \quad M^2$

And KC VALUE FOR TOTAL AREA $= \dfrac{(1.5\,W_3 + 2\,W_2 + 0.5\,W_1) + 0.5\,b\,W_1(1 - \frac{1}{3}b)}{(0.5\,W_3 + W_2 + 0.5\,W_1) + 0.5\,b\,W_1} \quad M$

THE 'KC' VALUE CAN THEN BE USED AS THE LEVER ABOUT THE KEEL FOR THE TOTAL SECTION AREA, IN THE MOMENTS OF AREA CALCULATION FOR DETERMINING THE KB0 VALUES AT DIFFERENT DRAFTS.

THE VOLUMES OF SOME PARTS OF THE HULL, SUCH AS THE OVERHANG OF A TRANSOM STERN OR A BULBOUS BOW, MAY BE BEST ESTIMATED AS APPENDAGE VOLUMES BY APPROXIMATING THEM TO THE CLOSEST REGULAR GEOMETRIC SHAPE

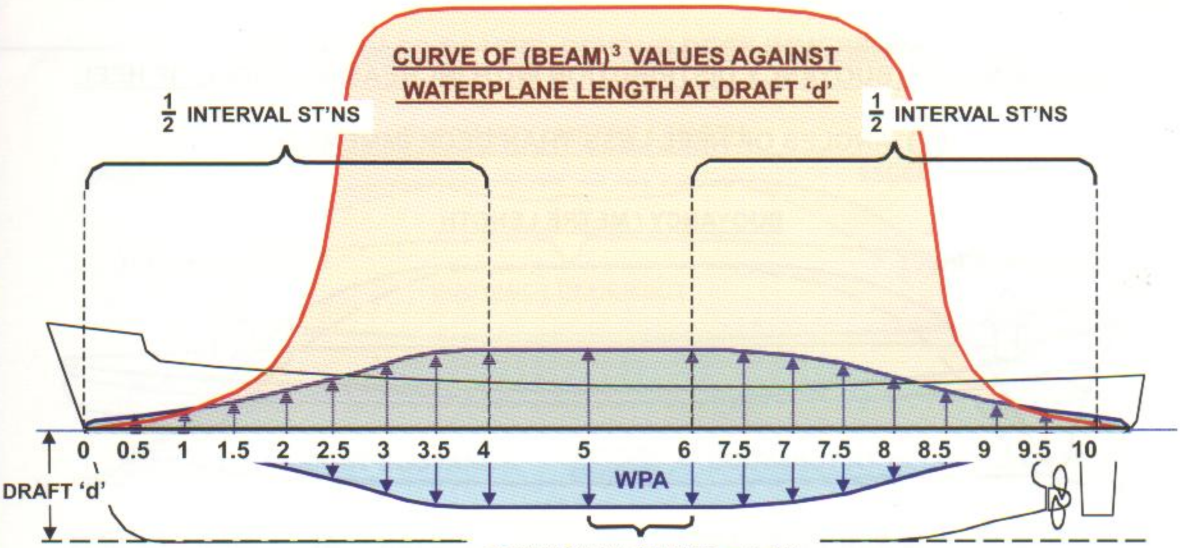
We now see that we can determine the underwater volume and KB0 values at different drafts by taking moments about the keel of either waterplane or transverse sectional areas. If both procedures are carried out on the same table of offsets and there are significant differences in the solution, then it is probable that there are insufficient half interval measurements in the data. It is important to define the hull shape well with such measurements in the regions close to the bow, stern and keel, where curvature changes quite markedly over short distances.

Quarter station intervals can be used at the extreme ends of the hull to improve the definition further. Such measurements can easily be incorporated into the approximate intergration methods by quartering the standard multipliers.

DETERMINING THE UPRIGHT 'BM0' VALUE OF A SHIP-SHAPED HULL

The Wall-sided model is used to determine the BM0 value for an upright vessel, at a given draft. This involves calculating the area under a graph of **(Beam)³ / Length** and errors, inherent in approximate integration methods, are greatly increased when we are dealing with the cubed values of ordinates, so half station measuring intervals are used wherever there is a large rate of change of beam with length.

ESTIMATING THE UPRIGHT BM0 VALUE OF A HULL AT A GIVEN DRAFT



CALCULATING THE 'BM0' VALUE BY THE TRAPEZIUM METHOD

ST'N	BEAM	(BEAM)³	MULTIPLIER	PRODUCT
0.5	$B_{0.5}$	$(B_{0.5})^3$	0.25	$0.25(B_{0.5})^3$
1	B_1	$(B_1)^3$	0.5	$+\ 0.5(B_1)^3$
1.5	$B_{1.5}$	$(B_{1.5})^3$	0.5	$+\ 0.5(B_{1.5})^3$
2	B_2	$(B_2)^3$	0.5	$+\ 0.5(B_2)^3$
2.5	$B_{2.5}$	$(B_{2.5})^3$	0.5	$+\ 0.5(B_{2.5})^3$
3	B_3	$(B_3)^3$	0.5	$+\ 0.5(B_3)^3$
3.5	$B_{3.5}$	$(B_{3.5})^3$	0.5	$+\ 0.5(B_{3.5})^3$
4	B_4	$(B_4)^3$	0.75	$+\ 0.75(B_4)^3$
5	B_5	$(B_5)^3$	1	$+\ (B_5)^3$
6	B_6	$(B_6)^3$	1	$+\ (B_6)^3$
9.5	$B_{9.5}$	$(B_{9.5})^3$	0.5	$+\ 0.5(B_{9.5})^3$
10	B_{10}	$(B_{10})^3$	0.25	$+\ 0.25(B_{10})^3$
			SUM OF PRODUCT	Σ (B³ PRODUCT)

$$\text{UPRIGHT BM}_0 = \frac{0.1\,L}{12 \times \Delta_T\ \text{VOLUME}} \Sigma\,(\text{B}^3\ \text{PRODUCT})\ \text{METRES}$$

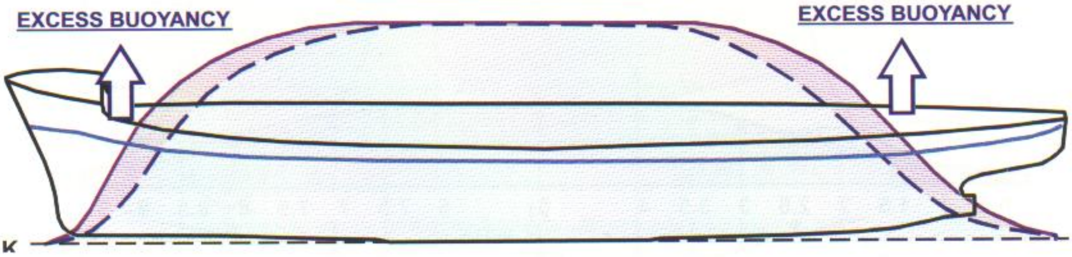
BUOYANCY DISTRIBUTION CHANGES WITH INCREASING HEEL

Now that the hullform has been thoroughly analysed to find the underwater volume and KB value in the upright condition, we now have to see how the buoyancy distribution changes and the Centre of Buoyancy shifts with increasing angles of heel at different drafts. Initially, as the hull is rotated about the waterplane axis, the waterplane area becomes progressively more asymmetrical about the centreline and excess buoyancy is produced at the fore and aft ends, due to the effects of flare. This trend continues up to the point of deck edge immersion and causes the hull to bodily rise whilst the rolling axis shifts away from the centreline towards the excess buoyancy on the low side of the hullform.

THE CHANGE OF BUOYANCY DISTRIBUTION WITH INCREASING ANGLE OF HEEL

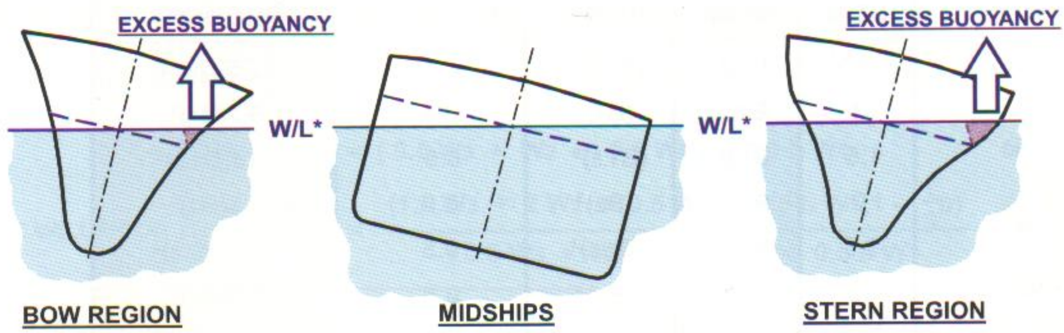
1 ANGLES OF HEEL LESS THAN DECK IMMERSION

BUOYANCY / METRE LENGTH



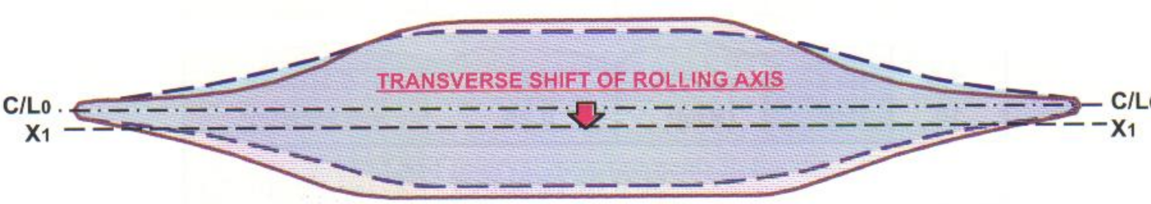
= UPRIGHT BUOYANCY = BUOYANCY AT $\theta_1°$ ANGLE OF HEEL

TRANSVERSE SECTIONS ROTATED $\theta_1°$ ABOUT THE WATERPLANE CENTRELINE



W/L* = W/L$_0$ ROTATED THROUGH $\theta_1°$ ABOUT THE C/L = EXCESS BUOYANCY

WATERPLANE AREAS

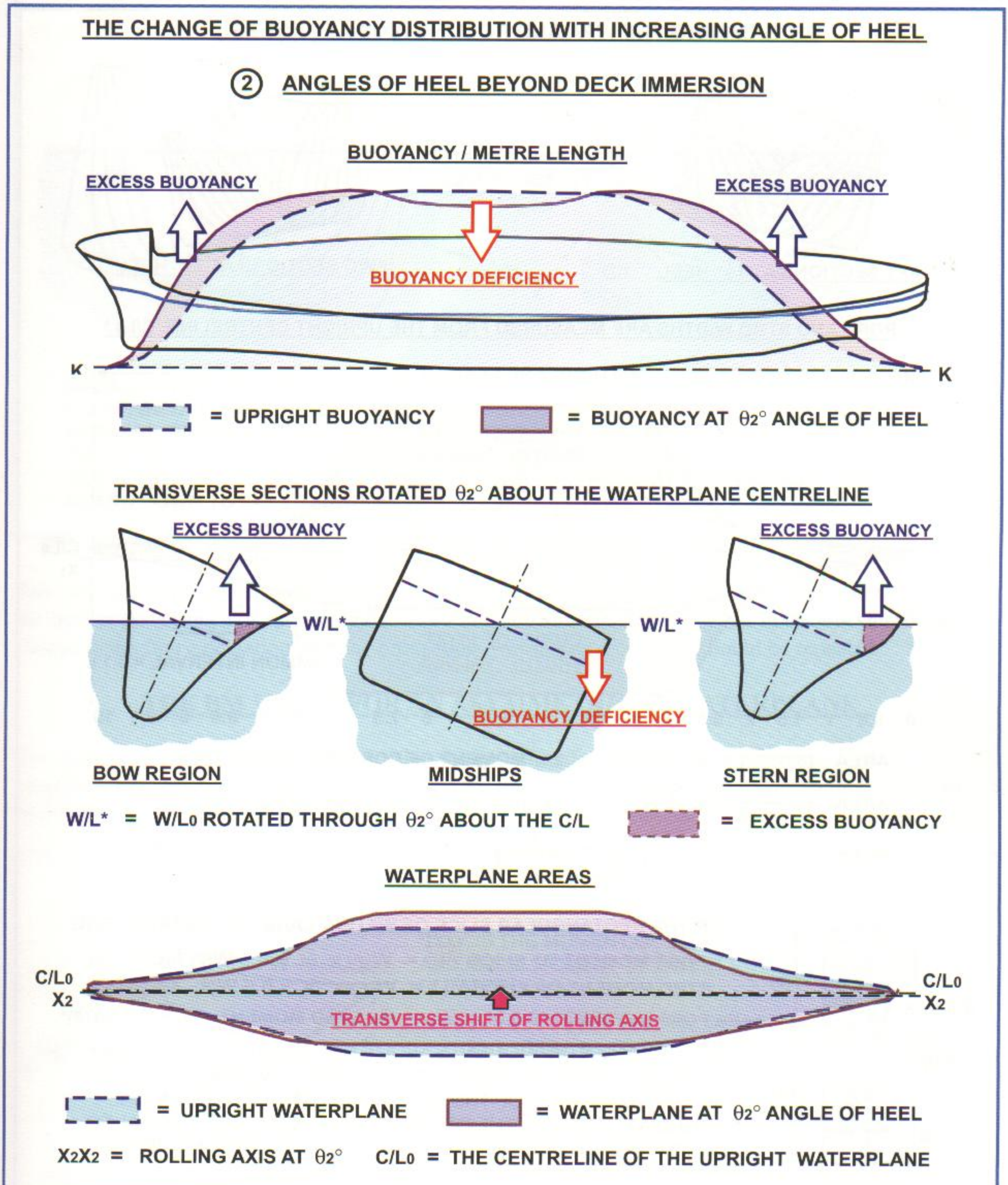


= UPRIGHT WATERPLANE = WATERPLANE AT $\theta_1°$ ANGLE OF HEEL

X_1X_1 = ROLLING AXIS AT $\theta_1°$ C/L$_0$ = THE CENTRELINE OF THE UPRIGHT WATERPLANE

BUOYANCY DISTRIBUTION CHANGES WITH INCREASING HEEL (Cont.)

When a ship-shaped hull is heeled over beyond the point of deck edge immersion, the trend of increasing effective waterline beam will tend to begin to reverse as extreme waterline beam will now start to reduce,creating a midships buoyancy deficiency, though flare and sheer will still produce excess buoyancy at the bow and stern . The bodily rise of the hull will first decrease and then start to reverse whilst the rolling axis will start to shift back towards the high side of the waterplane.

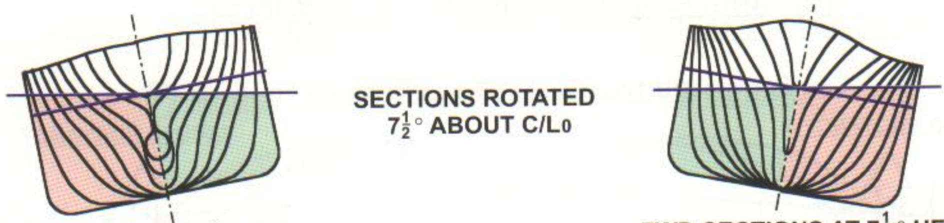


We must find the means of tracking the position of the Centre of Buoyancy through all these complex changes in the submerged hullform as it is rolled over from the upright condition to 90° of heel.

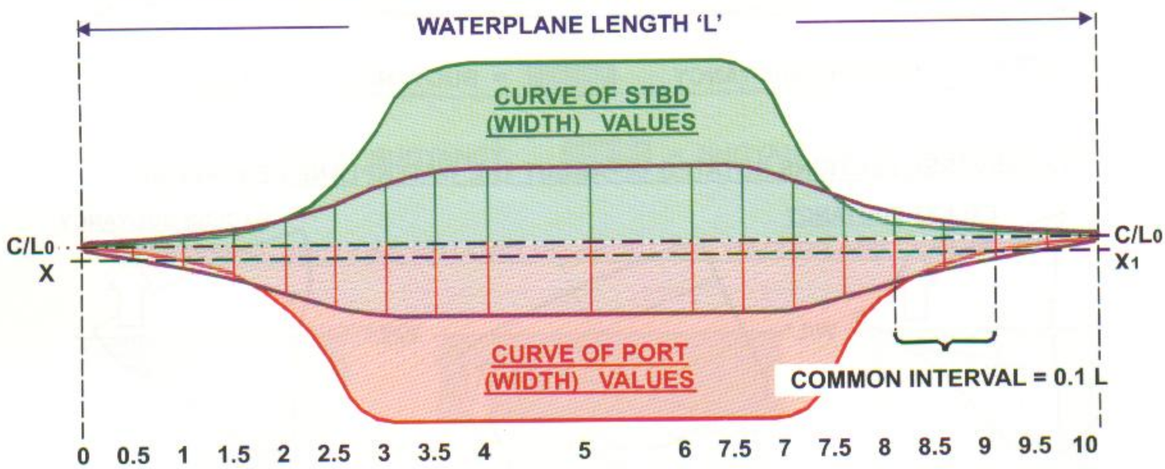
FIXING WPA AND ROLLING AXIS AT INCREASING ANGLES OF HEEL

This is carried out in distinct steps of, say $7\frac{1}{2}°$ of increasing heel. Initially, the upright sections of the hull at a given draft are rotated around the upright waterplane centreline. The port and starboard waterline widths at $7\frac{1}{2}°$ heel can then be measured to produce a new waterplane. Approximate integration is used to calculate this new asymmetrical waterplane area and then first moments of port and starboard areas are taken about the upright centreline to determine the shift in the rolling axis.

DETERMINING THE WPA AND ROLLING AXIS AT 7 ° HEEL



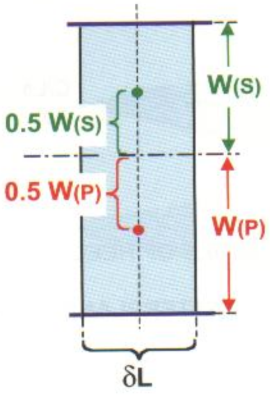
PORT AND STBD WIDTHS ARE MEASURED FROM THE UPRIGHT CENTRELINE 'C/L0'



AREA (red) $= \delta L \sum W_{(P)}^2$ = 2 x MOMENT OF PORT WPA ABOUT C/L0

AREA (green) $= \delta L \sum W_{(S)}^2$ = 2 x MOMENT OF STBD WPA ABOUT C/L0

AREA (outlined) $= \delta L \left[\sum W_{(S)} + \sum W_{(P)}\right]$ = TOTAL WATERPLANE AREA 'WPA'



FOR THE RECTANGULAR SLICE OF WATERPLANE 'δL, METRES LONG

PORT MOMENT OF SLICE 'δL' = $W_{(P)} \times \delta L \times 0.5\, W_{(P)}$ ABOUT C/L

STBD MOMENT OF SLICE 'δL' = $W_{(S)} \times \delta L \times 0{,}5\, W_{(S)}$ ABOUT C/L

So PORT MOMENT = $0.5\,\delta L \times W_{(P)}^2$ & STBD MOMENT = $0.5\,\delta L \times W_{(S)}^2$

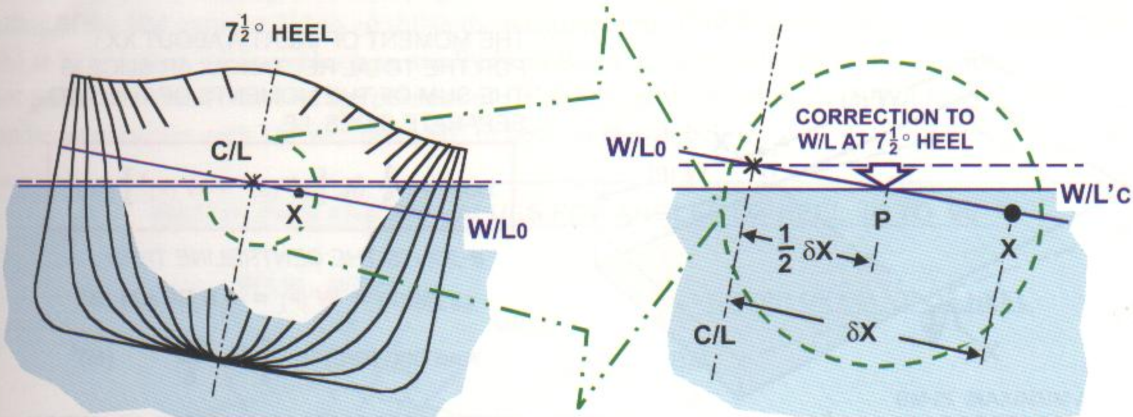
THE PORT AND STBD MOMENTS ABOUT THE C/L ARE EQUAL TO HALF THE AREAS UNDER THE RESPECTIVE LENGTH / (W/L WIDTH)² CURVES. THE CENTRE OF AREA OF THE ENTIRE WATERPLANE LIES ON THE ROLLING AXIS 'XX1, SO THE OFFSET OF XX1 FROM C/L0, IS EQUAL TO THE DIFFERENCE BETWEEN THE PORT AND STBD AREA MOMENTS, DIVIDED BY THE TOTAL WPA, I.E.:-

$$XX_1 \rightarrow C/L_0 = \frac{0.5\,\delta L \left[\sum W_{(P)}^2 - \sum W_{(S)}^2\right]}{\delta L \left[\sum W_{(P)} + \sum W_{(S)}\right]} \text{ METRES}$$

ADJUSTING THE HEELED WATERLINE FOR BODILY CHANGES IN DRAFT

Once the waterplane area and rolling axis have been determined at θ° of heel (as shown on the previous page), the waterline must be adjusted for any bodily change in draft that will have occurred in the incremental change of heel. Consider the caculations for $7\frac{1}{2}$° heel, shown on the previous page.

CORRECTING THE WATERLINE FOR BODILY RISE AT $7\frac{1}{2}$° HEEL



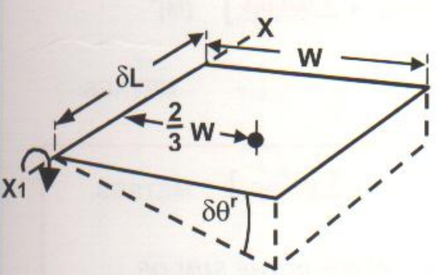
INITIALLY, WHEN THE VESSEL IS UPRIGHT, IT ROTATES ABOUT THE UPRIGHT CENTRELINE C/L. HOWEVER, AT $7\frac{1}{2}$° HEEL, THE ROLLING AXIS HAS MOVED OUTBOARD TO X, SO THE **AVERAGE POSITION OF THE AXIS (POINT 'P') IS MIDWAY BETWEEN THE C/L AND 'X'.** THE $7\frac{1}{2}$° W/L IS FIRST ESTIMATED BY ROTATING THE SECTION ABOUT C/L BUT NOW THAT THE OFFSET OF X HAS BEEN ESTABLISHED, **THE WATERLINE AT $7\frac{1}{2}$° HEEL MUST BE MOVED DOWN TO PASS THROUGH 'P'.**

Port and starboard widths should be measured from the centreline along this corrected waterline and the shift in the rolling axis re-calculated. If this adjustment to the waterline has made any significant change to the first calculation then the procedure of waterline correction should be repeated.

THE BM VALUE FOR AN ASYMMETRICAL WATERPLANE

The Wall-sided equation gives the BM value for a symmetrical waterplane, rotated about the centreline, but now we must find an expression for the Moment of Inertia about the rolling axis of an unsymetrical waterplane. We can start by considering a rectangle rotated about one of its edges.

THE MOMENT OF INERTIA OF A RECTANGULAR SLICE ABOUT AN EDGE



THE SLICE 'δL' METRES LONG AND 'W' METRES WIDE, IS ROTATED δθ RADIANS ABOUT THE XX1 AXIS

'I_{XX_1}' IS THE MOMENT OF INERTIA ABOUT THE XX_1 AXIS AND IS THE MOMENT OF SWEPT VOLUME / RADIAN OF ROTATION.

I.e. $$I_{XX_1} = \frac{\text{VOLUME OF WEDGE} \times \frac{2}{3}W}{\delta\theta^r} \quad M^4$$

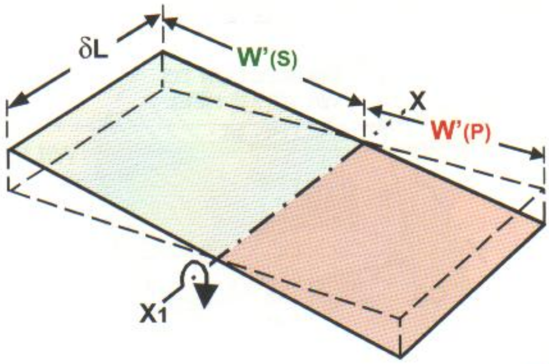
So $$I_{XX_1} = \frac{\delta L \times W \times \frac{1}{2} W \delta\theta^r \times \frac{2}{3} W}{\delta\theta^r} \quad M^4$$

$$\boxed{\text{Hence } I_{XX_1} = \frac{1}{3}\delta L\, W^3 \quad (\text{METRES})^4}$$

THE BM VALUE FOR AN ASYMMETRICAL WATERPLANE (Cont.)

If we calculate the **Moment of Inertia** about the **rolling axis 'XX_1'** for the port and starboard sides of the waterplane area separately, then the Moment of Inertia of the entire waterplane about **XX_1**, will be the sum of the port and starboard Moments.

MOMENT OF INERTIA OF A RECTANGULAR SLICE ABOUT ANY AXIS



THE MOMENT OF INERTIA ABOUT XX_1 FOR THE TOTAL RECTANGULAR SLICE IS THE SUM OF THE MOMENTS OF THE TWO SEPARATE SIDES. I.E. :-

$$\boxed{I_{XX_1} = \frac{1}{3}\,\delta L\left[W'_{(S)}{}^3 + W'_{(P)}{}^3\right]\ (M)^4}$$

IF XX_1 IS THE CENTRELINE THEN :-

$$W'_{(S)} = W'_{(P)} = 0.5 \text{ BEAM 'B'}$$

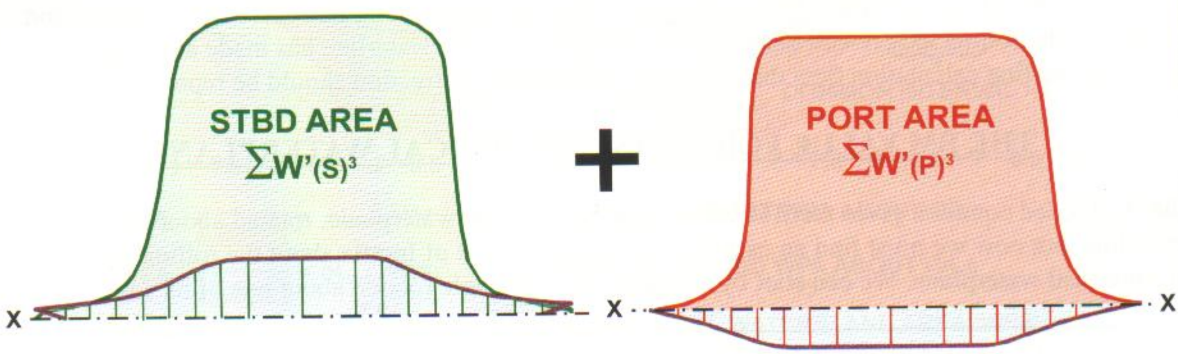
$$\text{And } I_{XX_1}(C/L) = \frac{2}{3}\,\delta L\,\frac{B^3}{8} \qquad (M)^4$$

WHICH AGREES WITH THE WALL-SIDED EQUATION

CALCULATING THE BM VALUE FOR THE WATERPLANE AT θ° OF HEEL

WE CAN USE THE ABOVE GENERAL EQUATION FOR THE MOMENT OF INERTIA TO CALCULATE ***I_{XX_1}*** *FOR THE PORT AND STARBOARD SIDES OF THE ASYMMETRICAL WATERPLANE.* ***NOTE THAT WE MUST RE-MEASURE THE WIDTH VALUES FROM THE ROLLING AXIS AND NOT THE CENTRELINE***

CURVES OF PORT & STBD (HALF WIDTH')^3 / WPA LENGTH



MOMENT OF WATERPLANE INERTIA 'I_{XX_1}' $= \frac{1}{3}\delta L\left[\Sigma W'_{(S)}{}^3 + \Sigma W'_{(P)}{}^3\right]\ (M)^4$

Now $BM_1 = \dfrac{I_{XX_1}}{V_{\Delta T}}$ **METRES**

So The radius of swing of the C of B, BM_1 $= \dfrac{\delta L}{3\,V_{\Delta T}}\left[\Sigma W'_{(S)}{}^3 + \Sigma W'_{(P)}{}^3\right]$ **METRES**

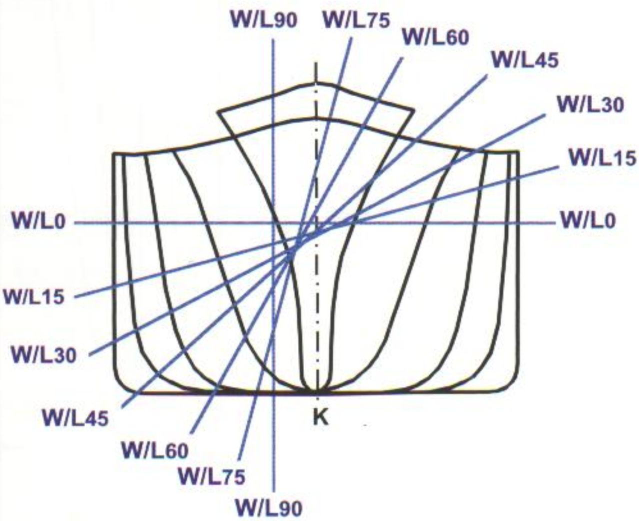
WHERE '$V_{\Delta T}$' IS THE DISPLACED VOLUME AND $\Sigma W'_{(S)}{}^3 + \Sigma W'_{(P)}{}^3$ IS THE SUM OF THE AREA'S UNDER THE (HALF WIDTH')^3 / WPA LENGTH CURVES FOR THE PORT AND STBD WIDTHS, MEASURED FROM THE ROLLING AXIS.

THESE AREA'S CAN BE DETERMINED BY APPLYING THE TRAPEZIUM METHOD OR SIMPSON'S RULES TO THE (HALF WIDTH')^3 ORDINATES

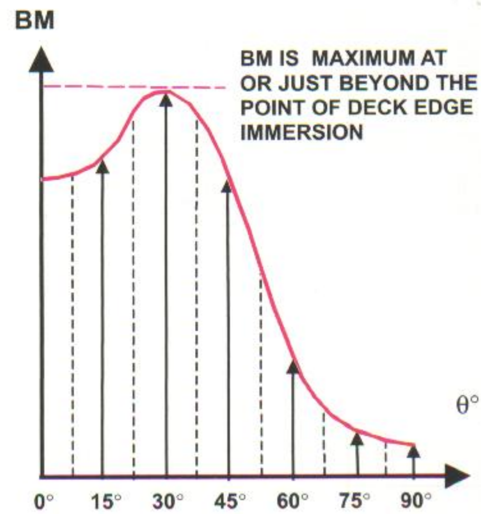
TRACKING THE CENTRE OF BUOYANCY AT INCREASING ANGLES OF HEEL

We can continue rotating the transverse sections in $7\frac{1}{2}°$ steps and determine the BM value at each step by repeating the process described in the last three pages, using the waterlines and rolling axis offsets, found previously, as the basis for the calculations of each successive step.
Page 28 shows that the Metacentre 'M' moves with changing angle of heel but it always remains in vertical alignment with C of B. We can estimate the track of both the Centre of Buoyancy and the Metacentre 'M' by starting with the upright condition. The C of B is swung 15° around M_0 with a radius of the BM value at $7\frac{1}{2}°$ to establish the position of the C of B at 15° (B_{15}). A new radius of the BM at $22\frac{1}{2}°$ is then projected back vertically from B_{15}, (i.e. perpendicular to the track of the C of B at that point) to estimate the average position of 'B' for the next 15° swing to 30° of heel. This procedure can be repeated for each 15° step up to 90° of heel, as shown below

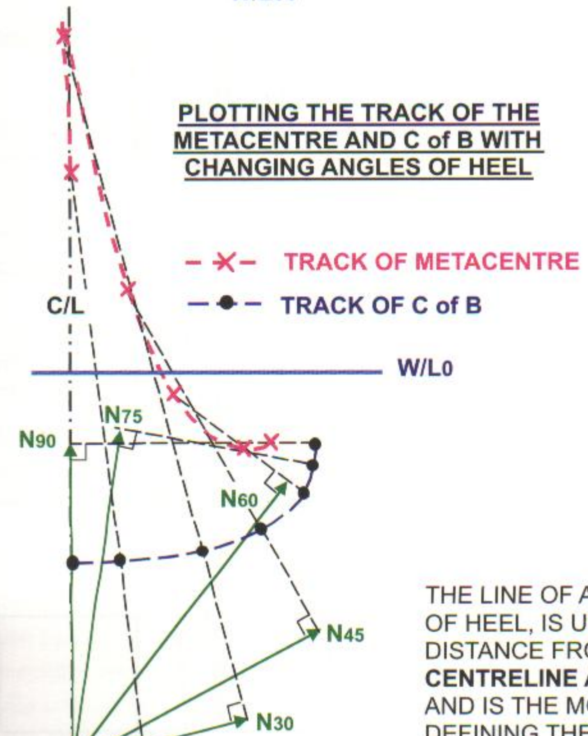
WATERLINES AND BM VALUES FOR ANGLES OF HEEL, 0° TO 90°



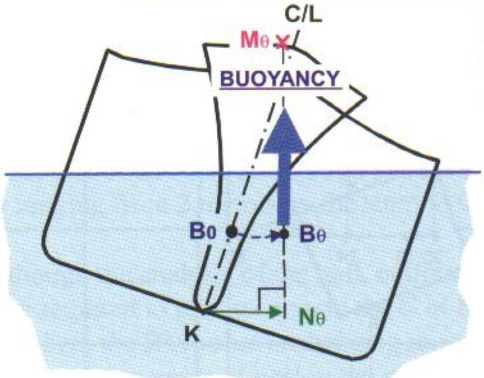
GRAPH OF BM / θ° OF HEEL



PLOTTING THE TRACK OF THE METACENTRE AND C of B WITH CHANGING ANGLES OF HEEL



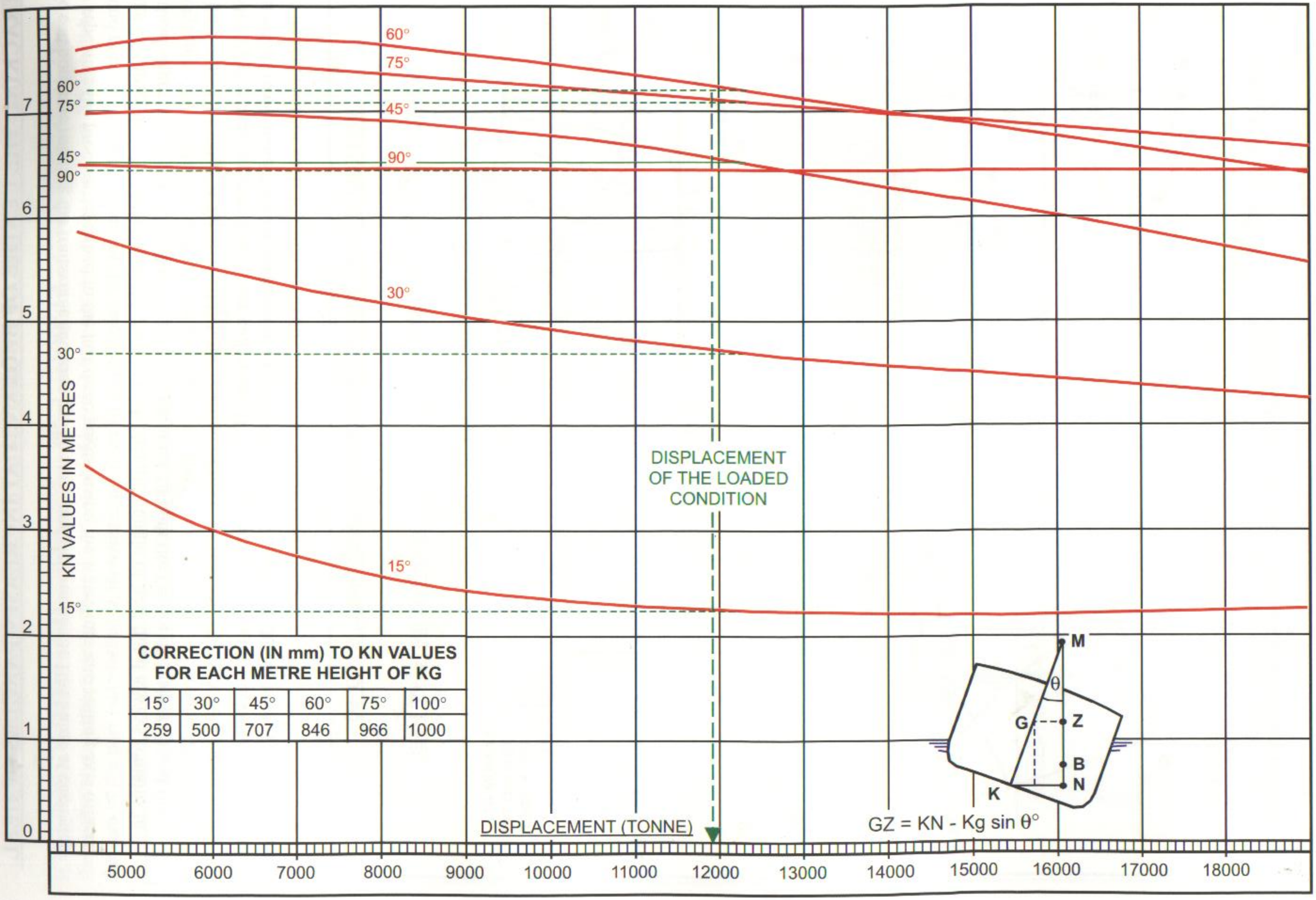
VESSEL AT θ° OF HEEL



THE LINE OF ACTION OF BUOYANCY AT ANY GIVEN ANGLE OF HEEL, IS USUALLY DEFINED BY ITS HORIZONTAL DISTANCE FROM THE SINGLE FIXED POINT **'K, ON THE CENTRELINE AT THE KEEL THIS IS CALLED ITS KN VALUE** AND IS THE MOST USUAL AND CONVENIENT WAY OF DEFINING THE LINE OF ACTION OF BUOYANCY

So. $KN_\theta = KM_\theta \times \tan\theta$

CROSS CURVES OF STABILITY (KN CURVES)



CORRECTION (IN mm) TO KN VALUES FOR EACH METRE HEIGHT OF KG					
15°	30°	45°	60°	75°	100°
259	500	707	846	966	1000

The Metacentre is not a very convenient reference for locating the line of action for the force of buoyancy, as it not a fixed point. The intersection between the vertical centreline and the keel, (point 'K'), is normally used instead and the measurements given are the horizontal distances between K and the line of Buoyancy for different conditions of draft and trim over the range of heel from 0 to 90°. These are called 'KN' values and the previous page shows the data presented graphically as KN values plotted against displacement.

The KN value at any instant, would be the Righting Lever 'GZ' at that particular draft and angle of heel, between the forces of Weight and Buoyancy if the Centre of Gravity of the ship was at the position 'K'. Consequentlty, the calculation of GZ is very simple

i.e RIGHTING LEVER 'GZ' at θ° of HEEL = KNθ - KG Sinθ

Other forms may show deadweight or draft scales in place of, or as well as, the displacement scale. The particular set of curves, shown on the page opposite, is a sample produced by the U.K Maritime Authority for use in certificate of competency examination questions, so it does not actually relate to a real ship but it is typical of what the builders will include in the stability and hydrostatic data book that every new vessel is supplied with. Alternatively, the information may be tabulated, as follows:-

TABLE OF KN VALUES

SW DRAFT METRES	DISPLACEMENT TONNES	D'EADWEIGHT* TONNES	KMo M	KNθ (METRES)			TRIM = 0.5M STERN		
				15°	30°	45°	60°	75°	90°
3.80	4800	0	13.14	3.52	5.77	7.00	7.65	7.48	6.50
4.00	5146	346	12.95	3.47	5.75	6.99	7.68	7.49	6.50
4.20	5502	702	12.80	3.43	5.73	6.99	7.69	7.50	6.50
4.40	5864	1064	12.69	3.40	5.71	6.98	7.69	7.51	6.50

A NEGATIVE DEADWEIGHT WOULD INDICATE A DRAFT CONDITION, LESS THAN THAT OF LIGHTSHIP. THIS WOULD BE REQUIRED IF THE SHIP IS TO BE LAUNCHED IN PARTLY COMPLETED STATE

Notice that the above extract of 'KN' tables includes a trim value, which will relate to the initial upright condition. Trim changes the distribution of buoyancy and will effect the shift in the C of B as the vessel heels over. If there is a significant difference in reserve buoyancy between the forward and aft ends of the hull, then the heeling of the hullform will actually produce a trimming effect which must be taken into account when considering transverse stability. The next chapter considers this 'Free Trim' effect in more detail,as it is particularly important to the type of ship which has a high fo'c'sle and low after deck (such as a rig supply vessel). In any case, sets of KN curves or tables should be provided for different states of trim of any vessel, usually ranging from 2 metres by the stern to 0.5m by the head.

It is normal to tabulate or plot KN values in steps of 15°, though the analysis of the hull, outlined on the previous pages, may indicate that intermediate steps are required to plot accurately the C of B's track. This would be particularly so over the range of heel where the BM value is changing most rapidly. Errors due to approximation will accumulate with increasing angle of heel in the process of projecting further the track of the C of B from the position estimated by the calculations of the previous step.

These calculations assume that the hull remains rigid as the buoyancy distribution changes with heel. The excess buoyancy produced by the flare of the fore and aft ends at increasing angles of heel, will produce bending moments to make the hull sag in the midships region. The structure deforms slightly as it generates extra molecular forces within the steel to withstand the bending moments. The resulting slight change of shape is generally ignored in the transverse stability hull analysis, though they are important in estimating the strength requirements of the hull.

CONCLUDING COMMENTS ON THE HULLFORM ANALYSIS FOR TRANSVERSE STABILITY CHARACTERISTICS

This chapter has outlined a means by which the transverse changes in the buoyancy distribution can be determined as a ship is heeled over. The previous pages should demonstrate that this involves the processing of a considerable number of measurements, which will require a rigorous set procedures to avoid confusion. In the past, a ship's lines plans were drawn out by hand, often to full scale, in a process known as 'lofting the lines' because the rigging loft in the shipyard was the only covered space large enough to carry out the operation. The success of the whole process relied upon the accuracy and skill of the draughtsmen and their drawing equipment but the use of modern computer techniques has improved precision, whilst taking a lot of the man hours out of the sheer task of 'number crunching' all the data. It should be possible to take measurements at much closer intervals and rotate the hull around smaller increments of heel relatively easily once the basic hull shape data has been put into a computer aided design (CAD) program.

It is not necessary to actually draw out curves of $(\text{beam})^2$ and $(\text{beam})^3$. They are shown in the text simply to illustrate the principles of the calculations and, in any case, modern computerised processing eliminates manual calculations. However, it should be appreciated that any errors in the integration program increase with measurements raised to higher powers, so sampling intervals of beam, which are adequate for calculating the waterplane area, may be insufficient for calculating moments of area and inertia. These require closer measuring stations where there is a significant change of beam, even if this appears to be almost linear over the length of an interval, as graphs of the same beam measurements, squared or cubed, will not be straight lines. This is particularly important for the design of smaller hullforms, such as pleasure yachts, which are considerably more curved and require more measuring stations to define their shape adequately. Large commercial cargo carrying hulls tend to have a moderate proportion of midships parallel body in their length and so are less sensitive to the errors inherent in the integration methods and wall-sided approximations used in the calculations.

A similar process of analysis is used to determine trim properties, by taking longitudinal moments of waterplane area and buoyancy about the aft perpendicular, whilst bending moments calculations require the buoyancy distribution over the ship's length. These topics are covered in later chapters but a full hull analysis would include determining all the hydrostatic properties of a ship. It is not always necessary, though, to go through this procedure with every new ship built as two geometrically similar shaped hulls will possess the same hydrostatic properties. If the offset measurements of a previously built vessel are all increased by 10% then the displacement will be 33% greater, resulting in a considerably larger ship, but the KN values will increase by the linear proportion of 10%. This means that if a shipyard analyses about six different hull shapes in detail, it can build a wide range of ships of varying size without the need for further analysis, providing that vessels all conform to one of the basic geometric shapes, stored in the yard's database.

Ship's officers tend to regard transverse stability problems as being concerned with the distribution of weight within the ship as this is the factor that they have some control over. However, the position of the ship's centre of gravity is only half of the equation and the hullform properties are equally important, especially in dictating the way a ship rolls. Most seafarers will know of ships, with a reputation for being comfortable and of ones that are said to be able to 'roll on wet grass'. Sometimes the ship's trade or fixed weights within the vessel make it is impossible to load the ship in a really satisfactory way to avoid violent rolling, but the actual shape of the hull is critical in how rapidly buoyancy distribution changes with increasing angles of heel. This is particularly relevant to the ship's behaviour at the ends of a roll where the accelerations of the motion are the greatest and most wearing on the crew. Comfort for those onboard who are not fare paying passengers is generally not considered as a top priority by the shipping industry but the human being is a fairly good motion sensor even if he is not very good at expressing exact measurements. If the ship's rolling is hard on the crew, it is almost certainly causing problems for the ship's structure as well, so it should be worth considering which features of the hull shape tend to promote a more comfortable motion in a seaway. Chapter 3 examines how various hullform features affect the shape of the righting curve for a given height of centre of gravity, which should give more insight to a ship's transverse stability characteristics, whilst chapter 6 considers rolling motion in more detail.

CHAPTER 3

TRANSVERSE STABILITY CHARACTERISTICS & THE GZ CURVE

SUMMARY

THIS CHAPTER EXAMINES THE ENERGY INVOLVED IN HEELING A SHIP OVER, HOW THE DESIGN OF THE HULL INFLUENCES A SHIP'S STABILITY CHARACTERISTICS AND THE CRITERIA FOR MEASURING TRANSVERSE STABILITY.

THE CHAPTER IS GROUPED INTO THE FOLLOWING TOPICS

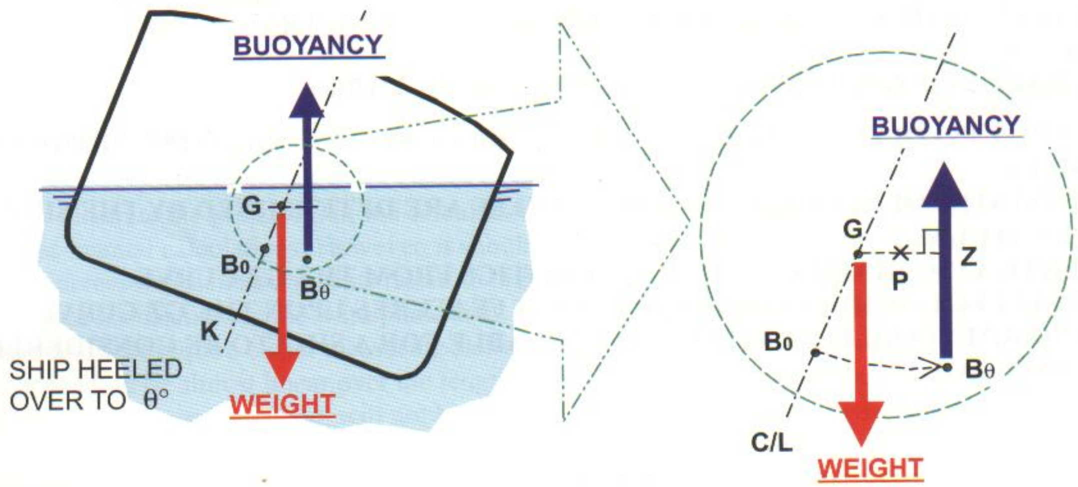
1) **ENERGY INVOLVED IN HEELING A SHIP, THE CURVES OF RIGHTING MOMENT AND GZ.**
2) **THE STATES OF STABILITY AND HOW THESE ARE DETERMINED BY THE SHIP'S UPRIGHT GM.**
3) **IDENTIFYING STABILITY CHARACTERISTICS FROM THE GZ CURVE.**
4) **THE EFFECT OF HULLFORM AND DESIGN FEATURES UPON THE GZ CURVE.**
5) **MINIMUM STABILITY CRITERIA ACCEPTABLE FOR A SHIP TO BE CONSIDERED SEAWORTHY.**

CONTENTS

ENERGY INVOLVED IN HEELING A VESSEL AND THE RIGHTING MOMENT

As Chapter 2 explained, when a vessel is heeled over from the upright condition, the forces of buoyancy and weight move out of vertical alignment as the Centre of Buoyancy 'B' shifts with the change of underwater hullform. This produces a **Righting Moment**, measured in Tonnes-Metres (T-M), as shown below

THE RIGHTING MOMENT



'P' IS MIDPOINT OF THE PERPENDICULAR DISTANCE 'GZ' BETWEEN THE TWO EQUAL AND OPPOSITE FORCES OF WEIGHT AND BUOYANCY. TAKING MOMENTS ABOUT 'P'

TOTAL RIGHTING MOMENT $= \frac{1}{2}$**GZ (WEIGHT + BUOYANCY)**

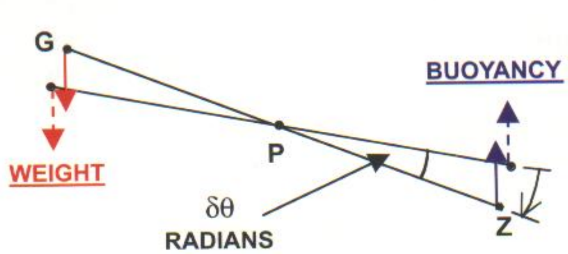
BUT WEIGHT = BUOYANCY, SO

TOTAL RIGHTING MOMENT = GZ x SHIP'S WEIGHT

IF THE EXTERNAL FORCE HEELS THE SHIP OVER THROUGH THE ADDITIONAL SMALL ANGLE OF δθ°, THEN THE ENERGY USED (OR WORK DONE) TO DO THIS IS GIVEN BY:-

WORK DONE = FORCE x DISTANCE FORCE IS MOVED

PROVIDED THAT δθ IS SMALL, THE TWO FORCES OF WEIGHT AND BUOYANCY BOTH WILL MOVE IN AN ARC OF CONSTANT RADIUS AND CENTRED ON POINT 'P'. IF δθ IS MEASURED IN RADIANS, THEN THE LENGTH OF THE ARC WILL BE EQUAL TO δθ x RADIUS OF THE ARC, SO THE WORK DONE IN INCREASING THE HEEL BY THIS SMALL ANGLE, IS AS FOLLOWS:-



RADIUS OF ARC $= \frac{1}{2}$**GZ**

SO WEIGHT AND BUOYANCY ARE BOTH MOVED THROUGH THE LENGTH OF ARC, GIVEN BY:-

ARC LENGTH $= \delta\theta \times \frac{1}{2}$**GZ**

WORK DONE BY HEELING SHIP THROUGH δθ = RIGHTING MOMENT x δθ RADS

WORK DONE BY HEELING SHIP THROUGH δθ = SHIP'S WEIGHT x GZ x δθ T-M-RADIANS

MEASURING THE ENERGY STORED IN THE RIGHTING MOMENT

Chapter 2 showed how analysis of the underwater hullform at different drafts and angles of heel, produces a set of Cross Curves of Stability, known as KN Curves. If the height of the Centre of Gravity above the keel (the 'KG' value) is known for a ship's particular loaded state, then the values of GZ (the Righting Lever) can be calculated for angles of heel given by the KN curves.

RIGHTING LEVER GZ AT $\theta°$ OF HEEL = KN AT $\theta°$ OF HEEL x KG sin θ

These values can then be used to plot a graph of GZ against angles of heel, known as **a GZ Curve**, or they can be multiplied by the ship's displacement, '**Δ't**, to produce a **Righting Moment Curve**. The basic shape of these two curves will be the same, as the displacement remains constant for a given loaded condition, and either can be used to indicate the hull's stability characteristics in that particular loaded state. The Righting Moment Curve, however, allows us to calculate the energy used in rolling a ship over to any particular angle of heel. This energy provides the Righting Moment and is known as the **Dynamic Stability**.

From the previous page,

WORK DONE BY CHANGE OF HEEL '$\delta\theta^r$' = Δ't x GZ x $\delta\theta$ T-M-RADIANS

So the total Work Done in heeling the ship over to θ **RADIANS**, would be the sum of all the separate small increases in heel, i.e.

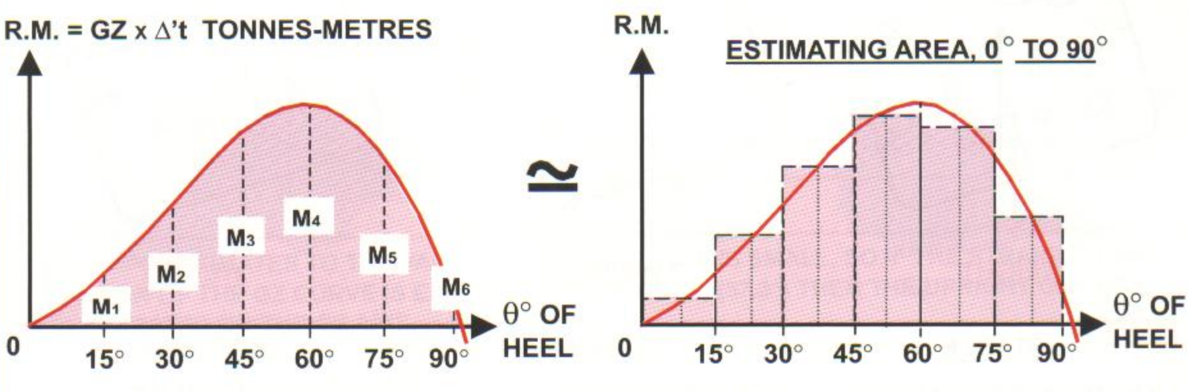
WORK DONE BY CHANGE OF HEEL from 0 to θ^r = Σ (Δ't x GZ) dθ T-M-RADIANS

This is equal to the area under the Righting Moment curve, from the upright to θ **RADIANS** of heel. Angles of heel are generally measured in degrees, so we need to convert this to Radians. **A Radian is the angle subtended at the centre of an arc, where the length of the arc is equal to its radius.** Now, a circle is an arc of 360° and its length (i.e. its circumference) so;-

CIRCUMFERENCE OF A CIRCLE = $2\pi r$, SO NUMBER OF RADIANS / 360° = 2π

SO, 1 RADIAN = $\frac{360°}{2\pi}$ SO, 1 RADIAN = 57.3°

ESTIMATING THE RANGE OF A SHIP'S DYNAMIC STABILITY



DYNAMIC STABILITY FROM 0° TO 90° HEEL = AREA UNDER R.M. CURVE BETWEEN 0° & 90°, WHERE M1, M2, M3, M4, M5 & M6 ARE THE MOMENTS, MEASURED AT 15° INTERVALS USING THE TRAPEZIUM METHOD OF APPROXIMATE INTEGRATION;-

$$\text{DYNAMIC STABILITY, } 0° \text{ TO } 90° \text{ HEEL} = \frac{15}{57.3}\left(M_1 + M_2 + M_3 + M_4 + M_5 + \tfrac{1}{2}M_6\right)$$

THE GZ AND RIGHTING MOMENT CURVES COMPARED

The most important requirement for any hull, is that it will return to the upright after being temporarily heeled over by the strongest forces of the wind and the waves that it is likely to encounter. There are, however, other characteristics of the hull that are also important, such as the extent to which it will heel over for a given seaway and the quickness of resulting rolling motion. A ship that heels over excessively due to a small righting moment is known to be **'tender'** and will have an increased risk of flooding due to waves sweeping the deck. A **'stiff'** vessel has a large righting lever and will roll with a quick and violent motion that increases the stress on the vessel's structure and is unpleasant for the crew. As stated on the previous page, a considerable amount of information can be gained about a ship's stability performance from either the GZ curve or the Righting Moment curve but it is important to appreciate the difference between the two curves.

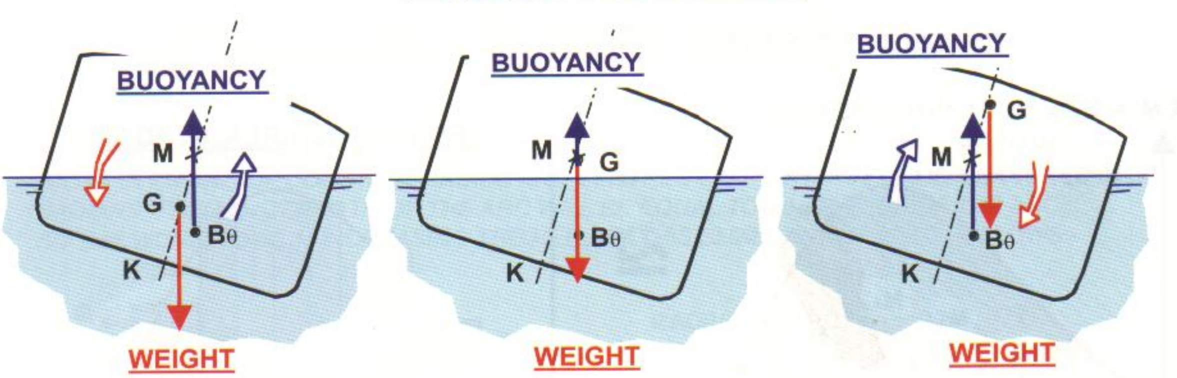
The GZ curve simply shows the changing value of the Righting Lever as the underwater hullform alters with the angle of heel, whereas the area under the Righting Moment curve measures the energy involved in changing the underwater hullform, although both curves have the same shape. Two similar shaped hulls of differing displacement can have the same GZ curve, if the upright GM is the same. The two vessels will share certain characteristics, such as the angle of heel at which the deck edge is immersed and at which the GZ reaches its maximum value. However, heeling the heavier ship will require more energy than that used in producing the same angle of heel in the smaller vessel, so for the same wave and wind conditions, the larger ship should be steadier and roll about less.

The various maritime authorities have produced guidelines and rules, which state minimum stability criteria which a loaded ship must comply with in different circumstances, depending upon the type of vessel, its cargo and the climate it is operating in. In some situations, the criteria simply relate to the GZ curve, whilst for other circumstances, they refer to the Righting Moment curve, which takes the ship's displacement into account as well as the changing value of the Righting Lever, 'GZ'.

The criteria quoted in this book will be those of 'The Marine Coast Guard Agency of the U.K.' (MCA-U.K.) and 'The International Maritime Organisation' (The I.M.O.), which are standards generally accepted by most major maritime nations.

TRANSVERSE STABILITY CHARACTERISTICS OF A HULL

BASIC STABILITY CONDITIONS



STABLE

KM GREATER THAN KG
KM – KG = POSITIVE GM
HEELING PRODUCES
RIGHTING MOMENT

NEUTRAL

KM EQUALS KG
KM – KG = ZERO GM
INITIAL HEELING
PRODUCES NO MOMENT

UNSTABLE

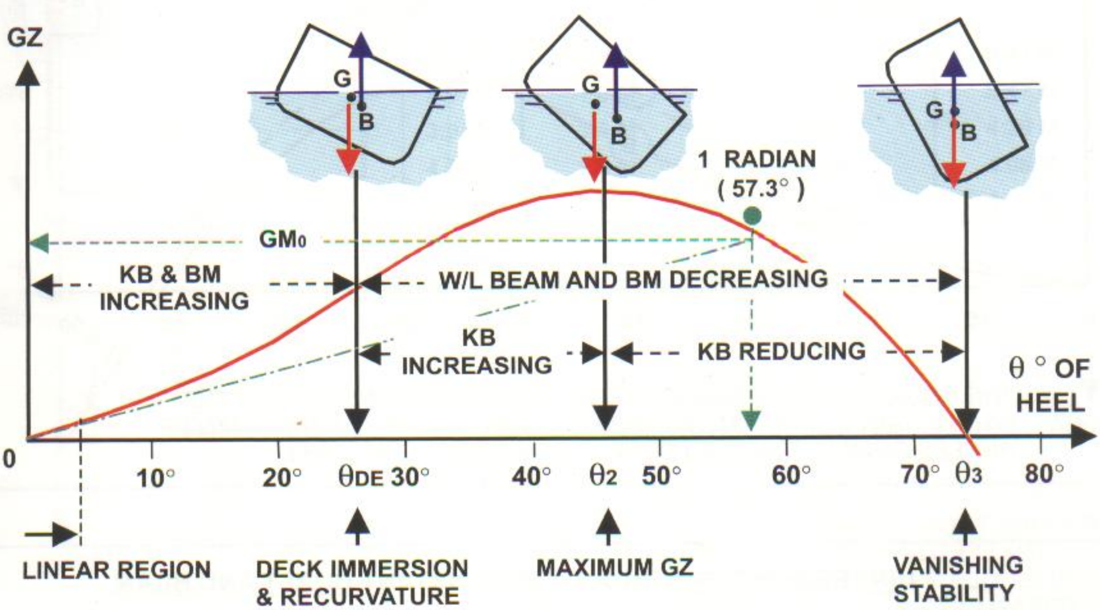
KM LESS THAN KG
KM – KG = NEGATIVE GM
HEELING PRODUCES
CAPSIZING MOMENT

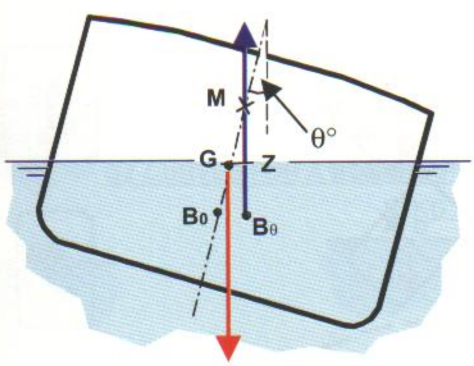
THE UPRIGHT GM VALUE IS THE MAIN INDICATOR OF A SHIP'S STABILITY CONDITION

TRANSVERSE STABILITY CHARACTERISTICS OF A HULL (Cont.)

A ship will have positive stability at any angle of heel, if the Centre of Buoyancy 'B' is outboard of the Centre of Gravity 'G'. Chapter 2 showed that, for heel angles within about 5° of the upright position, the Metacentre 'M' remains approximately in the same place but as the heel increases, so KB and BM increase with the increases of waterline beam and underwater asymmetry. This causes 'M' to rise up the centreline and move outboard to the low side at increasing rates until the deck edge becomes immersed, at which point waterline beam and BM start decreasing so the ship's positive stability will stop increasing. Further reduction in the waterline beam (and, hence, BM) at greater angles of heel, will lead to actual reduction of positive stability.

FEATURES OF THE RIGHTING LEVER 'GZ' CURVE





SMALL ANGLE OF HEEL APPROXIMATIONS

UP TO ABOUT 5° HEEL OF A BOX-SHAPED HULL, THE WATERLINE BEAM REMAINS APPROXIMATELY CONSTANT AND THE RISE OF 'B' IS NEGLIGIBLE, SO 'M' CAN BE CONSIDERED TO BE STATIONARY. THE INITIAL PART OF THE GZ CURVE IS LINEAR

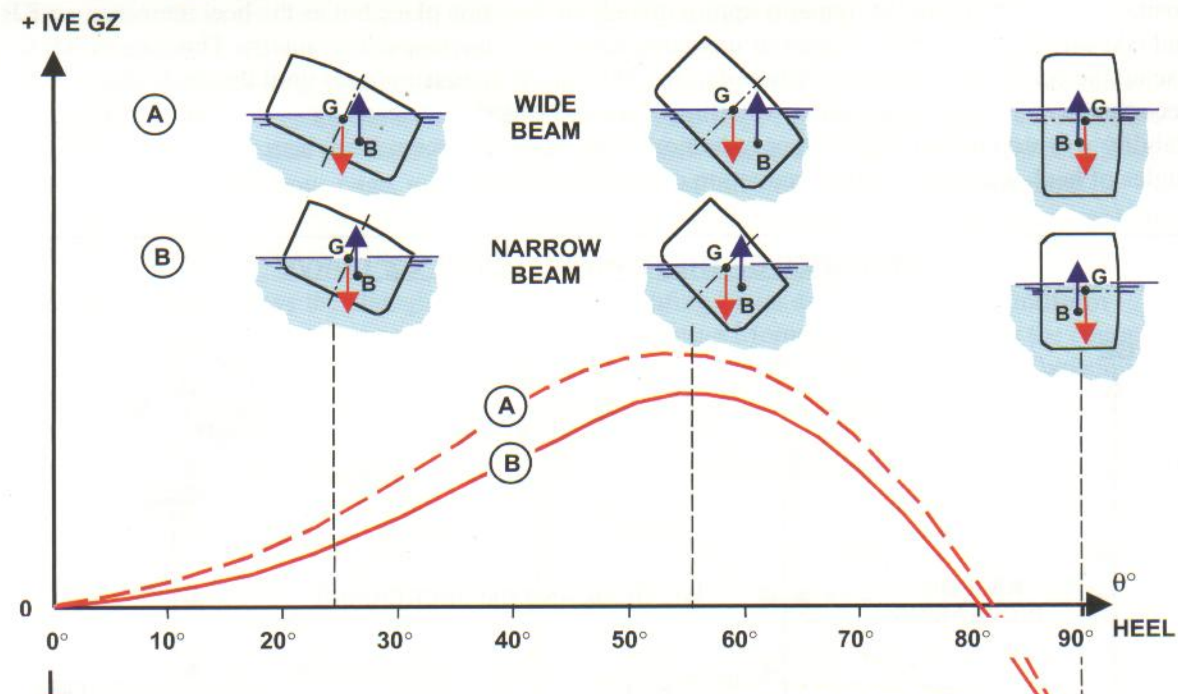
AT θ° OF HEEL, GZ = $GM_0 \times \sin\theta$

Sin θ APPROXIMATES TO THE VALUE OF θ RADIANS IF θ IS SMALL, SO IF THE STRAIGHT LINE PORTION OF THE GZ CURVE IS EXTENDED TO 1 RADIAN, (57.3°) THEN THE UPRIGHT GM_0 WILL BE INDICATED ON THE GZ AXIS

A ship with a negative upright GM will be unstable when upright and so will heel over at least 4° or 5° before there is any significant rise of 'M' up the centreline. The vessel will regain positive stability at larger angles of heel, if the rise in 'M' is sufficient to counter the initial negative GM. The normal range of acceptable GM values is in the order of 0.2 to about 2 metres, though this may be exceeded in the case of very large vessels. The diagrams in this book (and most others), tend to exaggerate the GM and shifts in the C of B, relative to the ship's dimensions, for the sake of clarity.

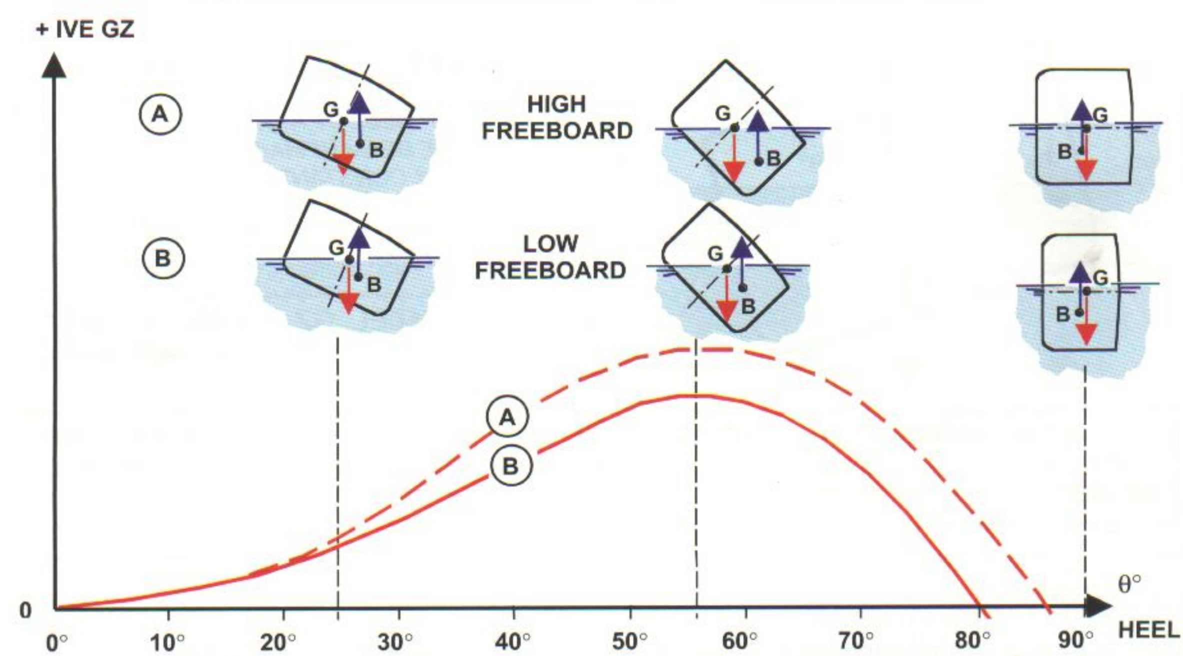
FEATURES AFFECTING THE GZ CURVE OF A BOX-SHAPED HULL

1 INCREASING BEAM AT CONSTANT DRAFT AND FREEBOARD



THE GREATER BEAM INCREASES THE VESSEL'S STIFFNESS OVER THE WHOLE RANGE OF POSITIVE STABILITY, PARTICULARLY AT THE SMALLER ANGLES OF HEEL. THE TWO CURVES CONVERGE AT 90° HEEL

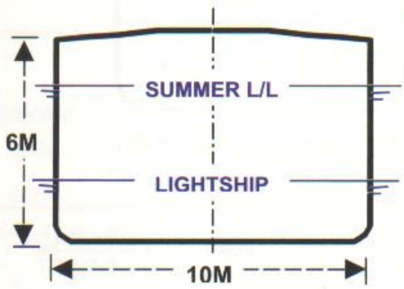
2 INCREASING FREEBOARD AT CONSTANT DRAFT AND BEAM



THE GZ CURVE IS IDENTICAL FOR THE TWO HULLS UP TO THE ANGLE OF IMMERSION OF SHIP 'B' DECK LINE. BEYOND THIS POINT, THE EXTRA FREEBOARD OF SHIP 'A' INCREASES ITS DYNAMIC STABILITY

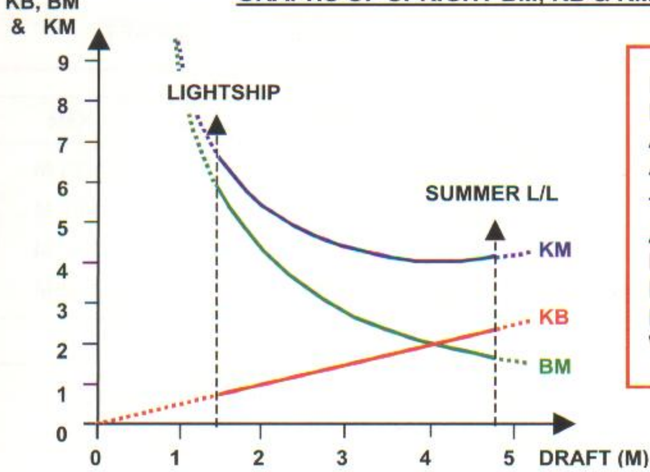
In the previous two features, the effects of beam and freeboard upon the GZ curve were considered for separate hulls. In this case, we are considering the stability changes that will occur if we progressively increase the loading of a box-shaped hull in such a way as to maintain a constant KG. As the draft is increased from lightship to the Summer loadline, so the KB increases but the BM is reduced because the volume of the wedge of buoyancy, transferred on heeling, becomes a progressively lower proportion of the total immersed volume.

(3) INCREASING DRAFT AND REDUCING FREEBOARD OF A HULL AT CONSTANT KG



DRAFT	KB = $\frac{\text{DRAFT}}{2}$	BM = $\frac{(\text{BEAM})^2}{12(\text{DRAFT})}$	KM = KB + BM
1.0 M	0.5 M	8.33 M	8.83 M
2.0 M	1.0 M	4.17 M	5.17 M
3.0 M	3.5 M	2.78 M	4.28 M
4.0 M	2.0 M	2.08 M	4.08 M
5.0 M	2.5 M	1.67 M	4.17 M

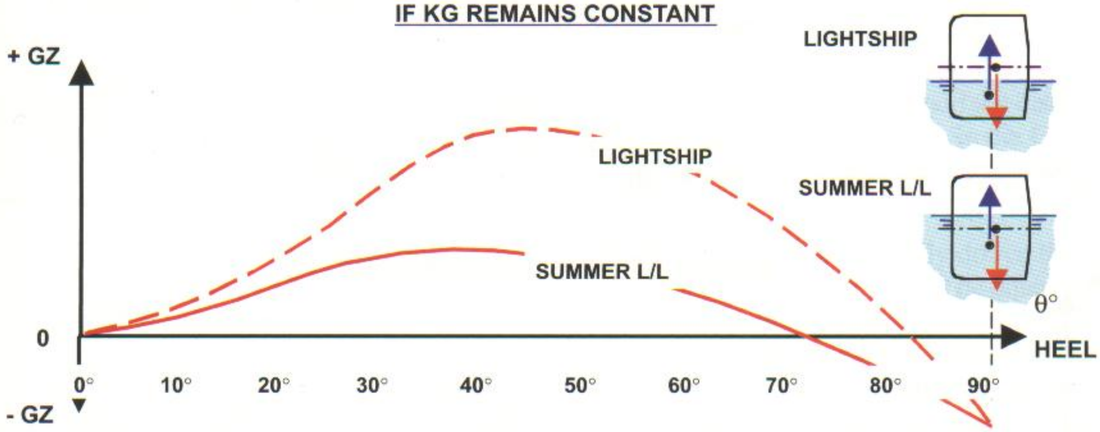
GRAPHS OF UPRIGHT BM, KB & KM AGAINST DRAFT



INCREASES OF DRAFT TEND TO PROGRESSIVELY DECREASE THE KM AND, HENCE, THE GM VALUES, BUT AT A CONTINUALLY REDUCING RATE.

THE FREEBOARD WILL ALSO REDUCE AS DRAFT INCREASES, SO AT SUMMER LOAD, THE BOX-SHAPED HULL WILL HAVE LOWER VALUE AND REDUCED RANGE OF POSITIVE GZ, COMPARED WITH THE LIGHTSHIP CONDITION.

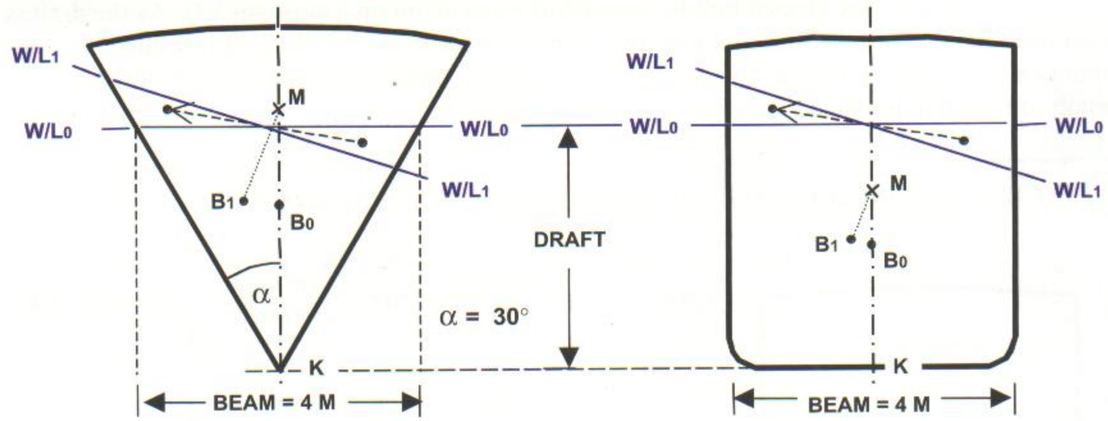
GZ CURVES FOR A BOX-SHAPED HULL AT SUMMER LOAD AND LIGHTSHIP IF KG REMAINS CONSTANT



THE RIGHTING MOMENT (i.e. GZ x DISPLACEMENT) CURVES WILL, HOWEVER, SHOW THAT THE EXTRA WEIGHT OF THE LOADED VESSEL DOES REQUIRE MORE ENERGY TO ROLL THE HULL THROUGH A PARTICULAR ANGLE OF HEEL AS DISPLACEMENT IS INCREASED

THE TRIANGULAR CROSS-SECTIONAL HULLFORM

KB₀ AND BM₀ OF TRIANGULAR AND BOX-SHAPED SECTIONS WITH EQUAL W/L BEAM



$KB = \frac{2}{3}DRAFT, \quad BM_0 = \frac{(BEAM)^2}{6 \times DRAFT}$

$BEAM = DRAFT \times 2\tan\alpha, \quad BM_0 = \frac{2}{3}DRAFT\tan^2\alpha$

$KM_0 = \frac{2}{3}DRAFT(1 + \tan^2 30°), \text{so } KM_0 = \frac{8}{9}DRAFT$

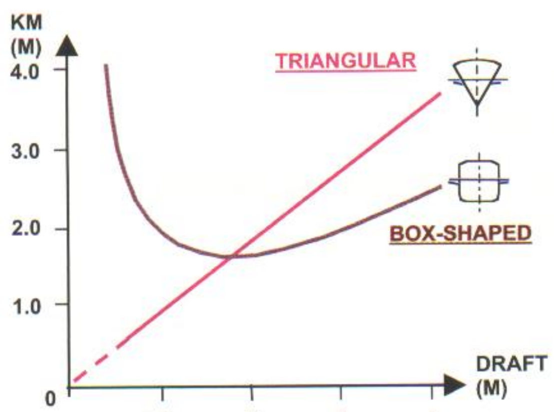
$KB = \frac{1}{2}DRAFT, \quad BM_0 = \frac{(BEAM)^2}{12 \times DRAFT}$

BEAM EQUALS 4 M

$KM_0 = \frac{1}{2}DRAFT + \frac{4}{3\,(DRAFT)}$

DRAFT	KM_0
1.0 M	0.89 M
2.0 M	1.78 M
3.0 M	2.67 M
4.0 M	3.56 M

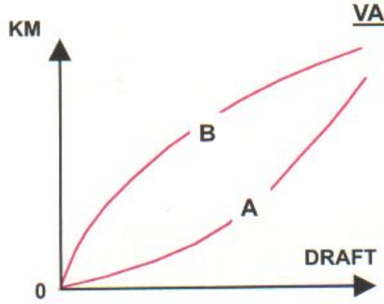
DRAFT	KB	BM_0	KM_0
1 M	0.5 M	1.33 M	1.83 M
2 M	1.0 M	0.67 M	1.67 M
3 M	1.5 M	0.44 M	1.94 M
4 M	2.0 M	0.33 M	2.33 M

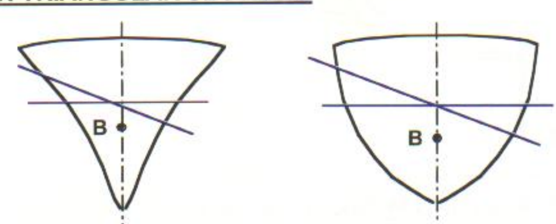


THE TRIANGULAR SECTION HAS HALF THE IMMERSED VOLUME OF A BOX-SHAPED SECTION WITH THE SAME W/L BEAM, SO ITS VALUE OF BM₀ IS TWICE AS GREAT.

AS DRAFT INCREASES. THE BOX-SHAPED SECTION RETAINS A CONSTANT BEAM, WHEREAS THE W/L WIDTH OF THE TRIANGULAR SECTION IS DIRECTLY PROPORTIONAL TO THE DRAFT. THIS RESULTS IN BOTH THE KB AND BM₀ VALUES INCREASING PROPORTIONALY WITH THE DRAFT OF A TRIANGULAR SECTION

VARIATIONS ON TRIANGULAR SECTIONS





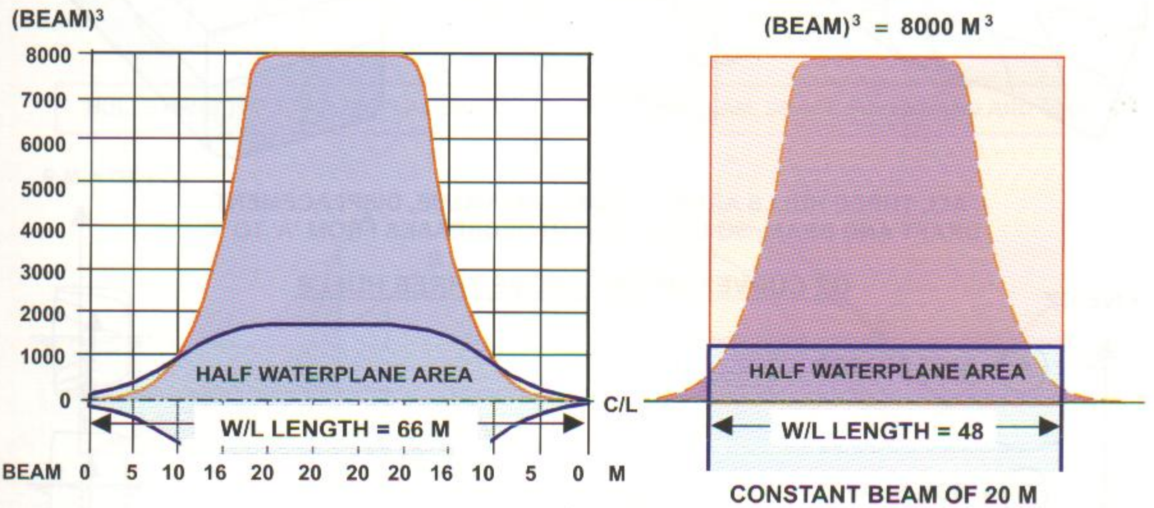
A = CONCAVE TRIANGLE (HURRICANE BOW)

B = CONVEX TRIANGLE (FULL STERN)

STABILITY CHARACTERISTICS OF DIFFERENT SHAPES OF HULL

Now that we see how changing different single characteristics of the hull form alters the GZ curve, we can start to build up a curve of an actual ship-shaped hull. Features, such as sheer, flare and block coefficient, greatly influence the shape of a GZ curve and, consequently, the rolling characteristics of a vessel. Ships' officers are inclined to concentrate their attention on the KG and GM values, as the determining stability factors of their vessel, which is understandable because they control the weight distribution onboard but are hardly in a position to alter the shape of the hull. However, understanding the influence of hullform upon the ship's transverse stability, will help them to develop a 'feel' for their own particular vessel and an appreciation of its limitations under service conditions.
The first step of this process is to compare the characteristics of the box-shaped hull to that of a wall-sided vessel, tapering at the fore and aft ends.

THE WATERPLANE MOMENT OF INERTIA



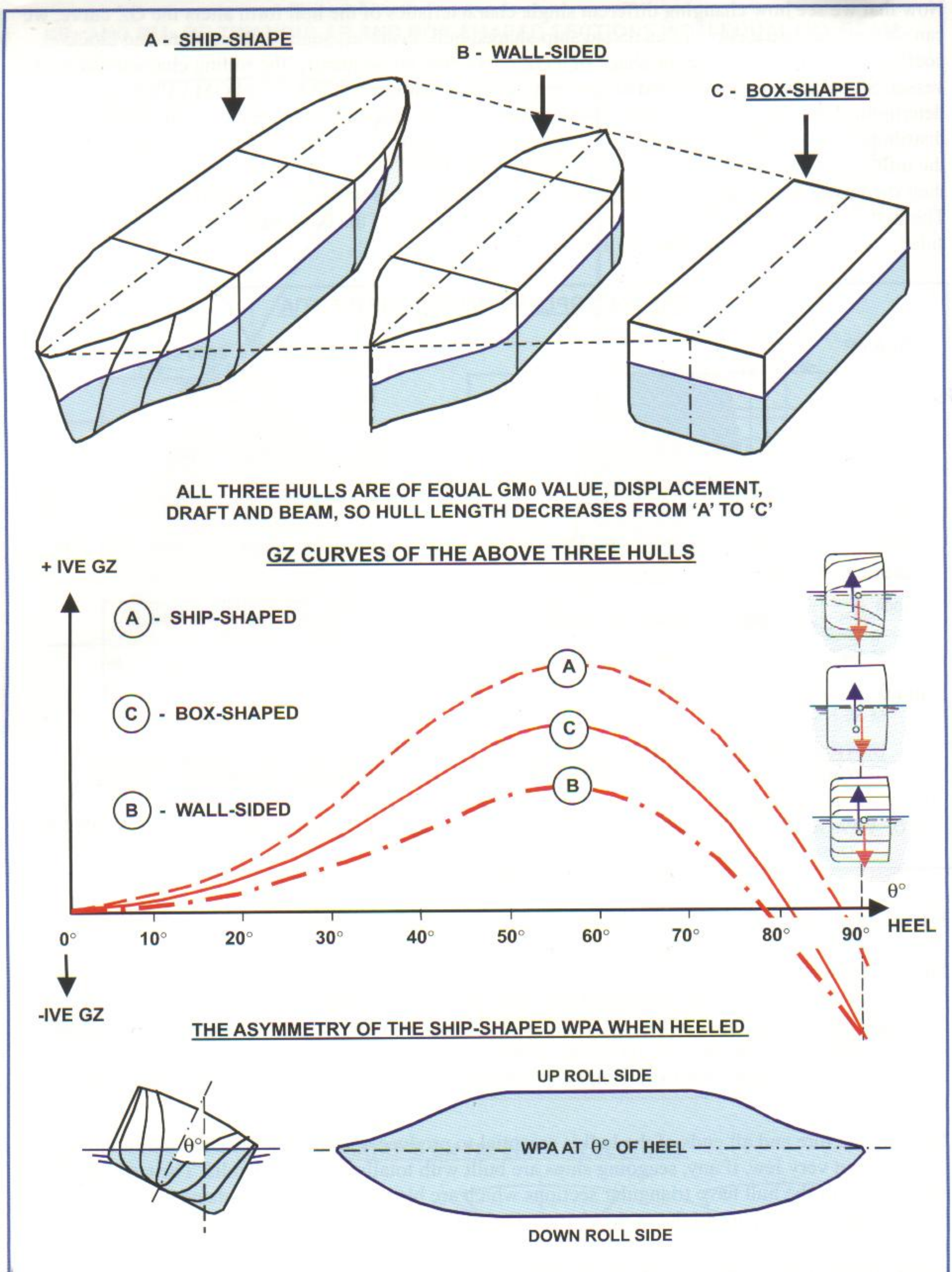
DISTRIBUTION OF $(BEAM)^3$ VALUES FOR A WALL-SIDED TAPERED WATERPLANE

DISTRIBUTION OF $(BEAM)^3$ VALUES FOR AN EQUIVALENT BOX-SHAPED VESSEL

'I' OF WPA $= \sum \frac{(BEAM)^3 \times \delta L}{12}$ so 'I' $= \frac{1}{12}$ SHADED AREA, 'I' OF WPA $= \frac{(BEAM)^3 \times L}{12} = \frac{8000 \times 48}{12}$

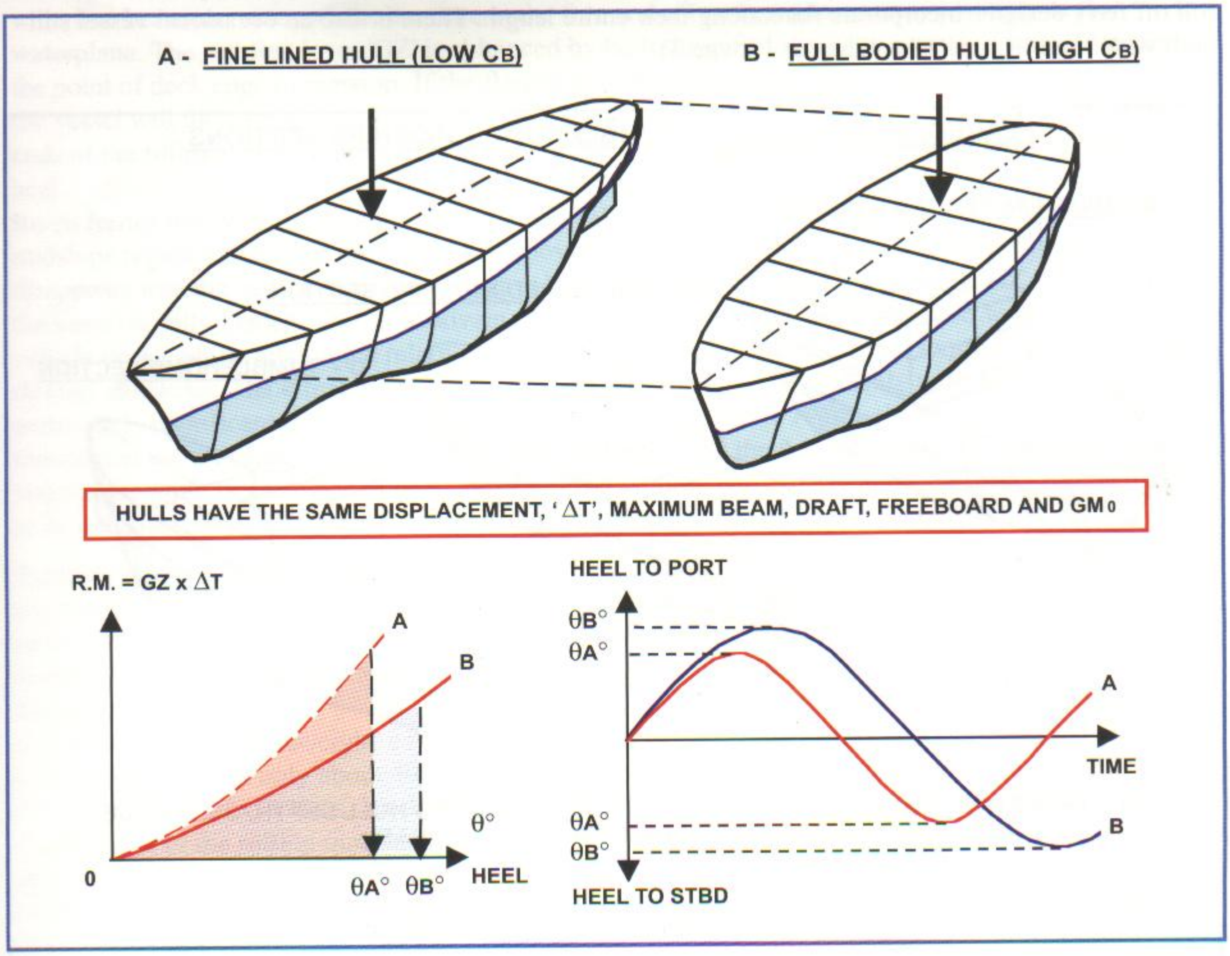
The draft and hence the KB values for the two hull forms will remain the same as both are parallel sided along their entire length and have the same displacement.
The Waterplane Moment of Inertia, 'I', and the BM value, are directly proportional to the sum of the $[Beam]^3$ values along the ship's length, which is the purple area under the curve in the above diagrams. Tapering the ends of the waterplane, whilst increasing its length to maintain the displaced volume, reduces the effective beam (i.e. the cube root of the average $[Beam]^3$ value).So the GZ curves will show that the tapered wall-sided hull has reduced stability compared with a box-shaped hull of the same displacement.

Tapering the fore and aft ends of the hull is essential to produce a shape that is easily driven through the water but very few, if any, seagoing ships are built with totally wall-sided hulls. The tapered fore and aft ends of the hull have triangular sections which are blended into the box-shaped parallel body in the midships region. This produces the characteristic '**Flare**' of the hull at the bow and stern. The easily driven tapered waterplane is maintained, but the transverse stability characteristics are enhanced by the considerable increase in waterplane beam in the flared regions when the ship is heeled over. Flare also greatly increases the hull's resistance to pitching as the extra reserve buoyancy it provides at the ship's ends, helps the ship ride over waves, rather than push its way through. Both of these characteristics are further reinforced by the feature of '**Sheer**', in which the freeboard is gradually increased at the ends of the hull as the upright waterline beam is reduced.



The tapered wall-sided hull has the lowest value of GZ for any particular angle of heel. Flare enhances stability by increasing the asymmetry of the waterplane as it is heeled. This is particularly so in the forward region where the flare is usually more concave in shape than at the stern.

A STABILITY COMPARISON BETWEEN FINE LINED AND FULL BODIED HULLS

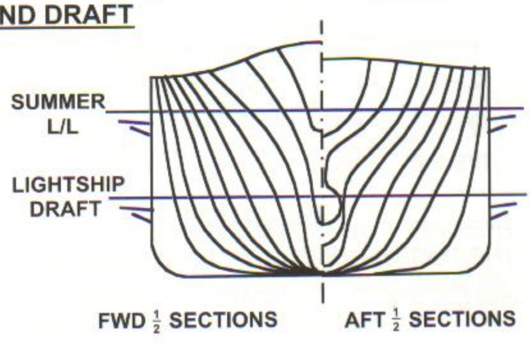


The two vessels above have the same upright GM and displacement values so the initial trends of GZ and Righting Moment curves will also be the same at small angles of heel. However, as heel is increased further, the asymmetry of the fine lined hull and, hence, its righting lever, will increase at a greater rate than that of the full bodied vessel. When a ship rolls in a seaway, wave energy is transferred to the work done in heeling the ship. This is equal to the area under the **Righting Moment** curve up to the angle of heel reached at the end of the roll. The fine lined hull will absorb the same amount of wave energy at a lesser angle of heel than the full bodied vessel, particularly if both ships are rolling through relatively large angles. The heel angle at the end of the roll is reduced but the roll will be quicker with higher accelerations. **In heavy seas, the increased stiffness of fine lined hulls at large angles of heel, can lead to a more violent and uncomfortable roll than a fuller bodied hull with the same displacement and GM0.**

BLOCK COEFFICIENT AND DRAFT

THE AVERAGE BEAM OF THE WATERPLANE AREA IS CONSIDERABLY REDUCED AT LIGHTER DRAFTS, PARTICULARLY IN THE CASE OF FINE LINED HULLS.

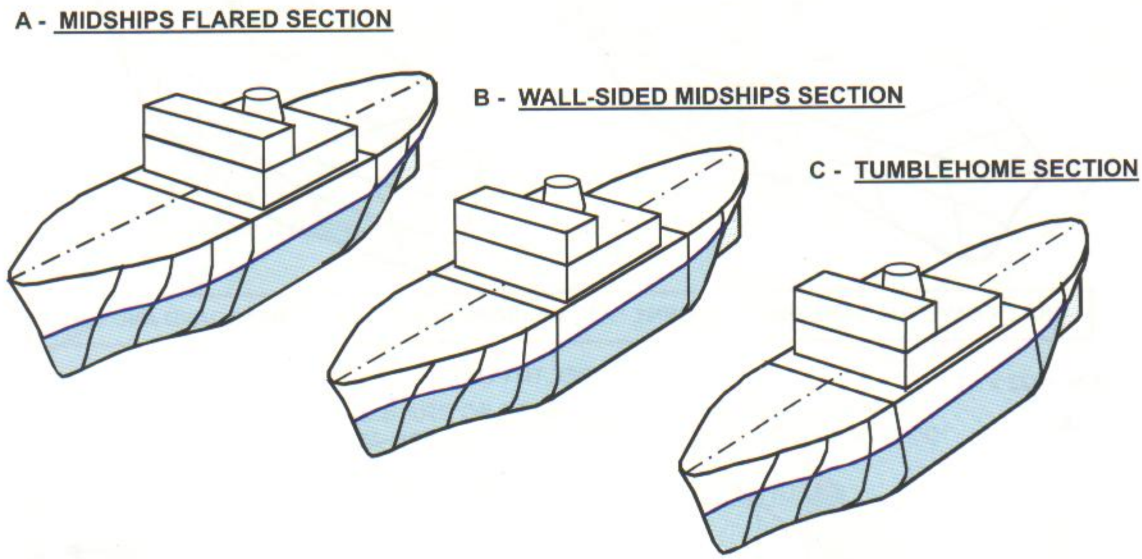
SHIPS WITH LOW C_b VALUES TEND MORE READILY TO LOSE STABILITY AS THE DRAFT DECREASES. THIS IS THE OPPOSITE TREND TO BOX-SHAPED HULLS



MIDSHIPS FLARE AND TUMBLEHOME

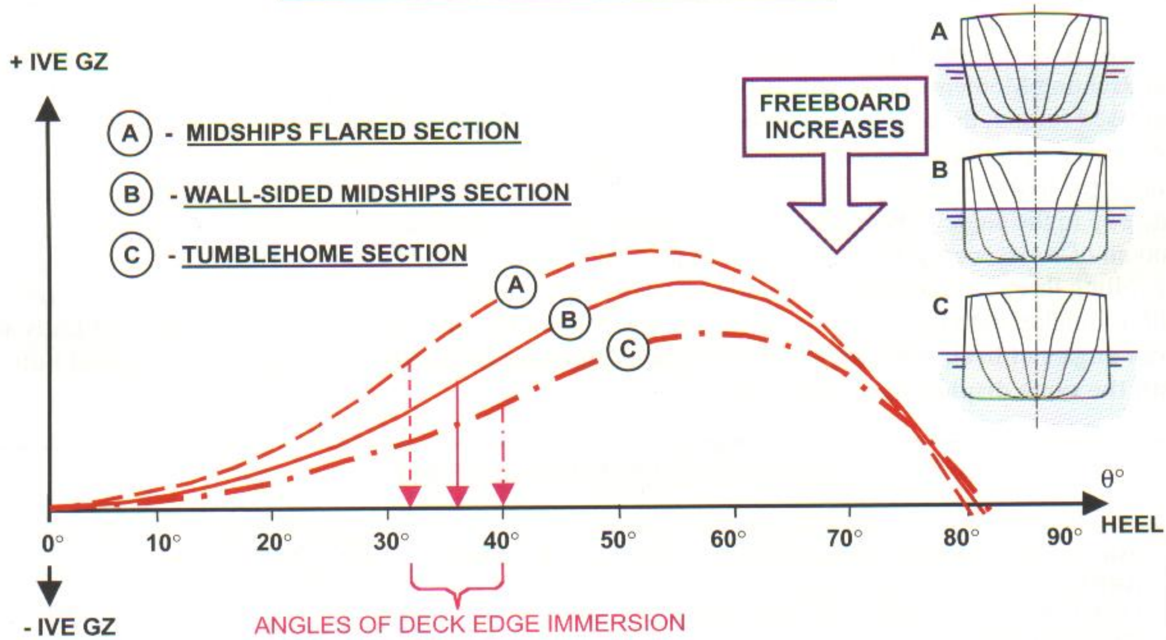
Most merchant ship hulls have a rectangular midships section but some vessels, notably some roll on - roll off ferry designs, incorporate flare along their entire length. There is also an occasional vessel still built with 'tumblehome' (See Chapter 1, page 3).

FLARED, WALL-SIDED AND TUMBLEHOMED MIDSHIPS SECTIONS



ALL THREE HULLS ARE OF EQUAL DISPLACEMENT, MIDSHIPS HULL DEPTH, GM 0 VALUE, LENGTH, AND UPRIGHT WATERLINE BEAM.

GZ CURVES OF THE ABOVE THREE HULLS



THE DRAFT DECREASES AS WE MOVE FROM THE FLARED MIDSHIP'S SECTION 'A' THROUGH TO THE TUMBLEHOME HULL 'C', SO THE FREEBOARD INCREASES FROM HULL 'A' TO HULL 'C'

HULL 'C' HAS THE GREATEST RANGE OF POSITIVE STABILITY DUE TO THE REDUCING BEAM WIDTH AND THE INCREASE IN FREEBOARD. THOUGH IT HAS THE LOWEST MAXIMUM GZ VALUE

MIDSHIPS FLARE AND TUMBLEHOME (Cont.)

When a normal hull with a wall-sided midships section is initially heeled from the upright, the waterline beam increases whilst the flare at the fore and aft ends produces an asymmetry in the waterplane. The righting lever 'GZ' is enhanced by both these features as the angle of heel increases to the point of deck edge immersion. If the flare is continued along the entire length of the hull, the GZ of the vessel will increase more quickly with heel so the 'stiffness' and accelerations experienced at the ends of the roll.will also be increased. However, the deck edge will be immersed at a lower angle of heel.

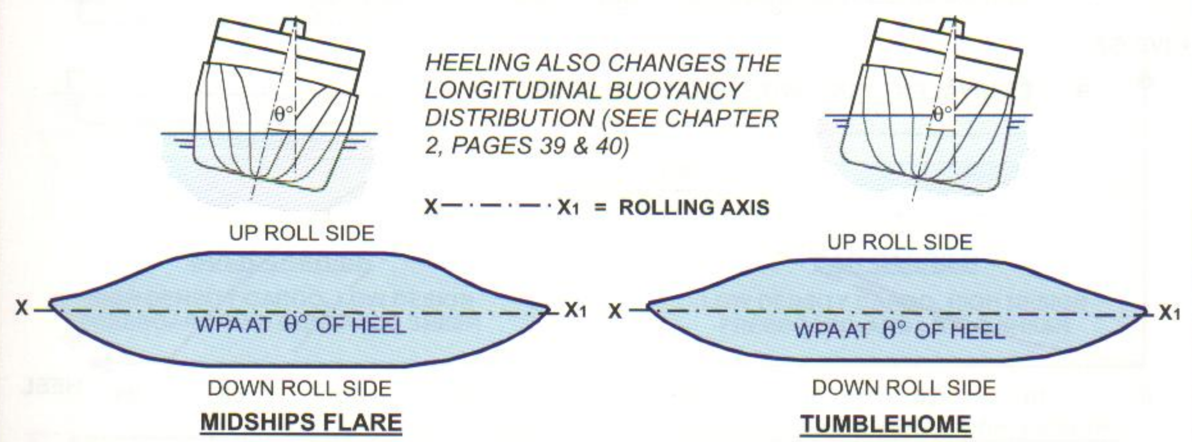
Ro-ro ferries that feature full hull length flare usually have a 'hard chine' hull, i.e. the flare in the midships region extends up the hull to just above the normal waterline where it turns a corner and disappears into the vertical sides of the upper hull. This reduces its effects upon the GZ curve when the vessel is fully loaded. The main advantage of such a design is to produce a hull with wide upper vehicle decks whilst retaining a relatively narrow waterline beam, which will improve the ship's speed through the water. The effect of this flare upon the ship's stability is probably unintentional and not particularly significant *except when the vessel is being operated at relatively light drafts.* In these circumstances, the chine will be well above the waterline and the full stiffening effect of the flare will occur up to quite large angles of heel. The resulting motion, particularly at the ends of a roll, is likely to be quick and quite violent if the ship also has a relatively generous upright GM value.

Tumblehome will have the opposite effect to midships flare as it reduces the increase in width and asymmetry of the heeled waterplane. It is usually considered to be a old fashioned hull feature that is rarely seen nowadays. However, in 1997, the 8,000 ton cableship 'Cable Retriever' was built with tumblehome despite the same yard producing an earlier near identical sister ship with a wall-sided midships section.

Cableships are not 'deadweight' carriers as the weight of cable, fuel, water and stores loaded onboard such vessels is often only about 40% of the ship's displacement so there may only be a limited range of normal operating KG values. Modifying a design by incorporating tumblehome can be used to reduce the violence of the rolling motion if the design tends to produce a ship that is too 'stiff' (i.e. it has excessive stability) in its normal loaded condition. This is particularly so if the design is also to include fine lines with generous flare at the bow and stern in order to retain a fast hull with good seakeeping capabilities. (See Chapter 7 for more information on the rolling behaviour of a ship.)

Both tumblehome and midships flare can cause problems in berthing a ship and keeping it securely alongside in ports that are exposed to any significant weather or swell. It is far easier to keep a ship alongside a jetty if it has a box-shaped midships section, particularly when the vessel is moving vertically up and down, either due to loading and discharging cargo or with the rise and fall of the tide.

THE EFFECT OF MIDSHIPS FLARE AND TUMBLEHOME ON THE HEELED WPA



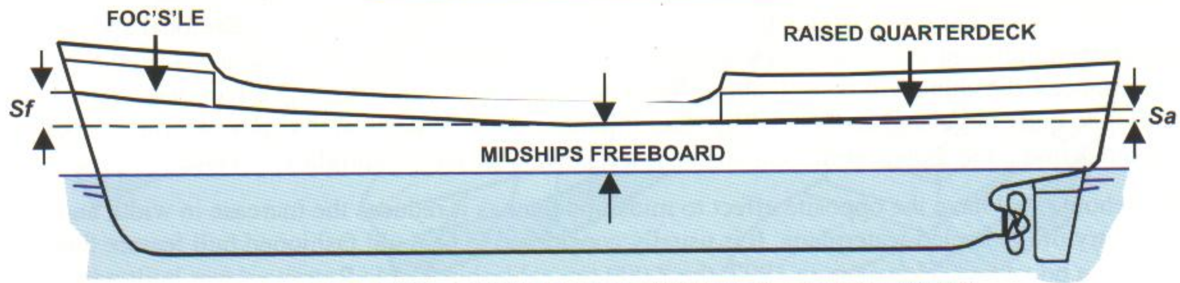
ENHANCED INCREASE IN WATERPLANE BEAM AND ASYMMETRY. HULL TENDS TO BE 'STIFF' THE FLARE CREATES EXCESS MIDSHIPS BUOYANCY WITH INCREASING HEEL

REDUCED INCREASE IN WATERPLANE BEAM AND ASYMMETRY. HULL TENDS TO BE 'TENDER' TUMBLEHOME CREATES A BUOYANCY DEFICIT MIDSHIPS WITH INCREASING HEEL.

EFFECT OF SHEER AND SUPERSTRUCTURE ON STABILITY

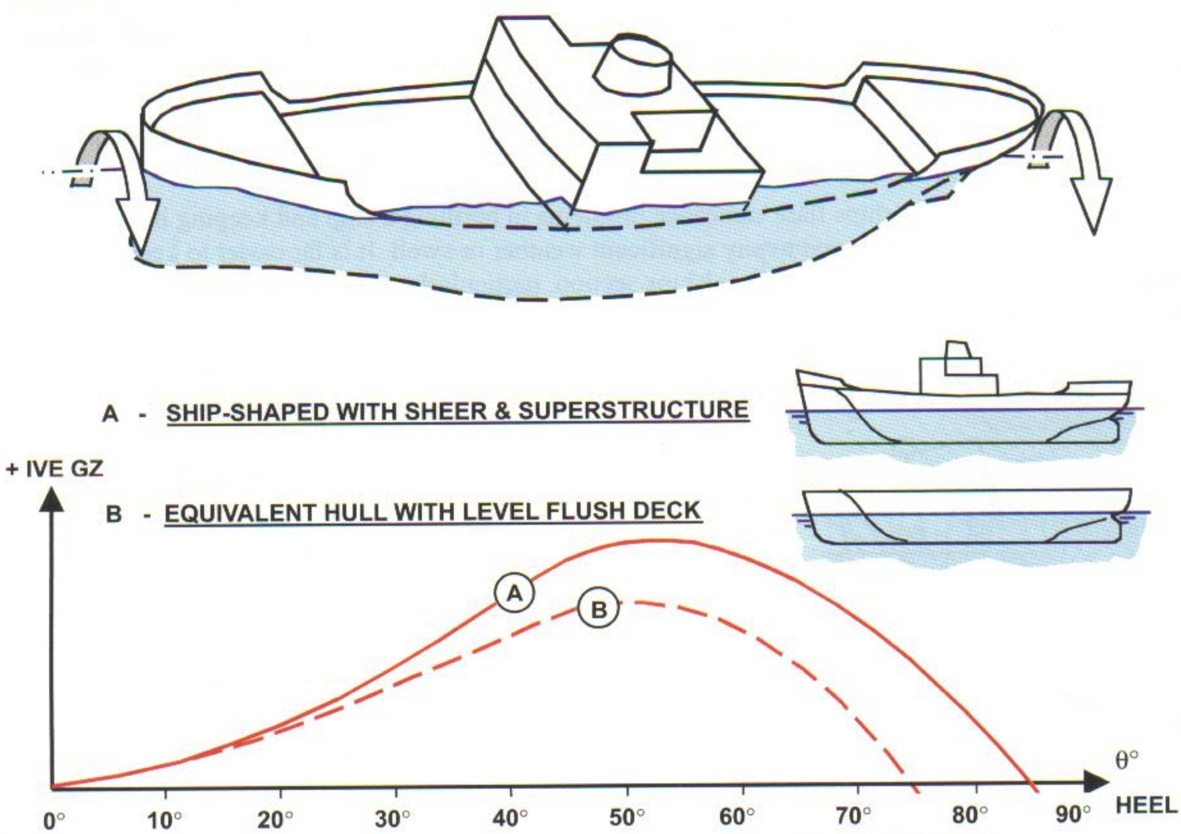
Sheer is the increase in freeboard at the fore and aft ends of the hull and, typically, the sheer at the bow is about twice that at the stern. Superstructures are watertight structures that usually extend over the full width of the hull. They are most commonly built into the bow as a forecastle and the stern as a poop or raised quarterdeck. In addition, some vessels have a midships superstructure, known as a centrecastle. Normal deckhousing is not considered watertight if it has doorways that are essential for access at sea and so is not accepted as superstructure. *(See Chapter 11 - 'The Load Line Regulations')* Both sheer and superstructures increase the reserve buoyancy at the fore and aft ends of the hull and so improve the hull's resistance to pitching and protection of the deck from shipping heavy seas, as well as enhancing the transverse stability characteristics.

SHEER AND SUPERSTRUCTURE



'Sf' AND *'Sa'* ARE THE FORWARD AND AFT SHEERS . RESPECTIVELY

THE EFFECT OF SHEER AND SUPERSTRUCTURE ON THE GZ CURVE



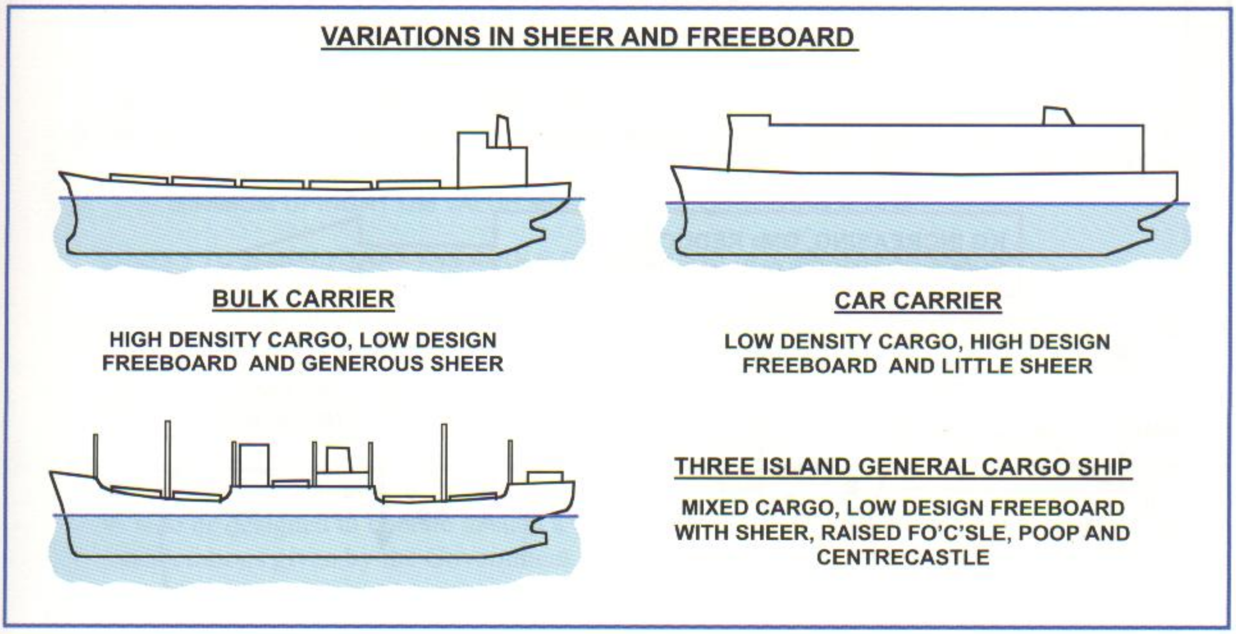
SHEER AND SUPERSTRUCTURE HAVE NO EFFECT UPON THE GZ CURVE AT SMALL ANGLES OF HEEL BUT, BY INCREASING RESERVE BUOYANCY, THEY IMPROVE THE RANGE OF A HULL'S POSITIVE STABILITY, FOR A GIVEN GM_0 VALUE.

HULLFORM VARIATIONS IN DIFFERENT VESSELS

Although most vessels share the same basic shape, there is considerable detailed variation in the hullforms of merchant ships. This is the result of ships being built to meet design criteria that cover a far wider scope than transverse stability characteristics. A shipowner will usually start by specifying the type (crude oil carrier, container ship, ro-ro ferry etc.), carrying capacity and speed of a new vessel but the requirements of particular trades impose restrictions on draft, beam and/or length. For example, many vessels are built to be the largest possible ship that can navigate the Panama Canal. Perishable and high value freight (such as fruit and manufactured goods) are generally carried in faster ships than bulk cargoes, so the hulls of such vessels tend to be fine lined, whilst the need for good hatch space, requires that the beam at maindeck level is as wide as possible along most of the ship's length. This will result in such hulls having considerable flare at the fore and aft ends.

These requirements influence hull features such as length to beam ratio, length to draft ratio, block coefficients and flare, which will be significant in the resulting stability characteristics. The variations of sheer and superstructure arrangements are a case in point. During a roll, the deck edge will be most vulnerable to shipping seas and immersed first at its widest part, (i.e. the midships region), so it seems curious that this should also be the region of lowest freeboard. The fore and aft ends require a freeboard sufficient to keep them clear of shipping water whilst pitching but why not maintain that freeboard for the entire length of the vessel and have no sheer? Some ships do have relatively little or no sheer but these are generally carrying low density freight which fill the cargo space before reaching the vessel's potential maximum deadweight capability. Car carriers are a good example of this type of design, which naturally have a high freeboard and so little requirement for sheer. Ships carrying high density freight, such as bulk cargoes, are in the opposite situation. The deadweight capacity is reached before the hold space is completely filled, so any additional underdeck midships volume would be unusable. It would however, increase the lightship weight, which would reduce the deadweight capacity and increase the building costs. It would also increase the registered tonnage values (these are based upon underdeck volume) which would tend to increase the port dues levied on the ship, so the operating costs would rise as well. The design freeboard of such vessels is generally quite small and sheer is essential to ensure good pitching characteristics.

Up to the 1970's, most bulk and manufactured goods were carried in the same type of general purpose cargo ship, typically of about 14,000T deadweight with five or six hatches. These ships were built to carry both high and low density goods and often incorporated a midships centrecastle, particularly if the trades they worked required additional segregated cargo spaces for small lock up special stows (usually of high value goods such as spirits or bullion). This produced the 'Three Island' design with both sheer and additional midships reserve buoyancy, though it has largely disappeared as ships have become more specialised and dry cargo handling is more mechanised through containerisation.

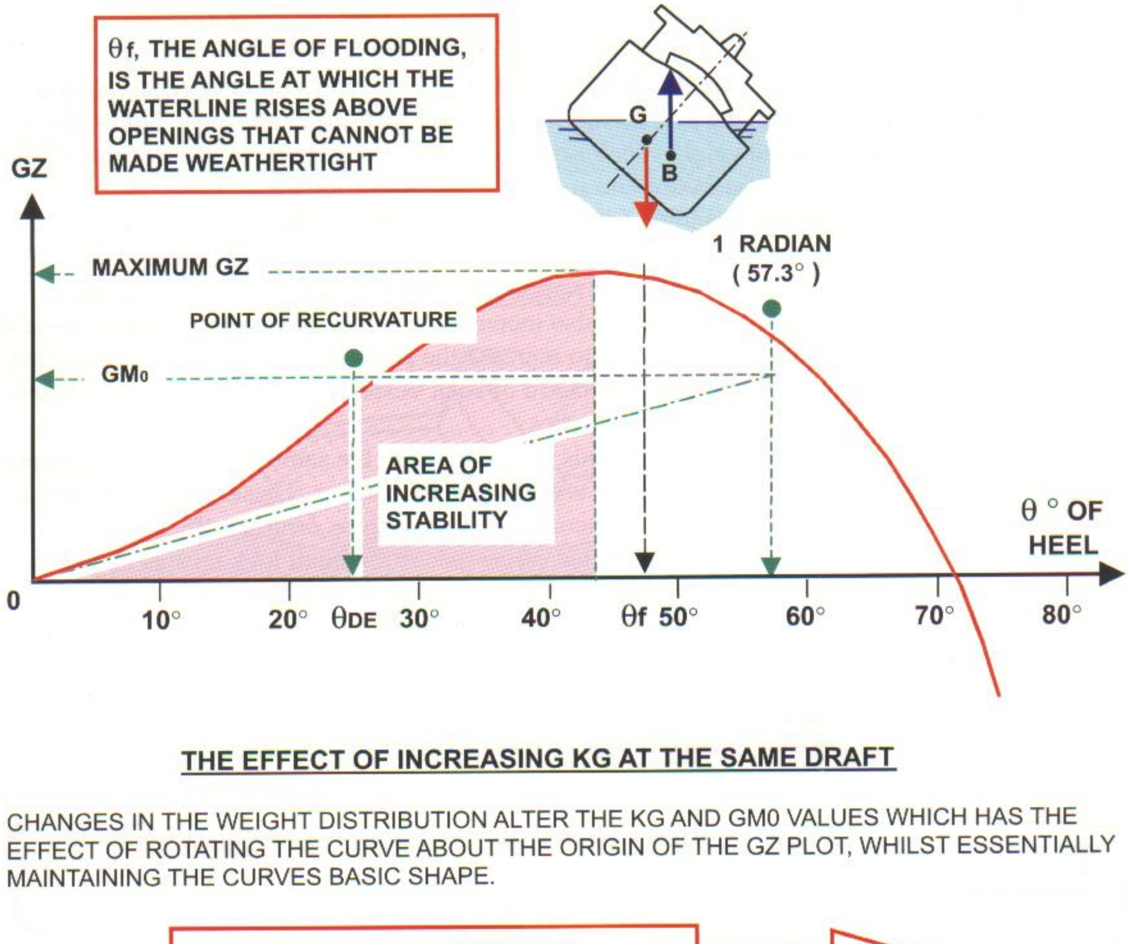


THE GZ CURVE SUMMARISED

The basic shape of a Righting Lever, or GZ, curve is obtained by correcting KN values for the KG value of the ship's loaded condition. The KN values are determined by the ship's hullform, draft, and trim. Certain features, such as the angle of deck edge immersion, the angle of maximum GZ value and the upright GM, can be identified by this shape. The area under the curve up to any angle of heel indicates the hull's resistance to be rolled over from the upright to that angle..

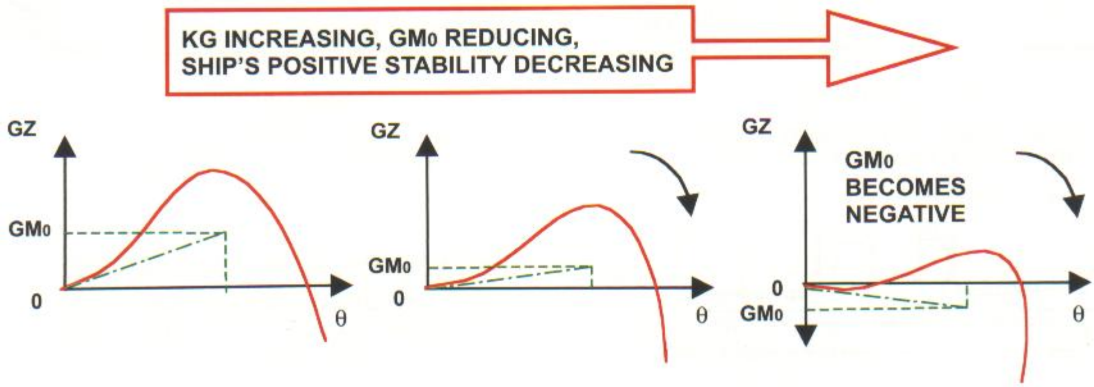
The Angle of Flooding, however, is not identifiable from the curve because the KN values are based upon the intact immersed hullform. This must be indicated separately over the range of the ship's operating drafts. Good design practice ensures that such vulnerable openings, such as hold ventilators and machinery air intakes, are mounted high enough or sufficiently inboard to mean that the Flooding Angle is well beyond the point of maximum GZ under normal operating circumstances.

A TYPICAL GZ CURVE



THE EFFECT OF INCREASING KG AT THE SAME DRAFT

CHANGES IN THE WEIGHT DISTRIBUTION ALTER THE KG AND GM0 VALUES WHICH HAS THE EFFECT OF ROTATING THE CURVE ABOUT THE ORIGIN OF THE GZ PLOT, WHILST ESSENTIALLY MAINTAINING THE CURVES BASIC SHAPE.

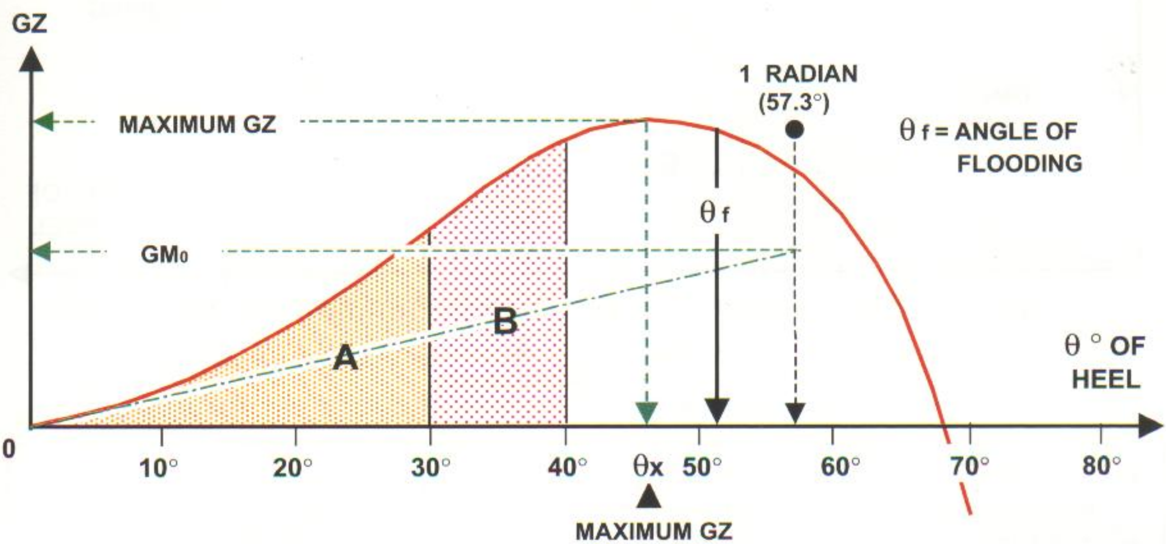


THE MINIMUM STABILITY CRITERIA FOR SEAWORTHINESS

The design team and builders fix the basic shape of a ship's GZ curves by deciding on such factors as block coefficient, beam, sheer etc, but the ships' officers determine the KG value by how they load the ship. To ensure that the ship is seaworthy by possessing adequate stability, the various maritime authorities have laid down guidelines in the form of minimum intact stability criteria. This book will primarily refer to those of the **Marine and Coastguard Agency of the U.K. (MCA,-U.K.), as given in the U.K. Stationary Office publication 'Load Line,- Instructions for the Guidance of Surveyors'.** The text will also refer to **'The International Maritime Organisation' (I.M.O.) as given in their Code on Intact Stability,** which is similar but covers some different types of ship.

There are six minimum transverse stability criteria of seaworthiness, that the MCA-U.K require a normal ship's GZ curve must meet at all times that the vessel is at sea.

MCA,-U.K.MINIMUM CRITERIA OF INTACT STABILITY



1) THE AREA, **A** , UNDER THE GZ CURVE, **0° TO 30° MUST NOT BE LESS THAN 0.055 METRE-RADIANS** (WHERE 1 METRE-RADIAN = 57.3 METRE-DEGREES)
2) THE AREA, **A + B** , UNDER THE CURVE, **0° TO 40°, OR TO θf**, WHICHEVER IS THE SMALLER, **MUST NOT BE LESS THAN 0.090 METRE-RADIANS**
3) THE AREA ' **B'** UNDER THE CURVE **30° TO 40°, OR TO θf**, WHICHEVER IS THE SMALLER, **MUST NOT BE LESS THAN 0.030 METRE-RADIANS.**
4) THE ANGLE OF HEEL, **θx,** FOR THE MAXIMUM RIGHTING LEVER GZ, MUST BE AT LEAST **25° AND PREFERABLY IN EXCESS OF 30°**
5) θx MUST **NOT BE LESS THAN 30°** AND THE **MAXIMUM GZ VALUE MUST NOT BE LESS THAN 0.2M**
6) THE **MINIMUM UPRIGHT GM** VALUE MUST NOT BE LESS THAN **0.15 M**

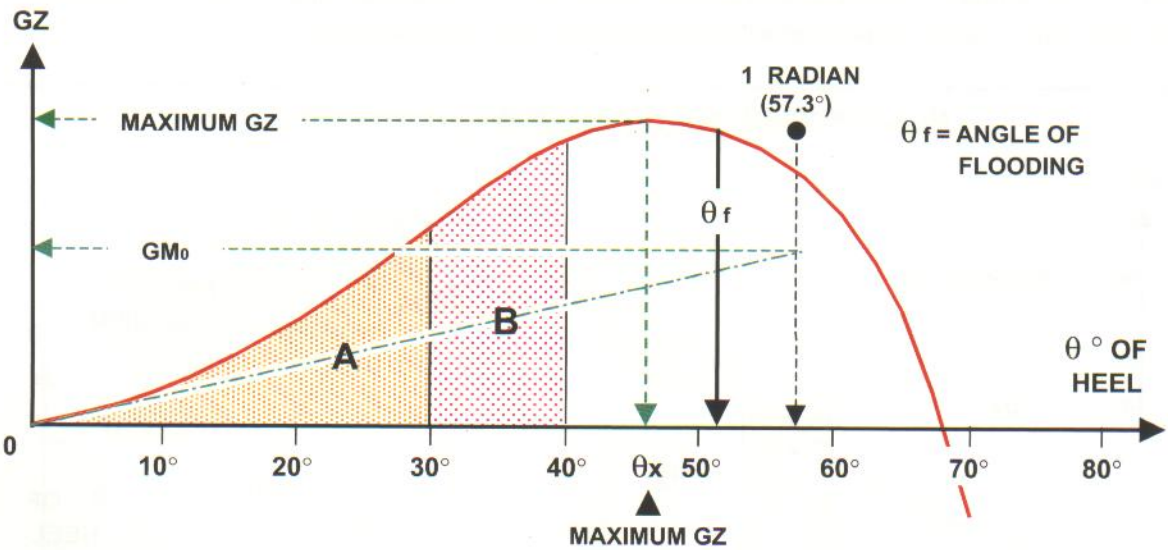
RULES 1 TO 3, IMPLY THAT THE ANGLE OF FLOODING, θ f, MUST EXCEED 30°

THE MCA ALSO REQUIRE THAT EVERY VESSEL SHOULD MAINTAIN A MINIMUM BOW HEIGHT, WHICH IS TO BE AGREED UPON AFTER CONSIDERING THE SHIP'S SIZE, AREA OF OPERATION AND GENERAL LAYOUT. THIS WILL LIMIT THE DEGREE OF TRIM BY THE HEAD WHICH IS ACCEPTABLE FOR DIFFERENT DRAFTS

If a vessel is to be designed to operate in such a way as to not comply with these criteria or any of the special considerations given by the Government, then the operators should seek advice from the MCA directly.

The I.M.O. Code also requires a normal ship's GZ curve to meet six minimum stability criteria at all times that the vessel is at sea.
These are almost the same as those of the MCA-U.K. but are more flexible by allowing ships to have the maximum GZ value occurring at an angle of heel as low as 25°. If, however, this angle is less than 30°, then having a maximum GZ greater than 0.2 metres must compensate for the reduction of positive stability range.

I.M.O. MINIMUM CRITERIA OF INTACT STABILITY



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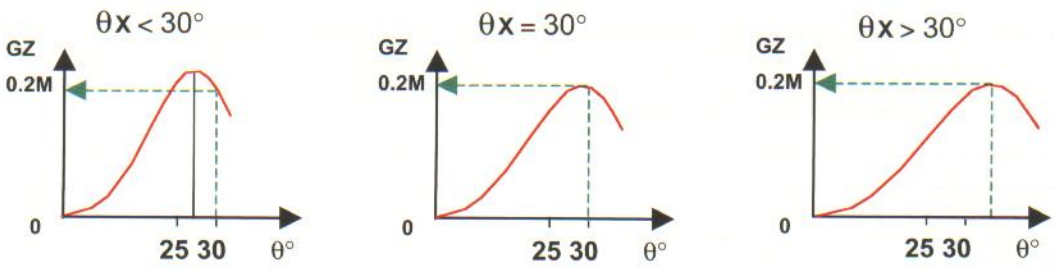
2) THE AREA, **'A + B'** UNDER THE CURVE, **0° TO 40°, OR TO** θf, WHICHEVER IS THE SMALLER, **MUST NOT BE LESS THAN 0.090 METRE-RADIANS.**
3) THE AREA, **'B'** UNDER THE CURVE **30° TO 40°, OR TO θf,** WHICHEVER IS THE SMALLER, **MUST NOT BE LESS THAN 0.030 METRE-RADIANS.**
4) THE ANGLE OF HEEL, **θx,** FOR THE MAXIMUM RIGHTING LEVER GZ, **MUST BE AT LEAST 25° AND PREFERABLY IN EXCESS OF 30°**
5) IF θ**x** IS LESS THAN 30°, THE GZ VALUE AT 30° MUST NOT BE LESS THAN 0.2 M. IF θ**x** IS EQUAL TO OR GREATER THAN 30°, **THE MAXIMUM GZ VALUE MUST NOT BE LESS THAN 0.2M**
6) THE MINIMUM UPRIGHT GM VALUE **MUST NOT BE LESS THAN 0.15 M**

RULES 1 TO 3 IMPLY THAT THE ANGLE OF FLOODING, θ f, MUST EXCEED 30°

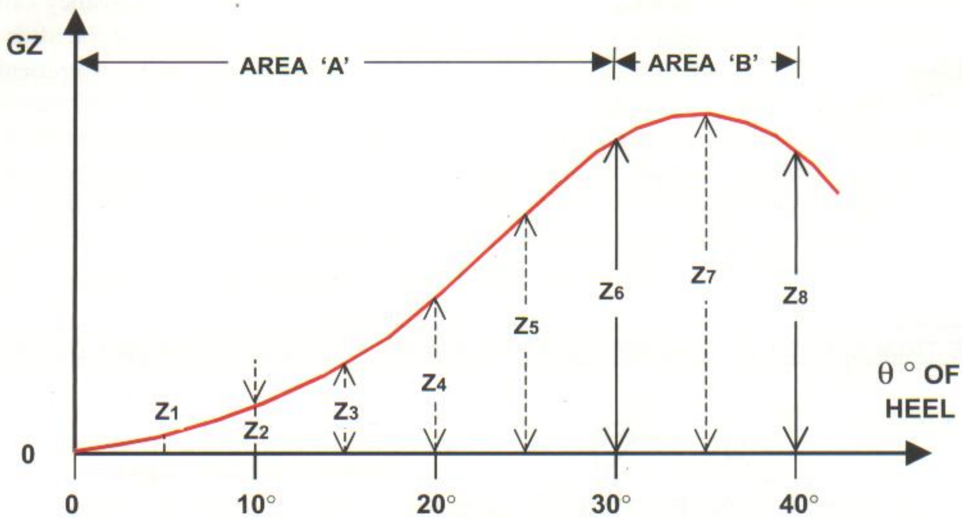
EXAMPLES OF MINIMUM COMPLIANCE WITH I.M.O. RULE 5



CHECKING FOR COMPLIANCE WITH THE MINIMUM STABILITY CRITERIA

The appropriate areas under a GZ curve can be estimated by using one of the approximate integration methods, such as Simpson's Rules or the Trapezium method.

AN EXAMPLE METHOD FOR ESTIMATING THE AREA UNDER THE GZ CURVE



BY THE TRAPEZIUM METHOD OF APPROXIMATE INTERGRATION;-

AREA 'A' = $\frac{5}{57.3}\left(Z_1 + Z_2 + Z_3 + Z_4 + Z_5 + \frac{1}{2}Z_6\right)$, AREA 'B' = $\frac{5}{57.3}\left(\frac{1}{2}Z_6 + Z_7 + \frac{1}{2}Z_8\right)$

Z_1 TO Z_8 ARE THE GZ VALUES AT 5° INTERVALS, AREAS ARE IN METRE-RADIANS

The illustrated example uses values of GZ measured from the plotted curve at 5° intervals. Simpson's Rules approximate the curve to a parabola between measurements rather than the straight line assumption above, so it is generally acceptable to use fewer co-ordinates when using Simpson's Rules. However, both methods are equally valid

The I.M.O. criteria are really concerned with the first 40° of heel, though it is not unusual for loaded states of a ship to produce GZ curves with positive intact stability up to 80° or more. In reality, however, flooding and shifts in the weight distribution will have occurred at a far lesser angle of heel, so, beyond about 50° of heel, a ship is in serious trouble and the curve is no longer particularly relevant. The purpose of the I.M.O. Code is to prevent the vessel ever reaching this state of affairs

It should be appreciated that the Code gives **absolutely minimum** criteria to be met at all stages of a voyage and a ship may well need an upright GM greater than 0.15 metres in order to satisfy the other requirements, such as minimum areas under the curve. This would be particularly so if a vessel is loaded to its minimum freeboard.

The required minimum upright GM value for adequate stability is considered to be independent of the size of ship. The resulting righting moment, which actually works against the forces of the sea, includes the ship's displaced weight and so will be much greater in a large ship than a smaller one.

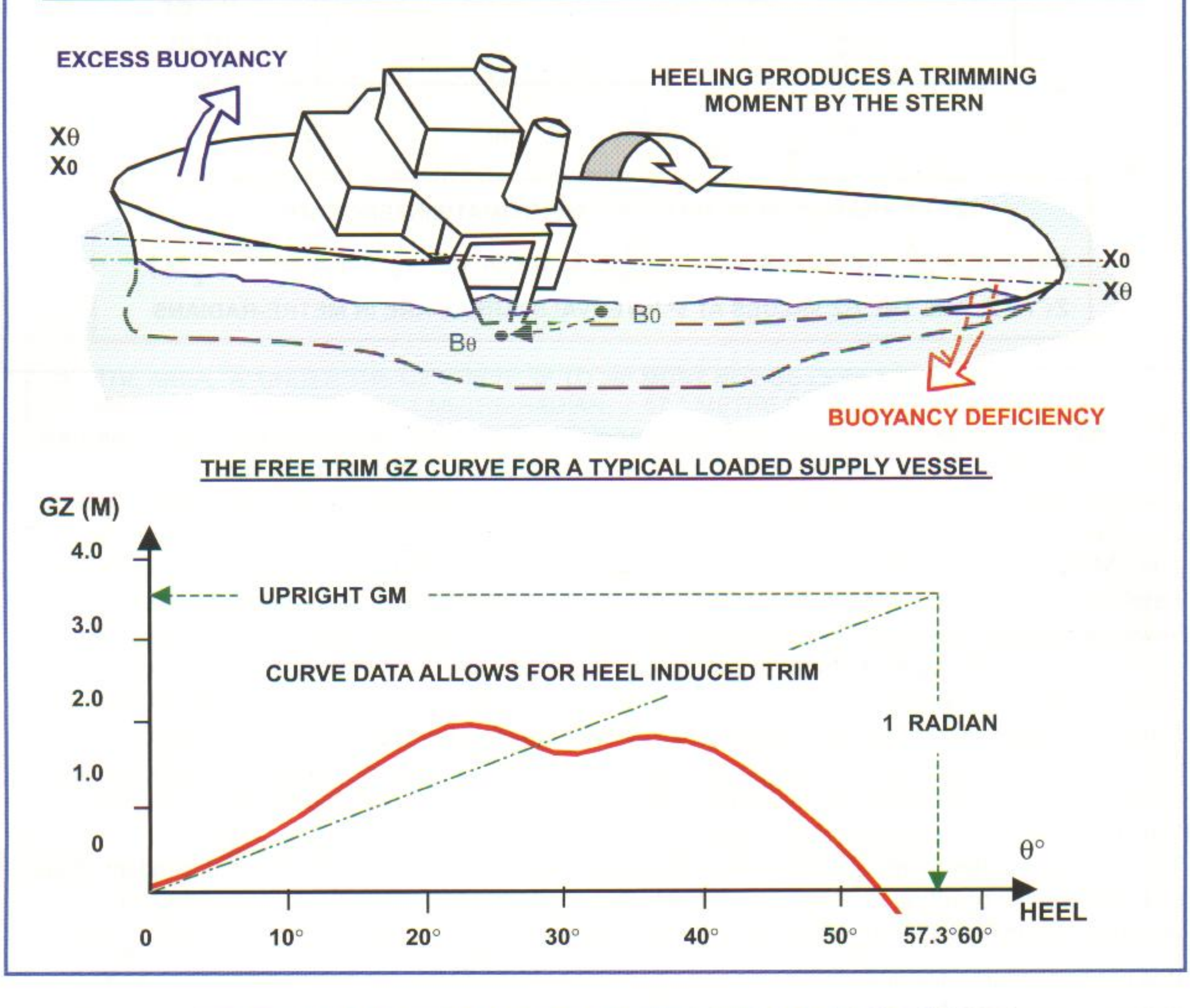
It is not particularly desirable to have an excessive GM either, as this tends to produce a violent rolling motion. A positive GM_0 of between 0.5 and 1 metre, is ideal for good stability in many typical sea-going vessels. GM_0 values much greater than 1 metre tend to produce a 'stiff' hull, characterised by an overquick roll, with high angular accelerations which tend to overstress the ship's structure. At GM_0 values much less than 0.3 metres, most ships will become too 'tender' with an excessively slow and reluctant roll.

Chapter 4 explains the practical calculations and procedures followed by the ship's officers, to ensure that the weight distribution in the vessel will provide acceptable stability.

STABILITY AND TRIM OF HIGH FO'C'SLE VESSELS

When most commercial ships heel over, the immersion of reserve buoyancy is about equally distributed between the fore and aft ends as there is sheer at the bow and stern. However, there are also many vessels nowadays operating in the offshore industries that are built with high fo'c'sles and a low working after deck. When such a ship is heeled over to immerse the after deckline, the fo'c'sle remains well above the waterline, consequently, there is considerably more reserve buoyancy at the bow than the stern. Further heeling will result in a significant forward shift of the Centre of Buoyancy causing a stern trimming moment which will submerge the stern further and leads to a danger of the after deck being flooded. Stability data for such vessels must allow for this change of trim and, consequently, GZ curves supplied by the shipbuilder, are said to be for the 'Free trim situation'. (i.e. The righting lever is calculated from the GM value which is based upon the vessel freely changing trim as it heels over). Oilrig supply ships are typical of the type of vessel that will suffer a significant trimming moment by the stern when heeled over beyond a certain point. The resulting loss of stability beyond this angle of heel is partly compensated for by the relatively large beam that such ships tend to be built with, which is an operational requirement to provide the ship with a large aft working deck for cargo stowage.

THE FREE TRIM EFFECT ON THE GZ CURVE FOR A HIGH FO'C'SLE/LOW AFT DECK SHIP



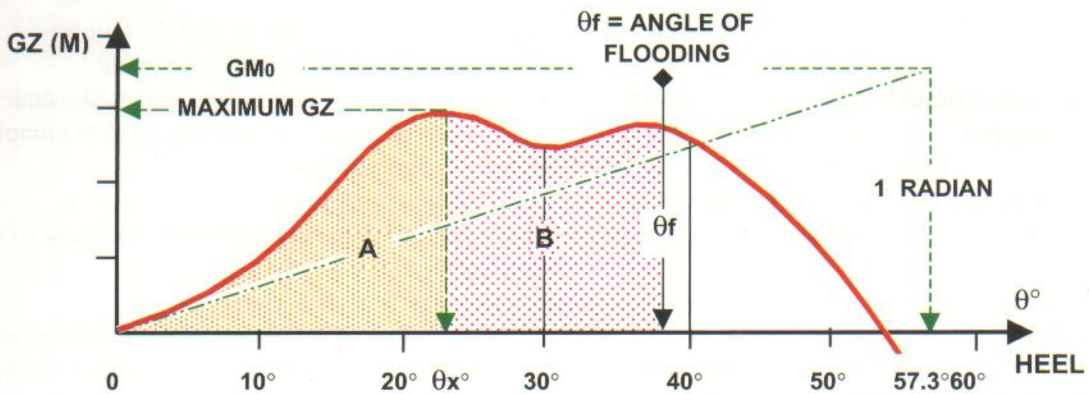
FEATURES OF AN OFSHORE SUPPORT VESSEL'S GZ CURVE

1) The vessel posses a large upright GM_0 value due to a generous beam to length ratio
2) There are two points of maximum GZ value due to the separate immersions of the aft deckline and the fo'c'sle deck.
3) The heel angles of maximum and vanishing stability are relatively low due to the free trim effect.

ALTERNATIVE STABILITY CRITERIA FOR RIG SUPPLY SHIPS

The necessity for rig supply ships to be built with a long low after deck in order to meet the needs peculiar to their trade, has led the authorities to modify the minimum stability criteria which must be satisfied for any condition of loading to be considered seaworthy.

I.M.O. AND MCA-U.K. ALTERNATIVE MINIMUM CRITERIA OF INTACT STABILITY



1) THE ANGLE OF HEEL OF MAXIMUM GZ, 'θx', MUST NOT BE LESS THAN 15°
2) AREA 'A', UNDER THE CURVE, 0° TO θx° ≥ 0.055 + 0.001 (30° - θx°) M- RADS *WHERE '(30 - θx)' IS TAKEN TO BE ZERO IF θx IS GREATER THAN 30°*
3) AREA 'B', UNDER CURVE, IS θx° (OR 30°, IF θx° >30°) TO θf° (OR 40°, IF θf° >40°) WHERE AREA 'B' ≥ 0.03 M- RADS
4) IF θx IS LESS THAN 30°, THE GZ VALUE AT 30° MUST NOT BE LESS THAN 0.2 M. IF θx IS EQUAL TO OR GREATER THAN 30°, THE MAXIMUM GZ VALUE MUST NOT BE LESS THAN 0.2M
5) THE MINIMUM UPRIGHT GM VALUE MUST NOT BE LESS THAN 0.15M
6) THE STERN FREEBOARD MUST NOT BE LESS THAN OF 0.5% OF THE VESSEL'S LENGTH.

A rig supply vessel is allowed a smaller range of positive dynamic stability, providing that the vessel is stiffer at low angles of heel than would be normal for a cargo ship of similar size. This is achieved by giving the vessel a generous maximum beam and maintaining this beam at deck level almost to the stern. In addition to the above stability criteria, the authorities also make the following requirements:-

1) Deck cargo Kg values used in stability calculations, must be reasonable estimates of actual heights of the Centres of Gravity of such stows and not simply the deck height.
2) Allowance must be made for the weight of water that can be trapped within any deck stow. Deck cargoes of pipe sections are particularly likely to trap water.
3) Deck cargo should be unloaded, preferably before discharging hull cargo tanks.
4) No openings, such as funnel intakes etc. should be in positions vulnerable to flooding due to the free trim effect.

An important consequence of meeting these requirements is that such ships usually have to operate with a trim by the head and this must be allowed for in the vessel's design.

In the early years of the North Sea oilfield operations, supply boat designs were often based upon the vessels used in the Gulf of Mexico. It was common for the funnels and engine room casings to be built on the outboard sides of the after deck, quite close to the stern. There were several incidents of such vessels being lost due to flooding of the engine room due to heavy seas sweeping up the after deck when the ship was rolling. Modern designs, however, are a great improvement and nowadays supply boats are much larger with funnels, vent intakes etc. mounted high up and well forward of the stern.

ADDITIONAL AND ALTERNATIVE STABILITY REQUIREMENTS

Both the MCA-U.K. and the I.M.O. Codes lay down further intact stability requirements, which certain categories of ship must meet, in addition to the six minimum stability criteria, explained in the previous pages.

There are extra requirements for:-

1) Passenger vessels (i.e.any ship which carries more than twelve passengers).
2) Ships carrying deck cargoes of timber and loaded to the Lumber Marks.
3) High-sided ships subjected to significant wind forces. The high sides can be due the ship's construction (e.g. car carriers) or a substantial deck cargo, such as timber or containers.
4) Ships transporting bulk cargoes, which are liable to shift in heavy weather (i.e. grain). The code does not specify the specific criteria that these vessels must meet but refers to the need to comply with the 1974 International Code for the Safe Carriage of Grain in Bulk.
5) Ships carrying out heavy lift operations at sea.
6) Ships operating in conditions where there is a danger of build up of ice on the exposed parts of the vessel's structure.

Chapter 5 will discuss these additional criteria, where they are required by conventional displacement ships, but there are some types of marine craft, which are totally different from the conventional single displacement hull. The IMO and the MCA-U.K give guidance for the following types of vessel:-

1) Offshore mobile drilling units (MODU's), as used primarily in oil exploration. These include both the semi-submersible and 'jack-up' types of drilling rigs
2) Seagoing Pontoons
3) Dynamically Supported Craft (DSC's), such as fast hydrofoil ferries.

There are also accepted stability requirements for ships in the damaged condition as a result of collision or going aground. These are stated in the **'Safety of Life at Sea' (SOLAS) 1974 Convention** and will be considered in the chapter concerned with bilging and subdivision.

COMPLIANCE WITH THE STABILITY CODE

The legal standing of these Codes varies from one nation to another and, in practice, it is not particularly convenient for Port State or Flag State authorities to check on a ship's compliance with them. U.K. registered ships must comply with the MCA's requirements and the carrying onboard of the necessary stability information in the **'Approved Stability Book',** is part of the Loadline regulations. (*See Chapter 11, page 267*) However, compliance with the stability code relies upon the accuracy of weight distribution estimations, used in the KG calculations and this, in turn, depends upon the judgement of the ship's officers who are responsible for loading of the vessel.
If it was found that inadequate stability caused the loss of a ship, the Master and the officers concerned with the loading, would be considered negligent in their duties and held, at least in part, responsible for the accident. More to the point though, is that they may also be dead as a result of the ship sinking. The whole purpose of the Codes is to provide both ship designers and operators with guidance as to how to prevent such situations arising. The actual figures in the minimum criteria requirements are derived empirically, that is to say they are obtained from measurements made on both real ships and model tests conducted over a long period. They apply to the complete diverse range of commercial vessels and, as such, may not be entirely adequate for a particular ship in a particular situation.

IT SHOULD ALWAYS BE APPRECIATED THAT THE MINIMUM STABILTY CRITERIA ARE **ONLY MINIMUM REQUIREMENTS** AND SIMPLY MEETING THEM DOES NOT TAKE AWAY THE SHIPS OFFICERS BASIC RESPONSIBILITY IN UNDERSTANDING THE BEHAVIOUR OF HIS OWN VESSEL. A GOOD DECK OFFICER SHOULD ALWAYS BE AWARE OF HIS VESSELS STATE OF STABILITY AND ADJUST IT ACCORDING TO THE CIRCUMSTANCES THAT THE SHIP IS IN AT THE TIME. IN DOING SO, HE SHOULD ALSO REMEMBER AND ALLOW FOR THE DEGREE OF ERROR INVOLVED IN THE CALCULATED KG VALUE, WHICH IS BASED UPON **ONLY ESTIMATIONS** OF WEIGHT DISTRIBUTION WITHIN THE SHIP.

CHAPTER 4

OPERATIONAL TRANSVERSE STABILITY

SUMMARY

THIS CHAPTER DESCRIBES HOW THE SHIP'S LIGHTWEIGHT HEIGHT OF CENTRE OF GRAVITY IS ESTABLISHED AND OUTLINES THE PROCEDURES FOR PRACTICAL TRANSVERSE STABILITY PROBLEMS

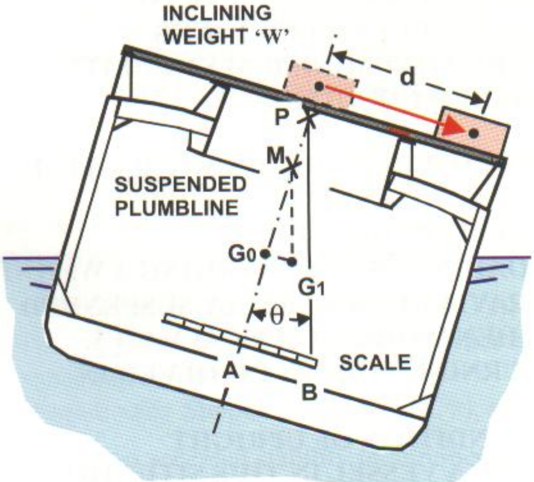
1) **THE INCLINING EXPERIMENT FOR MEASURING A SHIP'S LIGHTWEIGHT KG.**
2) **PROCEDURE FOR CALCULATING THE LOADED KG VALUE OF A VESSEL.**
3) **LIQUID FREE SURFACE EFFECTS DUE TO PARTIALLY FILLED TANKS.**
4) **USING SHIP'S HYDROSTATIC DATA TO ENSURE THAT MINIMUM STABILITY CRITERIA ARE MET FOR A PARTICULAR LOADED CONDITION**
5) **TRANSVERSE STABILITY OF TANKERS**
6) **CALCULATING THE ANGLE OF LIST PRODUCED WHEN THE 'C OF G' IS OFFSET FROM THE CENTRELINE OF THE VESSEL.**
7) **THE EFFECT OF A LIST UPON THE SHIP'S MIDSHIPS DRAFT.**
8) **CALCULATING THE SHIFT IN 'G' DUE TO LOADING OR DISCHARGING A WEIGHT**
9) **THE EFFECTIVE HEIGHT OF CENTRE OF GRAVITY FOR A FREELY SUSPENDED LOAD AND STABILITY CONSIDERATIONS WHEN WORKING A HEAVY LIFT.**
10) **THE ANGLE OF HEEL PRODUCED BY THE TURNING FORCES INVOLVED IN ALTERING COURSE**
11) **THE ANGLE OF LOLL RESULTING FROM A CONDITION OF UPRIGHT INSTABILITY AND ACTION TO TAKE ONBOARD A VESSEL IN THIS SITUATION**

CONTENTS

DETERMINING THE LIGHTSHIP GM VALUE BY THE INCLINING EXPERIMENT

Calculating the KG of a ship for any given distribution of weight is only possible if the Lightship KG value is known (i.e. the height of the C of G for the ship's structural weight). Naval architects will have estimated this early in the vessel's design but it must be determined by actual measurement before the ship is put into service. This is done by the **'Inclining Experiment'**, which measures the angle of heel produced by shifting a known weight through a measured athwartships distance (usually from the centreline to the deck edge), whilst the vessel in lightship conditions, is moored in smooth water and no wind. The vessel's displacement and KM values can be obtained from the hydrostatic hull particulars for the upright draft of the experiment, so the GM and KG values can be calculated.

THE INCLINING EXPERIMENT



VESSEL'S DISPLACEMENT ΔT INCLUDES THE WEIGHTS OF THE INCLINING EQUIPMENT AND MOVEABLE WEIGHT 'W'

WHEN THE INCLINING WEIGHT w IS MOVED THROUGH DISTANCE d, THE C OF G MOVES FROM G_0 TO G_1, CAUSING THE LIST θ°, WHICH IS MEASURED ON THE PLUMBLINE SCALE

Now $$G_0G_1 = \frac{W \times d}{\Delta T}$$

And $$G_0G_1 = GM \tan\theta^\circ$$

So $$GM = \frac{W \times d}{\Delta T \times \tan\theta^\circ}$$

But also $$\tan\theta^\circ = \frac{AB}{AP}$$

So $$GM = \frac{W \times d}{\Delta T} \times \frac{AP}{AB}$$

$$KG = KM - GM, \quad \text{So} \quad KG_{EXP} = KM - \frac{\text{INCLINING WEIGHT} \times \text{DISTANCE MOVED}}{\text{DISPACEMENT WEIGHT}} \times \frac{AP}{AB}$$

The angle of heel is usually measured by a plumb line, which, for accuracy, needs to be as long as possible so the weight is moved across an open hold or cargo tank space.

The angle of heel should be within about 4° to allow the small angle stability equations to be used (i.e. the Metacentre is assumed to remain stationary over the range of heel angle).

For convenience, the vessel is usually tied up alongside a jetty but conditions need to be as calm as possible to avoid windage and any weight coming on the mooring lines. Large floating fenders are placed between the ship and the quay so the ship is free to heel over without fouling the quayside

The procedure should be repeated several times and measurements recorded with the ship heeled over both to port and to starboard so that any wind effect can be averaged out.

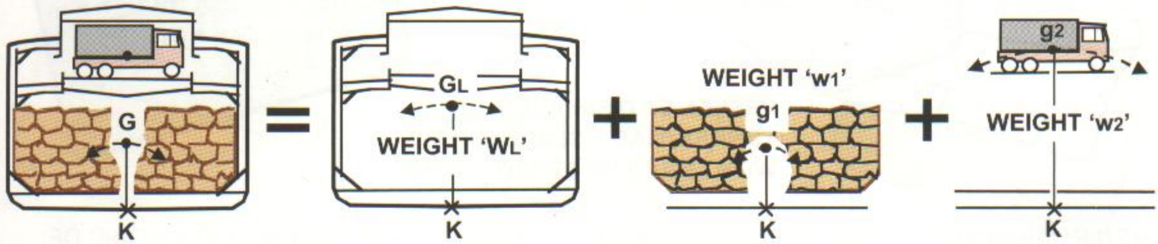
The ship should ideally be empty of any weights, additional to its structure, but this is not always possible as some ships are excessively trimmed or badly stressed in the lightship condition. The results of the experiment, including a precise record of any additional portable weights and their positions, must be kept with the ship's hydrostatic data in the stability book. Both the I.M.O. 'Code on Intact Stability' and the U.K. Authority's 'Instructions for the Guidance of Surveyors' describe the required procedures in detail.

Ships, particularly offshore support vessels, often tend to accumulate weight over their operational life as they are modified to meet new requirements or converted to carry out work completely different from their original designed purpose. Such changes should require the inclining experiment to be repeated periodically to re-determine the lightship KG.

CALCULATING THE LOADED KG VALUE OF A SHIP

The KG value for a ship and its load is calculated by summing the moments about the keel of each individual weight (including the lightship moment), and then dividing the total moment by the ship's displacement. This is demonstrated in the following diagram.

ADDING MOMENTS OF SEPARATE WEIGHTS ABOUT THE KEEL TO DETERMINE THE KG



LOADED VESSEL		LIGHTSHIP		LOWER HOLD STOW		T/D'K STOW
$\Delta_T \times KG$	=	$W_L \times KG_L$	+	$w_1 \times Kg_1$	+	$w_2 \times Kg_2$
DISPLACEMENT Δ_T	=	W_L	+	w_1	+	w_2

ITEM	WEIGHT (T)	HEIGHT ABOVE KEEL (M)	MOMENT (T-M)
LIGHTSHIP	W_L	KG_L	$W_L \times KG_L$
LOWER HOLD STOW	$+ w_1$	Kg_1	$+w_1 \times Kg_1$
TWEEN DECK STOW	$+ w_2$	Kg_2	$+w_2 \times Kg_2$
LOADED CONDITION	$\boxed{\Delta_T}$	KG	$\boxed{\Delta_T \times KG}$

TABLE OF WEIGHTS MUST INCLUDE ALL THE WEIGHTS OF ALL FUEL AND WATER IN TANKS

$$\text{LOADED KG} = \frac{\text{SUM OF THE MOMENTS ABOUT 'K'}}{\text{TOTAL DISPLACEMENT}} \text{ METRES}$$

FLUID AND SOLID VALUES OF KG AND THE GM

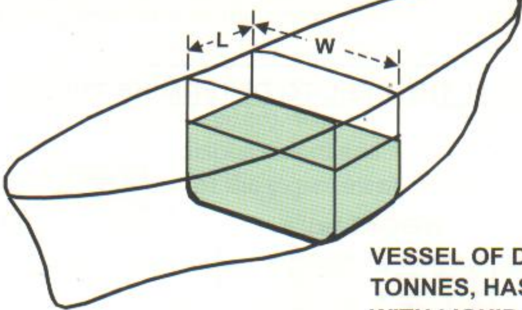
The calculation shown above produces a value known as the **solid KG.** This term is perhaps slightly misleading, as it will include the weights and vertical moments of all the fuel, water and liquid ballast onboard the vessel. The solid KG, however, is based upon the assumption that all the weights remain in a fixed position, even as the ship is rolling. If a tank or hold space is only partially filled with liquid, then that liquid surface will tend to remain horizontal as the ship's angle of heel changes. This results in a wedge of liquid being transferred from side to side when the ship rolls. This weight movement produces a capsizing moment, known as **the Free Surface Effect,** which reduces the ship's stability and so can be considered as an effective rise in the KG value. The Free Surface Effect can be calculated from the length and width of the slack tank space and the density of the liquid concerned. A correction factor can then be added to solid KG to give a **fluid KG** value.

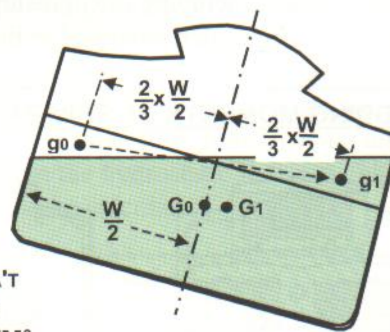
We can calculate the ship's fluid GM by taking the fluid KG from the KM value for the vessel's particular loaded mean draft and trim, as given in the ship's hydrostatic particulars.

KG (fluid) = KG (solid) + FREE SURFACE CORRECTION

GM (fluid) = KM – KG (fluid)

DETERMINING THE LOSS OF GM, DUE TO A RECTANGULAR FREE SURFACE

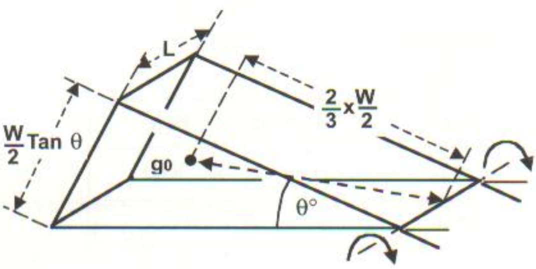




VESSEL OF DISPLACEMENT Δ'T TONNES, HAS A SLACK TANK WITH LIQUID OF DENSITY ρ T/M³

A SHIP HAS A PARTIALLY FILLED TANK OF LENGTH 'L' AND WIDTH 'W', CONTAINING A LIQUID OF DENSITY 'ρ'. WHEN THE SHIP HEELS OVER, A WEDGE OF THE LIQUID TRANSFERS FROM g_0' TO 'g_1' WHICH CAUSES THE C OF G TO MOVE FROM G_0 TO G_1. THIS IS AN EFFECTIVE LOSS OF STABILITY.

$$\text{SHIFT IN C OF G,- } G_0G_1 = \frac{\text{WEIGHT OF LIQUID WEDGE x } g_0 g_1}{\text{SHIP'S DISPLACEMENT, '}\Delta\text{'T}}$$



WEIGHT OF WEDGE = VOLUME x DENSITY

AND VOLUME 'V' = X-AREA x LENGTH

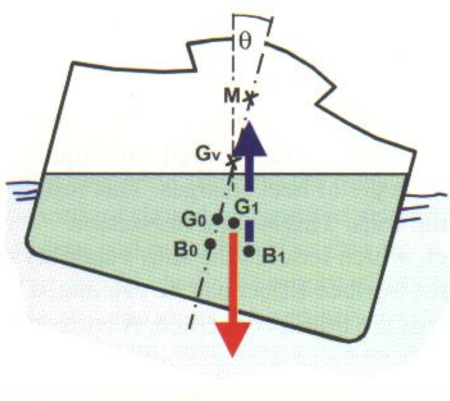
SO $$V = \frac{1}{2}\left[\frac{W}{2}\right]\frac{W}{2}\ L \tan\theta$$

WEIGHT OF WEDGE $$= \frac{W^2}{8} L\,\rho \tan\theta$$

ALSO $$g_0 g_1 = \frac{2}{3}W$$

$$\text{SO } G_0G_1 = \frac{2}{3}W\left[\frac{W^2}{8}\right]\frac{L\,\rho\tan\theta}{\Delta'T} \qquad \text{HENCE } G_0G_1 = \frac{L\,\rho\,W^3\ \tan\theta}{12\,\Delta'T}$$

FOR A SMALL ANGLE OF HEEL, θ°, THE SHIFT OF G_0G_1 CAN BE CONSIDERED AS A TRANSVERSE MOVE, SO THERE IS A NEGLIGIBLE RISE OF G AND KG CAN BE TAKEN TO REMAIN CONSTANT.



AS THE VESSEL HEELS, THE C OF B SWINGS OUT FROM B_0 TO B_1 WHILST THE FREE SURFACE EFFECT CAUSES THE C OF G TO MOVE FROM G_0 TO G_1 WHICH REDUCES THE RIGHTING LEVER AS WEIGHT NOW ACTS VERTICALLY BENEATH G_V.

G_0G_V IS THE EFFECTIVE LOSS OF GM AND IF θ IS SMALL THEN'-

$$G_0G_V = \frac{G_0G_V}{\tan\theta}$$

SO $$G_0G_V = \frac{L\,\rho\,W^3}{12\,\Delta'T}$$

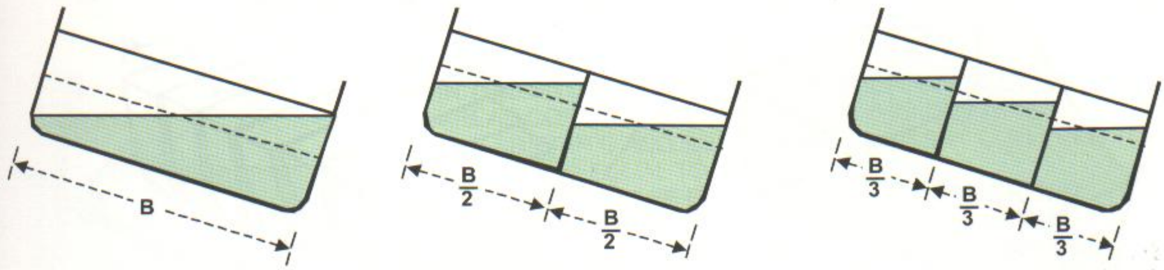
THE LOSS OF UPRIGHT GM FOR A SHIP OF DISPLACEMENT Δ'T **TONNES**, DUE TO THE FREE SURFACE EFFECT OF A LIQUID WITH DENSITY '**ρ'T/M³** IN A TANK '**L' METRES** LONG AND '**W' METRES** WIDE. IS GIVEN BY:-

THE VIRTUAL LOSS OF UPRIGHT GM, **'G_0G_V'** $= \dfrac{L\,\rho\,W^3}{12\,\Delta'T}$ **METRES**

REDUCING FREE SURFACE EFFECT IN A SHIP'S TANKS BY SUBDIVISION

The loss of GM due to free surface increases with the $(\text{width})^3$ of the tank containing the fluid. From this, we can see that the free surface of tank space can be greatly reduced by longitudinally dividing it into several smaller tanks. If a tank is longitudinally divided into two tanks, each half-width tank has a free surface moment equal to 1/8 of the original full width value. As there are now two tanks instead of one, the total free surface moment for the same volume of fluid, is a quarter of the full width tank.

THE REDUCTION OF FREE SURFACE BY SUBDIVISION OF A TANK



FULL BEAM WIDTH TANKS

$$G_0G_V = \frac{L\rho B^3}{12\,\Delta'_T}$$

HALF BEAM WIDTH TANKS

$$G_0G_V = 2 \times \frac{L\rho}{12\,\Delta'_T}\left[\frac{W}{2}\right]^3$$

$$G_0G_V = \frac{1}{4} \times \frac{L\rho B^3}{12\,\Delta'_T}$$

ONE THIRD BEAM WIDTH TANKS

$$G_0G_V = 3 \times \frac{L\rho}{12\,\Delta'_T}\left[\frac{W}{3}\right]^3$$

$$G_0G_V = \frac{1}{9} \times \frac{L\rho B^3}{12\,\Delta'_T}$$

LONGITUDINAL BULKHEADS CONSIDERABLY REDUCE TRANSVERSE FREE SURFACE EFFECT IN A TANK SPACE

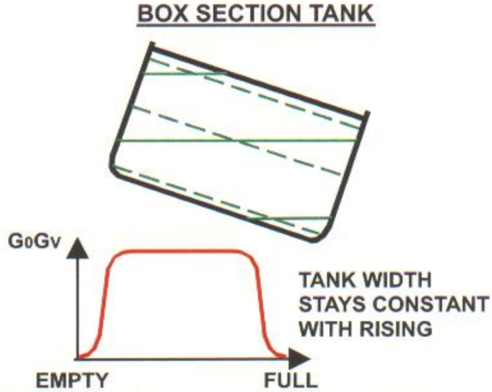
THE LOSS OF UPRIGHT GM FOR A SHIP OF DISPLACEMENT Δ'_T **TONNES**, DUE TO THE FREE SURFACE EFFECT OF A LIQUID WITH DENSITY 'ρ**'T/M^3** IN A TANK SPACE ' **L' METRES** LONG AND '**W' METRES** WIDE DIVIDED BY **N** EQUALLY SPACED LONGITUDINAL BULKHEADS. IS GIVEN BY:-

THE VIRTUAL LOSS OF UPRIGHT GM, **'G_0G_V'** = $\frac{1}{(1+N)^2} \times \frac{L\rho B^3}{12\,\Delta'_T}$ **METRES**

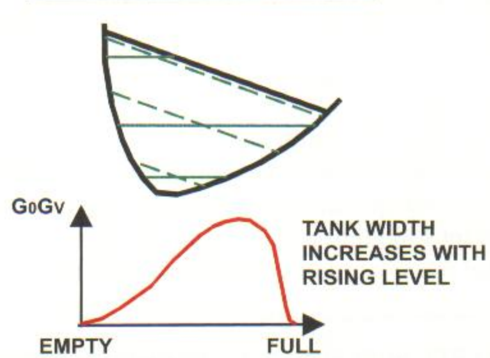
Free surface effect is independent of the level of fluid in tanks with a rectangular transverse section whilst the fluid surface extends across the entire width of the tank. (i.e. the tank is approximately within 5% and 95% full and the angle of heel is relatively small). The free surface effect is reduced when there is insufficient liquid or its flow is restricted by the tank top. Free surface does, however, increase with the depth of fluid in end tanks where the sectional area is triangular

FREE SURFACE EFFECT AND THE LEVEL OF LIQUID IN A TANK

BOX SECTION TANK



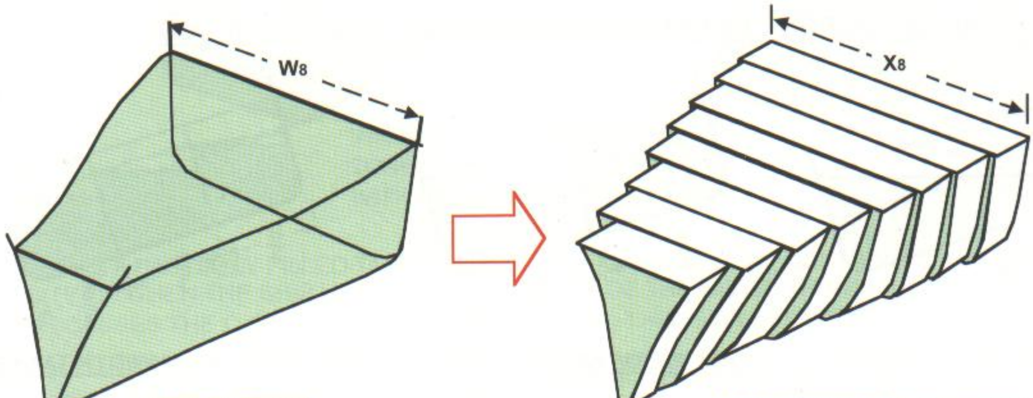
TRIANGULAR SECTION TANK



THE FREE SURFACE EFFECT OF A SLACK BOX SECTION TANK REMAINS CONSTANT BETWEEN ABOUT 5% AND 95% FILLED LEVEL, BUT THE FREE SURFACE OF A TRIANGULAR SECTIONED TANK INCREASES UP TO ABOUT 95% WITH THE LIQUID SURFACE WIDTH

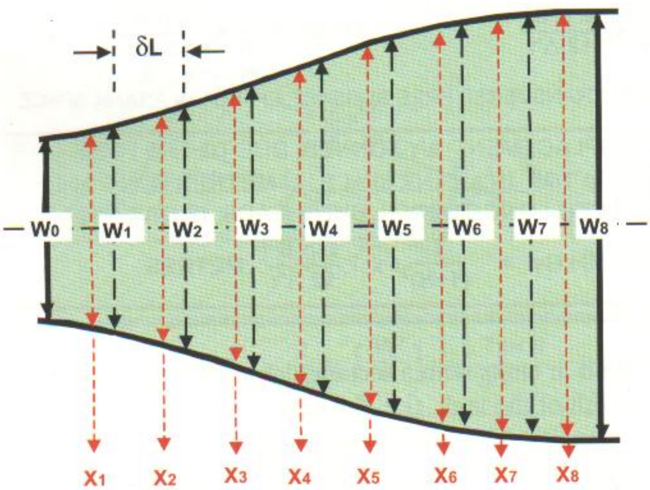
FREE SURFACE EFFECTS OF NON-RECTANGULAR PLANFORMS

A lot of tankspace in a ship's hull is not rectangular in planform, particularly at the fore and aft ends of the vessel. The free surface effect of any slack tank depends upon how the free surface area changes with the ship's angle of heel, i.e. the Moment of Inertia of the liquid surface and this can be estimated by the methods of approximate integration that were used in Chapter 2 for the ship's waterplane area.



ACTUAL TANK

APPROXIMATION OF TANK



W_0, W_1, ETC, ARE THE WIDTH ORDINATES MEASURED AT EQUAL INTERVALS OF δL ALONG THE CENTRELINE OF THE TANK

X_1, X_2 ETC, ARE THE EFFECTIVE FREE SURFACE WIDTHS OF EACH STRIP IN THE APPROXIMATION OF THE TANK, WHERE

$$X_1^3 \approx \frac{W_0^3 + W_1^3}{2} \quad M^3$$

AND $$\delta I_1 \approx \frac{X_1^3\ \delta L}{12} \quad M^4$$

WHERE δI_1 IS THE MOMENT OF INERTIA OF THE FORWARD MOST STRIP OF THE APPROXIMATED TANK FORM

THE TOTAL MOMENT OF FREE SURFACE INERTIA 'I', IS THE SUM OF ALL THE δI VALUES

I.E. $$I \approx \frac{\delta L}{12}\left(\frac{W_0^3}{2} + \frac{W_1^3}{2} + \frac{W_1^3}{2} + \frac{W_2^3}{2} + \frac{W_2^3}{2} + \frac{W_3}{2} + \text{--} + \frac{W_6^3}{2} + \frac{W_7^3}{2} + \frac{W_7^3}{2} + \frac{W_8^3}{2}\right) M^4$$

SO $$I \approx \frac{\delta L}{12}\left(\frac{W_0^3}{2} + W_1^3 + W_1^3 + W_2^3 + W_2^3 + W_3^3 + \text{--} + W_6^3 + W_7^3 + W_7^3 + \frac{W_8^3}{2}\right) M^4$$

The value of 'I' (the Moment of Inertia or second Moment of Area) is calculated for every tank and cargo space capable of carrying fluid in a ship and supplied by the builders in the ship's stability data book. This will also include other basic data, such as tank capacities and positions of centres of volume for different sounding levels. From this information, the loss of GM due to free surface can be calculated for any given loaded condition, by using the following equations.

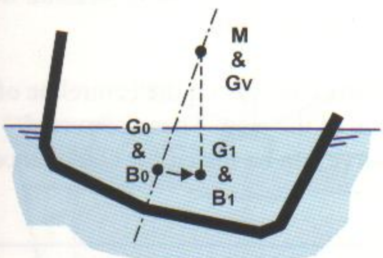
FLUID FREE SURFACE MOMENT ABOUT KEEL $= I\rho$ **TONNES-METRES, &** **VIRTUAL LOSS OF GM TO FREE SURFACE** $= \dfrac{I\rho}{\Delta T}$ **METRES**

WHERE 'ΔT' IS THE DISPLACEMENT OF THE VESSEL. A VESSEL'S FLUID UPRIGHT GM IS OBTAINED BY SUBTRACTED THE VIRTUAL LOSS OF GM FOR ALL THE SLACK TANKS IN THE SHIP FROM THE SOLID VALUE OF UPRIGHT GM.

THE EQUATIONS FOR BM AND LOSS OF GM THROUGH FREE SURFACE

The Free Surface virtual rise of 'G' is a measure of the shift in 'G' caused by the transfer of a fluid wedge shaped weight as the ship heels. The 'BM' value of a hull is a measure of shift of 'B' caused by a similar transfer of a wedge shaped volume of buoyancy. The swing in the Centre of Buoyancy of an underwater hullform and the Free Surface effect as a ship heels over, both depend upon the Moments of Inertia of the areas concerned. This is demonstrated by completely flooding an open buoyant boat, which produces a free surface area almost identical to that of the waterplane, as shown below.

THE LOSS OF STABILITY IN A FLOODED OPEN BOAT



IN THIS SITUATION, THE WATERPLANE AND FREE SURFACE AREAS COINCIDE, SO THE SHIFT OF B IS IDENTICAL TO THE SHIFT IN G AND SO THE VIRTUAL RISE IN G TO FREE SURFACE IS EQUAL TO THE BM.
THE FREE SURFACE EFFECT CANCELS THE RIGHTING MOMENT AND THE BOAT WILL REMAIN AFLOAT WITH NEUTRAL STABILITY (i.e. ZERO GM) AND SO IS DIFFICULT TO BOARD WITHOUT TURNING OVER

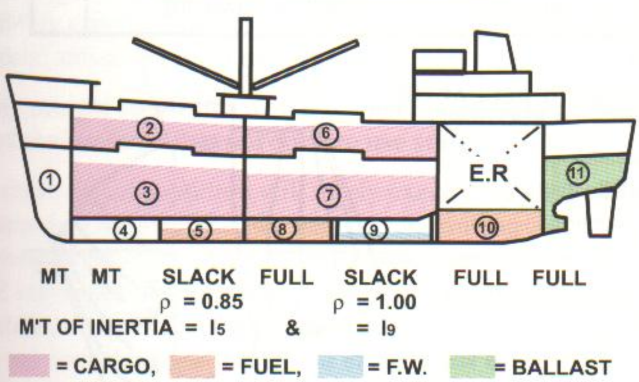
FREE SURFACE RISE IN 'G' TO Gv $= \dfrac{I_{(FSA)}}{\Delta T} \times \rho$ **METRES, & BM** $= \dfrac{I_{(WPA)}}{\Delta T} \times \rho$ **METRES**

WHERE BOAT'S DISPLACEMENT 'ΔT' = DISPLACED VOLUME '$V_{\Delta T}$' x WATER DENSITY 'ρ'

APPLYING FREE SURFACE MOMENTS TO CALCULATE A SHIP'S FLUID KG AND GM VALUES

The Moments of Inertia (value 'I') for any slack tanks is obtained from the ship's stability data book and then multiplied by the fluids' density to obtain Free Surface Moments for these tanks. These are then added to the total of the weight moments taken about the keel and the result is divided by the ship's total displacement to give a fluid KG value. This is subtracted from the KM value, as listed in the Hydrostatic data for that draft, to determine the ship's loaded fluid GM.

SHIP'S LOADED CONDITION



ITEM	WT (T)	VCG (M)	V. MOMENT (T-M)
LIGHTSHIP	W_L	KG_L	$W_L \times KG_L$
1	$+W_1$	Kg_1	$+(W_1 \times Kg_1)$
2	$+W_2$	Kg_2	$+(W_2 \times Kg_2)$
3	$+W_3$	Kg_3	$+(W_3 \times Kg_3)$
4	$+W_4$	Kg_4	$+(W_4 \times Kg_4)$
5	$+W_5$	Kg_5	$+(W_5 \times Kg_5)$
6	$+W_6$	Kg_6	$+(W_6 \times Kg_6)$
7	$+W_7$	Kg_7	$+(W_7 \times Kg_7)$
8	$+W_8$	Kg_8	$+(W_8 \times Kg_8)$
9	$+W_9$	Kg_9	$+(W_9 \times Kg_9)$
10	$+W_{10}$	Kg_{10}	$+(W_{10} \times Kg_{10})$
11	$+W_{11}$	Kg_{11}	$+(W_{11} \times Kg_{11})$
TOTALS	ΔT		WEIGHT M'T
FREE SURFACE M'T, TANK 5			$+ 0.85 \times I_5$
FREE SURFACE M'T, TANK 9			$+ 1.00 \times I_9$
TOTAL FLUID VERT. MOMENT			SUM OF ABOVE

GM (fluid) = KM – KG (fluid)

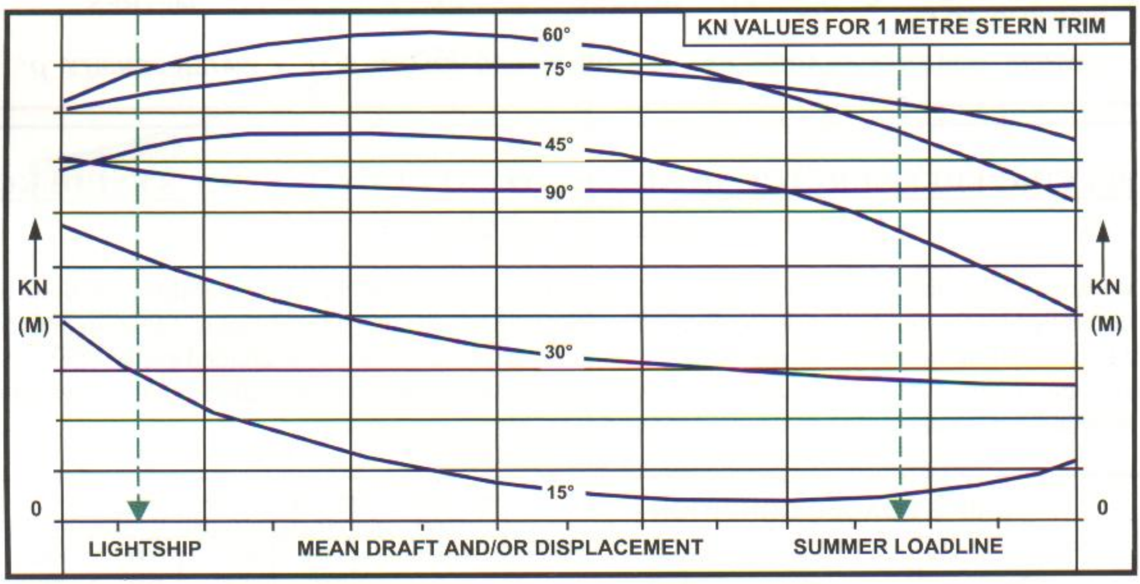
$$\text{KG (fluid)} = \frac{\text{WEIGHT MOMENTS + FREE SURFACE MOMENTS}}{\text{TOTAL DISPLACEMENT}}$$

PRODUCING A GZ CURVE FOR A SHIP'S LOADED CONDITION

The upright fluid GM value for a ship's particular loaded condition, is an important indication of the vessel's state of stability and is often used as the prime indication of whether or not the ship is loaded to a safe condition to sail. However, as Chapter 3 explained, the upright GM is not the only stability criterion to be considered when assessing a ship's seaworthiness. The other five minimum requirements concern the range of dynamic stability and, as such, need a GZ curve to be produced for the vessel's particular loaded state. Chapter 2 described in detail how the ship's hullform is analysed to produce the hydrostatic data in the ship's approved stability book, which include sets of KN values for the operating range of ship's displacement, from lightship to beyond the maximum loaded limit. These are either given in tabular or graphical form at intervals of 15° of heel angle. There should be separate sets of KN values for different values of trim, typically from 0.5m by the head to 2m by the stern.

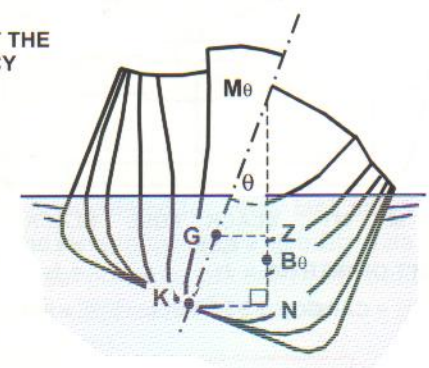
The KN value would be equal to the righting lever 'GZ', if the centre of gravity was on the centreline of the keel (i.e. KG = zero), which is corrected for the actual fluid KG value of the ship. These corrected GZ values are then plotted against heel angle so that the resulting GZ curve can be tested for compliance with the minimum stability criteria.

A TYPICAL SET OF KN CURVES



KN IS THE HORIZONTAL DISTANCE, MEASURED FROM THE KEEL AT THE CENTRELINE TO THE LINE OF ACTION OF THE FORCE OF BUOYANCY

$$GZ_\theta = KN_\theta - KG\sin\theta \text{ METRES}$$



ANGLE OF HEEL, θ	15°	30°	45°	60°	75°	90°
KN VALUES (M)						
- KGsinθ (M)						
=GZ (M)						

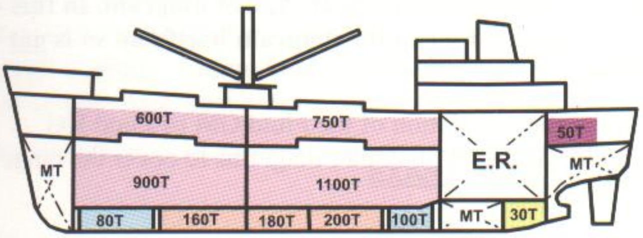
Sometimes, the hydrostatic data is in the form of 'GZ' values for different values of 'KG' at, for example, half metre intervals. In such a case, the actual 'GZ' values for a particular loaded condition can be obtained by direct interpolation between the appropriate tabulated figures.

ONBOARD INFORMATION TO ASSIST IN STABILITY ASSESSMENT

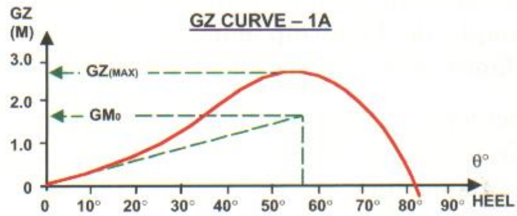
A ship's stability must remain acceptable during all stages of a voyage. In addition to the initial sailing condition, it must also be assessed at the end of a long passage when fuel and water have been consumed from the bottom of the ship and free surface moments may be present in most of the tanks. To assist in this process, the approved stability book includes a number of sample loaded conditions both at the beginning and the end of a voyage after consumption of fuel and water,

AN EXAMPLE OF A SAMPLE OF GZ CURVES

AT SUMMER DRAFT, BALE CARGO OF 3350 T, REL DENSITY 1.3, STOW FACTOR 0.8M³/T, BROKEN STOW 4%



CONDITION 1A,- DEPART WITH FULL FUEL, WATER & STORES



DRAFTS;- FWD 5.30 M, AFT 5.50 M, MEAN 5.41 M

DEADWEIGHT;- 4150 T, DISPLACEMENT;- 5208 T

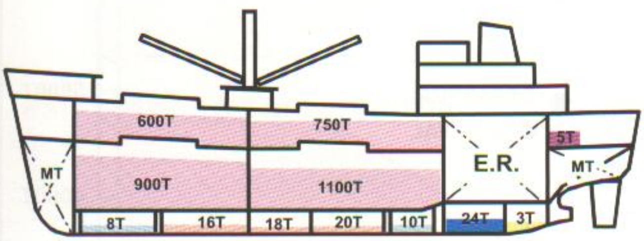
KG;- 4.510M, KM;- 6.110M, $GM_{0(SOLID)}$;- 1.600 M

FREE SURFACE LOSS OF GM;- ZERO

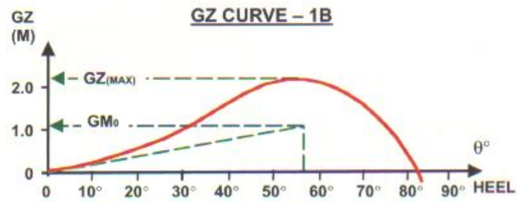
$GM_{0(FLUID)}$;- 1.600 M

= CARGO, = FUEL, = F.W. = BALLAST

= WASTE, = LUB. OIL = STORES



CONDITION 1B,- ARRIVAL WITH 10% FUEL, WATER & STORES



DRAFTS;- FWD 4.60 M, AFT 5.10 M, MEAN 4.81 M

DEADWEIGHT;- 3450 T, DISPLACEMENT;- 4508 T

KG;- 4.210M, KM;- 6.160M, $GM_{0(SOLID)}$;- 1.950 M

FREE SURFACE LOSS OF GM;- 0.952 M

$GM_{0(FLUID)}$;- 0.998 M

The range of sample conditions will include the lightship state, which usually fails to meet the minimum stability criteria. The lightship condition is the basis of all the other conditions and can be required for special situations, such as drydocking the vessel.

Any sample condition which is unseaworthy, must be clearly labelled as such in the stability book and the limits of its appropriate use stated.

A ship's actual condition will rarely exactly match the supplied examples but it is often acceptable to interpolate between the two nearest relevant curves at least in the planning stage of a voyage, to determine whether or not the proposed load will be acceptable. It would only be necessary to produce GZ curves for testing of acceptance, if the ship is to be operated close to the limits of the minimum stability criteria. In such circumstances, it is worth considering the accuracy of the fluid KG calculation. **In my experience, a margin of error must be made to allow for unaccountable weight distribution, either in the cargo or provisions and deck stores.** The accumulation of added weight through years of small additions of equipment and modifications is not easy to assess on an older vessel, particularly if the inclining experiment was carried out twenty years previously. Typically, the observed draft marks may indicate an additional 0.5 to 1 % weight to the calculated displacement.

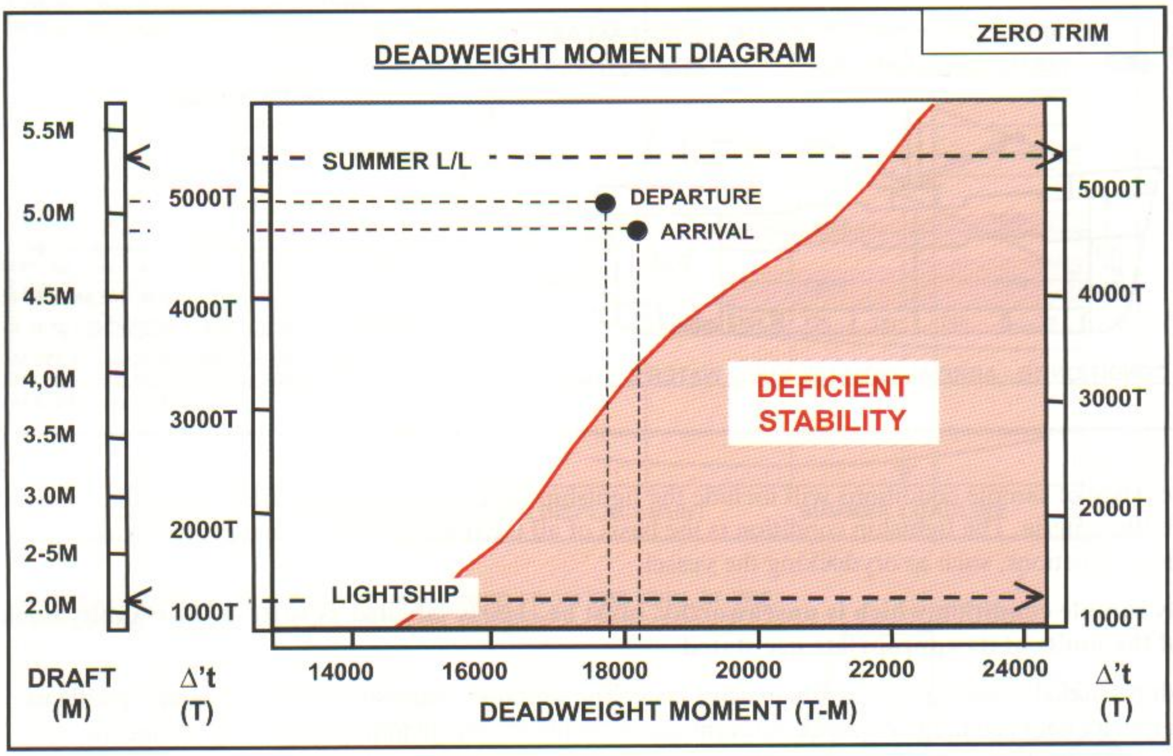
Many new vessels have an approved and dedicated stability computer with the hydrostatic information stored as a data base in the software. This makes stability assessment considerably easier and quicker than longhand calculations **but the margin of error in the weight distribution input, still applies.**

SIMPLIFIED STABILITY DIAGRAMS

Any ship at a given draft will have a maximum acceptable fluid KG value that will meet the minimum stability requirements. As such, it is possible to construct a diagram from the hull's hydrostatic data, which will indicate the boundary between acceptable and unacceptable loaded states. A ship's particular condition is plotted onto such a diagram and its position on the diagram, relative to the boundary, will indicate whether or not that particular condition complies with the minimum stability criteria. These diagrams can take several forms and the one shown below is known as a **Deadweight Moment Diagram.** The moments about the keel of all the weights of cargo, fuel, water, stores and ballast are added to the free surface moments to produce a **Total Deadweight Moment**. This then can be plotted on the diagram against the **ship's draft or displacement, 'Δ't** and if the plot lies outside the region of deficient stability, the particular loaded condition will meet the minimum stability criteria.
It is important to appreciate exactly what values are used in a particular type of diagram. In this example, the Lightship moment about the keel is accounted for in the diagram itself and so is <u>not</u> included in the calculation of deadweight moments.

Other forms of the simplified stability diagram require the calculation of the Fluid GM or Fluid KG value to enter against the Mean Draft. The ship should have a set of similar diagrams to cover the range of operating trim

SIMPLIFIED STABILITY DATA



IN THE ABOVE EXAMPLE, A VESSEL IS PLANNED TO DEPART ON A SEA PASSAGE. BOTH THE DEPARTURE AND ESTIMATED ARRIVAL CONDITIONS AT THE END OF THE VOYAGE HAVE BEEN PLOTTED ON TO THE DIAGRAM.

	MEAN DRAFT	DISPLACEMENT, Δ't	DEADWEIGHT MOMENT
DEPARTURE	5.10M	4850T	17700 T-M
ARRIVAL	4.85M	4600T	18100 T-M

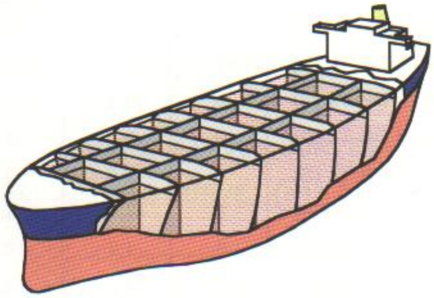
THE SHIP, AFTER THE CONSUMPTION OF 250T OF FUEL AND WATER AND INCREASED FREE SURFACE MOMENTS, ARRIVES IN A MORE TENDER CONDITION THAN THAT ON DEPARTURE, BUT IT STILL REMAINS IN THE REGION OF ACCEPTABLE STABILITY.

TRANSVERSE STABILITY AND TANKERS

MARPOL regulations (See Chapter 10, page 259), require new tankers at sea to comply with the I.M.O. minimum intact stability criteria explained in Chapter 3, page 63) and maimtain a minimum upright Gm value of 0.15 metres in port. Free surface can be minimal in its effect on the stability of dry cargo ships but it is unavoidable in tankers. Cargo tanks cannot be loaded much beyond 95% as space must be left to allow for expansion of the cargo and the construction of most tankers makes no provision for overspill, either through expansion or overloading.

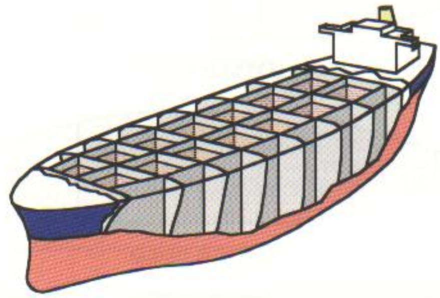
Tankers must retain adequate stability whilst having considerable free surface effects.and this is achieved by using longitudinal subdivision in the tank space. Before 1982, most tankers were built with a single skin layout in which the cargo spaces occupied most of the hull as three longitudinal rows of tanks. About 40% of these same cargo tanks were filled with seawater when the ship made a ballast passage. Since 1982, however, the MARPOL international anti-pollution regulations have required ballast tanks to be built around the cargo tanks in order to minimise leakage in the case of damage through collision or stranding.(See Chapter 10) The post 1982 tanker does not make sea passages with fluid in every tank as when the cargo tanks are full, the ballast tanks are empty and vice-versa.

EXAMPLES OF LOADED TANKERS



A PRE- 1982 BUILT TANKER

ALL THE TANK SPACES ARE LOADED OUT TO THE SHIPS SIDES AND THERE IS NO DOUBLE BOTTOM



A TANKER BUILT AFTER THE 1982 ANTI-POLLUTION RULES

LOADED CARGO TANKS ARE NOW PROTECTED BY EMPTY BALLAST SPACES ARRANGED AS WING AND DOUBLE BOTTOM TANKS

Post 1982 tankers are larger than their predecessors with the same deadweight. This increases the cost of building and designers have responded by increasing tank size, which reduces the steelwork and the number of tanks whilst also simplifying the pipework. However, a smaller number of larger tanks will have a greater free surface effect, particularly if the centrline bulkhead is ommited. Such ships are more likely to operate closer to the margins of acceptable stability than their prdecessors and require a more sophisticated method of calculating stability that relies on modern computer software.

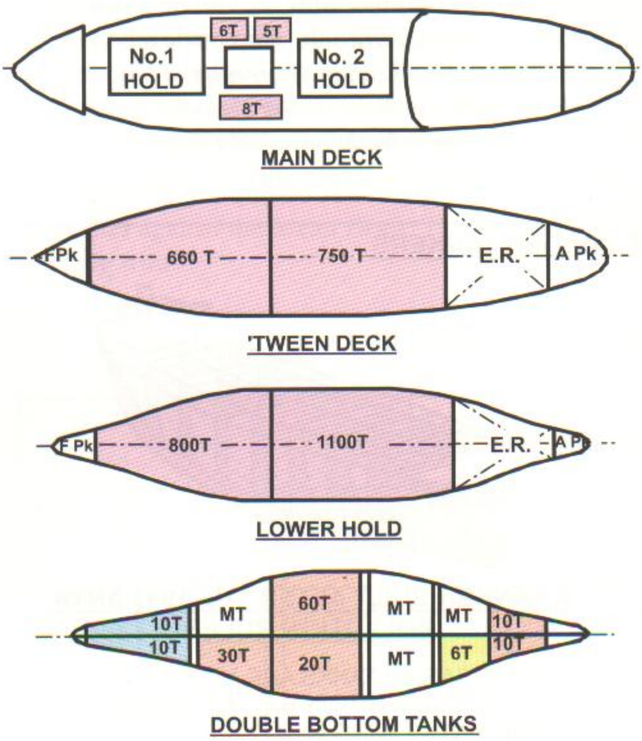
In the past, 'free surface' was allowed for by applying a single total fluid correction to the upright solid KG value, (see page 76) In reality though, free surface effects depend upon the changing shape of the fluid volume in the tanks just as a ship's BM value depends upon the changing underwater hullform and so varies with the angle of heel and level of liquid in the tank. The weight of the fluid wedge transfered increases with the angle of heel until the the tank top restricts further weight transfer. This is analogous to the point of deck edge immersion and its effect on the GZ curve.(see page 52) so the adverse stability effect of a free surface becomes progressively less significant at angles of heel beyond the point of being restricted by the tank top.(see page 74) Consequently, the free surface effect of a tank 95% full will be negligeable at all but the smallest heel angles, whereas it will continue to be significant at larger angles of heel if the same tank is only 50% full.

This will impose a limit on the number of tanks that can be partially filled at any one time, particularly during port operations when ballast and cargo are being worked at the same time. The stability book should give clear instructions as to the sequence in which cargo and ballast tanks can be loaded or discharged whilst retaining an adequate fluid GM.

HEELING DUE TO THE C OF G BEING OFFSET FROM THE CENTRELINE

So far, we have assumed that all the ship's weight is evenly distributed between the port and starboard sides of the vessel, resulting in the C of G being on the centreline and the ship upright. It is normally desirable to keep a ship as close to the upright condition as possible and so it is necessary to calculate the transverse position of 'G' to determine any list which might be produced by a given loaded condition.

AN EXAMPLE OF TAKING MOMENTS ABOUT THE CENTRELINE

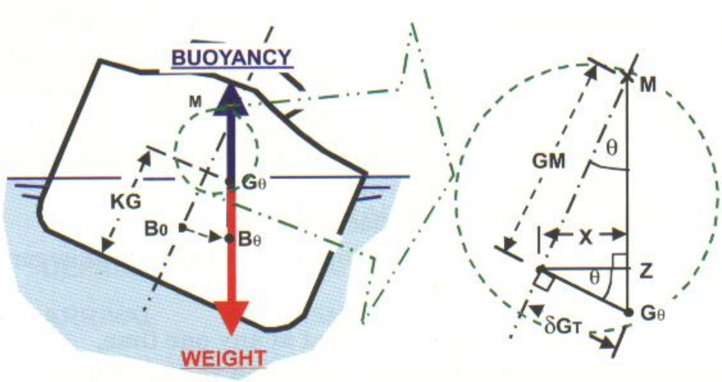


ITEM	WEIGHT (T)	OFFSET (M)	MOMENT (T-M)
DECK STOW (P)	11	+3.0	+33.0
DECK STOW (S)	8	-4.0	- 32.0
No.1 DB Tk (P)	5	+1.1	+ 5.5
No.1 DB Tk (S)	5	-1.1	- 5.5
No.2 DB Tk (P)	30	+2.5	+ 75.0
No.2 DB Tk (S)	0	-2.5	0.0
No.3 DB Tk (P)	60	+3.2	+192.0
No.3 DB Tk (S)	20	-3.2	- 64.0
No.4 DB Tk (P)	0	+2.8	0.0
No.4 DB Tk (S)	0	-2.8	0.0
No.5 DB Tk (P)	0	+1.7	0.0
No.5 DB Tk (S)	6	-1.7	- 10.2
No.6 DB Tk (P)	10	+1.2	+ 12.0
No.6 DB Tk (S)	10	-1.2	+ 12.0
TOTAL MOMENTS			+193.8 M
SHIP'S DISPLACEMENT			5000 T
SO OFFSET OF 'G' TO PORT			0.039 M

$$\text{OFFSET OF 'G' FROM THE C/L} = \frac{\text{TOTAL MOMENTS TO PORT OR STARBOARD}}{\text{SHIP'S TOTAL DISPLACEMENT '}\Delta\text{t'}}$$

Small angles of heel, or **list**, resulting from any transverse offset of 'G' from the centreline, can be calculated as follows

CALCULATING THE ANGLE OF LIST



'G' IS OFFSET 'δG_T' WHICH CAUSES A LIST $\theta°$ TO PORT

AT THE ANGLE OF LIST $\theta°$, THE C OF G IS IN VERTICAL ALIGNMENT WITH THE C OF B

HEELING LEVER $'X' = \delta G_T \cos\theta$

& RIGHTING LEVER $GZ = GM \sin\theta$

So $GM \sin\theta = \delta G_T \cos\theta$

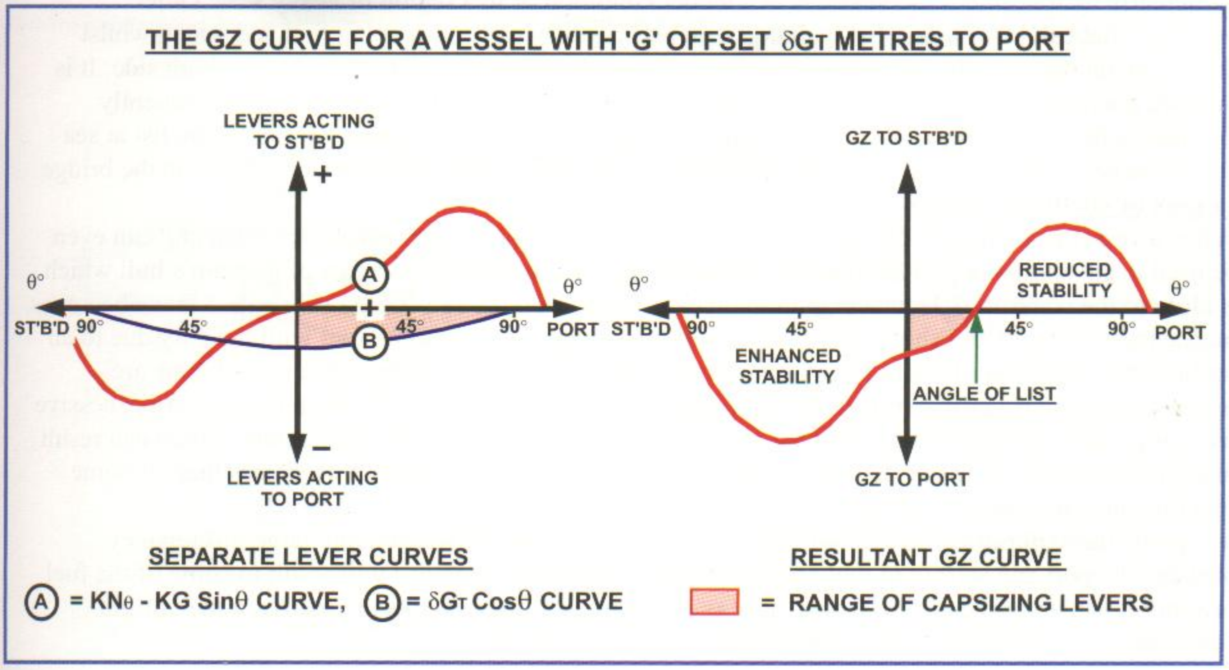
Hence $\dfrac{\sin\theta}{\cos\theta} = \dfrac{\delta G_T}{GM}$

SO LIST $\theta°$ IS GIVEN BY:-

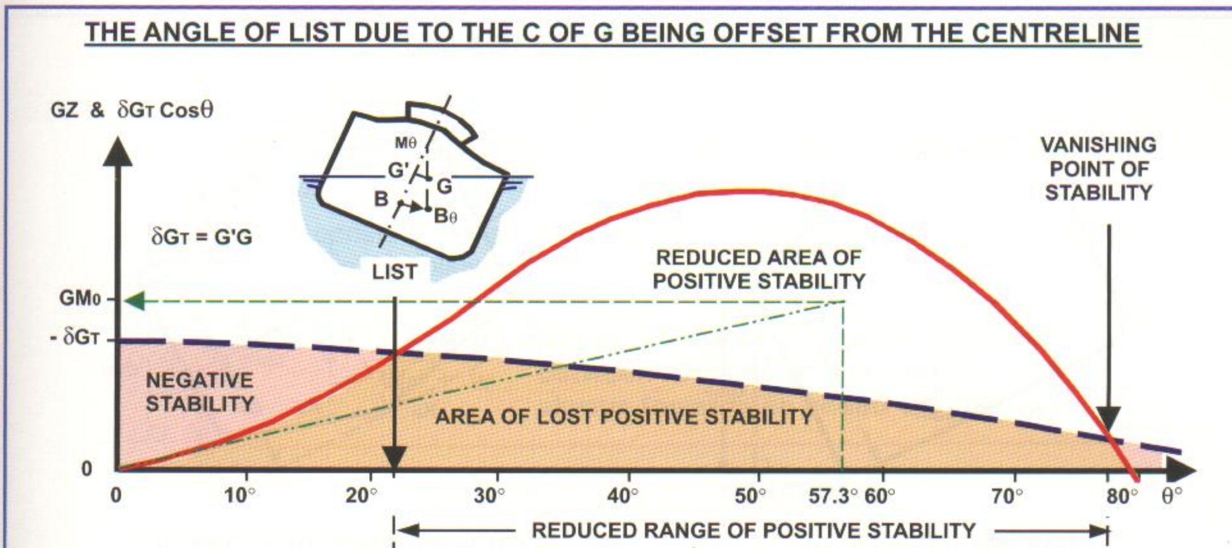
$$\tan\theta = \frac{\delta G_T}{GM}$$

ANGLE OF LIST AND A VESSEL'S GZ CURVE

A GZ curve, produced from a ship's KN curves and corrected for a particular KG value, will be symmetrical about the upright condition as it is based upon the C of G being on the centreline. If we superimpose a heeling lever curve onto this and then combine the two curves together, we can obtain a GZ curve for a ship in a listed condition, due to 'G' being offset from the centreline.



The combined curve above shows the range and extent of dynamic stability for the vessel heeled both to port and starboard, which now is no longer symmetrical. The angle of list is also indicated by the curve's intersection with the zero lever axis. This way of determining the list does not depend upon the assumption that the GM value remains constant and so is not restricted to small angles of heel. We are only really concerned with the side of reduced stability, so it is more convenient to superimpose the two curves in the following way.



IN EFFECT, THEN δGT Cosθ CURVE BECOMES THE DATUM LINE FOR MEASURING AREAS OF POSITIVE STABILITY UNDER THE GZ CURVE, WHEN CONSIDERING THE SHIP'S STABILITY CHARACTERISTICS

IF THE LIST IS LESS THAN 6°, THEN, $\tan(\text{LIST}) = \dfrac{\delta G_T}{GM_0}$ *WHERE GM_0 WOULD BE THE UPRIGHT GM VALUE*

THE CONSEQUENCES OF A SHIP DEVELOPING A LIST

At sea

When a ship has a list at sea, the minimum freeboard and range of dynamic stability is reduced, so the vessel's ability to resist heeling to the low side decreases and the risk of shipping seas onboard, with its associated risks to deck cargo, is increased. In general, the ship's seaworthiness is reduced, which is particularly undesirable if the vessel is in a tender condition or in a region of heavy seas. Other problems that can arise with a list include an increasing difficulty in pumping fuel and water, whilst there is an increasing reluctance for rain or seas to drain overboard effectively from the high side. It is therfore good practise to keep a ship as upright as possible whilst it is at sea, mainly by frequently alternating fuel and water consumption between the port and starboard sides. Normally, the list at sea should be kept within one or two degrees, which is quite perceptable by the watch officer on the bridge.

In port or sheltered waters

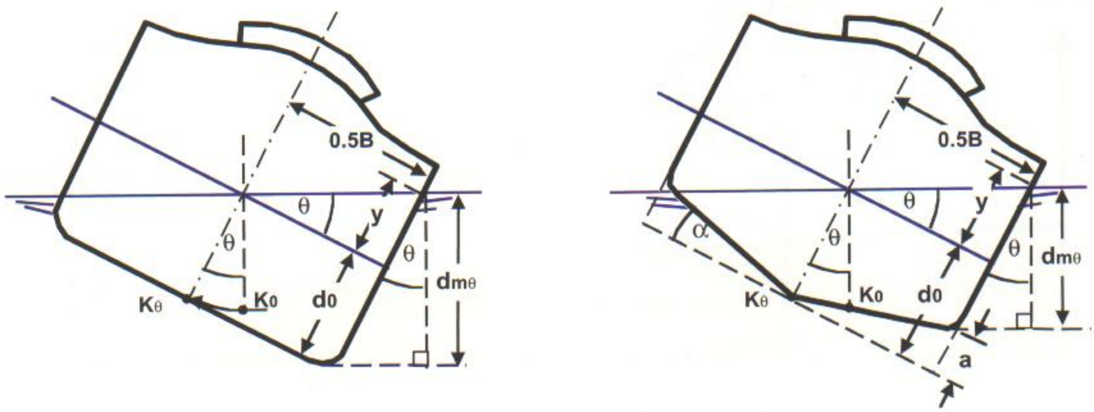
When a vessel is loading or discharging in port, larger angles of list are usually tolerated and can even be used to advantage (e.g.to facilitate the pumping dry of a tank or inspect part of the ship's hull which is close to the waterline). However, plumbing cargo spaces on the ship's high side with a crane becomes increasingly difficult and ship's side fittings will be damaged if they come foul with the jetty due to an onshore list, whilst an offshore list will put extra strain on the mooring lines. These problems are increased if there is vertical movement as well due to a significant rise and fall in the tide. An excessive list, particularly at light drafts, may also expose cooling inlets to the ship's generators, which can result in a power shut down. As a rule, list should be kept to a minimum, even in port, unless there is some specific purpose to be served.

It is particularly important to sail the ship very close to the upright without any large discrepancy between the port and starboard fuel and water tanks. Any such inbalance will result in some of the fuel or water being unusable during the voyage, as it will be required as ballast to maintain the upright condition.

CHANGES IN THE SHIP'S MIDSHIPS DRAFT DUE TO LISTING

Listing a ship alters the draft readings across the breadth of the vessel. Inevitably the maximum midships draft increases if a ship has a rectangular midships section and this can lead to grounding the ship on the low side. If a ship has 'rise of floor', then the midships maximum draft will increase if the angle of list is greater than the angle of the ship's bottom but decrease if it is less, as is shown by the following diagram:-

CHANGE OF DRAFT DUE TO A LIST



BOX-SHAPED SECTION

AT θ° HEEL, $d_{m\theta} = (d_0 + y)\cos\theta$

SO MAX. DRAFT = $(d_0 + 0.5B\tan\theta)\cos\theta$

SECTION WITH RISE OF FLOOR OF α°

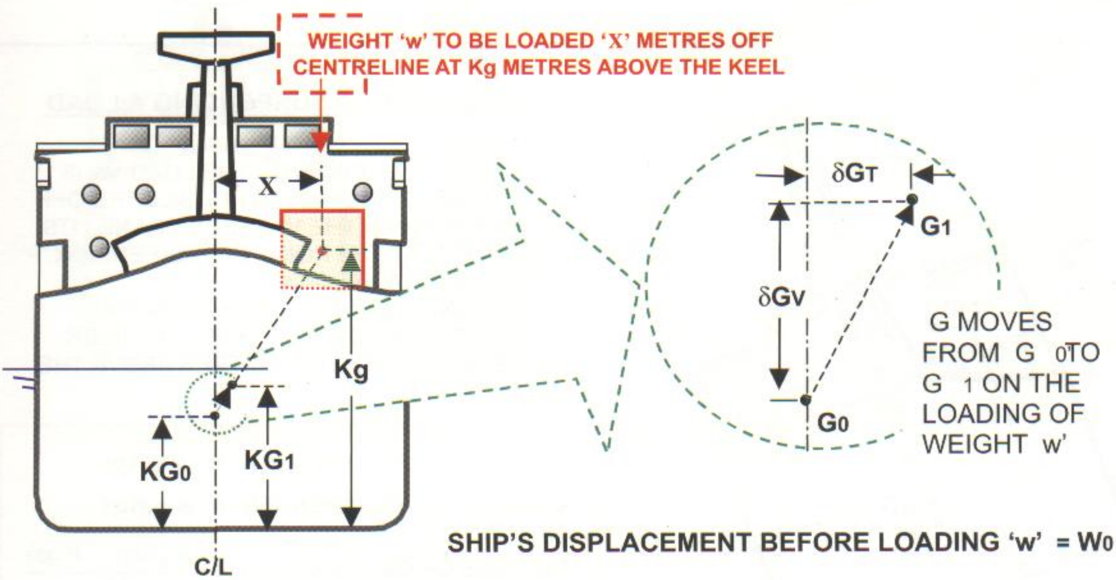
AT θ° HEEL, $d_{m\theta} = (d_0 + y - a)\cos\theta$

SO MAX. DRAFT = $(d_0 + 0.5B\tan\theta - \tan\alpha)\cos\theta$

THE CHANGE IN THE POSITION OF THE C OF G DUE TO LOADING OR DISCHARGING A SINGLE WEIGHT

Sometimes it is useful to be able to calculate quickly the change in the position of 'G' as a result of loading or discharging a single weight, whether it be cargo, fuel, water or stores, and this can be done in the following way. Note that it usual to consider vertical and transverse shifts in G separately.

THE SHIFT IN THE C of G BY MOVING A SINGLE WEIGHT



SHIP'S DISPLACEMENT BEFORE LOADING 'w' = W_0

CONSIDER THE VERTICAL MOMENTS ABOUT THE KEEL

$$KG_1 = \frac{Kg.w + KG_0.W_0}{W_0 + w} \qquad \& \qquad KG_1 = KG_0 + \delta G_V$$

$$\text{So } \delta G_V = \frac{Kg.w + KG_0.W_0 - KG_0.W_0 - KG_0.w}{W_0 + w}$$

$$\text{So } \delta G_V = \frac{w\,(Kg - KG_0)}{W_0 + w}$$

δG_V is positive as $Kg > KG_0$, This indicates that 'G' rises

Similarly, if we consider the transverse shift of G by taking moments about the C/L, we will obtain the following equation:-

$$\delta G_T = \frac{w.x}{W_0 + w}$$

Where 'x' is the transverse separation between 'G_0' and 'g'. The shift of 'G' is always towards an added weight

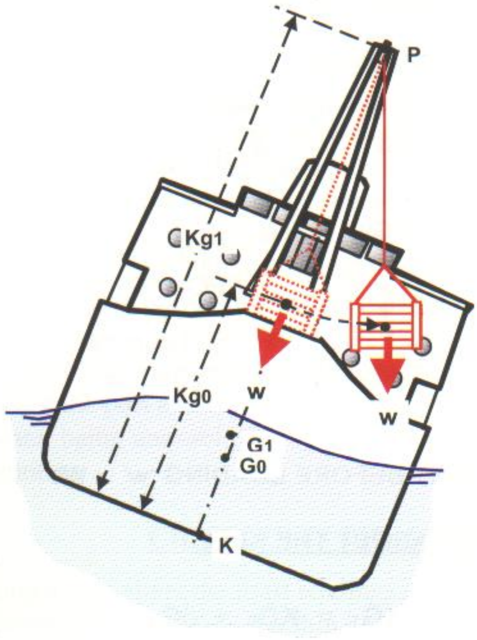
WHEN WEIGHT 'w' IS DISCHARGED

$$\text{So } \delta G_V = \frac{w\,(Kg - KG_0)}{W_0 - w} \qquad \& \qquad \delta G_T = \frac{w.x}{W_0 - w}$$

'w' now has a negative value and 'G' moves away from the point of discharge

THE EFFECTIVE Kg VALUE FOR A FREELY SUSPENDED WEIGHT

When a ship's crane or derrick supports a suspended load, then, as with the free surface effect, the weight of that load is free to swing as the ship heels. The weight will always act vertically downwards through its point of suspension, so that becomes its effective centre of gravity, regardless of how high the weight itself is actually lifted. This effective transfer of weight will occur the instant that the crane lifts the load off the deck of the ship and so there will be an immediate rise in the vessel's C of G in the direction of the crane head. This will cause a corresponding reduction in GM and, hence, a loss of transverse stability. Obviously, this is more serious , if the vessel is at sea, rather than in port, when the lifting operation is carried out.



THE EFFECT OF SUSPENDING A LOAD

IN THE DIAGRAM OPPOSITE, THE LOAD 'w', IS FREE TO SWING THE INSTANT IT IS LIFTED OFF THE FOREDECK BY THE VESSEL'S CRANE. ITS WEIGHT ACTS AT THE POINT OF SUSPENSION 'P' (THE CRANE HEAD BLOCK).
THIS WILL CAUSE A VERTICAL RISE IN THE VESSEL'S C OF G AS THE WEIGHT HAS BEEN TRANSFERRED FROM Kg_0 METRES ABOVE THE KEEL TO Kg_1.

MOMENT OF 'w' ON DECK = $w \times Kg_0$

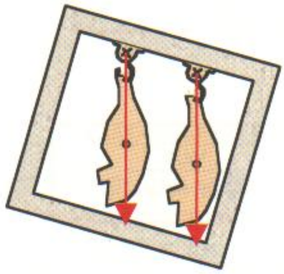
MOMENT OF w SUSPENDED = $w \times Kg_1$

So, LOSS OF GM (I.e. $G_0 G_1$) = $\dfrac{w\,(Kg_1 - Kg_0)}{\Delta_T}$

WHERE Δ_T IS THE SHIP'S TOTAL DISPLACED WEIGHT, INCLUDING 'w'

Provided that the rise in 'G' does not exceed the ship's initial GM before lifting the weight, then the ship will remain stable but with a reduced range of dynamic stability. If the lift is a significant weight, then the effect of the operation on the ship's stability must be considered beforehand and an estimate made of any transient list that will occur during the lift, The calculations for the effect upon the vessel's stability, is discussed more thoroughly in the following pages concerned with heavy lift operations but the swinging load is a hazard to the ship's structure and anyone working on deck. This can be minimised by lifting the load as close as possible to the crane head, which reduces the 'pendulum' length and tends to 'kill' the swing. **This will not cause further reduction in the ship's stability as the weight always acts around the pivot point, regardless of how high it is lifted.**
Some cargoes are actually stowed in suspension. Chilled meat is hung up on rails mounted to the deckhead of the chilled lockers or containers, to allow free circulation of the chilled air.

CHILLED MEAT CARGO

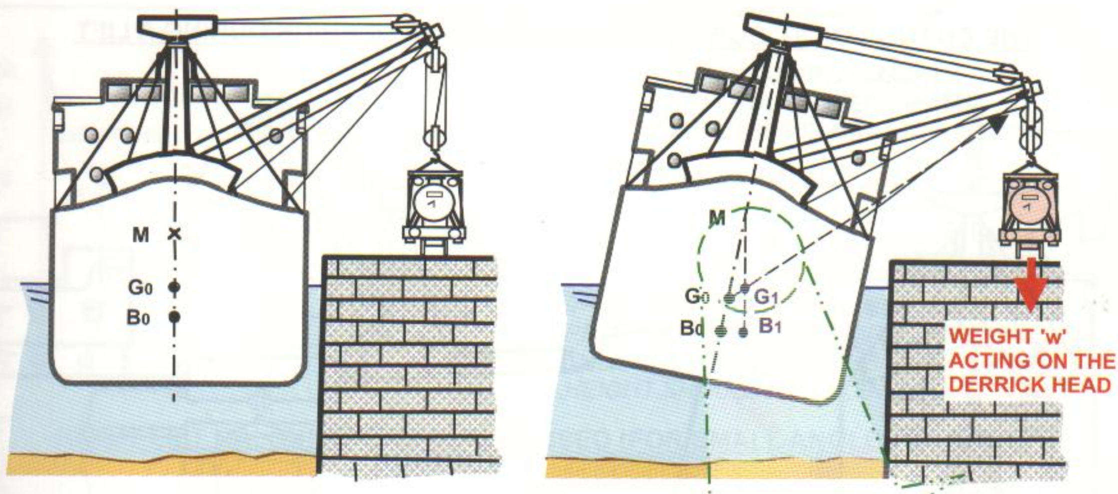


THE WEIGHT OF THIS CARGO ACTS AT THE POINT OF SUSPENSION ON THE DECKHEAD, SO THE Kg VALUE OF THE STOW, USED IN STABILITY CALCULATIONS, MUST BE THE HEIGHT ABOVE THE KEEL OF THIS SUSPENSION POINT

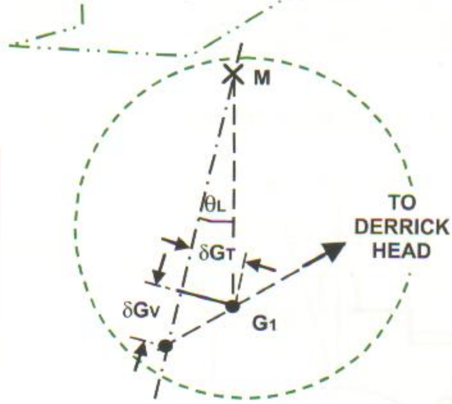
STABILITY CONSIDERATIONS WHEN WORKING A HEAVY LIFT

If a ship loads a heavy lift onboard from the jetty with its own derrick or crane, then there is a significant shift of 'G' upwards and outboard towards the derrick head, at the instant the derrick takes the weight. The ship will list towards the load on the jetty.

A VESSEL LIFTING A HEAVY LOCOMOTIVE ONBOARD



THE VESSEL IS INITIALLY UPRIGHT WITH G_0 ON THE CENTRELINE BUT AS THE WEIGHT 'w' OF THE LOCOMOTIVE IS TRANSFERRED TO THE DERRICK HEAD SO THE C OF G RISES BY δG_V AND MOVES δG_T OUTBOARD TO G_1

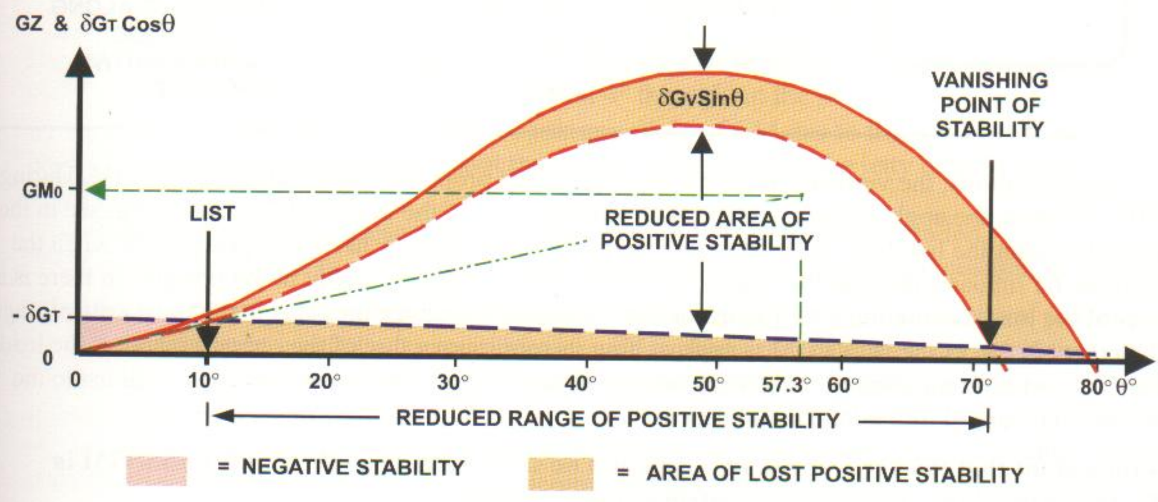


Now $\delta G_V = \dfrac{w\,h}{\Delta_T + w}$ & $\delta G_T = \dfrac{w\,x}{\Delta_T + w}$

AND BOTH BODILY SINKAGE AND LIST REMAIN SMALL, THEN THE METACENTRE 'M' CAN BE CONSIDERED TO REMAIN IN THE SAME POSITION DURING THE LIFT. SO THE LIST 'θ_L' IS GIVEN BY ;-

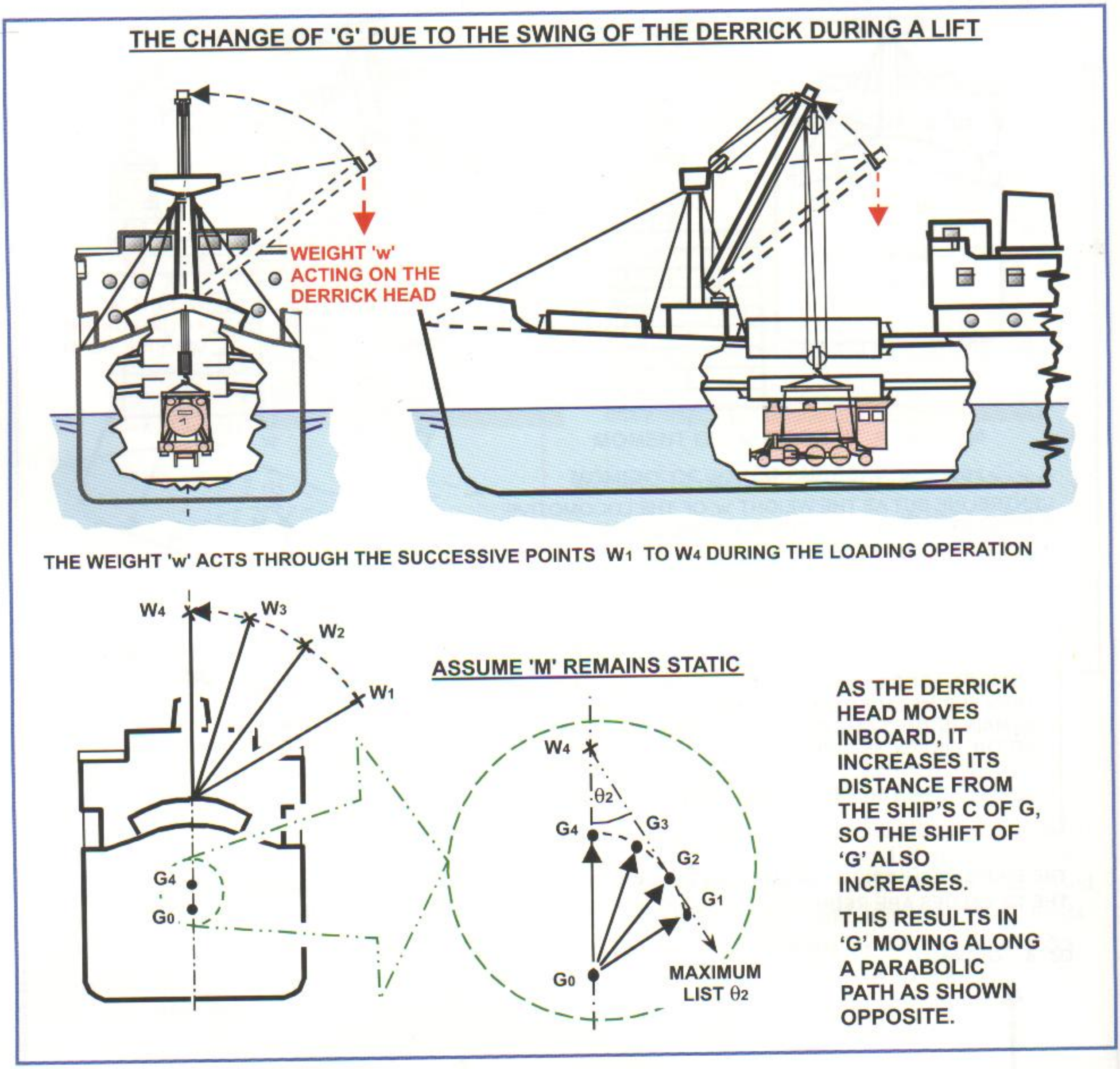
$$\text{Tan } \theta_L = \left(\frac{\delta G_T}{GM_0 - \delta G_V} \right)$$

THE SHIP'S RESULTING STABILITY CAN BE SHOWN ON THE GZ CURVE, IN THE FOLLOWING WAY. THE GZ VALUES ARE REDUCED BY $\delta G_V \sin\theta$, WHILST THE EFFECTIVE BASE IS THE - $\delta G_{VT} \cos\theta$ CURVE



STABILITY CONSIDERATIONS WHEN WORKING A HEAVY LIFT (Cont.)

The list caused by lifting a heavy load on the ship's gear must be kept as small as possible, so as to allow the movement of the lift to be kept under control during the whole operation. If we continue to look at the previous example then the derrick will have to be topped (i.e. raised) as well as swung, in order to move the locomotive across to plumb over its stowage position down the hold. This means that, although the transverse shift in 'G' will decrease, at the same time the rise in 'G' will be increasing. This is shown by the following diagram.

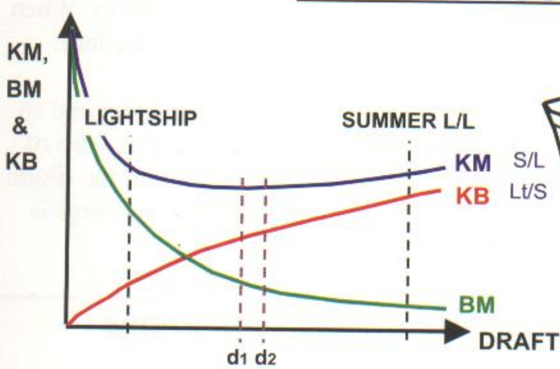


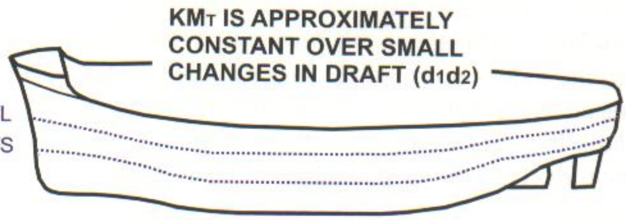
In the example above, the vertical rise of 'G' is a significant proportion of the ship's initial GM. During the lift operation, the angle of list is indicated by the direction of the line 'GM' and so we can see in the sketch above,that the list does not appreciably change until the derrick head is at position '3' when the lift is quite far inboard. Beyond this point, the vessel will come quite rapidly to the upright, so there is a danger of the load developing a swing just before it reaches the centreline. This is the most critical stage of the proceedings, as the ship's GM is approaching its minimum value of the operation and, if the load swings beyond the centreline or the derrick head is moved slightly too far, then the ship will list to the other side and control of the load could be lost.

It is important that the vertical rise in 'G' should be small relative to the ship's initial GM in order to minimise the risk of losing control of the operation.

It is usual to consider that during a heavy lift operation, a ship's stability and GM value are only affected by shifts in the position of 'G'. Loading or discharging a weight will alter the ship's draft and, consequently, will cause changes in the KB and BM values, so the position of the Metacentre 'M' may also move. However, as KB increases so BM decreases so providing the change in draft is relatively small, the Metacentre can be assumed to remain stationary, as the following diagram shows.

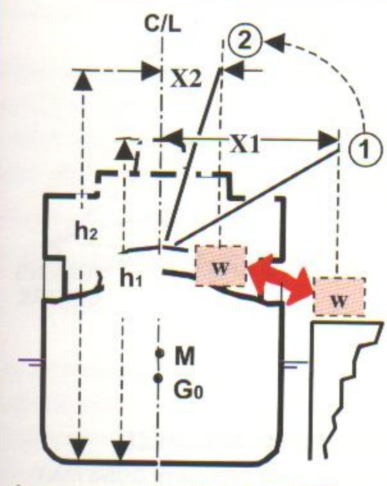
THE KM_T VALUES OF A SHIP SHAPED HULL





IF THE HEAVY LIFT IS A SIGNIFICANT PROPORTION OF THE SHIP'S DISPLACEMENT, THEN THE CHANGE IN THE METACENTRIC HEIGHT SHOULD BE TAKEN INTO ACCOUNT

HEAVY LIFT CALCULATIONS SUMMARISED (FOR SMALL ANGLES OF HEEL)



Δ'T = INITIAL DISPLACEMENT

WEIGHT 'W' IS LIFTED BY DERRICK HEAD 'h' M ABOVE KEEL AND 'X', M OFF THE CENTRELINE, THEN THE SHIFT OF 'G' IS GIVEN BY

$$\delta G_V = \frac{W.h}{\Delta'T + W} \quad \& \quad \delta G_T = \frac{W.X}{\Delta'T + W}$$

FOR A GIVEN GM_0, THE LIST IS GIVEN BY

$$\operatorname{Tan} \theta = \frac{W.X}{\Delta'T + W\,(GM_0 - w.h / [\Delta'T + W])}$$

FOR A REQUIRED MAXIMUM ANGLE OF LIST θ

$$GM_0 = \frac{W}{\Delta'T + W}\left(\frac{X}{\operatorname{Tan}\theta} + h\right)$$

PRECAUTIONS TO TAKE WHEN WORKING A HEAVY LIFT

1) Estimate the GM_0 values at positions '1' and '2' in the above diagram to ensure that the vessel has sufficient stability to keep the list to within about 4° throughout the lift.
2) Press up as many slack tanks as possible to minimise loss of GM due to free surface effects
3) Ensure that the ship is free to list without the hull or fittings fouling the jetty or grounding the midships region.
4) Ensure that the guy line arrangements can control the lift throughout the operation.
5) Avoid raising the derrick or crane more than is required to plumb the stowage position of the lift.

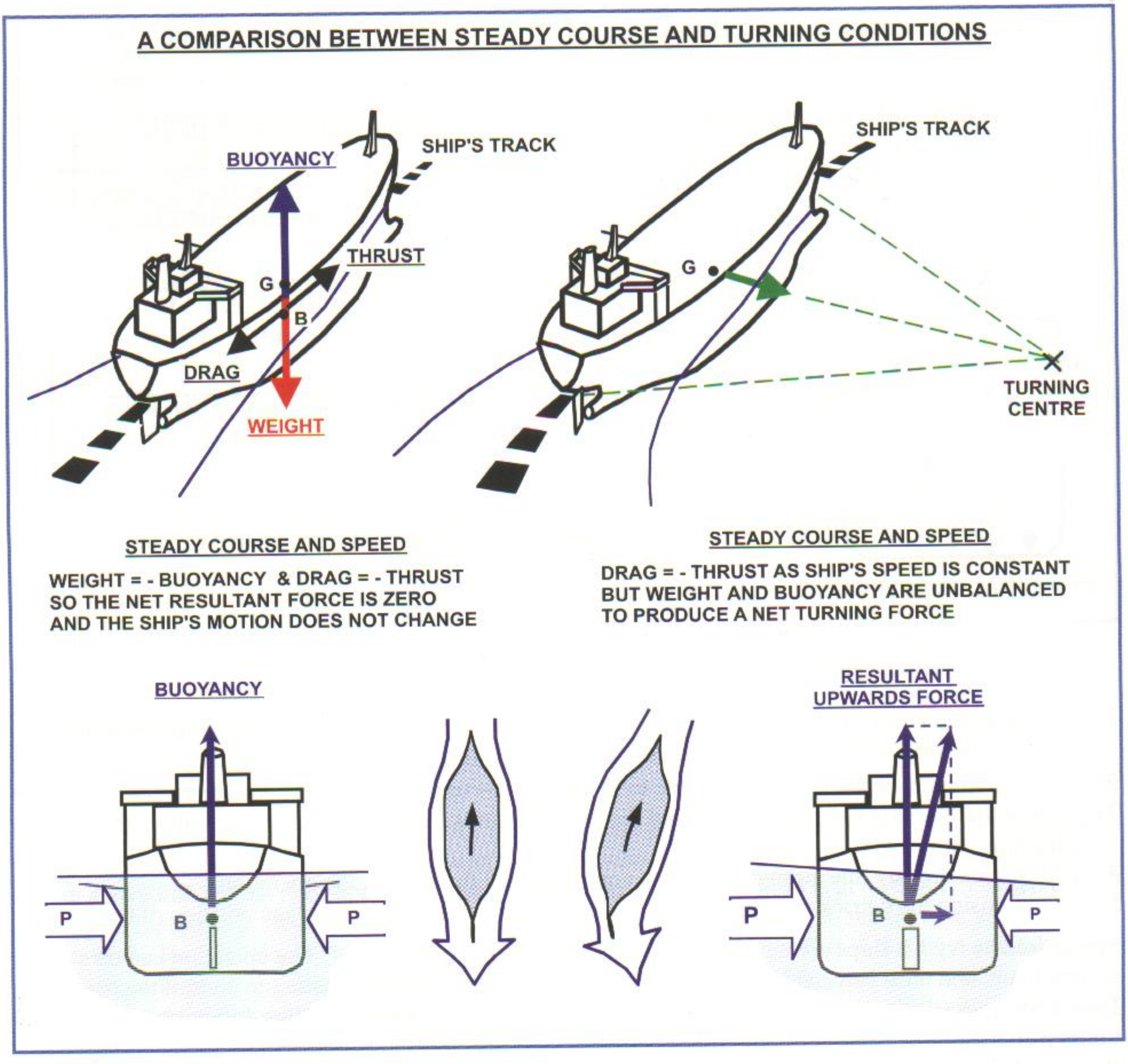
Specialised heavy lift ships are now being built to lift loads of up to 2,000 T. The operation can take several hours and must be carried out in conjunction with sophisticated ballasting to control the list. **Heavy lift operations are frequently carried out at sea, such as when handling buoys or underwater ploughs used in pipelaying. Fishing vessels also land nets that can be heavy, relative to the size of the boat. At sea, the stability requirements and the need to keep control of the lift, are even more essential and Chapter 5 outlines recommended minimum stability criteria for such lifts.**

THE HEELING EFFECT DUE TO TURNING FORCES ACTING UPON A SHIP

Newton's Laws of Motion state that when a body is at rest or moving at constant velocity, there is no overall force acting on it, so when a ship is moving at a steady course and speed, the forces of weight, buoyancy, drag and thrust are in a state of balance. However, if a ship alters course, these forces must be put out of equilibrium to produce a net force to act through the ship's Centre of Gravity towards the centre of the circular arc, which will be the ship's track as it changes direction. The size of this force depends upon the tightness of the ship's turn (i.e. the radius of the arc) and the speed of the ship through the water.

Buoyancy is produced by the pressure of displaced water acting on the submerged hull surfaces. When the ship is moving in a straight line in smooth water, the pressure is equal on both sides of the hull, producing a vertical upwards buoyancy force acting through the Centre of Buoyancy.

When the rudder is turned the ship's initial response is to slew its bow across the flow of water and so create an asymmetric flow around the hull which causes an imbalance of pressure on the two sides of the hull. Pressure increases on the hull to the outside of the turn and is reduced on the hull to the inside of the turn. The resultant upwards force of buoyancy is no longer vertical and a net sideways force is produced.

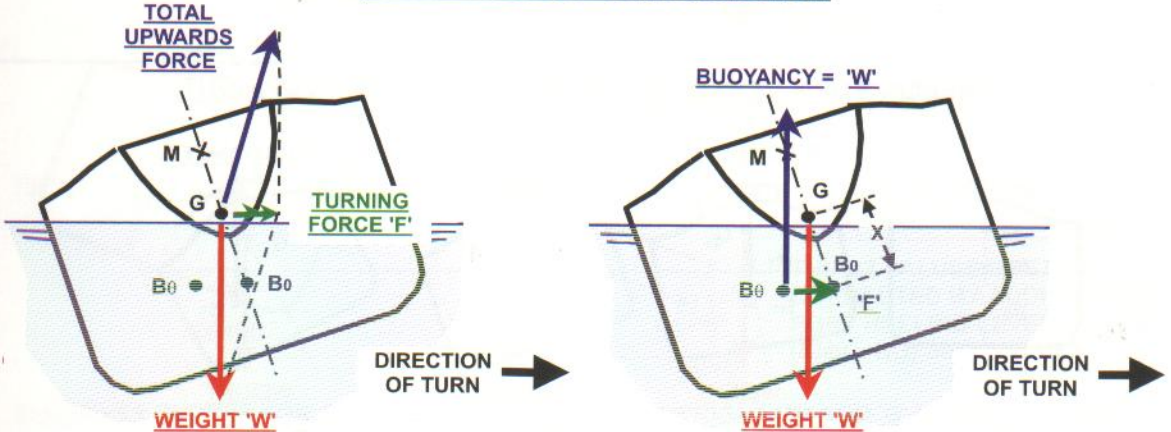


The Force of Buoyancy now no longer acts through the ship's Centre of Gravity, so an outboard heeling moment is created between the forces of Buoyancy and Weight.

DETERMINING THE ANGLE OF HEEL CAUSED BY A STEADY RATE OF TURN

As a ship turns, it heels outboard and the Heeling Moment reduces as the Centre of Buoyancy moves outboard until the Buoyancy Force's line of action passes through the Centre of Gravity. At this point, the Heeling Moment becomes zero and the two forces of Weight and Buoyancy produce a resultant force, acting through the ship's Centre of Gravity and towards the centre of the vessel's turning arc.

STEADY RATE OF TURN CONDITION



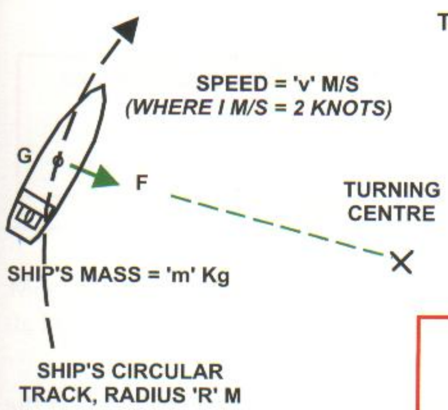
THE TRIANGLE OF FORCES ACTING THROUGH 'G' — THE COMPONENT FORCES

When we are relating a horizontal force, such as the turning force 'F', to the ship's weight 'W', we should return to basic physics and appreciate the difference between Mass, (measured in Tonnes or Kilograms) and Weight, (measured in KiloNewtons or Newtons) which is the force of gravity acting upon that mass. Gravitational and circular motion acceleration must be applied to the ship's mass.

1 Kilogram weighs 9.81 Newtons, where 9.81 m/s^2 is the acceleration due to gravity

The centripetal, or turning force 'F' (in Newtons) is related to the turning radius, (in Metres) and the ship's speed (in m/s). In order to solve the vector triangle, which includes Weight, Buoyancy and the Centripetal Force, we must express all the forces in the same unit of Newtons. Note for ship handling there is a difference between the centre of gravity and the turning centre and the pivot point and pivot centre.

SHIP TURNING IN A CIRCULAR ARC



TURNING, (OR CENTRIPEDAL) FORCE 'F' = $\frac{mv^2}{r}$ NEWTONS

SO $\left(\frac{mv^2}{r}\right) GB_0 = (W \sin\theta)\ GM_0$

HENCE $\left(\frac{mv^2}{r}\right) GB_0 = (9.81m \sin\theta)\ GM_0$

SO $\left(\frac{v^2}{r}\right) GB_0 = (9.81 \sin\theta)\ GM_0$

SO THE ANGLE OF HEEL 'θ' ON TURNING IS GIVEN BY;-

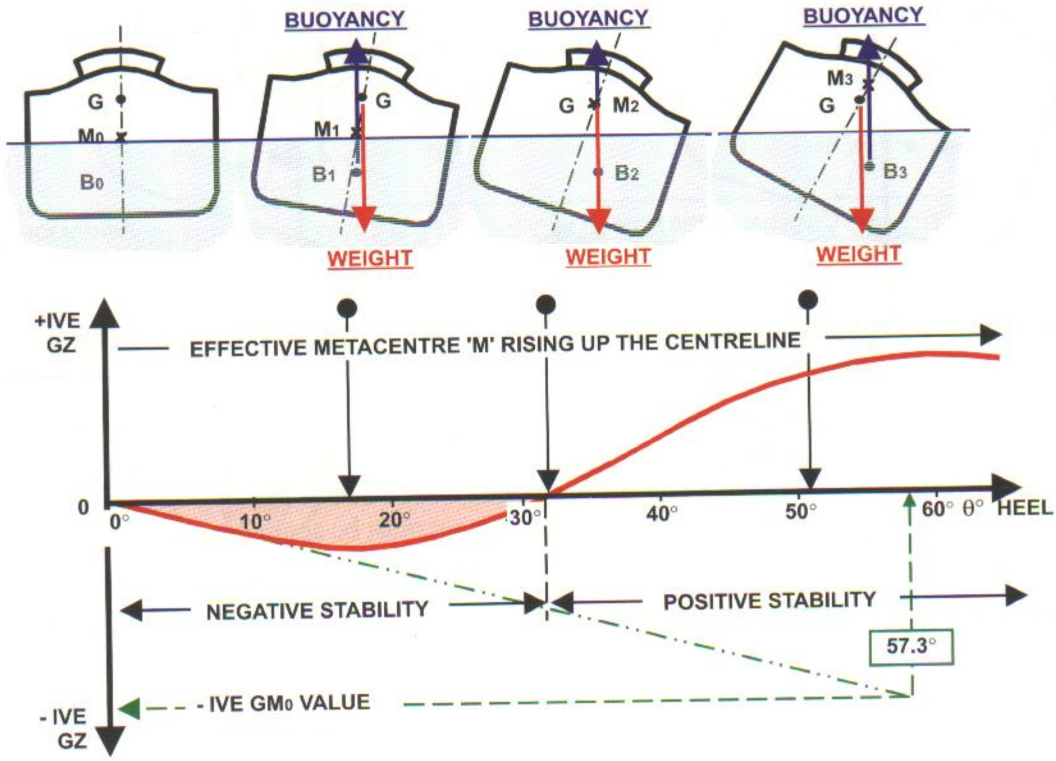
$$\sin\theta = \left(\frac{v^2}{9.81r} \times \frac{GB_0}{GM}\right)$$

WHERE 9.81 m/s^2 IS THE ACCELERATION DUE TO GRAVITY

THE LOSS OF TRANSVERSE STABILITY AND ANGLE OF LOLL

If a vessel losses stability to such an extent that the upright GM becomes negative, then it will experience a capsizing moment which causes it to heel over to either the port or starboard. As the ship heels over, the underwater hullform becomes more asymmetrical and the capsizing moment decreases. Provided that the upright GM value is small, an angle of heel is reached where the GM becomes positive and a restoring moment will act on the ship if it heels beyond this angle. Positive stability is recovered beyond this angle of heel which is known as the **angle of loll**, but the range and amount of positive dynamic stability are greatly reduced.

HEELING OF A VESSEL WHICH IS UNSTABLE WHEN UPRIGHT



THE WALL SIDED EQUATION (CHAPTER 2) SHOWS THAT FOR AN ANGLE OF HEEL θ° ;-

$$\text{VERTICAL HEIGHT OF METACENTRE 'KM'} = KB_0 + B_0M_0 + \frac{1}{2} B_0M_0 \operatorname{Tan}^2 \theta$$

So $$\text{THE RISE IN 'M' AT } \theta^\circ \text{ OF HEEL} = \frac{1}{2} B_0M_0 \operatorname{Tan}^2 \theta$$

So $$\text{AT THE ANGLE OF LOLL } \theta L^\circ, \; -GM_0 = \frac{1}{2} B_0M_0 \operatorname{Tan}^2 \theta$$

SO, THE ANGLE OF LOLL $\theta_L{}^\circ$ FOR A WALL SIDED HULL WITH A NEGATIVE GM_0, IS GIVEN BY;-

$$\operatorname{Tan} \theta L = \sqrt{\frac{2\, GM_0}{B_0M_0}}$$

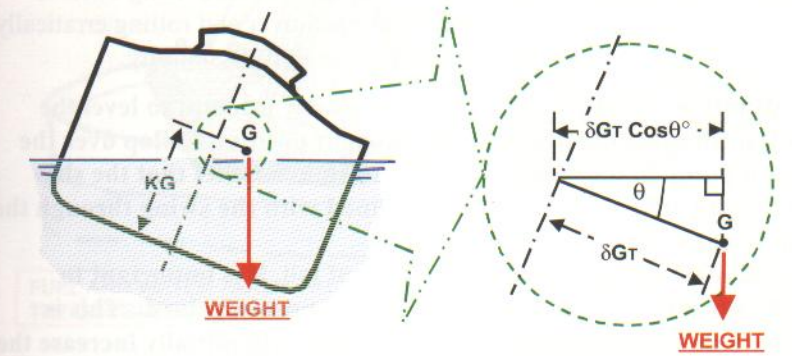
The wall-sided formula, shown above, can be used to estimate the loll angle if the negative upright GM is known. It will, however, over estimate the angle of loll for a ship shaped hull, as the flare fore and aft will cause the Metacentre to rise more rapidly than the wall-sided equation predicts.

The main danger to a ship with a negative GM_0 is that it can flop over to the other side and, if the roll through the upright positon is too violent, the vessel, with its reduced range and amount of dynamic stability, can roll right over, flood and capsize.

THE LOSS OF TRANSVERSE STABILITY AND ANGLE OF LOLL (Cont.)

The situation is futher complicated if the ship also has a list due to an imbalance of weight distribution between the port and starboard sides. In this case, the dynamic stability to the side of the list is even further reduced **but the ship is less likely to flop over to the other side, which would put the ship in greater danger of foundering.**

SHIP WITH A NEGATIVE UPRIGHT GM AND A PORT LIST



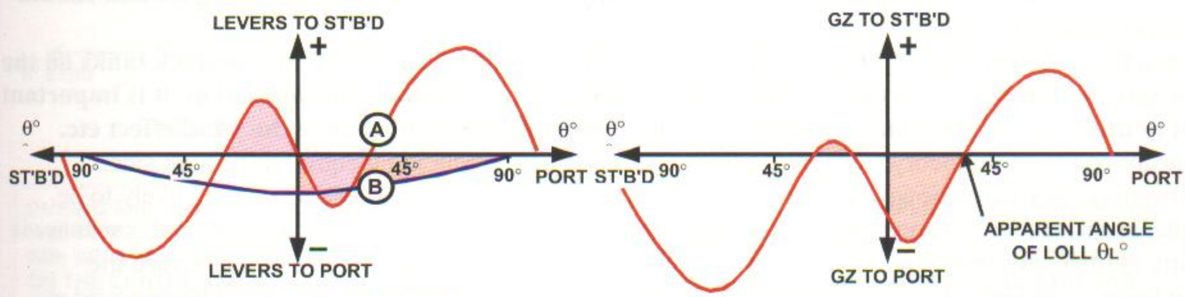
VESSEL HEELED AT θ°

THE HEELING LEVER, DUE TO 'G' BEING OFFSET FROM THE CENTRELINE BY δGT, IS GIVEN BY :-

$$\boxed{\text{LEVER} = \delta G_T \cos\theta^\circ}$$

THE GZ VALUES MUST BE CORRECTED BY ADDING THIS PORT HEELING LEVER TO THE (KMV – KG) Sin θ VALUES

THE LEVER CURVES FOR A VESSEL WITH A NEGATIVE UPRIGHT GM AND 'G' OFFSET TO PORT

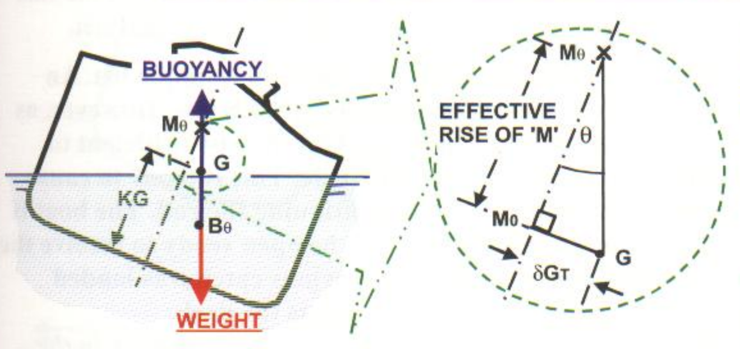


SEPARATE LEVER CURVES — A = KNθ - KG Sinθ CURVE, B = δGT Cosθ CURVE

RESULTANT GZ CURVE — = RANGE OF CAPSIZING LEVERS

> THE APPARENT ANGLE OF LOLL TO PORT, ' θL' IS DUE TO THE COMBINED EFFECT OF THE NEGATIVE VALUE OF GM0 AND THE OFFSET OF 'G' TO PORT, WHICH HAS INCREASED THE HEEL ANGLE AT WHICH THE SHIP SETTLES AND REDUCED FURTHER THE SHIP'S ABILITY TO RESIST RIGHT ITSELF FROM A PORT ROLL.
> HOWEVER, THE LIST PRODUCES AN INCREASED RESISTANCE TO ROLLING THROUGH THE UPRIGHT

COMBINED ANGLE OF LIST AND LOLL WITH A NEGATIVE GM 0 VALUE



THE EFFECTIVE RISE OF 'M' AS THE VESSEL HEEL θ°, IS GIVEN BY;-

$$M_0M_\theta = \frac{1}{2}B_0M_0\tan^2\theta$$

AND THE VESSEL SETTLES WHEN

$$M_0M_\theta = \frac{\delta G_T}{\tan\theta}$$

$$\boxed{\text{SO } \delta GT = \frac{1}{2}B_0M_0\tan^3\theta}$$

$$\boxed{\text{SO THE ANGLE OF LIST IS GIVEN BY;- } \tan\theta = \sqrt[3]{\frac{2\delta G_T}{\tan\theta}} \text{ WHEN THE SHIP'S } GM_0 \text{ IS NEGATIVE}}$$

RECOGNISING AND DEALING WITH A LOSS OF TRANSVERSE STABILITY

Many ships gradually lose stability as fuel and water are consumed from the double bottom tanks during a voyage as bottom weight is reduced and slack tanks create free surface effects. This should be allowed for in the planning of the departure loaded condition but severe weather can prolong the voyage and increase the fuel consumption, whilst some factors, such as icing, will lead to an increase in top weight. Timber ships are particularly prone to water being absorbed by their extensive deck cargo of wood. All this can lead to the ship becoming very tender after a longer than expected period at sea. The rolling motion will become progressively more sluggish and the vessel will tend to pause and 'hang over' to one side before starting the return roll. If the ship eventually remains flopped over and rolling erratically about an average angle of heel, then it has reached the condition of negative upright stability.

It is important not to confuse this situation with that of a simple list as any attempt to level the ship by ballasting the high side, will push it rapidly through the upright position to flop over the other way. If weight has already been pumped into this side in the mistaken belief that the ship required levelling, then it will produce an adverse list which, combined with the swing through the upright, could could cause the ship to capsize.
In conditions of negative upright GM with the ship lying at an angle of loll, it is important to avoid pushing the vessel through the upright until positive stability has been regained. This is achieved by adding ballast to the *low* side double bottom tanks first. This will initially increase the average angle of heel as it introduces a list but it will also tend to keep the vessel heeling to one side only and, as stability is restored, so the list will stop increasing and the rolling motion should become noticeably stiffer.
When it is felt that the ship has regained a positve upright GM and there are no slack tanks on the low side, ballast can be put in the high side to reduce the list. During this operation, it is important not to upset the ship's motion by actions such as altering course to change the wind effect etc.
The above action, in theory, can be supplemented by initially reducing weight on the high side. Jettisoning deck cargo however, particularly from the high side of a ship heeled over, is likely to be quite hazardous for the crew and is a last resort measure. Furthermore, loss of stability in this manner is fairly gradual and should allow action which avoids the GM_0 value going negative, provided the condition is recognised in time. Accumulated top weight, such as ice build up, should be removed as much as possible as it forms or the ship should steer for better conditions before it becomes unstable. Freeing ports on the weather deck should be kept clear.

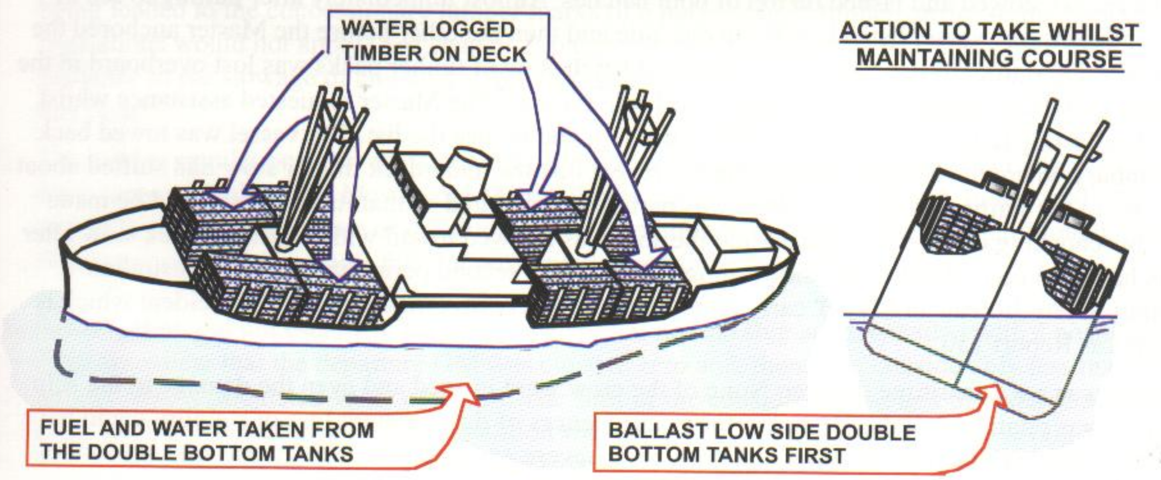
LOSS OF STABILITY WHILST HANDLING A HEAVY LIFT

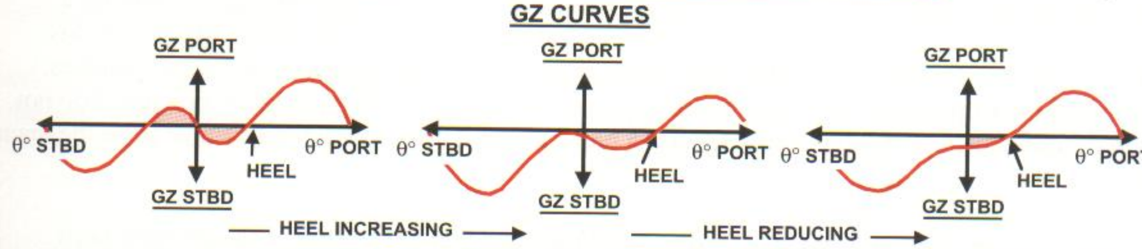
In this situation, the vessel's upright GM can go negative the instant it takes the weight of the lift on its derrick or crane head. It is particularly dangerous if the lift is being handled at sea and fishing vessels are prone to this hazard when landing their catch. The weight of the net is not actually known until it is brought onboard so the crew must be alert to any signs that the load is too heavy for the boat to handle. The weight on the net lines whilst it is being hauled alongside, should give some indication of the size of the catch as will the boat's initial angle of heel when the net starts being lifted out of the water.

Again, it is important to recognise the difference between loss of stability and a simple list. An overweight net may just cause a list whilst it is overside and the derrick head is low. However, as the derrick is raised to swing the catch inboard the additional loss of GM may be sufficient to create upright instability, causing the load to swing over to the other side. This is likely to cause the boat to suddenly lurch over to an opposite list with the risk of continuing the roll. The boat is particularly vulnerable to flooding at this time because the hatch will be open ready to receive the catch. The boat's stability may also have been further reduced if a previous catch was landed onboard with a significant quantity of wet fish that are free to slide across the deck.
If the boat's response to the initial lift indicates that it is too heavy, then it must be put back in the water and landed in smaller amounts.
On no account should the derrick be raised to swing the weight inboard in the mistaken belief that the boat will right itself when the net is over the hatch
Shifting boards should be used to restrict the movement of the wet fish as they are boarded to minimise any weight shift.

EXAMPLES OF LOSS OF UPRIGHT STABILITY AND ACTIONS TO BE TAKEN

1 TIMBER CARRYING SHIP AFTER A PROLONGED VOYAGE





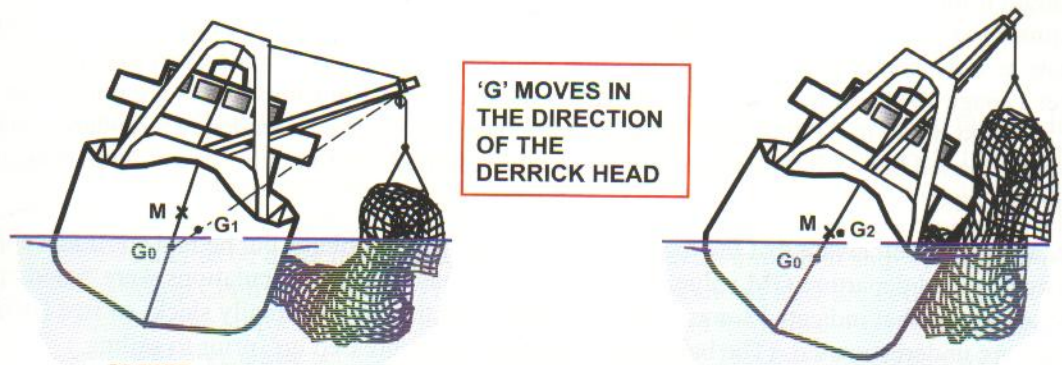
DURING THE VOYAGE, THE SHIP HAS DEVELOPED A NEGATIVE GM_0 WITH 'G' APPROXIMATELY ON THE CENTRELINE. THE VESSEL HEELS OVER TO THE ANGLE OF LOLL

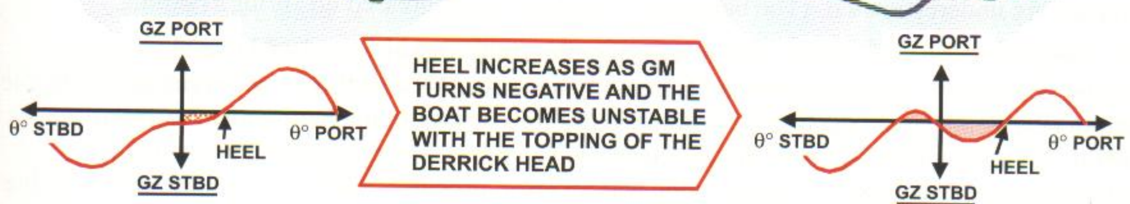
COMMENCE BALLASTING THE PORT DOUBLE BOTTOM TANKS AND THE NEGATIVE GM_0 REDUCES BUT THE HEEL ANGLE INITIALLY INCREASE AS 'G' MOVES TO PORT OF THE CENTRELINE

COMPLETION OF BALLASTING PORT DOUBLE BOTTOMS RESTORES A POSITIVE GM_0. THE SHIP'S ROLL STIFFENS AND THE PORT LIST IS REDUCED

THE VESSEL CAN BE BROUGHT UPRIGHT BY BALLASTING THE STARBOARD DOUBLE BOTTOM TANKS AFTER POSITIVE UPRIGHT STABILITY HAS BEEN RESTORED

2 A FISHING VESSEL LANDING AN EXCESSIVE CATCH





THE DERRICK MUST BE LOWERED IMMEDIATELY AND THE CATCH LANDED IN SMALLER LIFTS

A CASE STUDY INTO THE LOSS OF STABILITY OF THE 'SUN BREEZE'

In August 1999, the new two hatch general cargo ship 'Sun Breeze' sailed from the Australian port of Bunbury, loaded with approximately 6,830 T of pre-bundled sawn timber packs, filling the two holds with the excess stowed and lashed on top of both hatches. Almost immediately after putting to sea in calm conditions, the vessel listed heavily to one side and then the other before the Master anchored the ship with a 25° starboard list. Some of the forward hatch stow of timber packs was lost overboard in the proceedings, which damaged the ship's starboard side railings. The Master requested assistance whilst ordered ballasting to be carried out to stabilise the ship and reduce the list. The vessel was towed back into Bunbury where the hatches were opened to reveal that the underdeck timber stow had shifted about one metre to the starboard side. The cargo was partially discharged so that its stowage could be made secure by 'chocking off' the void spaces and the ship was allowed to sail with a reduced deck stow after repairs had been carried out on the side railings. During this second period in port, the 'Australian Transport Safety Bureau' or 'ATSB' carried out a marine safety investigation into the incident which is published as Report 150.

The affair was not a maritime disaster. None of the crew were injured and even the damage to the ship and cargo was relatively slight. However, the consequences of this apparent loss of stability could have been a lot more serious if the weather had been more severe or the crew not reacted as quickly as they did when things started going wrong. The report is particularly interesting in the way it shows up a series of errors and misjudgements that extend far beyond the decisions made by the Master and his officers. As with most accidents, a dangerous situation arose from the cumulative effect of mistakes, ignorance and over-optimistic decisions, rather than any individual act of gross incompetence. I do not intend to describe the report fully (it extends to 35 pages) but I will outline what I believe to be the main points.

1) The charter agreement appears to have originally intended that all the timber should have been carried under deck but both the owners and charterers greatly underestimated the stowage factor with the result that it became patently clear to the master that the cargo would not all fit in the holds.

2) The Master received neither a copy of the charter nor the promised support from the charterers with regard to stowage arrangements. Furthermore, though the charterer was prepared to authorise payment for deck lashings, no such allowance appears to have been made for securing the timber stowed in the holds. Hold cargo requires stowing so that it is jammed in place. Finished timber products have relatively little friction against a smooth steel tank top (as this vessel appears to have) and so void spaces must be packed out with timber shores strong enough to resist any movement.

3) No weight information was given with regard to the individual packs of timber. The Master was left to estimate the weight by observing the changes in the ship's draft during loading. This is not very accurate as it depends upon still water and good sighting of the marks. If the vessel sags (See chapter 1, page 19) to an increasing extent as it is loaded and the midships marks are not visible (they are most likely to be hidden by the quayside), then there will be a tendency to underestimate the weight loaded between draft observations and the resulting error is likely to accumulate as the load progresses.

4) The Master was well aware that the load could cause stability problems but he had confidence in the Mate's calculated departure GM value of 47 cm. Unfortunately, these calculations were based upon faulty soundings that indicated some tanks were full when they were actually slack, so free surface effects were underestimated. (The ballast tanks had not been 'pressed up', prior to sailing.)

5) The port authorities expressed concern regarding the ship's stability, so the Master decided to transfer fuel to No.1 double bottom tank from higher tanks. Though this would result in the double bottom tank being slack, the loss of GM due to the free surface moment given in the approved stability book would be less than the gain in stability by transferring weight downwards. Unfortunately this listed free surface value was an error, probably due to typing, (the correct value had been used in the sample stability conditions) and the overall effect of the fuel transfer was to decrease the GM. The investigation also found that the lightship KG value was also suspect due to the builders measuring the inclining experiment list with pendulums of very short lengths.

6) The ship was chartered to carry this timber cargo, which was not its usual trade, and there was no guidance in the ship's approved stability book regarding deck stows of lumber. The vessel was not being loaded to the concessionary lumber marks of a purpose built timber carrier so the lumber regulations would not apply (See chapter 5, pages 100 to 102). Nevertheless, these give useful guidance to the stowage of any deck cargo of timber, which proved to be inadequately secured in this case.

7) The lumber regulations also advise on the allowances that should be made for water absorption by a deck cargo of timber. The Master and Mate failed to take this into account and, in fact, did not appear to make any estimate of the ship's arrival condition at the end of the voyage.(though, as the ship never made the passage, this did not actually contribute to the accident.)

The investigation did not establish for certain whether or not the ship actually sailed with a positive GM, but it did determine that the departure GM was close to zero and, therefore, considerably less than the Master and Mate's calculated value of 47 cm. The shift of the considerable weight of underdeck timber combined with the vessel's very tender condition was more than sufficient to cause the severe list and this led to loss of part of the poorly secured deck cargo. If the ship had managed to sail further out to sea before developing problems, then the loss of stability would have had much more serious consequences.

This affair illustrates quite well how the various people involved in operating a ship can contribute to a potentially dangerous situation developing at sea. The problems in this case appear to start with a very poor charter agreement between the shipowners and the charterers which suggests that neither of the two parties had a real understanding of what the ship's carrying capacity actually was. The Master was put in the position of making the best he could out of a bad deal in which essential information, such as cargo weights and stowage factors, were either not supplied or inaccurate. Australian Marine Orders actually require individual packages in excess of 1 Tonne to be marked with their weights. The shipper, as a local firm, should have been aware of this responsibility but failed to comply with the regulation.

The stevedoring company compounded the problem with loose stowage. The Master was aware of this as the third Mate on cargo watch had informed him of the extent of broken stowage and the matter was brought to the stevedores' attention. Consequently, the packing became tighter but the voids in earlier poor stowage still remained to give the timber scope to shift. The Master should have delayed further loading until these voids were packed out but the stevedores also had a responsibility to load the cargo securely.

The Master's confidence in the stability calculations was unfortunately based upon some false data and, in particular, the Mate should have ensured that accurate soundings were made for all the tanks before calculating the GM.

It was also quite disturbing to discover an error in the approved stability book and also that the lightship KG value was derived from a questionable inclining experiment. The vessel was newly built in Japan, which is one of the leading maritime nations with a well established shipbuilding industry supervised by a reputable classification society (NK). However, the Society appears to have accepted procedures that the IMO consider to be quite inadequate for measuring such an important ship's particular as the height of the lightship Centre of Gravity. Page 71 of this chapter shows how the deflection of the test pendulum relates to the ship's KG in the inclining experiment and from this, it is obvious that a very small deflection will give a low accuracy in the KG measurement. The IMO guidelines recommend that a deflection of at least 150mm should be achieved by the experiment. As the list caused by the shift of test weight should also be relatively small (less than 2°), the pendulum used must be quite long.
The experiment carried out by the builders of the 'Sun Breeze' used pendulums of less than three metres in length and calculated the lightship KG from deflection measurements less than 15mm, which makes the results somewhat questionable.

The vessel was not constructed with the strengthened bulwarks and lashing points of a specialised timber carrier so the shipowner should have taken a more active interest in determining the limit of deck cargo that the ship could be reasonably expected to load.

Shipping is a commercial business and the shipowner must run his vessels profitably otherwise there is no money or credit to re-invest in the concern. However, to succeed, the shipping company must have a real appreciation of the practicalities and risks involved in operating its commercial vessels. Ignorance of shipboard operation will result in poor commercial judgement which can often lead to undue pressure for taking short cut measures in order to make any profit at all or at least minimise the financial loss resulting from a bad contract.

This does not relieve the Master of his ultimate responsibility for the safety of the ship but a shipping company's attitude towards its ships and marine staff will have a considerable influence on the decisions taken onboard a vessel. The job of the Master and his officers is to ensure that the ship operates without mishap and this presents a peculiar problem in that they can never be sure that their actions or decisions have definitely prevented something going wrong. If an incident or accident has not occurred, then it is difficult to say for certain why it has not happened or whether or not precautionary measures taken by the ship's officers were really necessary. On the other hand, it is much easier to determine errors of judgement when something does go wrong. A good shipping company should appreciate this and give its staff support when they are in difficult situations.It would appear that in the case of the 'Sun Breeze', the owners and charterers were relatively indifferent to the Master's problems.

Ultimately, though, a Master must be prepared to stand up against undue commercial pressure, if he believes it is going to cause serious risk to his ship. He and his crew can face the very real possibility of being drowned if the vessel is put into danger, regardless of whatever the financial costs of any such accident might be.

When incidents such as this lead to serious accidents, there is always a pressure to increase regulations but this is not always helpful and, in my opinion, can actually be counter-productive. The existing regulations were quite sufficient to ensure safe loading of the cargo onboard the 'Sun Breeze' if they had actually been followed.

Regulation usually creates paper-work that is time consuming and can distract attention from the immediate problems at hand. If existing rules are not being complied with, there seems to be little advantage in creating more regulation. There are increasing demands put onto the ship's officers' time and what with writing reports of past events and producing plans for future ones, there is often little time left to give proper attention to what the ship is actually doing at the present. Increased communications allow charterers and owners to request detailed information much more frequently than in the past with the result that it can be difficult for the Master and ship's staff to keep focused on immediate problems. It is, however, vitally important at times to put these issues to one side and concentrate on the job at hand.

The A.T.S.B. report also expressed some surprise and regret that such a modern ship as the 'Sun Breeze' was not equiped with a stability computer. It was felt that such an aid would have released the Mate and Master from time consuming hand calculations and so may have allowed them to concentrate more on the practical problems of stowage and weight assesment. It was, however, accepted that a computer would still have produced a dangerously misleading departure stability condition if erronous data was inputed so the lack of it was not considered to contibute directly to the ship sailing with inadequate stability.

I would certainly agree from experience that a stability computer is very useful but it should always be remebered that the answers it gives are only as good as the information put into it. There is still the need to be very cautious of any loaded condition that appears to have only marginally aceptable stability.

CHAPTER 5

STABILITY REQUIREMENTS FOR SHIPS OPERATING UNDER SPECIAL CIRCUMSTANCES

SUMMARY

THIS CHAPTER OUTLINES ADDITIONAL STABILITY REQUIREMENTS FOR THE FOLLOWING CATAGORIES OF VESSEL

1) **SHIPS CARRYING PASSENGERS**
2) **SHIPS CARRYING TIMBER CARGOES ON DECK**
3) **SHIPS CARRYING BULK CARGOES, INCLUDING GRAIN.**
4) **WINDHEELING AND SHIPS CARRYING HIGH DECK STOWS OF CONTAINERS.**
5) **SHIPS OPERATING IN HIGH LATITUDES WHERE ICE BUILD UP IS A DANGER.**

CONTENTS

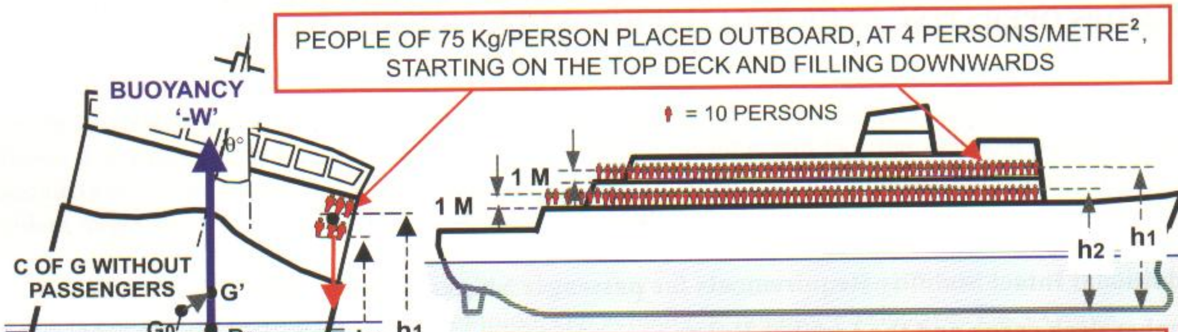
ADDITIONAL INTACT STABILITY REQUIREMENTS FOR PASSENGER SHIPS

Passenger ships (i.e. ships carrying more than twelve fare-paying persons) effectively carry a continually shifting weight as the passengers move about randomly within the public spaces available to them. The IMO code on Intact Stability recommends that a weight of 75 Kg is to be allowed for each passenger onboard and that they should be assumed to be located in the least favourable position for the purposes of calculating the ship's KG. The Code then recommends that any angle of heel, resulting from such a weight distribution, should not exceed 10°.
The IMO Code also specifies a maximum passenger ship turning heel of 10°, where the turning moment is calculated on the basis of the rudder being put hard over at full speed. The code assumes that the speed is maintained and the turning radius is 5 x vessel length.

ADDITIONAL INTACT STABILITY REQUIREMENTS FOR PASSENGER VESSELS

1 - HEEL DUE TO PASSENGER WEIGHT DISTRIBUTION

PEOPLE OF 75 Kg/PERSON PLACED OUTBOARD, AT 4 PERSONS/METRE², STARTING ON THE TOP DECK AND FILLING DOWNWARDS

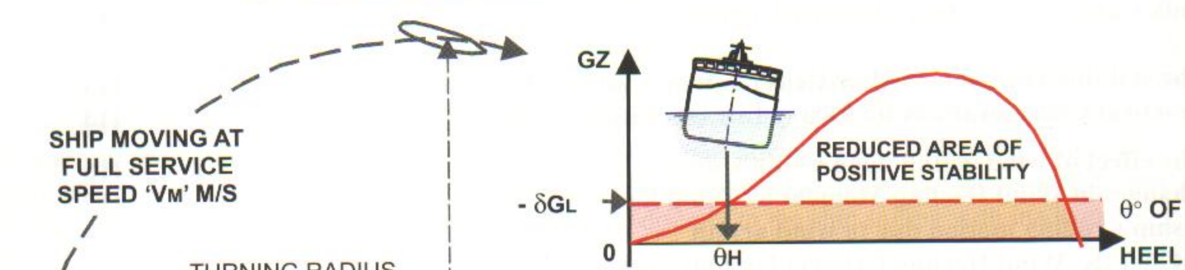


THE ANGLE OF LIST, θ°, PRODUCED BY THE LEAST FAVOURABLE DISPOSITION OF PASSENGER WEIGHT (AS SHOWN ABOVE) MUST NOT EXCEED 10°

THE HEIGHT ABOVE DECKLEVEL FOR PASSENGER WEIGHT CAN BE REDUCED TO 0.3M IF IT IS SEATED

THE IMO CODE REFERS ONLY TO PASSENGER WEIGHT BUT MANY CRUISE SHIPS AND FERRIES CARRY A LARGE NUMBER OF ANCILLARY STAFF (STEWARDS, ENTERTAINERS, SHOPKEEPERS ETC) AND IT WOULD BE WISE TO INCLUDE THESE PEOPLE IN THE PERSON WEIGHT AS WELL

2 - HEEL DUE TO TURNING AT FULL SPEED



TURNING MOMENT $= \left(\dfrac{\Delta_T V_M^2}{9.81 \times 5L}\right)\left(KG - KB_0\right)$

WHERE KB_0 CAN BE APPROXIMATED AS ½ DRAFT, SO

HEELING LEVER, $\delta_L = 0.02\ \dfrac{V_M^2}{L}\left(KG - \dfrac{1}{2}\,DRAFT\right)$

THE ANGLE OF HEEL, θ_H, PRODUCED BY TURNING AT FULL SPEED SHOULD NOT EXCEED 10°
PASSENGER WEIGHT TO BE PLACED AS HIGH AS POSSIBLE BUT ACROSS THE SHIP'S FULL BEAM SO THE VESSEL IS UPRIGHT PRIOR TO THE TURN

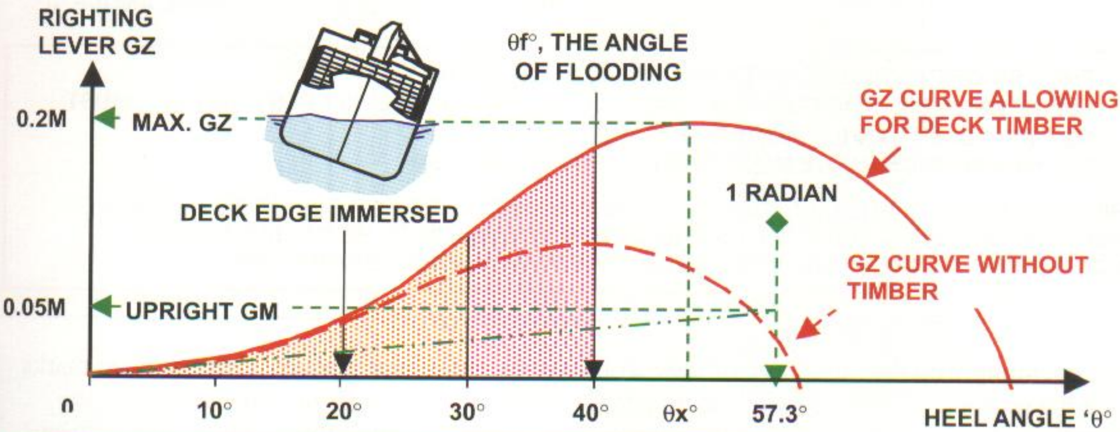
TIMBER DECK CARGO AND THE LUMBER RULES

Deck stows of timber can be considered as additional reserve buoyancy and so, providing a ship meets certain conditions, regarding the ship's construction and the securing of the deck stow, it is allowed to load to reduced minimum freeboard and upright GM value when carrying timber on deck. Furthermore, a proportion of the deck timber volume can be included in the calculations to construct the loaded condition's GZ curve. **This concession, however, does not apply to deck cargoes of wood pulp or products with a similarly large capacity to increase weight by water absorption.**

The '1966 Load Line Convention' gives the reduced lumber freeboards that a ship is allowed, if it meets the 'special timber conditions of freeboard assignment' but the minimum stability criteria are not specified in detail. However both I.M.O. and the U.K. Authorities do recommend the following.

The MCA of the U.K lay down the following minimum stability criteria, which must be met throughout the voyage of a ship loaded to its lumber marks. **KG calculations, made to ensure compliance with these criteria, must allow for 15% increase in the weight of the deck timber, due to water absorption during the voyage, and ice accretion, if the vessel is operating in an area of risk of icing. Consideration should also be given to the effect of strong beam winds.**

MCA-U.K. MINIMUM STABILITY CRITERIA TO MEET LUMBER REGULATIONS



1) **THE UPRIGHT GM VALUE MUST NOT BE LESS THAN 0.05M**

Remaining requirements are the same as for any other vessel

2) **AREA UNDER THE CURVE, 0 TO 30° MUST NOT BE LESS THAN 0.055 METRE RADIANS**
3) **AREA UNDER THE CURVE, 30° TO 40° OR θf° MUST NOT BE LESS THAN 0.03 METRE RADIANS**
4) **AREA UNDER THE CURVE, 0 TO 40° OR θf° MUST NOT BE LESS THAN 0.09 METRE RADIANS**
5) **THE ANGLE OF HEEL OF MAXIMUM GZ VALUE, 'θx' MUST NOT BE LESS THAN 30°**
6) **THE MAXIMUM GZ VALUE MUST NOT BE LESS THAN 0.2M**

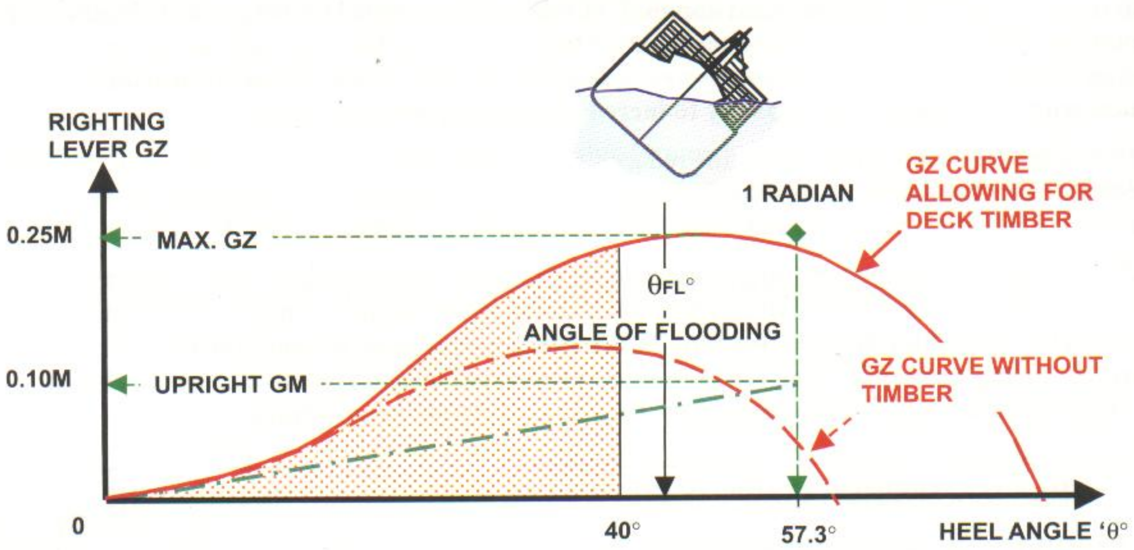
THE STABILITY DATA MUST INCLUDE ALTERNATIVE KN CURVES FOR SPECIFIED HEIGHTS OF TIMBER DECK STOWS. **ONLY 75% OF THE TIMBER VOLUME IS TO BE CALCULATED AS RESERVE BUOYANCY AS 25% MUST BE ALLOWED FOR WATER ABSORPTION**

Note that there are two corrections applied to calculations concerning the deck stow of timber, both being due to wood's ability to absorb moisture.

1) **The weight of deck timber must be increased by 15% of its dry weight**
2) **The volume available for reserve buoyancy is only 75% of the total deck timber.**

The IMO Code of Intact Stability allows a smaller area of positive stability from 0 to 40° (or the Angle of Flooding, θf°, whichever is the lesser) but requires greater minimal values of upright GM and maximum GZ. The code also highlights the point of avoiding excessive stability whilst carrying a deck timber cargo, as this can lead to violent rolling which puts unacceptably high strain upon the lashing points. In general, the upright GM should not be greater than about 3% of the ship's midships beam.

MINIMUM IMO STABILITY CRITERIA TO MEET LUMBER REGULATIONS



1) THE UPRIGHT GM VALUE MUST NOT BE LESS THAN 0.1M
2) THE AREA UNDER THE CURVE, 0 TO 40°, OR TO θFL°, IF θFL° IS LESS THAN 40°, MUST NOT BE LESS THAN 0.08 METRE RADIANS
THE MAXIMUM GZ VALUE MUST NOT BE LESS THAN 0.25M

THE STABILITY DATA MUST INCLUDE ALTERNATIVE KN CURVES FOR SPECIFIED HEIGHTS OF TIMBER DECK STOWS. **ONLY 75% OF THE TIMBER VOLUME IS TO BE CALCULATED AS RESERVE BUOYANCY AS 25% MUST BE ALLOWED FOR WATER ABSORPTION**

A vessel, operating with the advantage of these concessions, must have a set of 'Lumber' load marks etched on the midships region of each side, in addition to the normal loadline marks and meet the requirements laid down in the 'Lumber Regulations' part of the Loadline Rules. These cover various aspects of the ship's construction and the means of stowing the deck timber cargo.

1) The deck cargo is protected from the sea by a raised forecastle and, if under 100m in length, a raised superstructure aft. ('length' and 'superstructure' are defined in Chapter 11)

2) The ship is built with additional longitudinal subdivision in the midships double bottom tanks, in order to minimise the loss of stability through free surface effects due to slack tanks

3) The timber stow extends over the entire effective length of the weather deck, both forward and aft of the centrecastle for those ships with midships accommodation. This ensures that the reserve buoyancy of the stow is evenly distributed along the ship's length and that there is no trimming effect due to the immersion of a partial stow, either near the bow or stern, occurring at the ends of a roll.

4) The deck stow of timber is adequately secured and built up evenly to a height sufficient to provide reserve buoyancy but is not excessive for the voyage weather conditions.

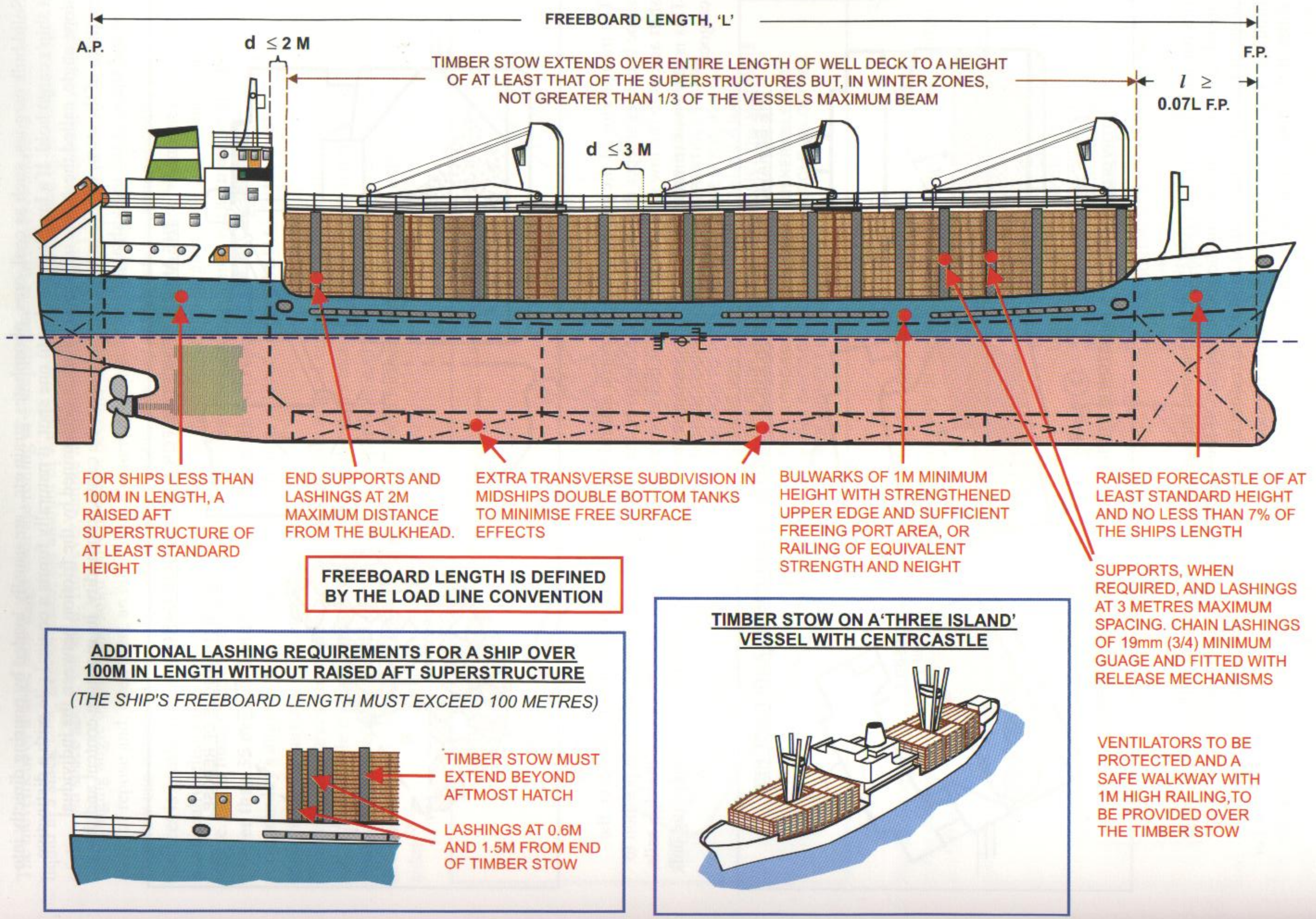
5) The deck cargo should not interfere with the ship's navigation and should be jettisonable.

6) The crew should have safe access across the deck stow.

7) Ventilators should be protected against damage resulting from a shift of the cargo.

These requirements are stated in Regulations 41 to 44 of International Convention on Load Lines (1966), available from the IMO and are summarised in the diagram on the following page. A ship carrying timber on deck that does not meet all of the above conditions must comply with the normal load line and minimum stability criteria.
Regulation 45 specifies how the lumber freeboards are determined and is explained in chapter 11 (See page 282).

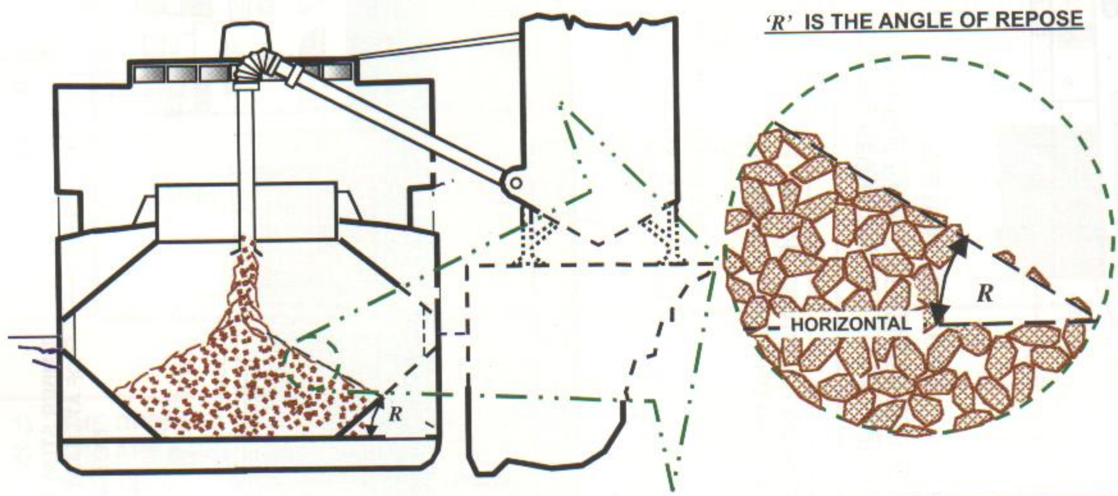
REQUIREMENTS FOR A SHIP LOADING TO THE LUMBER LOAD LINES MARKS



SOLID BULK CARGOES AND THEIR TENDENCY TO SHIFT

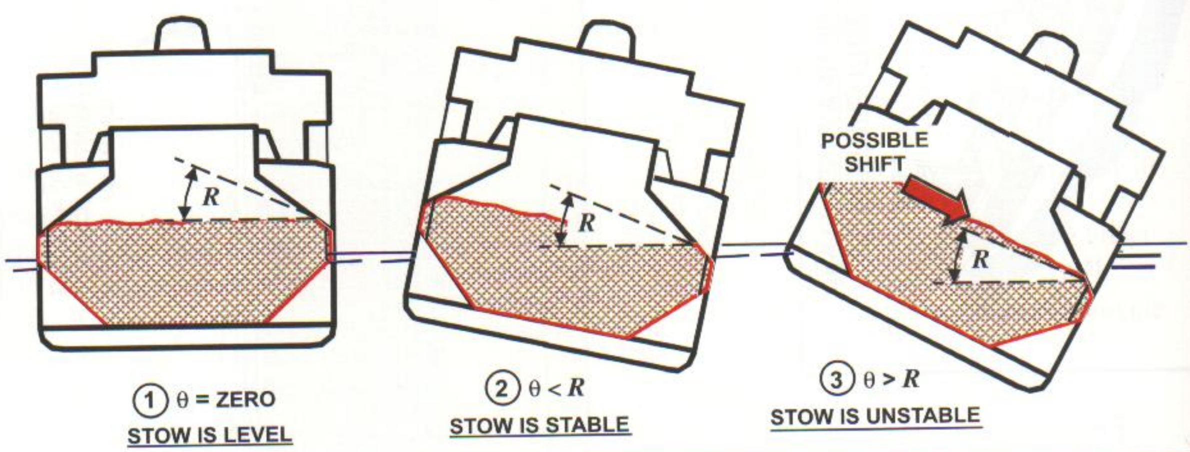
Solid bulk cargoes, such as coal, ores, phosphates, grain etc., are usually loaded by pouring directly into a ship's cargo hold. If a load is poured onto one spot, it naturally forms a conical pile with a distinctive slope angle, called the Angle of Repose. This is determined by the friction between the individual particles of the stow, which, in turn, depends upon the cargo commodity, its moisture content and the size and shape of the individual particles.

THE ANGLE OF REPOSE FOR A BULK SOLID CARGO



The interlocking of the particles will support a natural slope '*R*' from the horizontal. If, however, the slope becomes steeper due to the ship heeling over, the stow will become unstable and is in position to shift across to the low side of the hold. It is therefore essential to level off the stow before the ship sails. This is known as trimming the stow and is an important precaution to be carried out when shipping bulk cargoes.

THE BEHAVIOUR OF A TRIMMED BULK CARGO WITH INCREASING ANGLES OF HEEL



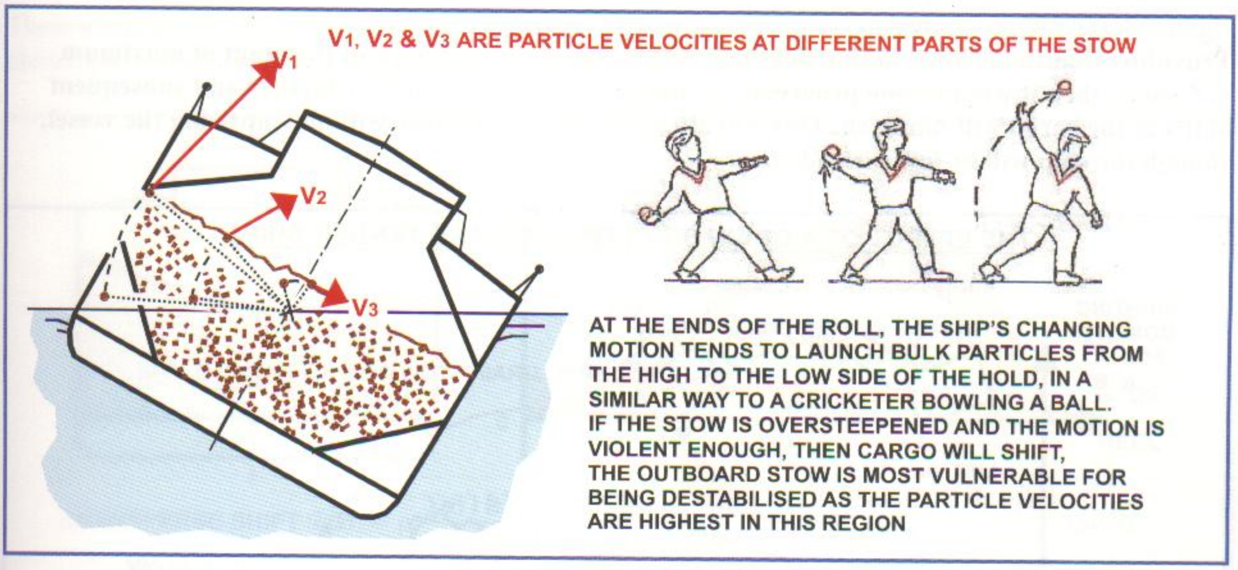
If a particularly heavy roll heels a vessel beyond the cargo's angle of repose, then the stow becomes unstable, as in condition 3 in the above diagram. If the shift occurs, then the ship will roll about an angle of list so the return roll is unlikely to restore the cargo to the level state. Further rolling will produce even greater angles of heel towards the side of shifted cargo. This, in turn, can lead to further shifts of the stow which causes the list to progressively increase. The process will either capsize the ship or reach a stable listed state, depending upon the vessel's transverse stability characteristics.

FACTORS CONTROLLING THE SHIFTING OF A BULK CARGO

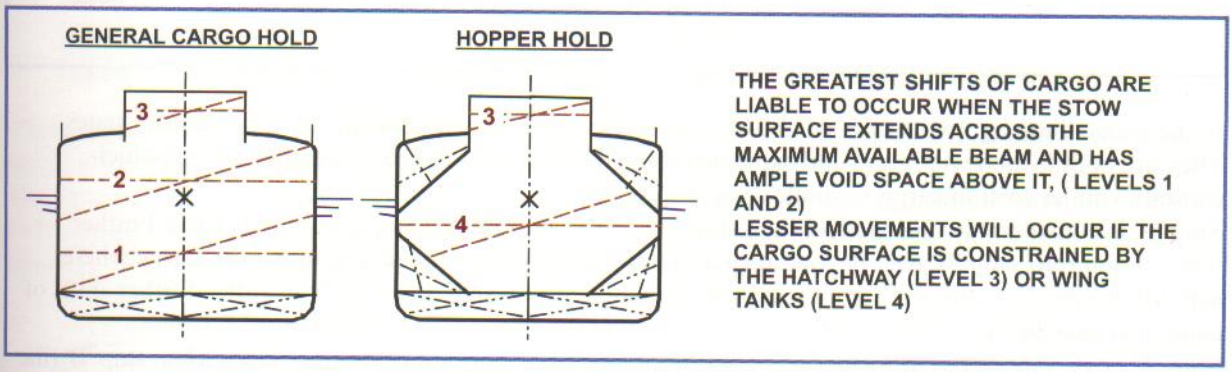
The movement of a solid bulk cargo with the rolling of a ship is similar to the Free Surface Effect of a liquid in a slack tank. In both cases, the amount of weight transferred is determined by the dimensions of the void space above the stow. However, unlike a liquid which moves freely and continuously with the changing angle of heel, the solid stow does not become unstable until the heel angle exceeds the angle of repose and, even then, an additional triggering force is required to actually cause it to move. The stow can withstand being over-steepened without moving, as the static friction between solid surfaces is greater than the dynamic friction. Once it has shifted though, the same argument means that it is equally reluctant to shift back. This behaviour is most marked in cargoes with a large angle of repose. **Solid bulk cargoes are divided into two categories, namely those with an angle of repose greater than 35 degrees and those with smaller angles of repose.**

A commodity's angle of repose and other relevant data is listed in the **I.M.D.G. 'Blue Book',** concerning the carriage of hazardous cargoes.

The triggering forces most likely to move an unstable stow of bulk cargo, occur at the end of a ship's roll. Considerable energy is involved in reversing the ship's motion as it begins its return roll in heavy weather, which reduces the effective weight of the particles and so the friction between them also is decreased. This is most marked in the regions of the hold furthest from the rolling axis, i.e. at the ship's sides.



The above argument is used to zone a cargo space into levels of stow of increasing susceptibility to cargo shifting. This depends upon the width and geometry of the void space available to allow the cargo to shift into.



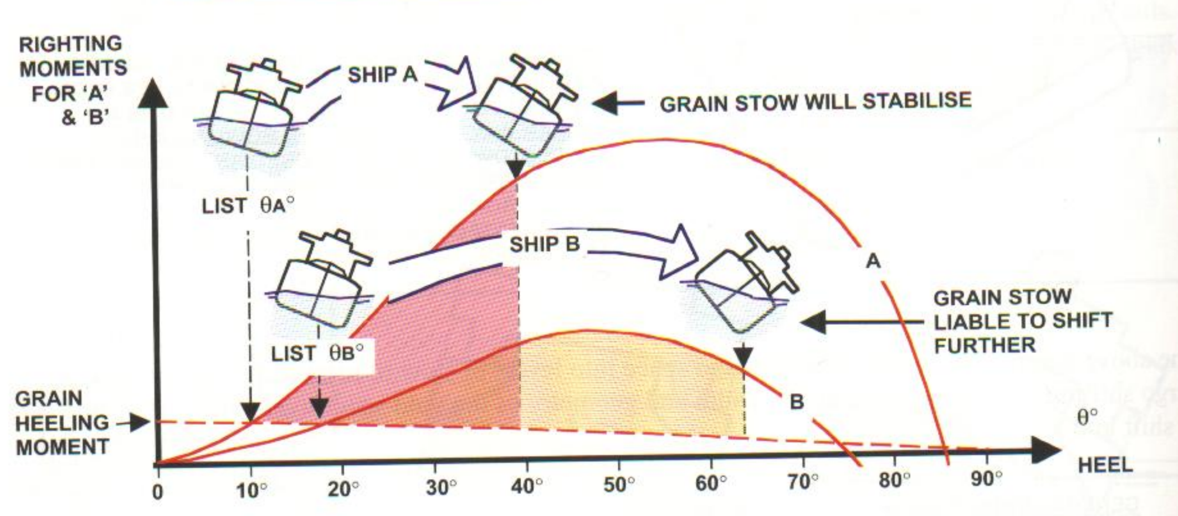
The diagram above shows that hopper shaped holds reduce the extent to which a cargo can shift as well as providing tank space which allows adjustments in the stability by ballasting.

THE CARRIAGE OF BULK CARGOES WITH ANGLES OF REPOSE LESS THAN 35 DEGREES

This is the category of bulk cargoes most liable to shift and includes grain cargoes which have a long history of contributing to the loss of ships by shifting in heavy weather. There is an almost equally long history of regulations for the carriage of grain designed to avoid such dangers. As ships and trading patterns have changed with the years; so have the grain rules. In the past, emphasis was placed on measures to prevent the grain shifting. Temporary wooden partitions had to be erected in the holds and loose grain stow surfaces of partly filled spaces had to be strapped down or secured by loading bagged grain on the top. However, some of these past practices were less than completely successful and today, most grain cargoes are moved in large specialised bulk carriers so the current rules accept that a shift in the cargo can occur but lay down stability criteria to ensure that the ship can survive this.

To understand the rules, we must start by considering the extent by which the cargo will shift in heavy weather. A typical grain stow has an angle of repose of about of 23 degrees whilst maximum angles of heel in the order of 30 degrees are quite possible during severe rolling in very heavy weather. Consequently, in such conditions, we can expect the cargo surface to shift about ten degrees from its level stowed position.. This will produce a list so subsequent rolls will lead to even greater angles of heel and possibly produce even further shifts in cargo, which can eventually cause the ship to capsize. However, a ship's righting moment progressively increases with angles of heel until the maximum righting lever, GZ, is reached.
Providing that the rolling motion does not take the angle of heel beyond the point of maximum GZ value, then it will become progressively harder to heel the ship over further and subsequent shifts in the cargo will diminish. This will allow the stow to stabilise without capsizing the vessel, though the ship will be left with a list.

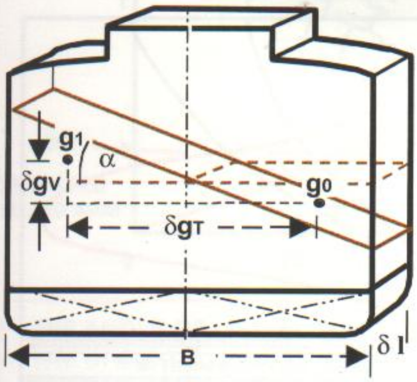
THE EFFECT OF A GRAIN SHIFT ON STIFF AND TENDER SHIPS



In the above diagram, two ships, 'A' and 'B' are similar in all respects except for their stability states. They are both loaded with grain and subjected to the same period of heavy weather rolling which causes a similar shift in cargo for both vessels.
Ship 'B' is considerably more tender than ship 'A', so the shift of cargo causes a greater list. Further wave action rolls ship 'B' well beyond its maximum GZ point into the region of the GZ curve where stability diminishes with further increases in angle of heel. So ship 'B' is likely to suffer further shifts of cargo and capsize.
Ship 'A' is also subjected to further wave action of a similar energy level to that imposed on ship 'B' (the shaded areas under the two GZ curves are the same) but remains in the region of increasing stability with angle of heel, so its cargo will stabilise and the ship should survive the further rolling.

THE GRAIN HEELING MOMENT FOR DIFFERENT STOWS IN A HOLD

We can derive equations for the heeling moments due to a shift of cargo in a rectangular hold by using the same process of geometry that gave us the wall-sided equation and the equation for Free Surface effects. This is summarised below:-



THE WEDGE OF GRAIN OF WEIGHT 'W' AND LENGTH 'δl' IS TRANSFERRED FROM 'g_0' TO 'g_1' SO THAT THE CARGO SURFACE SLOPES AT $\alpha°$ FROM THE ORIGINAL LEVEL STOW.

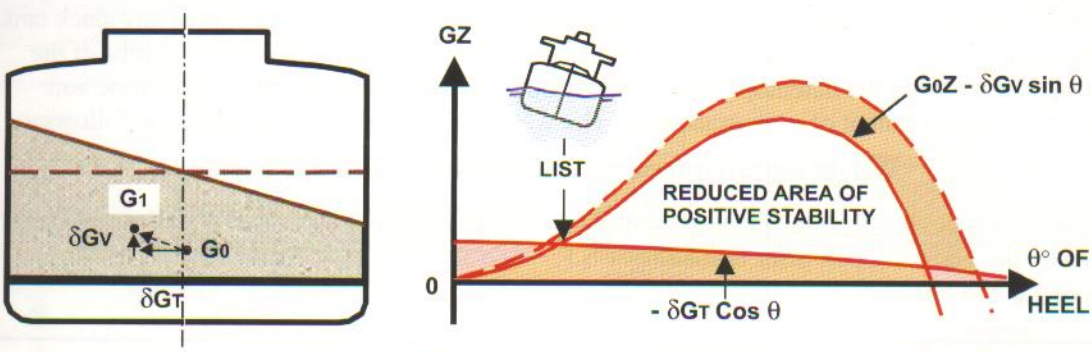
THIS PRODUCES BOTH TRANSVERSE AND VERTICAL HEELING MOMENTS.

TRANSVERSE MOMENT $= W\delta g_T = \rho \dfrac{\delta l B^3}{12} \text{Tan } \alpha°$

VERTICAL MOMENT $= W\delta g_V = \rho \dfrac{\delta l B^3}{24} \text{Tan}^2 \alpha°$

WHERE 'ρ' IS THE BULK DENSITY OF THE GRAIN STOW

These moments are caused by transverse and vertical shifts in the ship's Centre of Gravity and can be plotted on the GZ Curve as a heeling lever and reduction in the righting lever.



TRANSVERSE SHIFT 'δG_T' $= \rho \dfrac{\Sigma \delta l B^3}{12 \Delta T} \text{Tan } \alpha$, & VERTICAL RISE '$\delta G_V$' $= \rho \dfrac{\Sigma \delta l B^3}{24 \Delta T} \text{Tan}^2 \alpha$

WHERE $\dfrac{\Sigma \delta l B^3}{12 \Delta T}$ *IS KNOWN AS THE SUM OF THE VOLUMETRIC GRAIN HEELING MOMENTS*

Studies of grain cargoes and their shifting have lead the authorities to make the following stipulations with regard to the value of angle *'alpha'* to be used in stability calculations.

'Alpha' **= 25 degrees for full width partially filled holds**
'Alpha' **= 10 degrees for full compartments where the stow extends into the hatchway.**

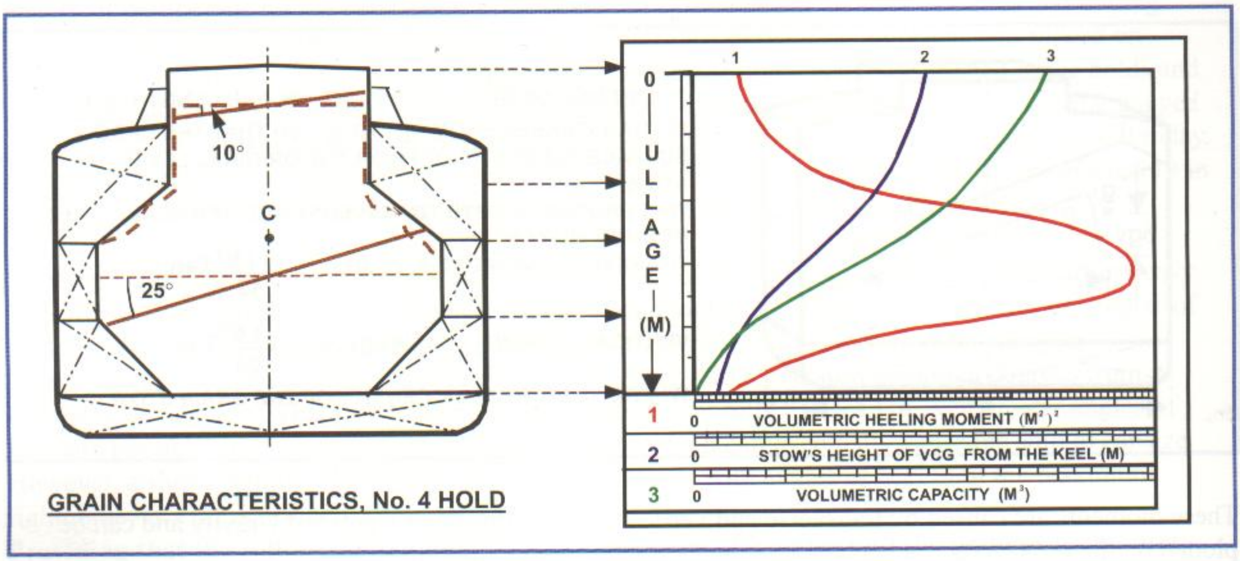
The latter requirement is to allow for settlement of the grain on passage, which will create a void space above it and also to take into account any underdeck voids left in the stow by poor trimming.
The vertical shift in the stow's C of G is less significant than its transverse shift so the rules allow this to be approximated by applying a correction factor to the measured depth of stow in the hold, which effectively increases the calculated loaded KG value for the vessel.

A factor of 1.12 is applied to measured Kg values of partially filled full width holds.
A factor of 1.06 is applied to measured Kg values of holds filled to the hatchway.

These corrections avoid the need to adjust the ship's loaded GZ curve, providing that they have been used in the calculations of its KG value and Volumetric Heeling Moment.

STABILITY CALCULATION WITH REGARD TO MEETING THE GRAIN RULES

Any ship that is to load grain, must have data, regarding the hold spaces, so that the Kg and volumetric heeling moment for each stow can be calculated. This information is supplied by the shipbuilder in the form of tables or diagrams, for each cargo space, as shown by the following illustration

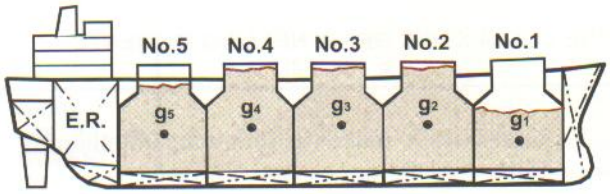


The fluid KG of loaded vessel in the upright condition is calculated in the normal way by taking moments of individual weights about the keel and allowing for free surface effects of any slack tanks. Heights of cargo stows in the holds are obtained by measuring their **ullages** (i.e. the depths of the stow's top surface from the hatchtop.). The ullage values are used to obtain the Kg, volume and volumetric heeling moment of each grain stow. The weight of each stow is calculated as follows:-

Weight of Cargo stow = Volume of stow / Stowage Factor

The value of the stowage factor, S.F. is generally used instead of Bulk Density, 'ρ' and should be supplied by the grain shipper, prior to loading. (S.F. = $1/\rho$)
Values of Kg and Volumetric Heeling Moments are corrected for all partially filled holds.

TOTAL GRAIN HEELING MOMENT 'δG_T'



CORRECTED HEELING LEVER 'δG_T' IS GIVEN BY THE SUM OF THE INDIVIDUAL VOLUMETRIC HEELING MOMENTS DIVIDED BY THE S.F. AND DISPLACEMENT 'Δ'_T'

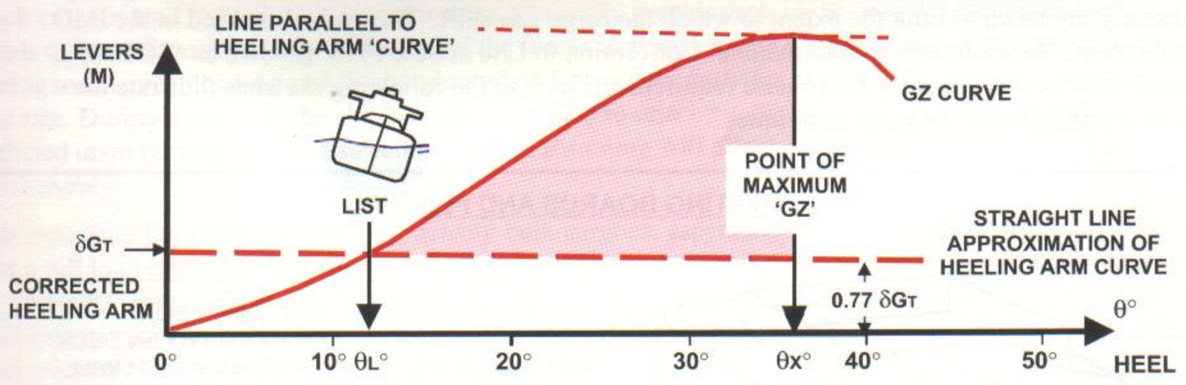
$$\delta G_T = \frac{1.12\,M_1 + M_2 + M_3 + M_4 + 1.06\,M_5}{\Delta'_T \times S.F.}$$

In the sketch above, nos. 2,3 & 4 holds are completely full and so the holds' height of centroid is used for the values of Kg_2, Kg_3 & Kg_4.
Nos. 1 & 5 holds are part filled, so their Kg values are obtained from the measured ullages and corrected with the appropriate factors of 1.12 & 1.06 respectively before being used in the KG calculation.
The total Volumetric Heeling Lever is then applied to ship's GZ curve. Where these two 'curves' intersect, will indicate the angle of heel which the ship will develop after a cargo shift.

If the resulting graph of levers meets certain criteria stipulated by the Grain Regulations then no additional precautions are required to be carried out in order for the ship to make its voyage with the cargo. Modern bulk carriers are built to meet these requirements over a wide range of loaded conditions.

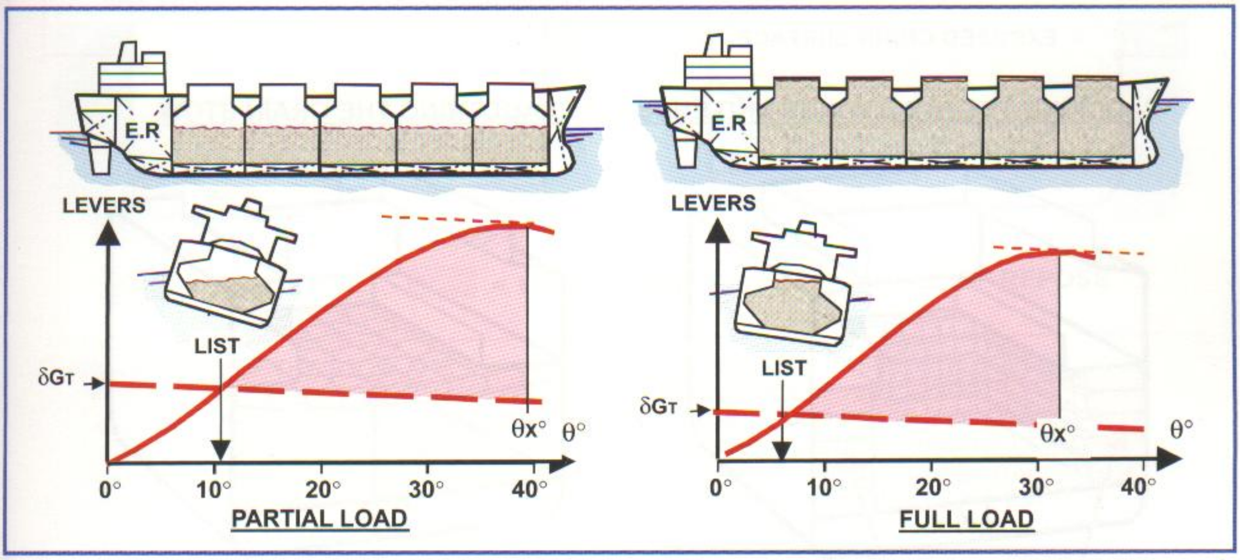
THE STABILITY CRITERIA REQUIRED BY THE GRAIN REGULATIONS

STABILITY CRITERIA REQUIRED TO WITHSTAND A GRAIN CARGO SHIFT



1) THE ANGLE OF LIST, θ_L, MUST BE LESS THAN 12°
2) THE UPRIGHT GM MUST NOT BE LESS THAN 0.3M
3) THE AREA OF POSITIVE STABILITY, θ_L° TO θx° MUST NOT BE LESS THAN 0.075 M-RADIANS
WHERE 'θx' IS THE ANGLE OF MAXIMUM GZ VALUE OR THE ANGLE OF FLOODING, WHICHEVER IS THE LESSER HEEL ANGLE

The Grain regulations allow considerable flexibility in how a well designed ship may be loaded, as the following sketches illustrate;-

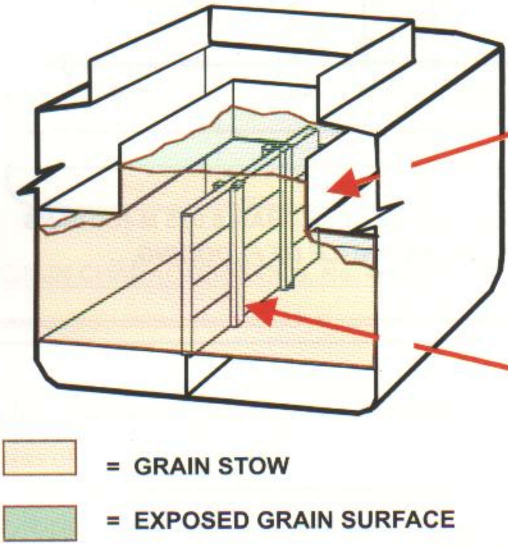


The shift of grain and its effect on a ship's KG value, is greater in the partially loaded vessel. The surface of the stows extend over the ship's entire beam so the grain has a large potential to shift and the weight of shifted grain, relative to the ship's displacement, is also large. However, the freeboard and, hence, the reserve buoyancy are relatively large whilst the KG value is relatively low and these two factors will produce a stiff ship with an extensive range of positive stability. A shift of cargo may result in a large list, close to the permitted maximum of 12 degrees, but the ship's increasing stiffness at larger angles of heel, allows the residual dynamic stability to meet the required minimum value of 0.075 metre-radians. With the fully loaded vessel, the above situation is reversed. Its range of positive stability will be less than that for the part-loaded ship, due to the reduced freeboard and greater KG value. However, the cargo shift is smaller and less significant compared to the ship's displacement, so the resulting list is also smaller than in the case of the part loaded vessel. A well-designed ship can meet the Grain Regulation criteria in both conditions.

ADDITIONAL MEASURES REQUIRED BY THE GRAIN REGULATIONS FOR SHIPS WHICH DO NOT MEET THE MINIMUM STABILITY CRITERIA

Grain is still carried on some vessels, which have insufficient stability unless additional temporary measures are taken to limit the extent to which the cargo can shift. These are described in the IMO publication, 'Amendments to International Convention to Life at Sea, 1974' (known as **'SOLAS'**), which must be consulted for the precise requirements though the following sketches illustrate the general principles of the main techniques

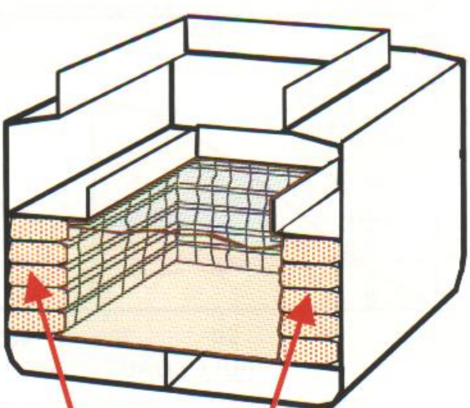
SHIFTING BOARDS AND FEEDERS



THE FEEDER IS A WOODEN OPEN BOX ERRECTED AROUND THE OPEN TWEEN DECK HATCH AND FILLED WITH GRAIN WHICH WILL SETTLE DOWN INTO THE UNDERDECK VOID SPACES AS THE VOYAGE PROGRESSES. IT SHOULD CONTAIN GRAIN TO AT LEAST 2% OF THE LOWER HOLD CAPACITY

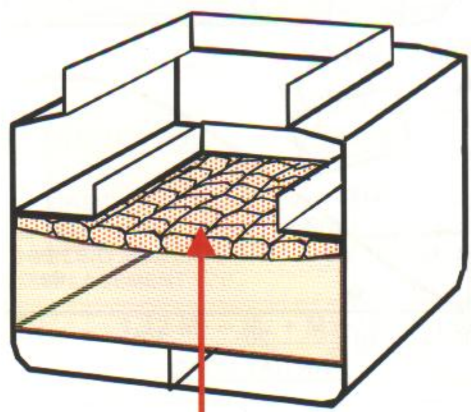
THE SHIFTING BOARD IS A TEMPORARY WOODEN LONGITUDINAL BULKHEAD FITTED ON OR CLOSE TO THE CENTRELINE. IT MUST EXTEND THE FULL LENGTH OF THE HOLD AND BE STRONG ENOUGH TO RESIST A SHIFT IN THE CARGO ACROSS THE FULL BEAM OF THE VESSEL. THE VOLUMETRIC HEELING MOMENT OF THE SPACE WILL BE QUARTERED BY THE SHIFTING BOARDS

BLOCKING THE HOLD WINGS AND ENDS



THE WINGS AND ENDS OF THE HOLD ARE PACKED TIGHT WITH BAGGED OR BALED CARGO SO THE EFFECTIVE WIDTH OF THE GRAIN STOW IS LESS THAN 60% OF THE HOLD WIDTH

SAUCERING THE GRAIN STOW



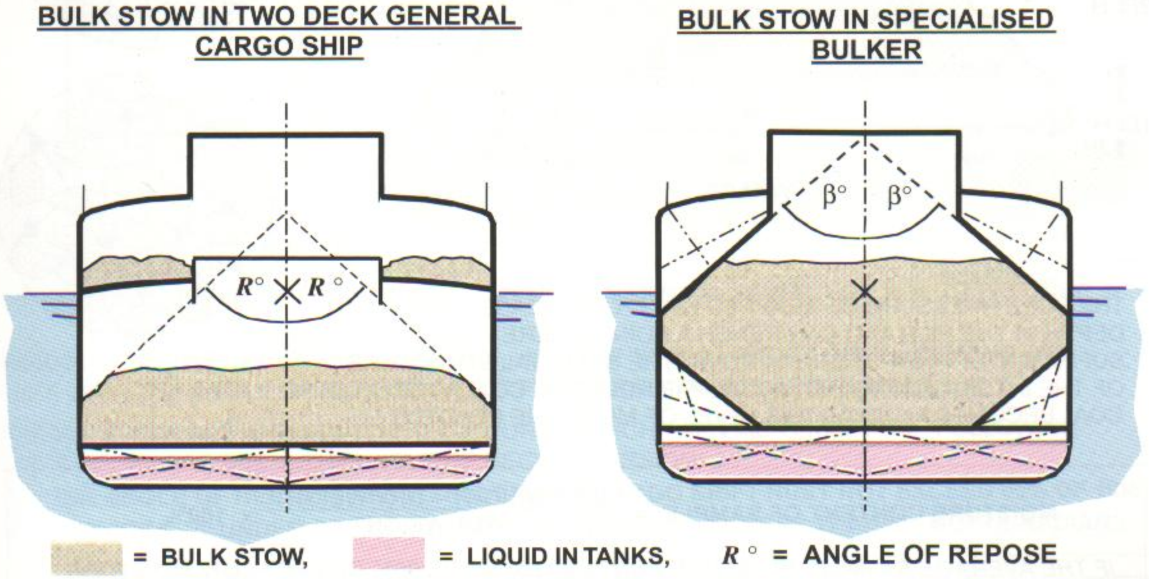
THE TOP SURFACE OF THE GRAIN STOW IS COVERED WITH A TARPAULINE AND THEN OVERSTOWED WITH BAGGED GRAIN TO A DEPTH OF AT LEAST 1.22M (BEAM < 9.14M) OR 1-83M (BEAM >18.43M)

These additional measures tend to be mainly required on smaller older ships, particularly if the vessel has only a part load, and they require considerable time and care to ensure their effectiveness. As with all bulk cargoes, **good trimming and maintaining an upright condition is essential for minimising the risk of a cargo shift.**

STABILITY AND DRY BULK CARGOES WITH LARGE ANGLES OF REPOSE

There are many bulk cargoes, such as metal ores, which have angles of repose greater than 35º and, as such, have a low risk of shifting, **provided that they are adequately trimmed before the ship sails.** Such cargoes tend to have high bulk densities (generally greater than $2T/M^3$) so the ship's structure needs to be sufficiently strong to withstand the forces imposed by the weight of the cargo, particularly during mechanical loading and discharging when heavy bulk particles are fed in or out of the holds at a fast rate. During a passage, the ship's motion can lead to unacceptably high weight being locally inflicted upon parts of the hold structure, though trimming will alleviate this by spreading the weight more evenly.

The main stability problem with high density bulk cargoes, especially for the non-specialised ship, is that a full load often does not fill the hold spaces and so the concentration of bottom weight produces an 'overstiff' ship. This excessive stability results in a quick and violent roll with higher stresses on the hull structure and an increased risk of cargo shifting locally in the stow. The load distribution, including fuel, water etc, should be arranged as much as possible to reduce the upright GM value to a more acceptable level. Tanks can be kept slack to further reduce an excessive GM



THE HEIGHTS OF THE STOW ARE RESTRICTED BY THE STRENGTH OF THE TANK TOP IN THE LOWER HOLD AND THE TWEEN DECK IN THE UPPER SPACE (A STRONG DECK HAS TYPICALLY, A LOAD BEARING CAPACITY OF $10T/M^2$).
THE LOWER HOLD STOW HAS BEEN TRIMMED OVER MOST OF ITS EFFECTIVE BEAM AND THE TWEEN DECK STOW IS JUST SUFFICIENT TO PRODUCE AN ACCEPTABLE UPRIGHT GM
TANKS CAN BE MADE SLACK TO REDUCE THE GM FURTHER IF THIS IS REQUIRED

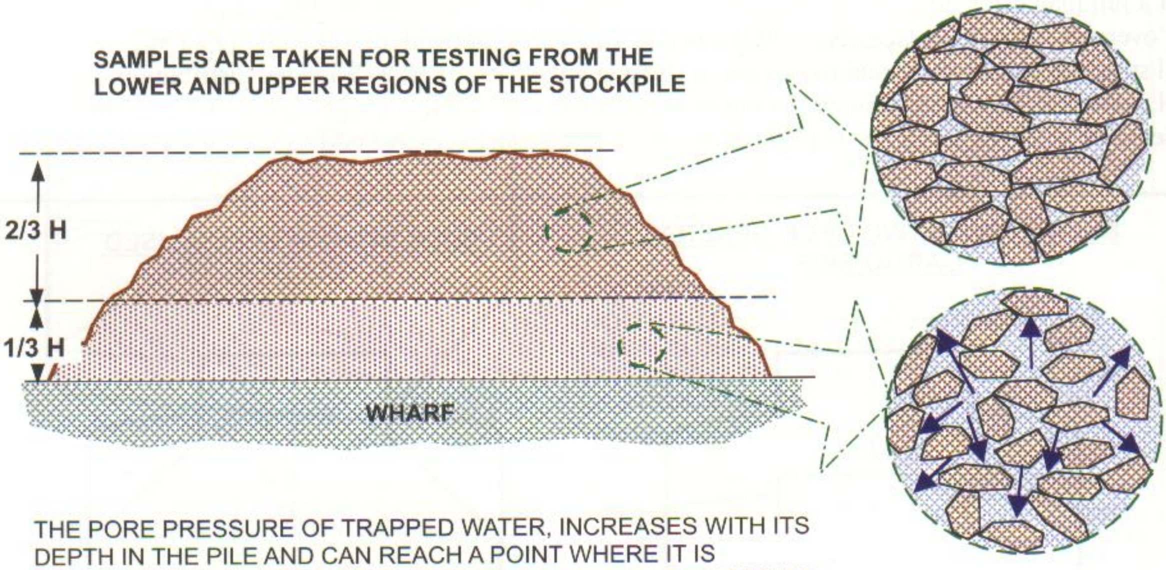
THE VESSEL IS BUILT WITH HOPPER SHAPED HOLDS AND A DEEP STRENGHENED DOUBLE BOTTOM. THIS WILL RAISE A STOWS CENTRE OF GRAVITY, RELATIVE TO THE ORDINARY CARGO SHIP AND THE UPPER WING TANKS PROVIDE BALLAST SPACE HIGH UP, WHICH CAN BE USED TO FURTHER REDUCE THE GM. PROVIDED THAT THE HOPPER SLOPE ' β' IS EQUAL TO OR GREATER THAN THE CARGO ANGLE OF REPOSE, THEN THE STOW WILL SELF TRIM ACROSS MUCH OF ITS WIDTH DURING LOADING

Some cargoes with normally large angles of repose can still carry the risk of shifting under certain conditions. Bulk cement and other similar fine powders, become very aerated during loading which makes such stows very 'free flowing' until the trapped pressurised air in the pores has leaked away. The change in the fluidity is quite remarkable and the MCA-U.K. recommends that a ship carrying such cargo should wait for about an hour for the cargo to settle, before sailing after completion of loading.

BULK CARGOES WITH HIGH MOISTURE CONTENT

The 'fluidity' of some bulk cargoes, such as coal and mineral concentrates, is very dependent upon the moisture content within stow. The water trapped in the pore spaces between particles, produces a pressure which acts to force the particles apart, so when it reaches a certain level, called the **'flow' or critical moisture content'**, the cargo becomes very fluid and is likely to shift when carried at sea. The IMO lists the values of this and other important properties for common bulk cargoes in the **'IMO Code of Safe Practice for Bulk Cargoes'**, and recommends that the **Safe Transportable Moisture Limit is 90% of the Flow Moisture Content.**

SAMPLING A STOCKPILE OF COAL



THE PORE PRESSURE OF TRAPPED WATER, INCREASES WITH ITS DEPTH IN THE PILE AND CAN REACH A POINT WHERE IT IS SUFFICIENTLY HIGH TO BREAK DOWN THE SKELETAL STRUCTURE OF THE INTERLOCKING INDIVIDUAL PARTICLES OF COAL. THE COAL BECOMES FLUID AT THIS LEVEL OF MOISTURE CONTENT

$$\text{MOISTURE CONTENT OF SAMPLE} = \frac{\text{WET WEIGHT} - \text{DRY WEIGHT}}{\text{WET WEIGHT}} \times 100\%$$

IF THE AVERAGE SAMPLE MOISTURE CONTENT EXCEEDS 90% OF THE FLOW MOISTURE POINT, THEN THE CARGO CANNOT BE TRANSPORTED WITHOUT TAKING SPECIAL PRECAUTIONS AGAINST SHIFTING

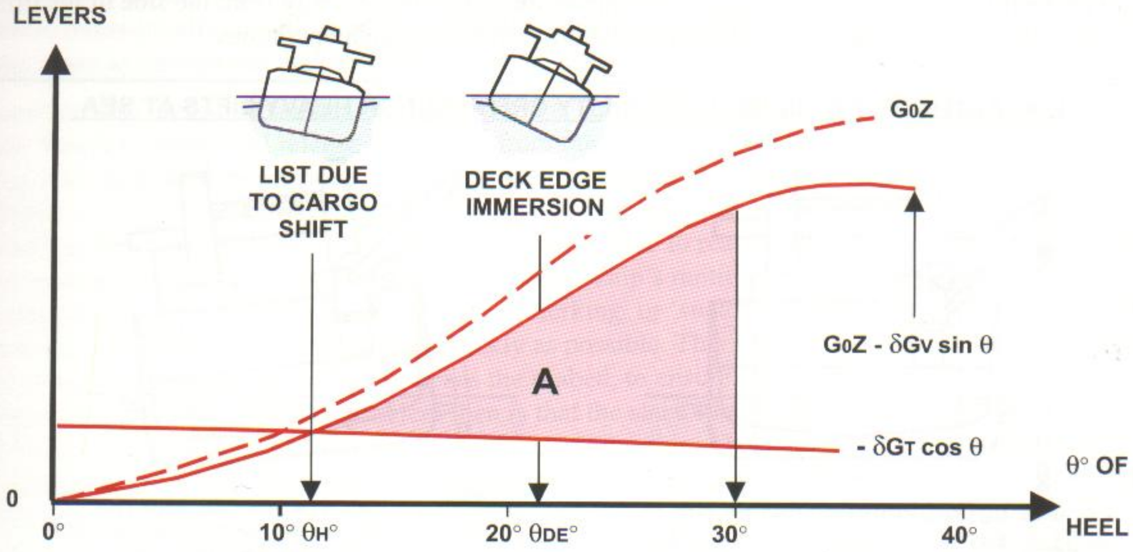
Material of small particle size is more prone to turning fluid due to the restricted drainage but, unlike the air trapped in cement, water will not leach out in an hour or so and the stow can remain fluid for several days. Any open stockpile that has been exposed recently to rainfall should be considered as suspect and proper testing of a range of samples is the only sure way of determining whether or not a cargo is safe to load. A quick indication of excessive moisture content can be seen if a can full of sample cargo turns wet on the surface after being vigorously knocked down several times on a hard surface. (This is similar to the effect seen on beaches where the sand is saturated and turns wet on the surface if agitated by a person repeatedly stamping their foot on it). Guidance for the testing and loading procedures of such cargoes, is given in the U.K. merchant Shipping Notice No. M746, entitled **'The Shipping of Mineral Products in Bulk'**

If the testing of samples taken from both the upper two thirds and lower one third of a stockpile, indicate that the moisture content is below the safe transportable moisture limit, then the cargo can be loaded with no special stability considerations. However, hold bilge lines should be checked for proper functioning, before loading into any space, so that moisture which collects in the bottom of the stow can be pumped out during the voyage.

BULK CARGOES WITH HIGH MOISTURE CONTENT (Cont,)

Bulk cargoes, other than grain, that have a risk of shifting due to their moisture content, can be loaded if the ship's stability meets certain minimum criteria.

MCA-U.K. MINIMUM STABILITY CRITERIA FOR BULK CARGOES LIKELY TO SHIFT



TRANSVERSE LEVER, $\delta G_T = \rho \dfrac{\Sigma\, \delta\, l\, B^3}{12\Delta T} \tan\alpha$, **& VERTICAL LEVER,** $\delta G_V = \rho \dfrac{\Sigma\, \delta\, l\, B^3}{24\Delta T} \tan^2\alpha$

WHERE 'ρ' IS THE CARGO BULK DENSITY, ΔT = THE SHIP'S LOADED DISPLACEMENT AND 'α' = 30° - THE ANGLE OF REPOSE OR 20° FOR CARGO THAT IS ABOVE THE S.T. M. L.

THESE LEVERS ARE APPLIED TO THE GZ CURVE WHICH THEN MUST MEET THE FOLLOWING CRITERIA, WHERE θ_{DE} IS THE ANGLE OF DECK IMMERSION

1) **THE ANGLE OF HEEL θ_H, RESULTING FROM A CARGO SHIFT NUST NOT EXCEED 65% OF θ_{DE}**
2) **THE AREA .'A' UNDER GZ CURVE, θ_H TO 30° MUST NOT BE LESS THAN 0.1 METRE-RADIANS**

The criteria are similar to those required for grain, though the bulk densities of such cargoes as mineral concentrates, can be considerably higher and it may be necessary to use longitudinal shifting boards or storage bins in order to restrict the heeling moment and meet these criteria.

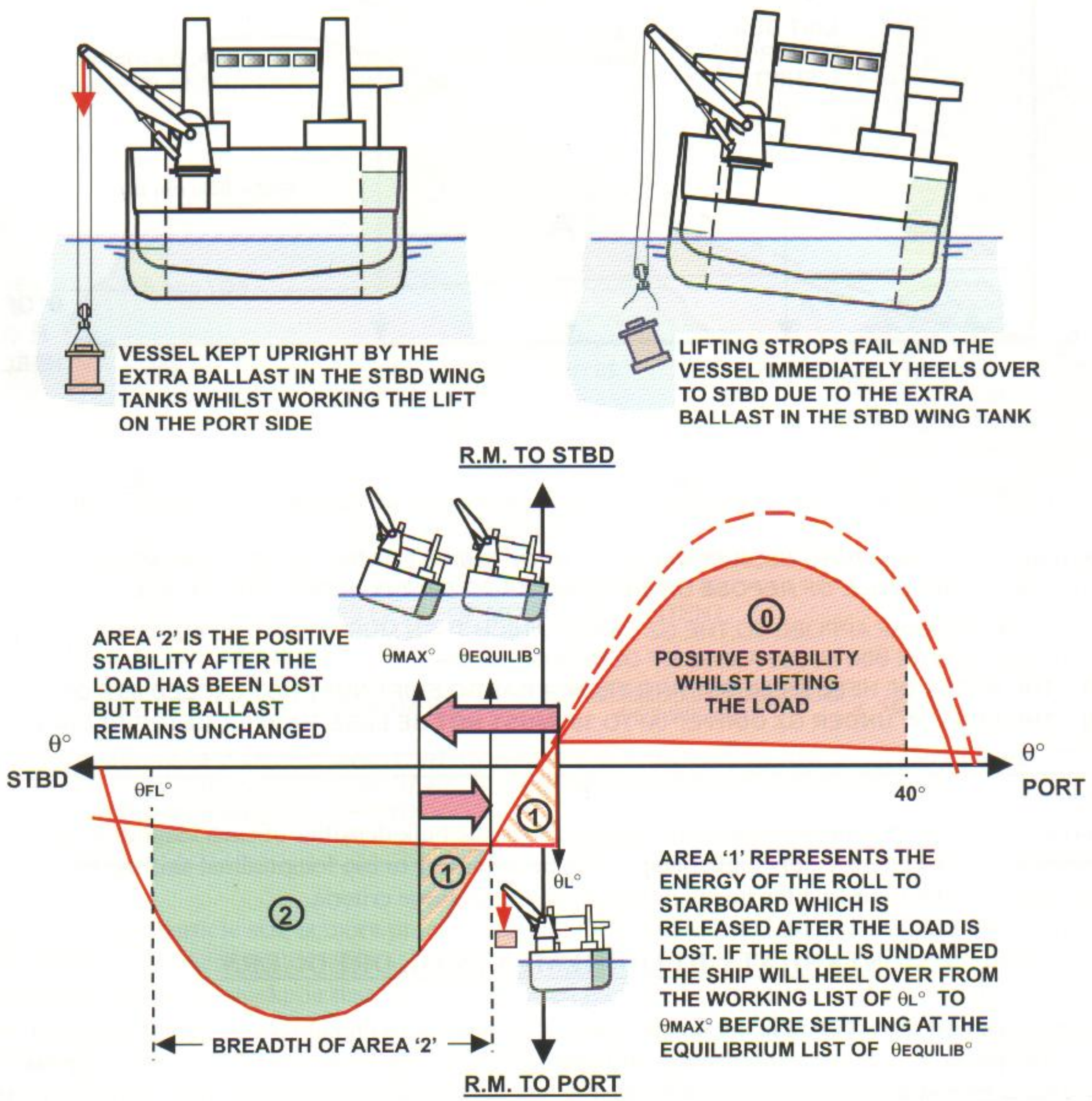
STABILITY REQUIREMENTS OF DREDGERS

Dredging is carried out either to maintain navigation channels, or to prepare a seabed site for a civil engineering project, such as pipeline laying, or to excavate building grade materials, such as gravels etc. By the nature of their work, dredgers invariably handle wet bulk cargo and so must comply with the above criteria on the basis that the cargo has a moisture content above the safe transportable limit. Furthermore, loading is carried out at sea and the precise properties of the cargo may not be known before it taken onboard, though sampling and testing should take place by conducting a site survey prior to the dredging operation.

Dredgers vary considerably in design type and size with some types operating with open hatches whilst at sea. The normal operational practice is to fill the holds until they spill over and, consequently, the risk of overloading such ships is considerable. The MCA of the U.K. lays down comprehensive guidelines to the assessment of dredgers' seaworthiness and stability, in their publication, **'Load Line-Instructions for the Guidance of Surveyors'**

In Chapter 4, we looked at the stability implications of working a heavy lift and these are considerably more critical when such a lift is being carried out at sea, such as in off-shore oilfield construction work. Many vessels involved in this kind of operation are equipped with a quick acting ballast system which can pump water between wing tanks in order to counter the changing list that would occur during the operation. Such a system, however, cannot expect to cope with the instantaneous loss of the load due to failure of the lifting gear, which will cause the ship to immediately roll away from the side of the lift. The U.K. 1968 Loadline Rules give minimum stability criteria for such operations

U.K. AUTHORITY'S MINIMUM STABILITY CRITERIA FOR HEAVY LIFTS AT SEA



AREA 2 IS MEASURED BETWEEN THE FIRST AND THE SECOND INTERCEPTS OF THE GZ CURVE WITH THE HEELING MOMENT OF THE BALLAST OR BETWEEN THE FIRST INTERCEPT AND THE ANGLE OF DOWNFLOODING, 'θ_{FL}°', WHICH EVER IS THE SMALLER

AREA 0 IS MEASURED BETWEEN θ_L° AND THE SECOND INTERCEPT OF THE HEELING MOMENT DUE TO THE BALLAST AND LOAD OR 40°, WHICHEVER IS THE SMALLER

1) AREA 2 MUST BE EQUAL TO OR GREATER THAN AREA 1 + 0.037 METRE- RADIANS
2) THE HEEL ANGLE '$\theta_{EQUILIB}$°' MUST BE LESS THAN THE ANGLE OF DECK EDGE IMMERSION
3) AREA 0 MUST BE EQUAL TO OR GREATER THAN 0.10 METRE- RADIANS

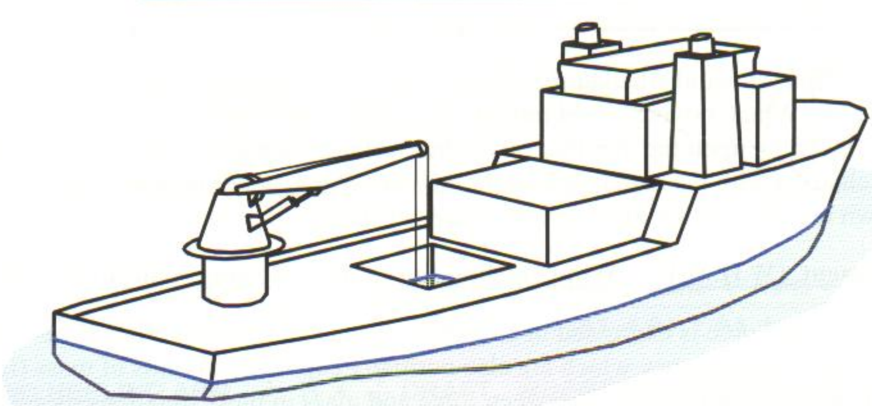
PRACTICAL CONSIDERATIONS FOR HEAVY LIFT OPERATIONS AT SEA

Meeting the stability criteria, as given on the previous page, is only one aspect of the planning required for a heavy lift operation to be carried out at sea. Most oilfield construction work will require that such lifts are placed quite precisely on the seabed to position tolerances of less than a metre, so the ship's own position keeping must be of a similar order. Usually, only vessels equipped with a 'Dynamic Positioning' or D.P. capability can carry out this type of work. Such a system requires the ship to be fitted with side acting thrusters at the bow and stern as well as a main propeller. The entire propulsion syetem is automatically co-ordinated by a computer, which is also receiving position data from various sources (such as underwater acoustic beacons, short-range radar fixing or satellite navigation).

The most critical part of such a heavy lift operation at sea, is the actual landing of the load on the seabed. This, of course, will release the weight from the crane head and lead to the same sequence of heeling over, described on the previous page. If the crane has a sophisticated tension control and the sea conditions are very calm, it may be possible to gradually release the weight over a minute or two after the load has been landed and so allow the ballast system time to react to the changing heeling moment. But in most situations, this is probably not an option. A ship's motion in a seaway is quite difficult to fully compensate for and if there is any chance of 'jerking' or 'snatching' the lift once it is landed, then the best option is to land and unhook it as quickly as possible. This requires paying out sufficient slack on the crane wires as soon as the load touches the seabed, to ensure that any movement of the crane head does not effect the load. The problem then is that the slack block and hook can damage or entangle itself with the lift as it rises and falls with the motion of the ship. This danger can be reduced by having long lifting wires between the hook and the load itself, but such an arrangement is limited by how high the crane jib can be topped to lift the load over the ship's side. Consideration must also be given as to how the load is actually released from the hook. The best method would use remote control, such as release mechanisms triggered by an underwater acoustic signal but in some cases, divers may also be involved and a lot of thought must be given to minimise the risk to them from swinging blocks etc.
All these aspects of the operation must be considered before carrying out a heavy lift operation at sea and an appreciation of the limits of the ship and its equipment is vital before deciding upon the maximum sea conditions in which the operation can take place.
The proof testing of lifting equipment actually used at sea must take the 'dynamic' loading into account. The effective weight of any mass hanging off a crane at sea can easily be momentarily increased by 20% due to the ship's motion. Loads should be lifted with equipment that has been tested with weights that are at least 30% in excess of the load's own static weight

One way of reducing the ship's reaction to the release of the load, is to work the lifting operation through a 'moonpool', which is vertical trunking in the centre of the ship and is open to the sea. This keeps the ship's centre of gravity close to the centreline and so limits the range of heel that the ship will go through when it lands the load on the seabed, though the lift will still be effected by rolling, pitching and bodily heave. Dive support vessels are generally built with a moonpool dedicated for the operation of the dive bell.

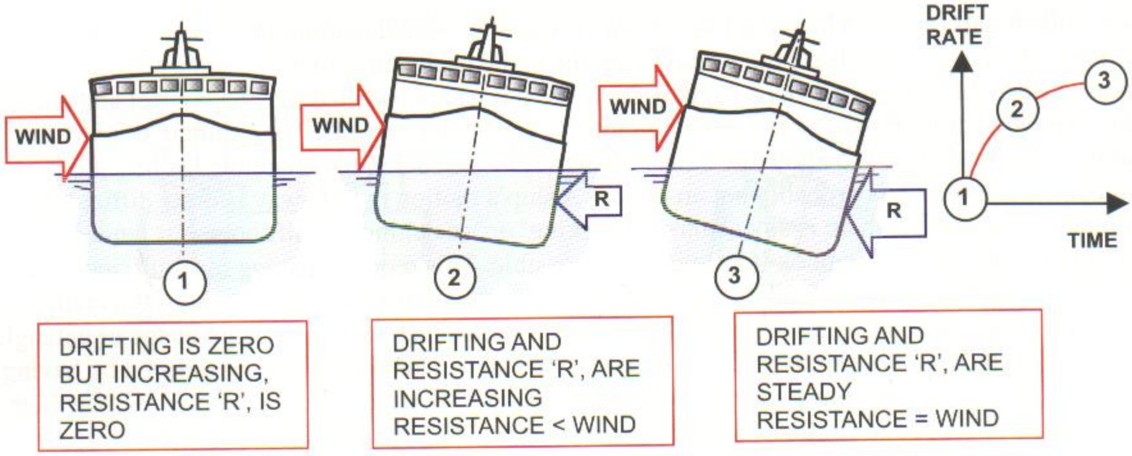
WORKING A HEAVY LIFT THROUGH A MOONPOOL



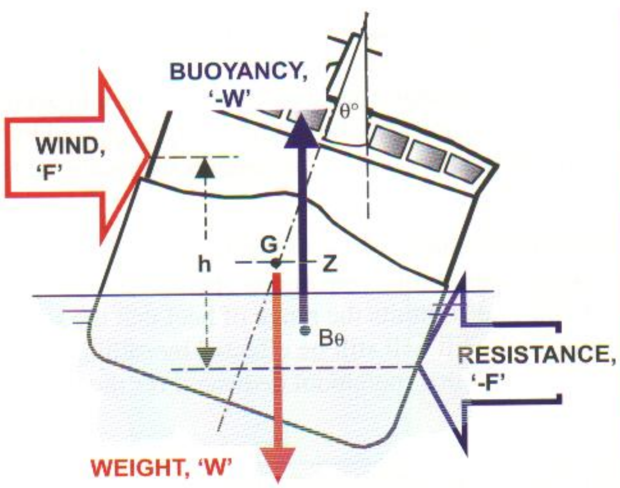
THE EFFECT OF WIND ACTING UPON A SHIP'S SIDE

If a ship is lying stopped in the water and subjected to the onset of a beam wind against its side, then it will start to drift downwind. This will be opposed by an increasing force of water resistance acting upon the leeside submerged hull. The ship will reach a steady downwind drift speed when the forces of windage and water resistance are equal and opposite. These two forces, acting upon different heights of the hull above the keel, create a heeling moment, as shown by the following diagrams;-

WIND HEELING



THE BALANCE BETWEEN WIND HEELING AND RIGHTING MOMENTS



STEADY WIND HEEL CONDITIONS

WHEN THE VESSEL REACHES THE STEADY STAGE **3**, THE HEELING MOMENT IS BALANCED BY THE RIGHTING MOMENT.

I.E. $\quad F \times h = W \times GZ$

THE HEELING MOMENT IS PRODUCED BY THE EQUAL AND OPPOSING FORCES OF WINDAGE, ACTING THROUGH THE CENTRE OF THE EXPOSED AREA, AND THE HULL RESISTANCE TO SIDEWAYS MOVEMENT, WHICH ACTS THROUGH THE CENTRE OF THE IMMERSED HULL

The above argument is equally valid whether the ship is simply drifting downwind or making its way ahead through the water. When moving ahead through the water, a ship may or may not make significant leeway, depending upon the distribution of side surfaces exposed to the wind, the way the immersed hullform changes with the resulting list and how the ship's rudder is applied. However, a list will develop as shown above, whether the ship makes leeway or not.

Wind Heeling Moment, (W.H.M.) = **Wind Heeling Force (F)** x **Heeling Arm (h)** **Kg- Metres**

Note that the Wind Heeling Force is expressed in Kilograms (a unit of mass) in order to be compatible with the forces of weight and buoyancy which are also normally expressed in units of mass (i.e. Tonnes and Kilograms). This failure to differentiate between force and mass in most situations, is a long-standing practice in naval architecture.

THE EFFECT OF WIND ACTING UPON A SHIP'S SIDE (Cont.)

The wind speed determines the dynamic pressure of the air as it strikes the ship's side

$$\text{DYNAMIC PRESSURE, } P_0 = \frac{1}{2}\rho v^2 \text{ NEWTONS / M}^2$$

WHERE ρ ***= AIR DENSITY OF 1.3 Kg / M³ AND*** v ***= AIRFLOW VELOCITY IN M / S***

The dynamic pressure is a measure of the air molecules' Kinetic Energy and so is the pressure required to stop the airflow completely. (i.e. remove all its Kinetic Energy).

Kinetic Energy and, consequently, dynamic pressure are proportional to the square of the wind speed. When wind encounters an obstruction, such as the ship's side, it is diverted in direction, rather than stopped, so it will only have lost a fraction of its Kinetic Energy. This lost energy is absorbed into the work done by heeling the ship over and produces the Effective Wind Pressure, which acts upon the ship's side. The extent to which an obstruction slows down the air flowing past it, depends upon the degree to which its shape is streamlined. However, most ships are vertical sided and a single fractional constant, or drag factor, can be assumed to apply to the air flow around all vessels.

A single equation, incorporating drag and air density, relates the Effective Pressure to wind speed as follows;-

$$\text{EFFECTIVE SIDE WIND PRESSURE} = 0.0035\ (\text{WIND SPEED IN KNOTS})^2\ \text{Kg / M}^2$$

The Effective Pressure is about 22% of the wind's Dynamic Pressure for any given wind speed. Note that the constant '0.0035' in the equation also converts the pressure units into the form of equivalent weight per area (Kg/metre squared) to allow the resulting wind force to be expressed in Kilograms, as follows;-

Wind Heeling Force (F) = Effective Wind Pressure (P) x Exposed side Area (A) Kg

And, as has been illustrated on the previous page ;-

Wind Heeling Moment, (W.H.M.) = Wind Heeling Force (F) x Heeling Arm (h) Kg- Metres

The exposed side area of the ship can be determined by applying the methods of approximate integration to horizontal strips of the ship's profile above the waterline.

The height above the waterline of the centre of this area can be calculated by various methods of approximation, such as summing moments of each strip about the waterline and dividing the resulting total moments by the total area.

The centre of water resistance is assumed to be at a depth of half the draft below the waterline, so the Heeling Arm, h, is the sum of half the draft plus the height of the centre of area above the waterline.

DETERMINING A VESSEL'S WIND HEELING MOMENT

h1, h2, etc, ARE THE HEIGHTS ABOVE THE W/L OF AREAS A1, A2 etc

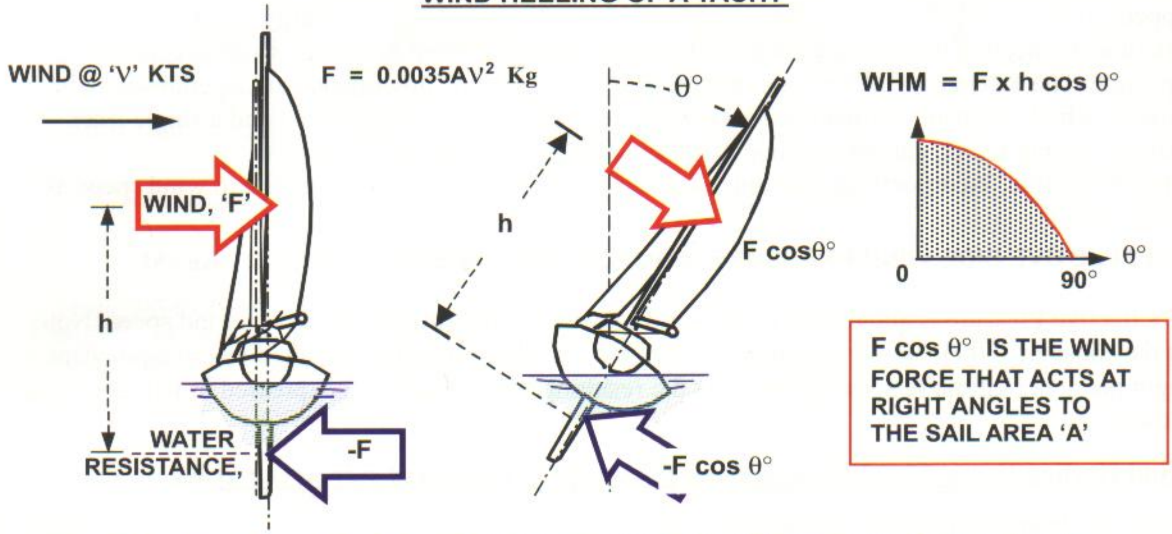
$$\text{HEIGHT 'H' OF C of A ABOVE THE W/L} = \frac{(A_1h_1 + A_2h_2 + A_3h_3 + A_4h_4 + A_5h_5 + A_6h_6 + A_7h_7)}{(A_1 + A_2 + A_3 + A_4 + A_5 + A_6 + A_7)}$$

$$\text{WIND HEELING ARM} = H + \frac{1}{2}\text{ DRAFT, } \textit{WHERE 'H' IS ESTIMATED AS ABOVE}$$

CHANGES IN WIND HEELING MOMENT AS A SHIP HEELS OVER

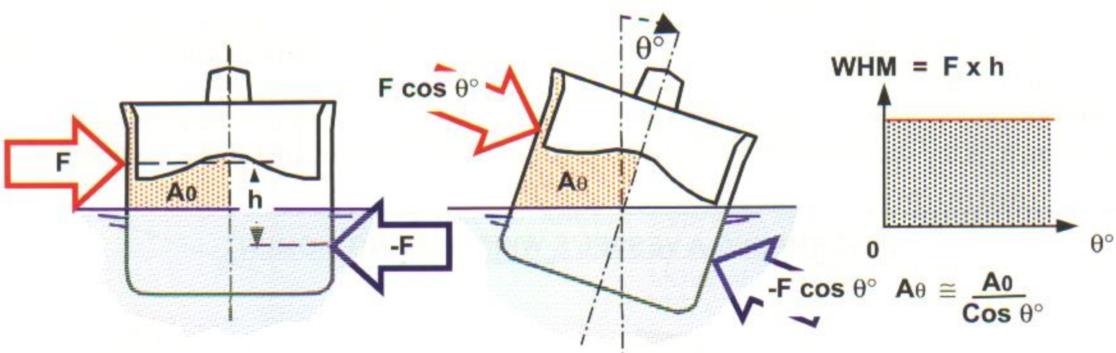
As a vessel heels over due to windage, the force of the wind strikes the exposed side area at an increasingly oblique angle so we would expect the Wind heeling Moment acting upon a ship to decrease by a factor of the cosine (angle of heel). This is approximately true for sailboats, where the windage and hull resistance surfaces lie predominately on the vessel's centreline.
However, when these two surfaces are separated by the ship's beam, heeling increases the area exposed to wind by approximately the same factor, whilst the separation between the centres of these areas also tends to increase, though the situation is complicated by the changing of the effected surfaces' shapes.

WIND HEELING OF A YACHT



FORCES ACT UPON AREAS ON THE CENTRELINE, SO AREA REMAINS CONSTANT AS THE BOAT HEELS BUT HEELING COMPONENTS OF FORCES REDUCE BY **cos θ°**

WIND HEELING OF A COMMERCIAL SHIP



FORCES ACT UPON AREAS SEPARATED BY THE SHIPS BEAM. EXPOSED WINDAGE AREA INCREASES AS HEELING COMPONENTS OF FORCES DECREASE BY **cos θ°**.

THE WIND HEELING MOMENT IS CONSIDERED AS UNCHANGED BY ANGLE OF HEEL.

Guide lines making stability recommendations for vessels subjected to wind heeling, use the following Effective Pressure values to represent a hypothetical Worst Service conditions.

Steady wind pressure for wind speed of 118 knots = 48.5 Kg per metre squared
Maximum wind pressure for wind speed of 177 knots = 72.8 Kg per metre squared

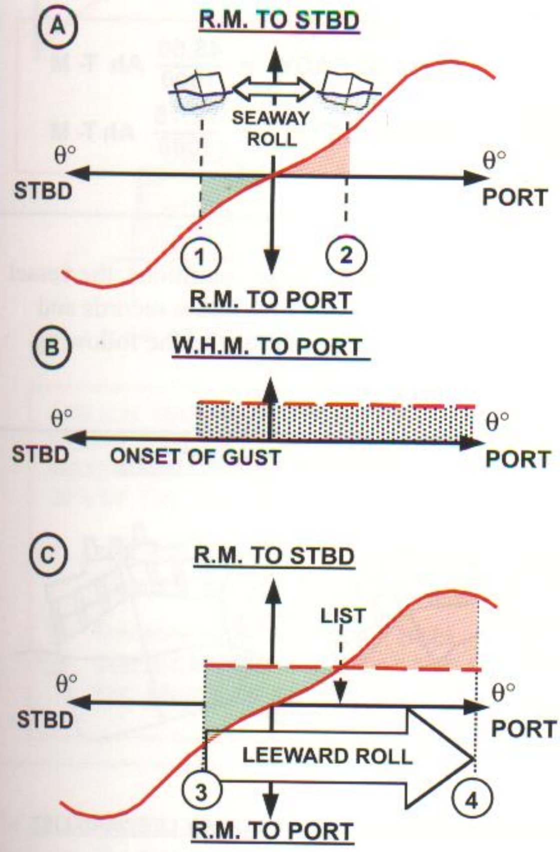
These winds are an estimate of the most extreme circumstances likely to be encountered. The I.M.O. recommend that slightly stronger windforces are allowed for in these calculations.

A SHIP'S ROLLING MOTION DUE TO WIND ACTION

When a ship is subjected to a change in heeling moments, energy is put into the vessel, causing it to roll. This energy of motion, called Kinetic Energy, continues to increase whilst the heeling moment is greater than the opposing righting moment, so the vessel is continuing to gain angular momentum. As the hull heels over, however, the righting moment is increasing so that a point is reached where the two opposing moments are equal. This is the angle of heel at which there is no further increase in the ship's angular momentum. It still possesses kinetic Energy, so it will continue to roll over to greater angles of heel until the righting moment (which is now larger than the heeling moment) has absorbed all that Kinetic Energy. Provided that the vessel has sufficient Dynamic Stability to do so, this Kinetic Energy will have been converted into the Potential Energy to start the return roll.

If the heeling moment that started this rolling motion, remains constant and there is no friction involved, then the ship would remain oscillating about the angle of heel. The original energy input would be continually transformed between Potential Energy at the ends of the roll to Kinetic Energy at the angle of wind heel. In reality, friction between the hull and the water drains this energy away, so each successive roll will be diminished after the onset of the wind until the vessel settles with a steady list or is set in motion again by the next change in heeling moments.

THE ROLL DUE TO A SUDDEN GUST OF WIND



(A) THE VESSEL IS ROLLING DUE WAVE ACTION ALONE. THE ROLL IS SYMMETRICAL ABOUT THE UPRIGHT

= ENERGY TO STBD

= ENERGY TO PORT

ENERGY TO STBD = ENERGY TO PORT

(B) A SUDDEN GUST OF WIND STRIKES THE SHIP'S STARBOARD SIDE AT THE END OF THE STARBOARD ROLL. THE AREA UNDER THE WIND HEELING MOMENT CURVE IS THE ENERGY INJECTED INTO THE SHIP'S RETURN LEEWARD ROLL TO PORT

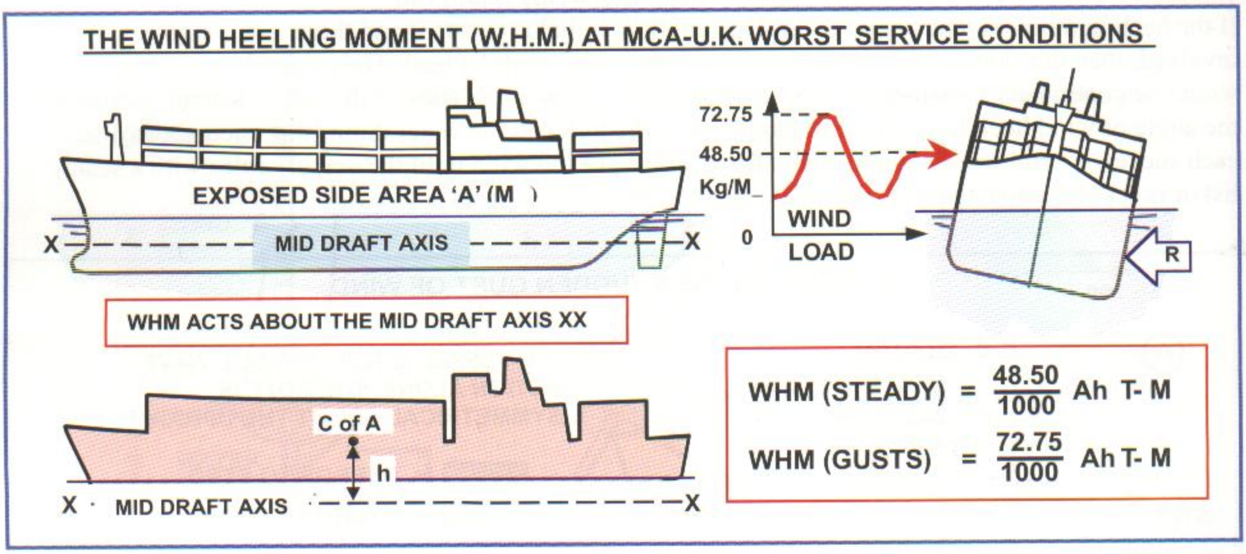
(C) THE WIND HEELING MOMENT CURVE IS INVERTED AND SUPERIMPOSED UPON THE RIGHTING MOMENT CURVE SO THAT THE AREA BETWEEN THE TWO CURVES IS THE TOTAL ENERGY FORCING THE SHIP TO PORT.
THE SHIP WILL ROLL TO A POINT WHERE IT SWEEPS OUT AN EQUAL AREA BEFORE RETURNING TO ROLLING ABOUT A LIST (INDICATED BY THE INTERSECTION), IF THE WIND PERSISTS

This rolling cycle can be illustrated by superimposing the wind heeling moment onto the ship's righting moment curve. The area beneath such curves between any two points, equals the energy involved in that moment rolling the ship through that particular range of heel. We use the Righting Moment curve, rather than the GZ curve, when considering wind heeling because the ship's displaced weight is not a common factor in the two opposing moments.

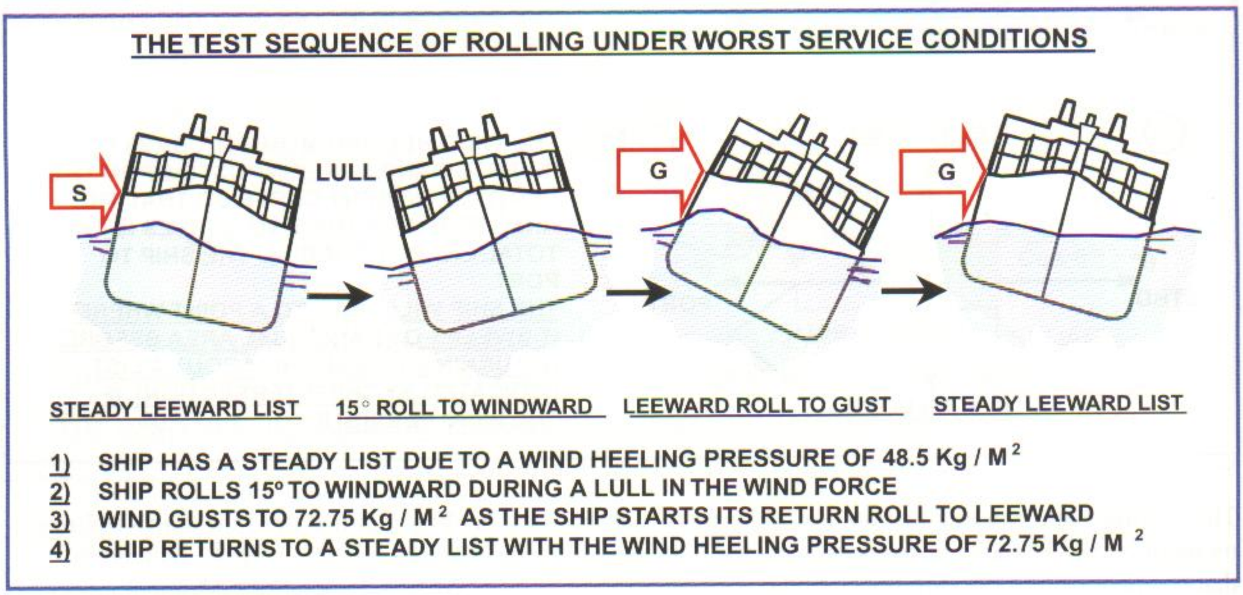
The wind Heeling Moment depends upon the exposed surface area, which would increase, *relative to the volume of shape*, if we made the ship smaller, consequently, small vessels are in greater danger from the wind than large ones, if they are similar in general overall proportions,

MCA-U.K. WIND HEELING CRITERIA FOR CONTAINER SHIP STABILITY

Several modern types of ship are relatively high sided for their displacement and draft, so beam winds can produce significant listing to leeward. There is particular concern for ship's carrying high stows of containers on the deck and hatches. Containers are watertight and so are likely to be buoyant but they are not part of the ship's structure. If subjected to partial immersion due to severe rolling, they are vulnerable to being washed overboard, which can cause damage to the ship's structure and result in the ship developing a list due to loss of weight from one side. The U.K. Marine Coastguard Agency has produced stability recommendations for such vessels, based upon the ship experiencing the Worst Service conditions, in which wind gusts exceed the average wind speed by 50%



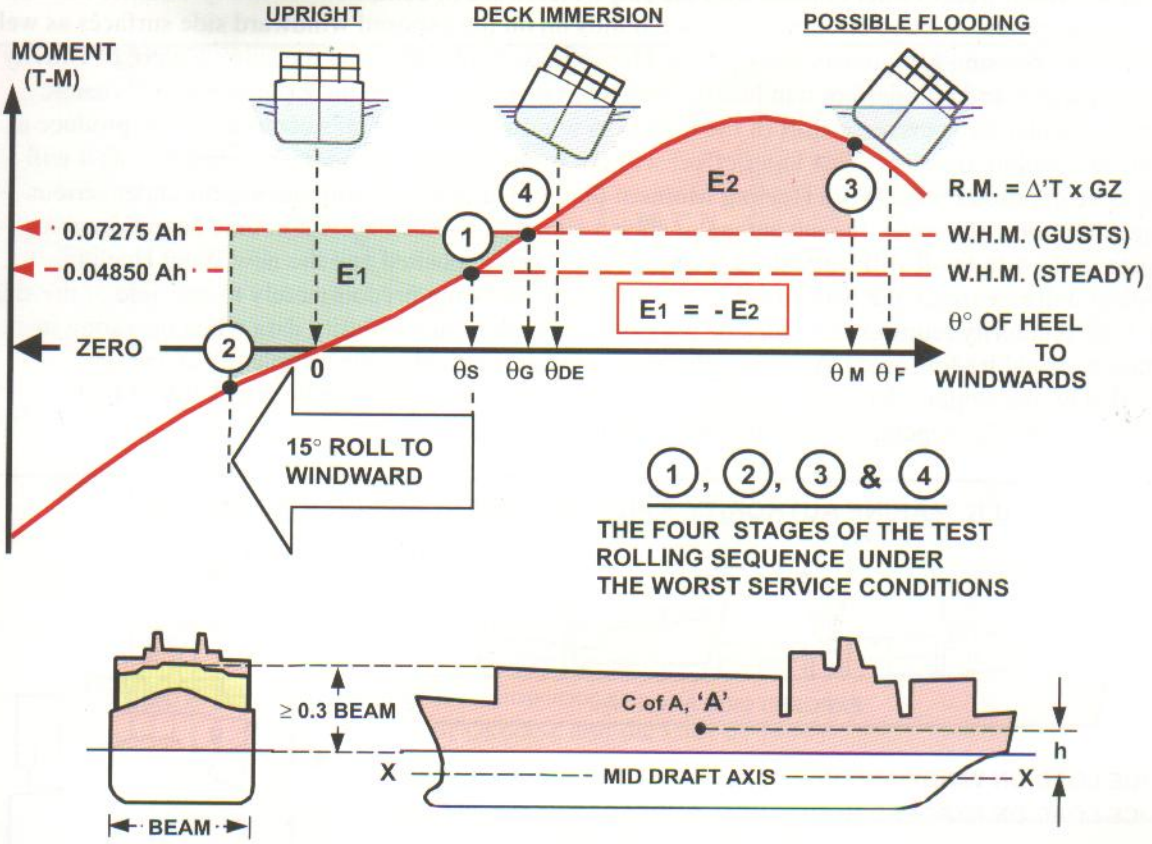
The exposed windage area must include the deck stow of containers. Under these conditions, the vessel is assumed to undergo a sequence of rolling which is based upon the analysis of weather records and records of data, regarding ships' motions in high winds. This test sequence is shown by the following sketches



The response of the ship to this test sequence is determined by superimposing the Wind Heeling Moment line onto the ship's Righting Moment curve, as described on a previous page. The stability recommendations specify particular criteria that the resulting graph should meet, in order for the ship to be considered seaworthy for that particular loaded state.

MCA-U.K. WIND HEELING CRITERIA FOR CONTAINER SHIP STABILITY

WIND HEELING AND RIGHTING MOMENTS



THE U.K. MARINE AUTHORITY STIPULATES ADDITIONAL STABILITY REQUIREMENTS TO RESIST WIND HEELING, FOR **CONTAINER VESSELS WHERE THE HEIGHT ABOVE THE WATERLINE OF THE TOP OF THE UPPERMOST DECK CONTAINER STOW IS GREATER THAN 30% OF THE SHIP'S MAXIMUM BEAM.** THESE EXTRA CRITERIA ARE;-

1) **THE STEADY WIND LIST, θ_S, UNDER WORST SERVICE CONDITIONS, SHOULD NOT EXCEED 65% OF THE ANGLE OF DECK IMMERSION, θ_{DE}.**

2) **THE MAXIMUM ANGLE OF HEEL, θ_M, WHICH RESULTS FROM THE SHIP BEING SUBJECTED TO A GUST AT THE END OF THE WINDWARD ROLL, SHOULD NOT EXCEED THE ANGLE OF FLOODING, θ_F.**

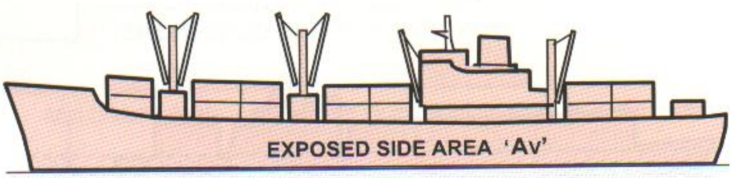
The above criteria are not absolute but if a ship fails to comply with them for a particular loaded condition, the owner must present a good argument for allowing the vessel to sail. He should base his case on the additional measures that have been taken to ensure the securing of the deck stow, such as extra lashings and rigorous battening down procedures which are to be followed as routine practice. If the Maritime and Coastguard Agency considers these alternative measures to be inadequate for the vessel and its trading pattern, then restrictions will be imposed on the allowable height of deck stow or on the ship's trading area and season.
The stability criteria for container ships, with regard to wind heeling, are in addition to the normal stability requirements that the GZ curve of every ship must satisfy.

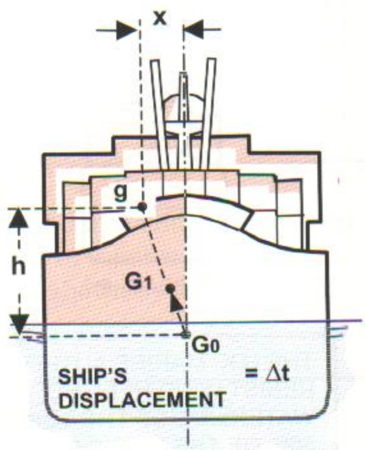
The IMO lays down similar windheeling criteria applicable to all ships, but calculates the windward roll angle, using an equation which allows for factors such as a ship's roll period, Cb value etc.

THE EFFECT OF ICE ACCRETION UPON A SHIP'S STABILITY

The build up of ice, or ice accretion, on a ship's upper works and rigging, has long been known as a serious hazard to ships operating in high latitudes. Some ice can accumulate from snowfall, sleet or freezing fog, but these fresh water sources are relatively minor, compared with the ice build up from sea spray which freezes after contact with the ship's steelwork in conditions of heavy weather and sub-zero temperatures. In these circumstances, ice builds up on the exposed windward side surfaces as well as the decks housing tops, masts and rigging. This extra weight will cause the ship's Centre of Gravity to rise upwards and outboard to windward. The rise in KG value will reduce the range of Dynamic Stability, whilst the transverse shift of the C of G to windward of the hull's centreline, will produce a windward heeling moment. This latter effect will not necessarily be immediately apparent, as it will tend to be opposed by the Wind Heeling Moment that will almost certainly be present under serious icing conditions. However, the advantage of such a situation is illusionary. A course alteration or a change in the wind will still leave the ice where it has been deposited and the new Wind Heeling Moment will re-enforce the list caused by the weight of ice being predominately to one side of the ship. The U.K. Authority requires that ships be provided with additional stability data when operating in certain seasonal trading areas. An icing allowance must be applied to the sample of GZ curves, provided by the shipbuilder to cover the ship's normal range of loaded conditions. A full or half allowance is made, depending upon the particular area of trade.

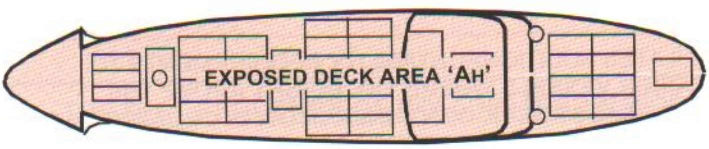
U.K. MARINE AUTHORITY'S FULL ALLOWANCES FOR ICE BUILD UP





ICE LOAD ON WINDWARD VERTICAL SURFACES, 'AV' = 15 Kg/M^3

ICE LOAD ON EXPOSED HORIZONTAL SURFACES, 'AH' = 30 Kg/M^3

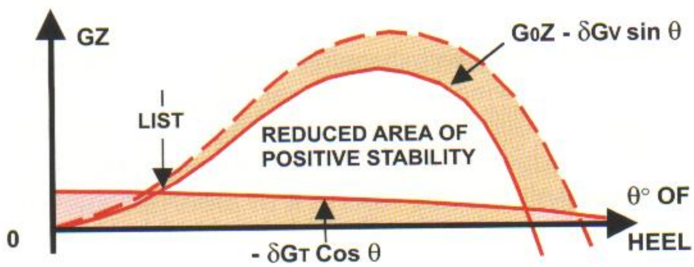


TOTAL WEIGHT OF ICE, 'W' = $1.05\ (0.03\ A_H + 0.015\ A_V)$ T

VERTICAL RISE OF G, 'δG_V' $= 1.1 \left(\frac{W\,h}{\Delta t + W} \right)$ M

TRANSVERSE SHIFT OF G, 'δG_T' $= 1.1 \left(\frac{W\,x}{\Delta t + W} \right)$ M

ALLOWANCES OF 5% EXTRA WEIGHT AND 10% EXTRA LEVERAGE ARE MADE FOR ICE TRAPPED ON THE SHIPS MASTS, DERRICKS, CRANES AND RIGGING



MODIFIED GZ CURVE

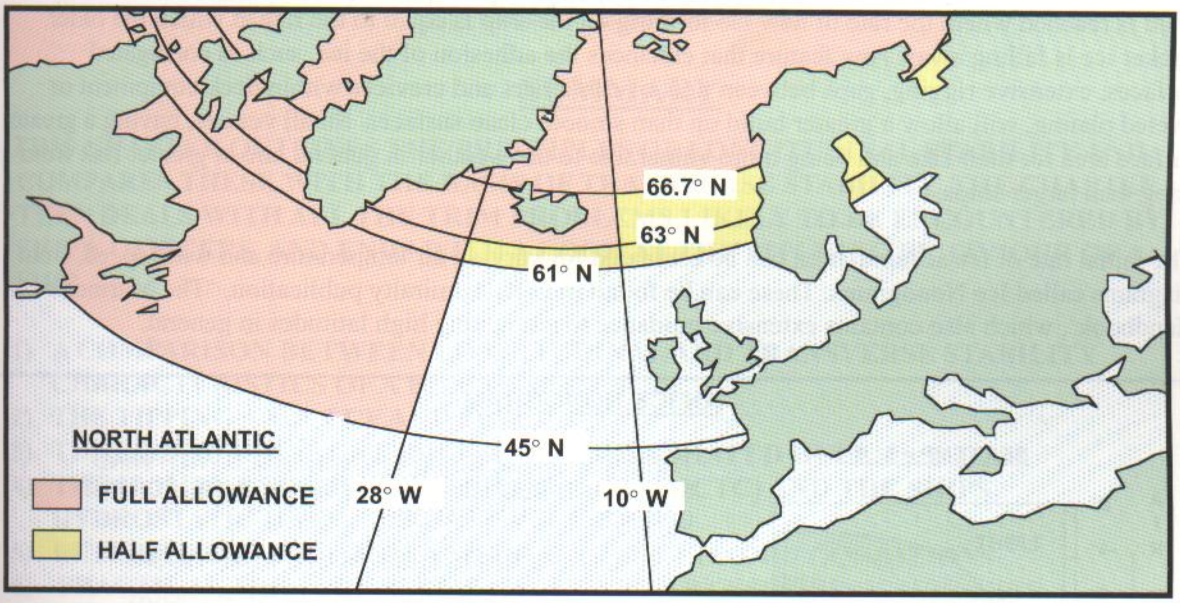
THE AREA AND RANGE OF POSITIVE STABILITY ARE REDUCED BY THE SHIFTS 'δG_V' AND 'δG_T', DUE TO ICE BUILD UP

A ship's GZ curve, modified to allow for ice build up, must meet the normal basic criteria of safety prior to sailing in trading areas where ice accretion is considered to be a likely hazard.

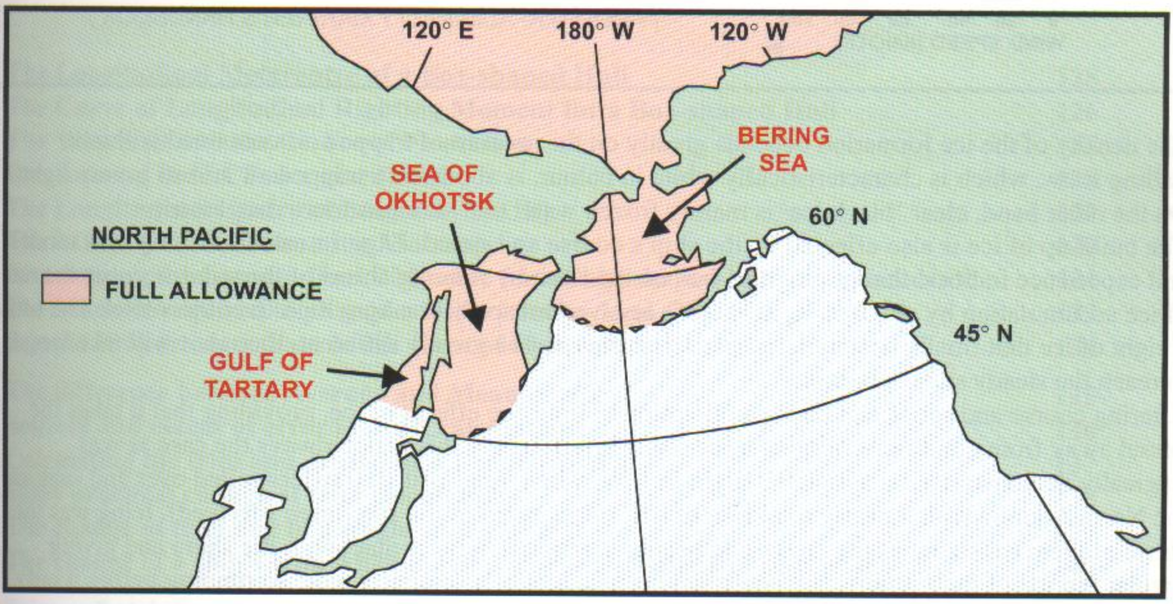
MCA-U.K. TRADING AREAS WHICH REQUIRE ALLOWANCE FOR ICING

The U.K. Maritime and Coastguard Agency defines the following regions as trading areas that require icing allowances to be applied to a ship's stability calculations. They consider three global regions.

1) **NORTH ATLANTIC.** Areas for full and half allowances are shown on the following map.



2) **NORTH PACIFIC.** Areas of full allowance are
The Sea of Okhotsk and the Gulf of Tatary.
The Bering Sea.



3) **SOUTHERN OCEANS**. Full allowance must be applied to all areas south of 60 degrees.

Areas requiring half allowance, must be determined by consultation between the Department of Transport and the shipowner involved, excepting the North Atlantic, where such areas are defined in the previous map.

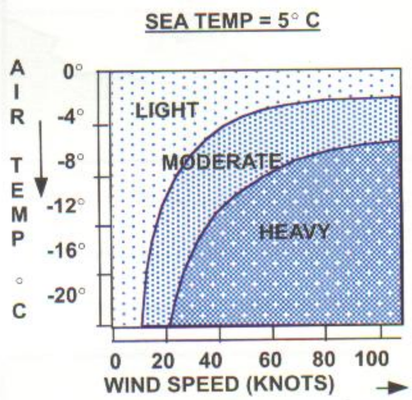
In general, any winter seasonal zone, as defined by the international loadline rules, should be considered an area of potential ice accretion.

THE RATE OF ICE ACCUMULATION

The rate at which ice builds up on a ship is highly variable and depends upon factors such as the wind speed and direction relative to the ship's own track, the air temperature and the sea temperature. Features of the ship also influence how effectively spray is broken up and trapped onboard long enough to freeze. In steady conditions, the rate of ice build up will generally decline as the ice thickness increases because thicker pieces of ice are more likely to slide off the ship's structure. Eventually a point is reached where the rate of new ice forming on the ship is equal to the rate at which the older thicker ice is falling away. Any feature that enhances the adhesion of the ice, such as corrugated surfaces, extensive rigging, gaps between tiers of containers and crevices within deck equipment or riveted plating, will allow a greater build up than smooth clean surfaces. Small vessels, having a greater surface area for their size and being more vulnerable to the weather in general, are at greater risk from icing than larger ships.

The initial rate of ice accumulation can be estimated for varying air temperature and wind speed, from diagrams called Ice Nonograms. These can be found in U.K. Admiralty publication, 'The Mariner's Handbook', which also contains extensive guidance to navigating high latitudes in general.

AN ICE NONOGRAM



THE RATE OF ICE BUILD-UP INCREASES WITH WIND SPEED AND REDUCING AIR TEMPERATURE, FOR THE GIVEN SEA TEMPERATURE

LIGHT ICING IS LESS THAN 0.7 CM / HOUR

HEAVY ICING IS MORE THAN 2.0 CM / HOUR

1 CM OF ICE WOULD RESULT IN AN ADDED WEIGHT OF ABOUT 20 KG / $(\text{METRE})^2$ OF EXPOSED SURFACE

THE TERMS LIGHT , MODERATE AND HEAVY ARE REFERED TO IN U.K. SHIPPING FORECASTS.

(FROM U.K. ADMIRALTY MARINER'S HANDBOOK)

The density of the ice formation depends greatly on the amount of trapped air contained within it. Falling snow, which is characteristically white in colour, is about 90% trapped air and so is very light. On the other hand, clear 'black ice' is mainly frozen water and so much more dangerous.
The build-up of ice is also effected by the ship's course and speed. Experiments on fishing boat models and experience indicate that putting the seas on the stern by running ahead of the wind, reduces the rate of ice accumulation by about 50%, as there is less violent spray breaking high over the vessel. The total weight of ice that finally accumulates, however, may not be greatly different, though it will take longer to reach the steady state conditions.
Altering course and speed can be a useful option to limit the effects of ice, providing that it takes the vessel away from conditions of greater ice risk. Unfortunately this is not always the case as was tragically demonstrated on a winter's night in 1968 when three British trawlers rolled over and sank in the North Atlantic with a nearly total loss of life. (Only three of the sixty men survived). Icing was the cause of them capsizing as the crews had been in continual radio contact with the shore and giving regular reports of their plight. The boats had been caught in a southerly gale, so running before it only took them further north into generally colder conditions and even though two of them reached comparative shelter of the coast of Iceland, the accumulation of ice still overwhelmed the ships.
Modern fishing vessels tend to be 'cleaner' in design with less rigging and clutter on the exposed deck and, consequently, the ice accumulation is reduced.

CHAPTER 6

LONGITUDINAL STABILITY AND PRACTICAL TRIM CALCULATIONS

SUMMARY

THIS CHAPTER OUTLINES THE PRINCIPLES OF LONGITUDINAL STABILITY AND COMPARES THESE WITH THE BASIS OF TRANSVERSE STABILITY. THE CHAPTER THEN DEALS WITH ALL THE TRIM PROBLEMS LIKELY TO BE ENCOUNTERED BY SHIP'S OFFICERS AND ENDS WITH A LOOK AT THE PITCHING BEHAVIOUR OF A SHIP

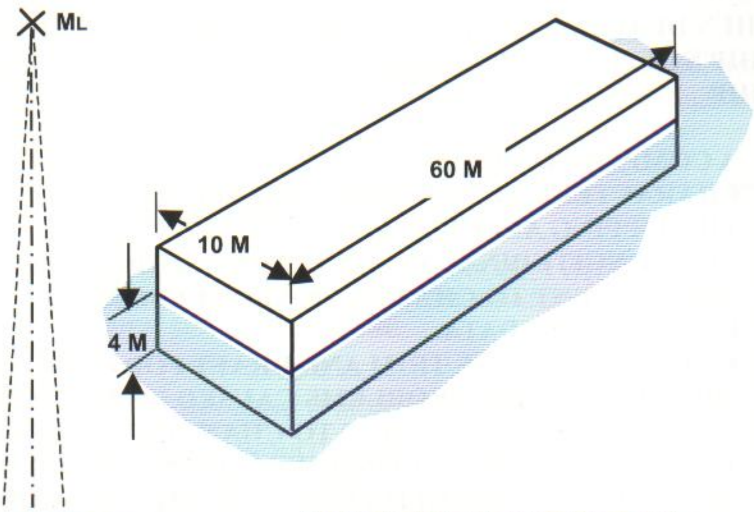
1) **A COMPARISON BETWEEN LONGITUDINAL AND TRANSVERSE STABILITY CHARACTERISTICS OF A BOX-SHAPED HULL**
2) **THE TIPPING AXIS, LCF AND LCB.**
3) **THE LONGITUDINAL STABILITY OF A SHIP-SHAPED HULL.**
4) **THE MOMENT TO CHANGE TRIM BY 1 CM (MCTC) AND THE SHIPS HYDROSTATIC DATA.**
5) **DETERMINING THE TRIM AND DRAFT FOR LOADED CONDITIONS.**
6) **PRACTICAL TRIM PROBLEMS CAUSED BY TRANSFERRING WEIGHTS.**
7) **THE CHANGE OF TRIM DUE TO CHANGING WATER DENSITY.**
8) **TRIM AND STABILITY CONSIDERATIONS FOR DRYDOCKING.**
9) **BEACHING A DAMAGED VESEL AND THE CONSEQUENCES OF STRANDING**

CONTENTS

THE LONGITUDINAL METACENTRE OF A BOX-SHAPED HULL

The fore and aft movement of the Centre of Buoyancy, relative to a vessel's Centre of Gravity, controls the ship's pitching motion and trim in exactly the same way that heel and rolling are the result of transverse shifts of the C of B. However, the hull's resistance to pitching is much greater than its resistance to rolling because its length is much greater than its beam, which results in the longitudinal metacentric radius (BM_L) being much longer than its transverse radius (BM_T). Furthermore, the height of the Centre of Buoyancy above the keel (the KB value) is relatively insignificant and so the BM_L approximates to the GM_L. This is shown below for a box-shaped hull.

THE LONGITUDINAL AND TRANSVERSE METACENTRES OF A BOX-SHAPED HULL



LONGITUDINAL METACENTRE ' M_L'

$$KM_L = \frac{(\text{LENGTH 'L'})^2}{12 \times \text{DRAFT}} + \frac{1}{2}\,\text{DRAFT}$$

$$KM_L = \frac{60^2}{12 \times 4} + 2 \quad \text{METRES}$$

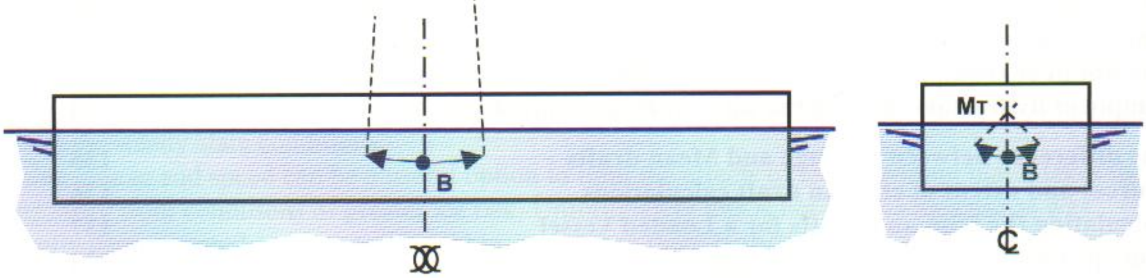
$$KM_L = 77 \quad \text{METRES}$$

TRANSVERSE METACENTRE ' M_T'

$$KM_T = \frac{(\text{BEAM 'B'})^2}{12 \times \text{DRAFT}} + \frac{1}{2}\,\text{DRAFT}$$

$$KM_T = \frac{10^2}{12 \times 4} + 2 \quad \text{METRES}$$

$$KM_T = 4.08 \quad \text{METRES}$$

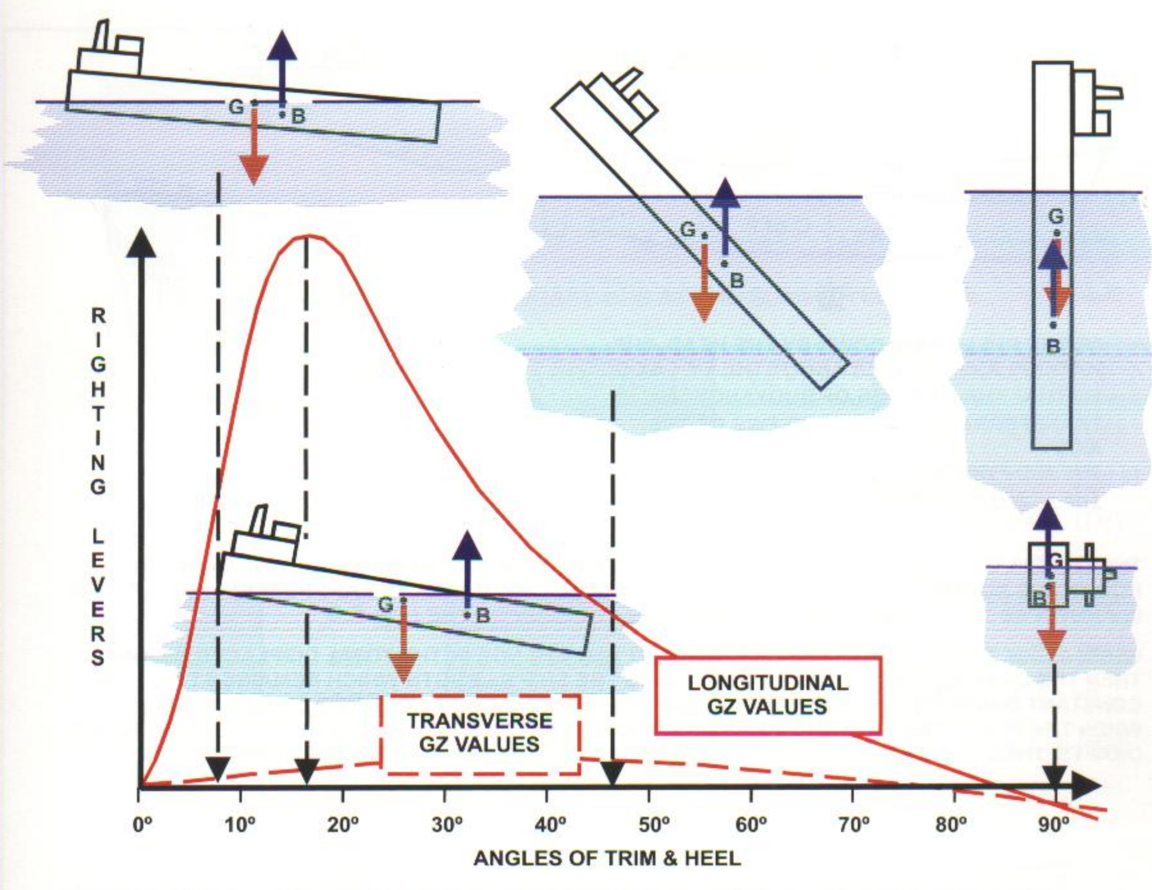


THE RATIO OF $BM_L : BM_T$ = (LENGTH : BEAM)^2 WHICH, IN THIS EXAMPLE, = 36 : 1

THE KM_L IS MUCH GREATER THAN THE KM_T AND ALSO MUCH GREATER THAN ANY POSSIBLE KG VALUE. THIS MEANS THAT, IN NORMAL CIRCUMSTANCES, THE GM_L WILL ALWAYS BE POSITIVE SO THE VESSEL WILL NOT LOSE LONGITUDINAL STABILITY AND IS, EFFECTIVELY, INDEPENDENT OF THE HEIGHT OF THE C of G (I.E. THE KG). FURTHERMORE, THE GM_L IS SO LARGE THAT TRIM ANGLES ARE USUALLY LIMITED TO ONLY ONE OR TWO DEGREES. OVER THIS RANGE, THE BM_L AND, HENCE, THE GM_L CAN BE CONSIDERED TO REMAIN CONSTANT FOR A BOX-SHAPED VESEL.

THE CURVE OF LONGITUDINAL RIGHTING MOMENT FOR A BOX-SHAPED HULL

Although the ship's hull has considerably more initial resistance to pitching than rolling, the fore and aft ends are immersed at relatively small angles of trim, so the pitching resistance peaks at a comparatively small angle of trim. The following diagram compares the longitudinal and transverse Righting Lever (GZ) curves for a box shaped vessel, assuming the weight distribution and, hence, the position of the C of G, remain constant up to 90° of pitch and heel.



THE LONGITUDINAL RIGHTING LEVER CURVE AT 90° OF TRIM IS EQUAL TO THE TRANSVERSE RIGHTING LEVER AT 90° OF HEEL

A ship can lose longitudinal stability if weight is taken at one end of the vessel, which is sufficient to trim the hull beyond the point of maximum Righting Moment. When the British transatlantic liner 'Titanic', collided with an iceberg in calm conditions, the length of underwater damage in the forward region of the hull was considerable and progressive flooding occurred. Gradually the bow was pulled further under the water until it reached a trim angle of about 55° at which point the ship broke in two and sank. (It is a testament to the strength of the vessel that it didn't break in two earlier). The ship appears to have maintained transverse stability until just before it sank.

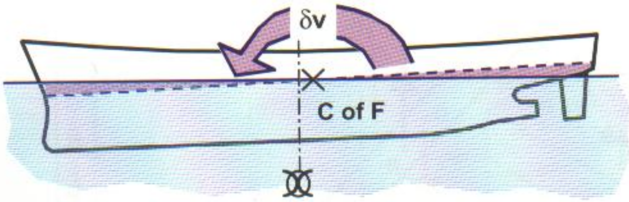
Much more recently, the British bulk carrier 'Derbyshire' sank in a typhoon. Subsequent investigations appear to indicate that a hatch, right forward, was broached by heavy seas coming over the bow and sufficient flooding occurred to prevent the bow rising properly to the next heavy waves. This seems to have led to further damage and flooding which again eventually caused a loss of longitudinal stability and the ship sank, tragically, with all hands.

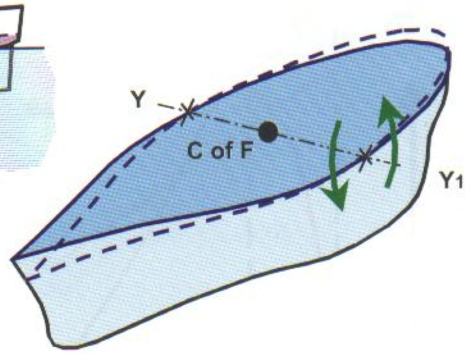
THE TRIM AXIS AND THE CENTRE OF FLOATATION, 'C of F'

When the trim of a vessel is changed by a redistribution of weight, the newly immersed volume of the hull must equal the original displaced volume, as the ship's total weight has remained constant. The waterplane area rotates around a tipping axis, which transfers a wedge of buoyancy from one side of the axis to the other. This is basically the same as occurs when a ship is heeled over around the centreline, except that the waterplane of a ship-shaped hull is not usually symmetrical about the midships axis. Generally, at deeper drafts, the hull becomes slightly fuller in the stern regions than the bow, so the tipping, or **trimming axis** will be a little bit aft of the midships point. Before we can calculate the BM value and, hence, the trimming characteristics of a hullform at a particular draft, we must determine the position of this trimming axis.

The position of the 'trimming axis' on the waterplane, is known as the 'Longitudinal Centre of Floatation'

THE LONGITUDINAL CENTRE OF FLOATATION, 'LCF'



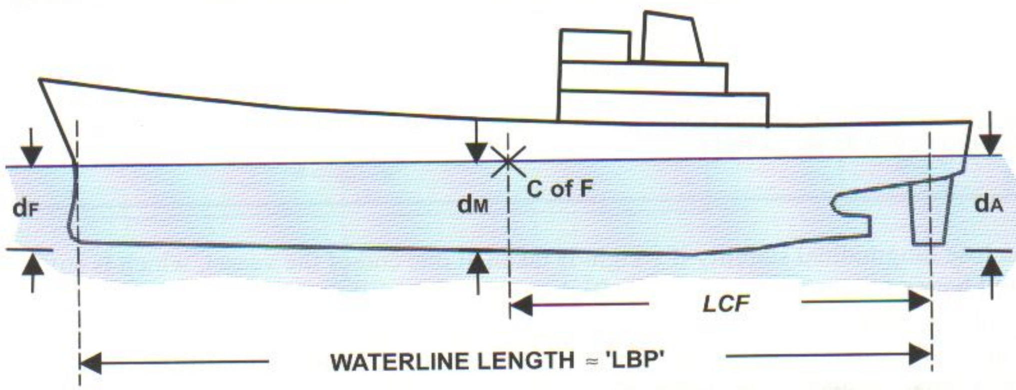


AS THE HULL TRIMS DOWN BY THE HEAD, THE WATERPLANE ROTATES ABOUT THE YY $_1$ AXIS. SO THAT THE A WEDGE OF BUOYANCY ' δv ' IS TRANSFERRED FROM AFT TO FWD OF THE AXIS

THE MID-POINT OF THIS AXIS IS KNOWN AS THE CENTRE OF FLOATATION OR THE C of F

THE CENTRE OF FLOATATION IS THE GEOMETRIC CENTRE OF THE WATERPLANE AREA AND ITS POSITION IS NORMALLY EXPRESSED AS ITS DISTANCE FORWARD OF THE AFT PERPENDICULAR. THIS IS KNOWN AS THE *LONGITUDINAL CENTRE OF FLOATATION OR THE LCF*

WHEN A SHIP TRIMS ABOUT THE C of F, WITHOUT ANY CHANGE IN THE TOTAL DISPLACED WEIGHT, THEN THE C of F AXIS IS THE ONLY POINT ALONG THE SHIP'S LENGTH WHICH REMAINS AT A CONSTANT DRAFT. *THIS IS THE POINT WHERE THE TRUE MEAN DRAFT SHOULD BE MEASURED*. WHEN THE PREDICTED TRIM AND MEAN DRAFT ARE BEING USED TO CALCULATE THE FORE AND AFT DRAFTS, THEN THE TRIM SHOULD BE PROPORTIONED FROM THE CENTRE OF FLOATATION.



TRIM $= d_A - d_B$ METRES BY THE STERN

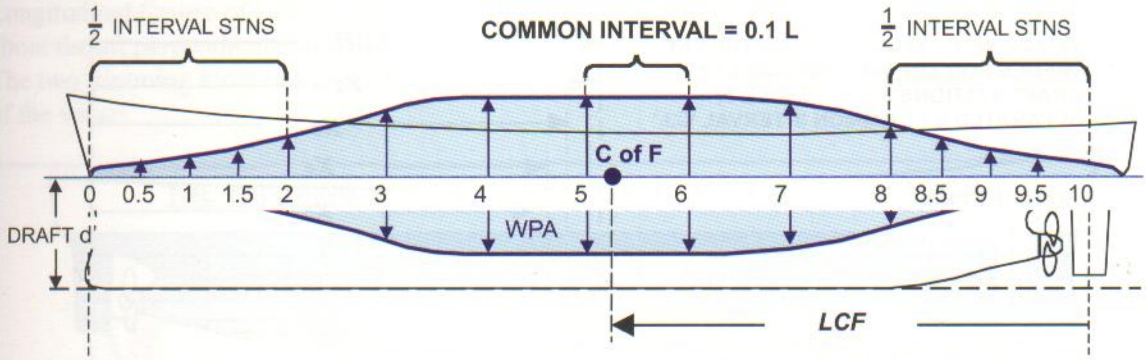
M'N DRAFT 'd_M' $= d_A - \text{TRIM}\ \dfrac{LCF}{LBP}$ M OR M'N DRAFT 'd_M' $= d_F + \text{TRIM}\ \dfrac{LBP - LCF}{LBP}$ M

AFT DRAFT 'd_A' $= d_M + \text{TRIM}\ \dfrac{LCF}{LBP}$ M OR FWD DRAFT 'd_F' $= d_M - \text{TRIM}\ \dfrac{LBP - LCF}{LBP}$ M

DETERMINING THE LCF AT A GIVEN DRAFT

The Distance of the LCF forward of the aft perpendicular (the LCB) for any given draft, is determined by taking the sum of the Moments about the aft perpendicular, of waterline beams, measured at regular station intervals along the waterline at that draft, and dividing it by the waterplane area. We can use the procedures of approximate integration, as described in Chapter 2 and apply either Simpson's Rules or the Trapezium Method.

DETERMINING THE LCF FOR A GIVEN DRAFT BY THE TRAPEZIUM METHOD



IN THIS PARTICULAR CASE, DRAFT 'd' IS AT THE SUMMER LOADLINE

STN	BEAM	MULTIPLIER	AREA PRODUCT	LEVER	MOMENT PRODUCT
0	B_0	0.25	$0.25(B_0)$	10.0 x C.I.	2.5 x C.I. (B_0)
0.5	$B_{0.5}$	0.5	$+\ 0.25(B_{0.5})$	9.5 x C.I.	+ 2.375 x C.I. $(B_{0.5})$
1	B_1	0.5	$+\ 0.5(B_1)$	9.0 x C.I.	+ 4.5 x C.I. (B_1)
1.5	$B_{1.5}$	0.5	$+\ 0.5(B_{1.5})$	8.5 x C.I.	+ 4.25 x C.I. $(B_{1.5})$
2	B_2	0.5	$+\ 0.5(B_2)$	8.0 x C.I.	+ 4.0 x C.I. (B_2)
3	B_3	0.5	$+\ 0.5(B_3)$	7.0 x C.I.	+ 3.5 x C.I. (B_3)
4	B_4	0.75	$+\ 0.75(B_4)$	6.0 x C.I.	+ 4.5 x C.I. (B_4)
5	B_5	1	$+\ (B_5)$	5.0 x C.I.	+ 5.0 x C.I. (B_5)
6	B_6	1	$+\ (B_6)$	4.0 x C.I.	+ 4.0 x C.I. (B_6)
9.5	$B_{9.5}$	0.5	$+\ 0.5(B_{9.5})$	0.5 x C.I.	+ 0.25 x C.I. $(0_{9.5})$
10	B_{10}	0.25	$0.25(B_{10})$	0 x C.I.	ZERO
			Σ AREA PRODUCT		Σ MOMENT PRODUCT

WATERPLANE AREA 'WPA' AT DRAFT 'd' = C.I. x Σ AREA PRODUCT M^2

So MOMENTS OF WPA ABOUT A.P = C.I. x Σ MOMENT PRODUCT M^3

So C of F LEVER ABOUT THE A.P. (THE *'LCF'*) = $\dfrac{\Sigma \text{ MOMENT PRODUCT}}{\Sigma \text{ AREA PRODUCT}}$ METRES

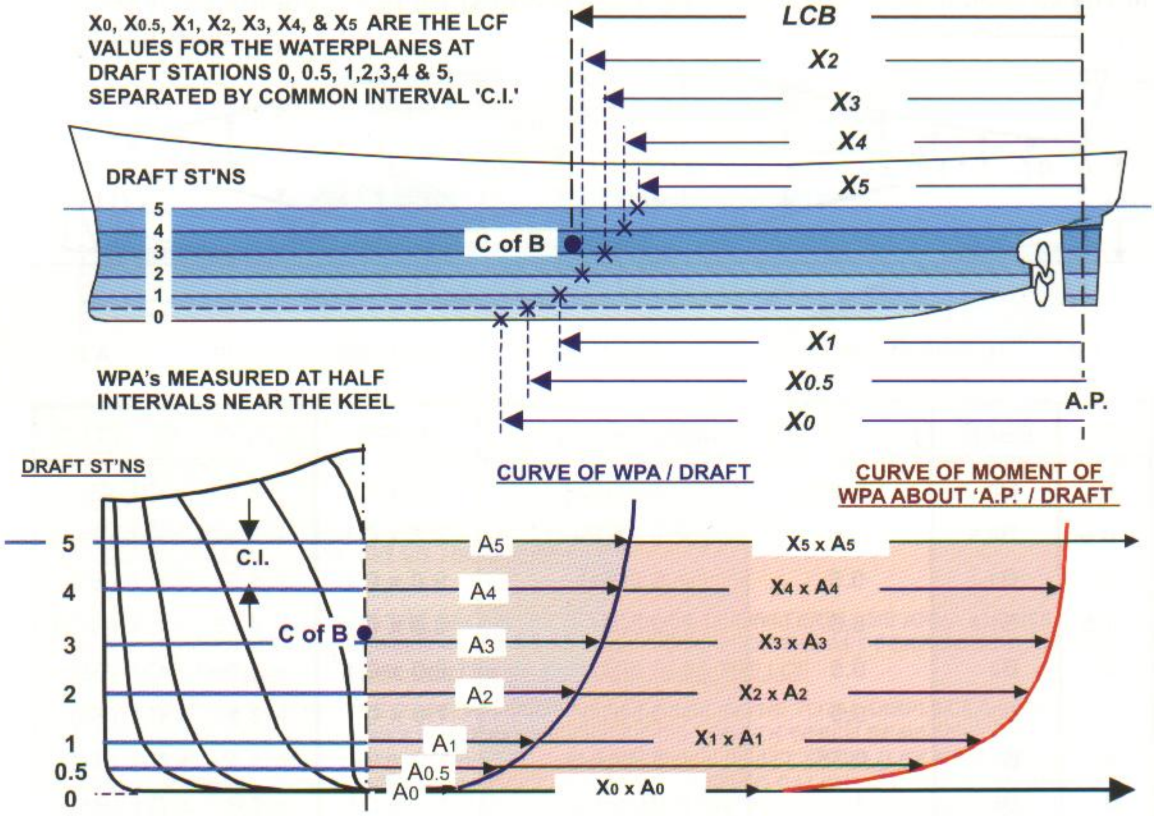
ANY APPENDAGES FWD OF ST'N 0 OR AFT OF ST'N 10 MUST BE CALCULATED SEPARATELY AND BE ADDED TO THE TWO SUMMATIONS. NOTE THAT THE LEVER OF ANY SUCH PART OF THE WATERPLANE AREA, AFT OF ST'N 10, WILL HAVE A NEGATIVE VALUE

IF WE ARE CALCULATING THE UPRIGHT VALUE OF THE LCF, WE COULD USE THE HALF BEAM VALUES, AS THE WATERPLANE AREA IS SYMMETRICAL ABOUT THE CENTRELINE

THE LONGITUDINAL CENTRE OF BUOYANCY (LCB)

The Centre of Floatation of a waterplane area is, in effect, the centre of buoyancy of a very thin horizontal slice of the submerged hull. If we divide the sum of the moments, about the aft perpendicular, of waterplane areas at regular draft intervals, from the keel up to the ship's waterline, by the displaced volume, we will obtain the distance, forward of aft perpendicular, of the Centre of Buoyancy for the entire underwater hullform. This distance is known as the 'LCB' and can be, again, determined by the methods of approximate integration.

DETERMINING THE LCB VALUE FOR A GIVEN DRAFT, BY THE TRAPEZIUM METHOD



THE SHADED AREAS ENCLOSED BY THE TWO CURVES GIVE THE DISPLACED VOLUME AND MOMENT OF VOLUME ABOUT THE AFT PERPENDICULAR 'A.P', SO, WE CAN USE THE TRAPEZIUM METHOD AS FOLLOWS

STN	WPA	MULTIPLIER	VOLUME PRODUCT	LEVER	MOMENT PRODUCT
0	A_0	0.25	$0.25(A_0)$	X_0	$0.25\ X_0\ (A_0)$
0.5	$A_{0.5}$	0.5	$+\ 0.5(A_{0.5})$	$X_{0.5}$	$+\ 0.5\ X_{0,5}\ (A_{0.5})$
1	A_1	0.75	$+\ 0.75(A_1)$	X_1	$+\ 0.75\ X_1\ (A_1)$
2	A_2	1	$+\ (A_2)$	X_2	$+\ X_2\ (A_2)$
3	A_3	1	$+\ (A_3)$	X_3	$+\ X_3\ (A_3)$
4	A_4	1	$+\ (A_4)$	X_4	$+\ X_4\ (A_4)$
5	A_5	1	$+\ (A_5)$	X_5	$+\ X_5\ (A_5)$
			Σ VOLUME PRODUCT		Σ MOMENT PRODUCT

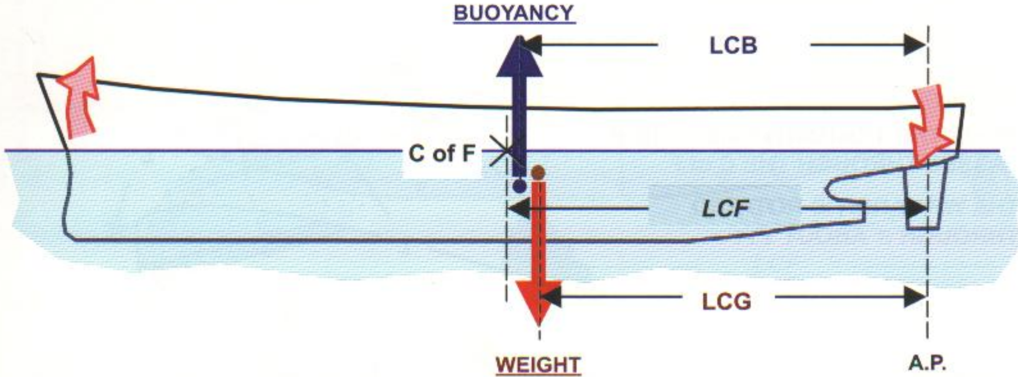
C of B LEVER ABOUT THE A.P. (THE *'LCB'*) $= \dfrac{\Sigma \text{ MOMENT PRODUCT}}{\Sigma \text{ VOLUME PRODUCT}}$ METRES

THE LCB, LCG AND TRIMMING MOMENTS

The LCB tends to be forward midships at light drafts, for most commercial hulls, due to the cutaway for the propeller aft. As draft increases, though, the LCB usually moves aft with further immersion of the fuller stern, particularly in the case of fine-lined hulls. However, for many large full-bodied ships with large bulbous bows, such as large tankers, the LCB may remain forward of the midships point. In these hulls, the 'pointed end' is actually the stern rather than the bow, at least as far as the underwater shape is concerned.

The force of buoyancy, acting through the LCB, will produce a trimming moment about the LCF, which will interact with the moment due to the weight distribution. This acts through the Longitudinal Centre of Gravity, the 'LCG', which is determined by summing the individual weights about the aft perpendicular and dividing the total moment by the total weight.
The two trimming moments about the LCF, due to weight and buoyancy, determine the actual trim of the vessel.

THE TRIMMING MOMENT ABOUT THE CENTRE OF FLOATATION



SUM OF MOMENTS ABOUT C of F = BUOYANCY (*LCF* - LCB) + WEIGHT (*LCF* - LCG)

But BUOYANCY = - WEIGHT

So MOMENTS ABOUT C of F = BUOYANCY (*LCF* - LCB - *LCF* + LCG)

So MOMENTS ABOUT C of F = BUOYANCY (LCG - LCB)

I.E. THE TRIMMING MOMENT = DISPLACED WEIGHT x (LCG - LCB) T-M

DETERMINING THE LIGHTSHIP LCG

When a new build is completed, the Lightship condition often has a considerable stern trim, which usually lies outside the normal hydrostatic data. The C of G must always be in vertical alignment with the C of B and so the LCG equals the LCB. However, the LCB can only be determined with an acceptable precision by placing sufficient weight onboard to bring the ship to even keel when the LCB can be accurately known from the hydrostatic data.
The trimming weight and its lcg are measured, whilst the LCB and displacement are obtained from the hydrostatic data for the observed draft. The trimming weight and its position should not cause significant bending moments to avoid excessive error in the draft readings.

$$\text{LCB} \times \text{Displaced weight } '\Delta t' = \text{LCG (lightship)} \times ('\Delta t' - 'w') + \text{lcg} \times 'w' \quad \text{T-M}$$

$$\text{So,} \quad \text{LCG (lightship)} = \frac{\text{LCB} \times '\Delta t' - \text{lcg} \times 'w'}{'\Delta t' - 'w'} \quad \text{Metres}$$

where 'w' is the trimming weight producing even keel, when placed at a known lcg from the A.P.

DETERMINING THE LONGITUDINAL BM VALUE FOR A SHIP-SHAPED HULL AT A GIVEN DRAFT

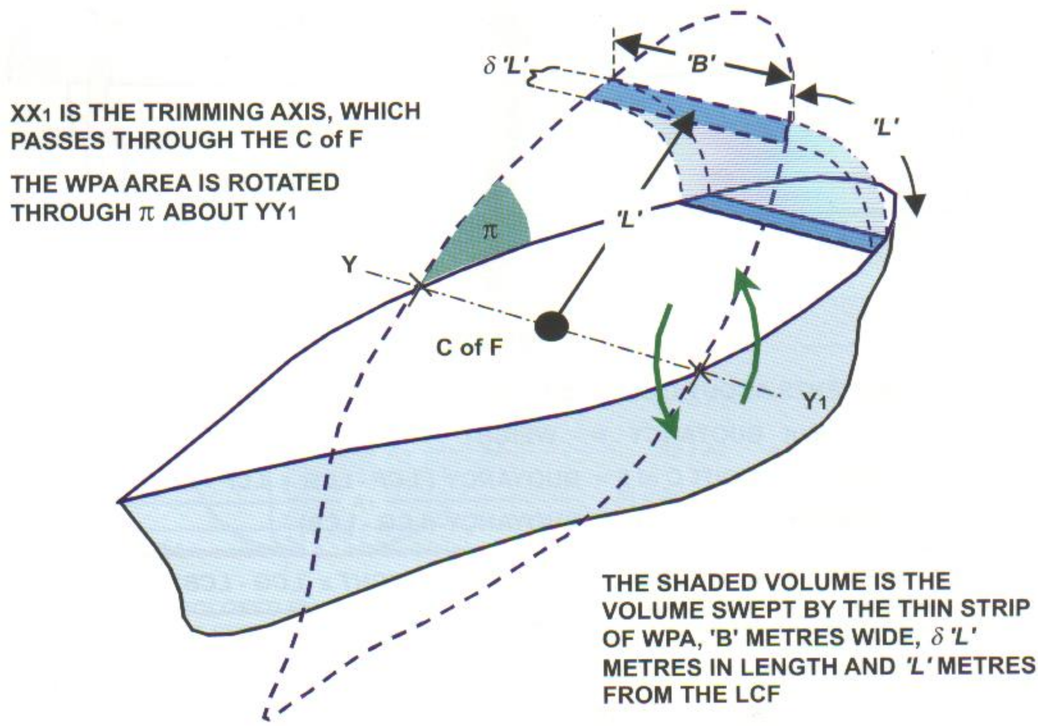
Chapter 2 (page 32) shows how the Transverse Moment of Waterplane Area, **'$I_{WPA(T)}$'**, relates to the Transverse BM value by the equation ;-

The radius of swing of the C of B, the 'BM' value $= \dfrac{I_{WPA}}{V\Delta_T}$ **Metres**

Where 'I_{WPA}' is the Moment of Waterplane Inertia <u>***about the axis of rotation***</u> ***and measured in (Metres)^4. '$V\Delta_T$' is the Displaced Volume of the hull, measured in (Metres)^3.***
The Moment of Inertia is also defined on page 32 by the following :-
The Rotational Volumetric Moment of Inertia of an area about any axis, is the moment of 'THE SWEPT VOLUME' / RADIAN OF ROTATION, of that area about that axis

We can use the above definition of 'Moment of Inertia' to find an expression for the Longitudinal Trimming Moment of Waterplane Inertia and so be able to determine the Longitudinal BM value.

THE LONGITUDINAL MOMENT OF WATERPLANE INERTIA



SWEPT VOLUME OF WATERPLANE STRIP = $B \times 'L' \times \delta 'L'$ M^3

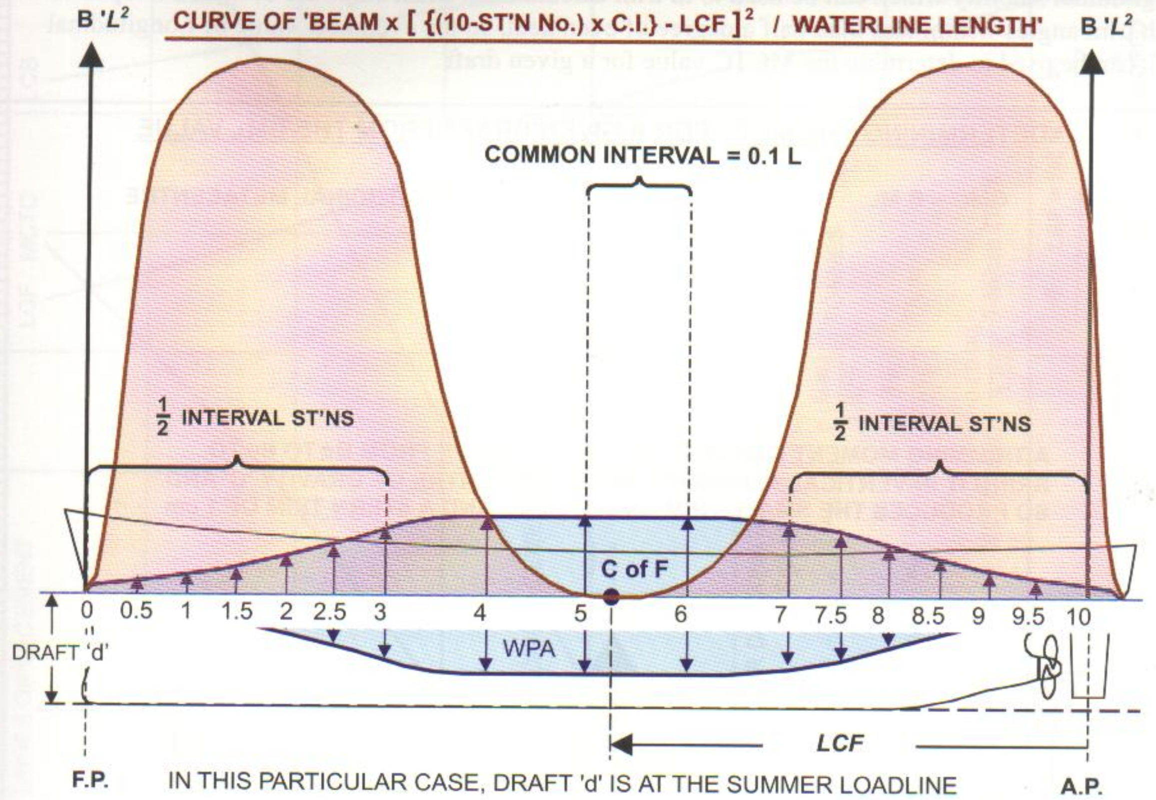
MOMENT OF SWEPT VOLUME ABOUT XX1 = $B \times 'L^2' \times \delta 'L'$ M^4

So, THE TOTAL MOMENT OF WATERPLANE INERTIA ABOUT XX1 IS THE SUM OF THE ALL THE MOMENTS ABOUT XX1 OF THE STRIPS WHICH MAKE UP THE TOTAL WATERPLANE AREA. I.E. :-

THE LONGITUDINAL MOMENT OF WPA INERTIA = $\displaystyle\int_{A.P.}^{F.P.} B\,'L^2'\,d'L'$ M^4

So the Longitudinal Moment of WPA Inertia for a ship-shaped hull, at a given draft, is equal to the area under a curve of 'B x L^2 over the ship's waterline length. This can be estimated by applying either Simpson's Rules or the Trapezium Method of approximate integration, to 'B' and 'L' values, measured at regular station intervals along the length of the waterplane at the required draft.

DETERMINING THE LONGITUDINAL BM VALUE FOR A SHIP-SHAPED HULL AT A GIVEN DRAFT (Cont.)



APPLYING THE TRAPEZIUM METHOD TO THE BEAM AND LENGTH MEASUREMENTS

ST'N	BEAM	(LEVER 'L')² = [C.I. (10-ST'N No.) -LCF]²	MULTIPLIER	MOMENT PRODUCT
0	0	x [10 C.I. -LCF]² x	0.25 =	0
0.5	$B_{0.5}$	x [9.5 C.I. -LCF]² x	0.5 =	+ 0.5[9.5 C.I. -LCF]² x $B_{0.5}$
1	B_1	x [9.0 C.I. -LCF]² x	0.5 =	+ 0.5[9.0 C.I. -LCF]² x B_1
1.5	$B_{1.5}$	x [8.5 C.I. -LCF]² x	0.5 =	+ 0.5[8.5 C.I. -LCF]² x $B_{1.5}$
2	B_2	x [8.0 C.I. -LCF]² x	0.5 =	+ 0.5[8.0 C.I. -LCF]² x B_2
2.5	$B_{2.5}$	x [7.5 C.I. -LCF]² x	0.75 =	+0.75[7.5 C.I. -LCF]² x $B_{2.5}$
3	B_3	x [7.0 C.I. -LCF]² x	1 =	+ [7.0 C.I. -LCF]² x B_3
4	B_4	x [6.0 C.I. -LCF]² x	1 =	+ [6.0 C.I. -LCF]² x B_2
9.5	$B_{9.5}$	x [0.5 C.I. -LCF]² x	0.5 =	+ 0.5[0.5 C.I. -LCF]² x $B_{9.5}$
10	B_1	x [ZERO. -LCF]² x	0.25 =	+ 0.25[-LCF]² x B_{10}
				Σ MOMENT PRODUCT

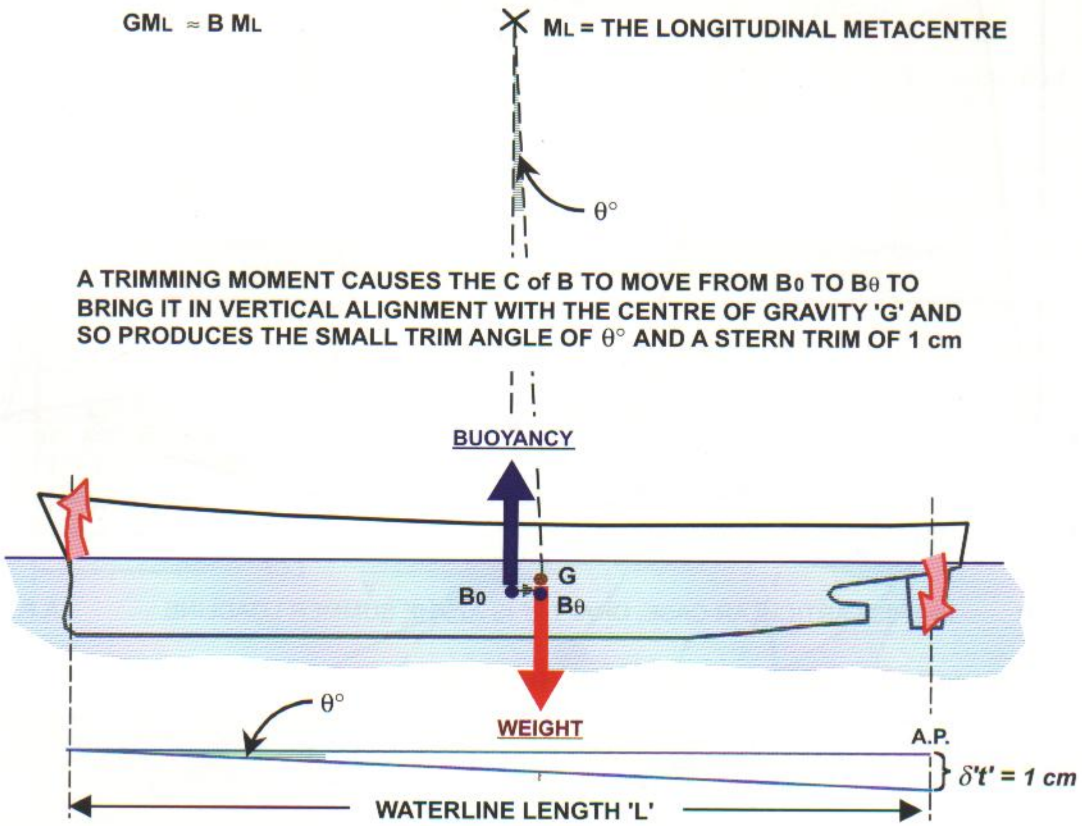
THE LONGITUDINAL BM VALUE AT THIS DRAFT $= \dfrac{\text{C.I. } \Sigma \text{ MOMENT PRODUCT}}{\text{DISPLACED VOLUME}}$ **METRES**

The shape of the ends of the waterplane have the greatest effect upon the value of the Longitudinal BM and half, or even quarter, station measurements should be taken in these regions. Bow and stern flare will significantly change these at even small angles of trim, so the BML value and the ship's resistance to changing trim, will increase considerably with relatively small increases in trim.

THE MOMENT REQUIRED TO CHANGE A HULL'S TRIM BY 1 CM (M.C.T.C.)

The Moment required to produce a one centimetre change in trim is a common measure of a ship's longitudinal stability which can be used to in trim calculations. Most ships are designed to operate with trim angles within less than half a degree of even keel, so the even keel value of Longitudinal BMl can be used to determine the MCTC value for a given draft

DETERMINING THE MCTC FOR A GIVEN DRAFT FROM THE BML VALUE



TRIMMING MOMENT = DISPLACEMENT WEIGHT 'W' x SHIFT IN THE C of B

So TRIMMING MOMENT = W x BM_L Sin θ ***WHERE Sin θ =*** $\dfrac{\text{CHANGE OF TRIM } \delta't'}{\text{W/LINE LENGTH } 'L'}$

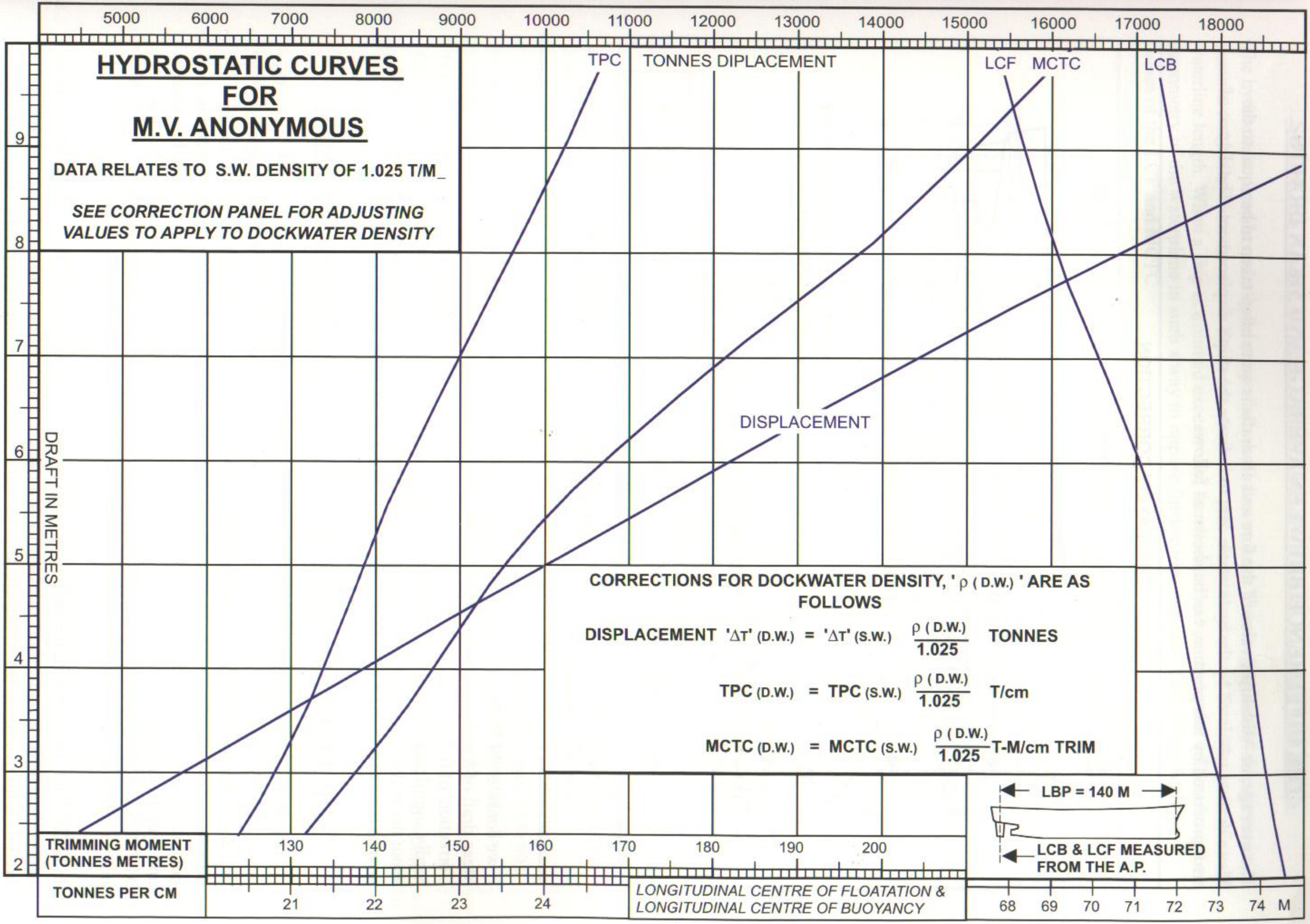
AS θ IS VERY SMALL AND Sin θ APPROXIMATES TO Tan θ

So TRIMMING MOMENT = $\dfrac{W \times BM_L \times \delta't'}{'L'}$ **TONNES - METRES**

IF δ't' IS EQUAL TO 1 CENTIMETRE, THEN THE TRIMMING MOMENT IS THE MCTC

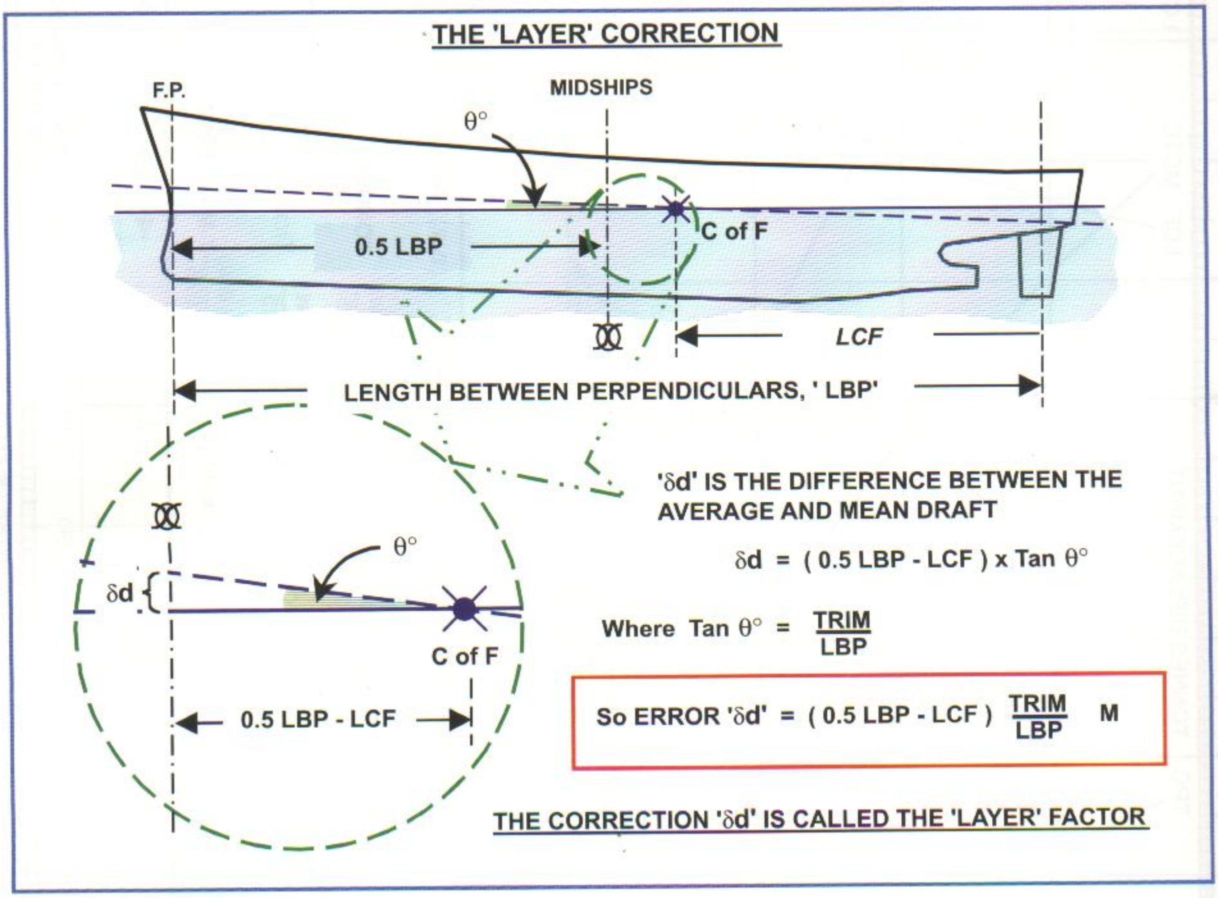
$$\boxed{\text{So} \quad MCTC = \frac{W \times BM_L \times \delta't'}{'L'} \quad \text{T-M / cm CHANGE OF TRIM}}$$

The ship's hydrostatic data provides Values of MCTC, LCB, LCF, TPC (Tonnes per cm, *See Chapter 1, page 17*) and Salt Water Displacement at regular draft intervals. The data must cover survivable flooded conditions, so it will range from the Lightship condition to just beyond the deepest allowable draft (Usually the Tropical Fresh Water mark). It is usually given in the form of tables at draft intervals of typically every 10 centimetres, but it may be provided in graphical form. The LCB and LCF used to be measured from the midships mark and so they were expressed in metres forward or aft of midships. This practise has given way to using the aft perpendicular as the reference point, but some older vessels may still have the data in this form.


HYDROSTATIC CURVES FOR M.V. ANONYMOUS
DATA RELATES TO S.W. DENSITY OF 1.025 T/M_
SEE CORRECTION PANEL FOR ADJUSTING VALUES TO APPLY TO DOCKWATER DENSITY
TONNES DIPLACEMENT
5000 6000 7000 8000 9000 10000 11000 12000 13000 14000 15000 16000 17000 18000
TPC
LCF
MCTC
LCB
DISPLACEMENT
DRAFT IN METRES
2 3 4 5 6 7 8 9
CORRECTIONS FOR DOCKWATER DENSITY, ' ρ (D.W.) ' ARE AS FOLLOWS
DISPLACEMENT 'ΔT' (D.W.) = 'ΔT' (S.W.) ρ (D.W.) / 1.025 TONNES
TPC (D.W.) = TPC (S.W.) ρ (D.W.) / 1.025 T/cm
MCTC (D.W.) = MCTC (S.W.) ρ (D.W.) / 1.025 T-M/cm TRIM
LBP = 140 M
LCB & LCF MEASURED FROM THE A.P.
TRIMMING MOMENT (TONNES METRES)
130 140 150 160 170 180 190 200
TONNES PER CM
21 22 23 24
LONGITUDINAL CENTRE OF FLOATATION & LONGITUDINAL CENTRE OF BUOYANCY
68 69 70 71 72 73 74 M

THE DIFFERENCE BETWEEN AVERAGE AND MEAN DRAFTS

The **average, or Midships,** value of the fore and aft drafts is quite often taken to be equal to the **Mean** draft as tabulated by the hydrostatic data to indicate the ship's displacement. The degree of error produced by this practice, can be shown as follows:-



If we consider 'M.V. Anonymous' with an LBP of 140 metres and a '1' metre trim by the stern at a Mean draft of 8 metres, then the LCF is 1 metre aft of midships. This results in a calculated 'Layer Correction' of 0.7 cm, which represents about 16 Tonnes for a total displacement of 16,800 T. This is hardly significant and, in most situations, it is reasonable to assume that commercial cargo ships trim about the midships point. Hence, the average draft approximates to the mean draft. **However, when answering examination questions, candidates must apply trim about the LCF.**

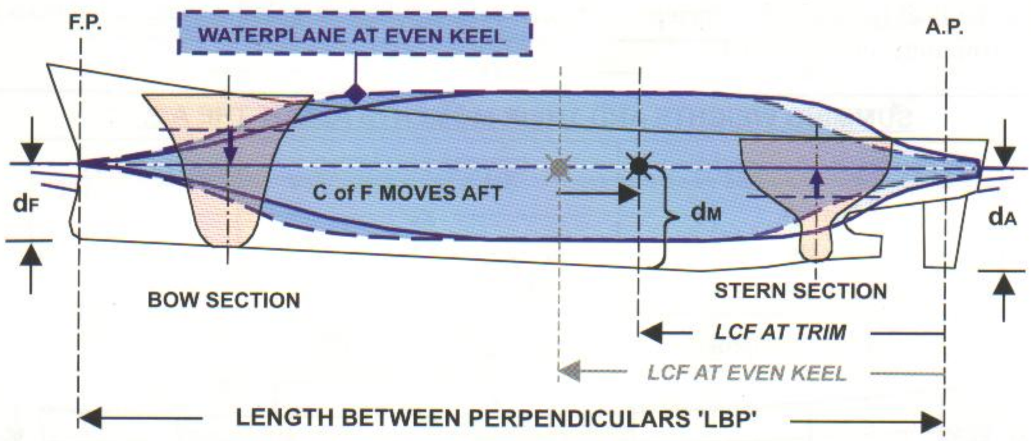
THE EFFECT OF WATER DENSITY ON THE HYDROSTATIC DATA

Hydrostatic data almost always relates to the vessel floating in saltwater of density 1.025 T/M^3. The values of LCB and LCF at any given draft, are purely determined by the underwater geometry of the hull and so are not effected by water density. However, Displacement 'Δ_T', TPC and MCTC all decrease proportionally with water density and the tabulated values should be corrected for this, if dockwater density is less than 1.025 T/M^3. Put simply, lower density water provides less buoyancy and is easier to push out of the way, so at any given draft, the values of Displacement, TPC and MCTC will be proportionally less than the tabulated S.W. values, if the dockwater is less dense than normal saltwater. The correction equations are shown with the Hydrostatic Curves on page 134.

In reality, any error incurred through using uncorrected saltwater values of MCTC and TPC is almost negligible. **However, questions in examination papers for certificates of competency often include trim problems involving changes in water density. Failure to correct hydrostatic data for density has, in the past, been severely penalised by some examination schemes.**

The hydrostatic data also usually relates to the vessel at even keel. This is quite acceptable for most commercial hulls when the trim is limited to about 0.6°, i.e. about 1 metre trim for 100 metres of waterline length. When a ship is trimmed excessively, the sheer at the bow and stern will increase the asymmetry of the waterplane in such a way to oppose further trim increase. This effects both the values of the LCF and MCTC.

THE CHANGE OF WATERPLANE WITH INCREASES IN STERN TRIM



THE WATERPLANE BECOMES MORE ASYMMETRICAL WITH INCREASING STERN TRIM. IMMERSED FLARE AT THE STERN INCREASES THE WPA IN THE AFT REGIONS, WHILST FLARE AT THE BOW REDUCES IT IN THE FWD REGION, SO THE LCF MOVES AFT. THE MCTC ALSO INCREASES, WITH REGARD TO RESISTING FURTHER STERN TRIM

This effect will result in the hull adopting a smaller trim than will be predicted by calculations based upon the even keel value of the MCTC. The resulting error is unlikely to be significant except in cases of extreme trim conditions or unusual hullforms. There are some specialised vessels with exceptional asymmetry in the hull lines between the bow and the stern. This is usually to produce a wide working aft deck, such as in the case of oil field support vessels (see Chapter 3, pages 65 to 66). This design of ship requires a greater analysis of the hullform to provide information regarding its trimming characteristics and how these interact with the rolling motion of the ship to effect transverse stability (the 'Free Trim' effect).

Trim calculations are only as accurate as the estimation of the longitudinal weight distribution within the ship. This is particularly imprecise for ships loaded with break bulk (or general) cargo.

The calculations also assume that the hull remains rigidly straight, regardless of the weight distribution. Most cargo carrying commercial hulls are designed to sag (see Chapter 1 page 19 and Chapter 10) when near to Lightship conditions and to hog (i.e. droop in the bow and stern) when fully laden. These bending moment effects will result in the calculations over-estimating the actual fore and aft drafts when the hull is sagged and under-estimating the drafts for a hogged hull.

Calculated drafts should be compared to the observed readings of the hull marks whenever possible, but these observations are unlikely to be better than within about 2 cm of the actual draft, even when the marks are clearly visible in still water conditions.

It should be appreciated that draft calculations and observed readings in port, indicate the **static draft.** There are dynamic effects, known as **Squat**, which bodily draws the hull deeper into the water if the ship is sailing in very shallow water (i.e. where the bottom clearance is less than half the deepest draft). **Increases in draft of up to 10% can occur if the ship is moving at speed in such a situation.**. The onset of significant squat is indicated by excessive vibration about the stern accompanied by sluggishness in the ship's response to extra power and helm actions.
Squat is reduced by slowing down the ship's speed in very shallow water conditions.

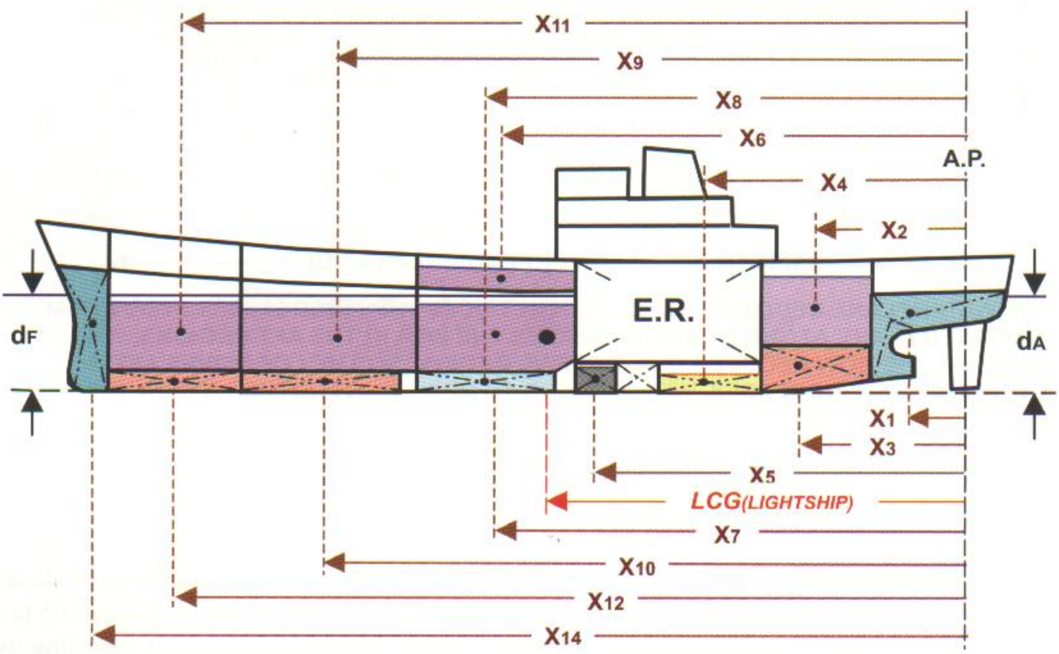
CALCULATING THE TRIM AND DRAFT FOR A LOADED VESSEL

A trimming moment exists if there is a horizontal separation between the Longitudinal Centres of Gravity and Buoyancy, i.e.

TRIMMING MOMENT = (LCB - LCF) x DISPLACEMENT T-M

We can calculate the LCG by summing up the individual weight moments about the Aft Perpendicular. The sum of the weights will give the displacement and then we can look up the mean draft, LCF, MCTC and LCB in the hydrostatic tables, applying a dock water (D.W.) correction as required. The difference between the LCB and LCG will give the trimming lever for the vessel in this condition, if it is at even keel. The resulting trimming moment is then divided by the MCTC (D.W.) to provide the final trim, which is then proportioned between the bow and the stern on the basis of the vessel trimming about the LCF.

SUMMING WEIGHTS AND THEIR MOMENTS ABOUT THE A.P.



WEIGHT	LEVER ABOUT A.P.	TRIMMING MOMENT ABOUT A.P.
LIGHTSHIP Δ_T	$LCG_{(LIGHTSHIP)}$	LIGHTSHIP Δ_T x $LCG_{(LIGHTSHIP)}$
+ W_1	X_1	+ $W_1 \times X_1$
+ W_2	X_2	+ $W_2 \times X_2$
+ W_3	X_3	+ $W_3 \times X_3$
+ W_{13}	X_{13}	+ $W_{13} \times X_{13}$
+ W_{14}	X_{14}	+ $W_{14} \times X_{14}$
Σ (WEIGHTS)		Σ (TRIMMING MOMENTS)

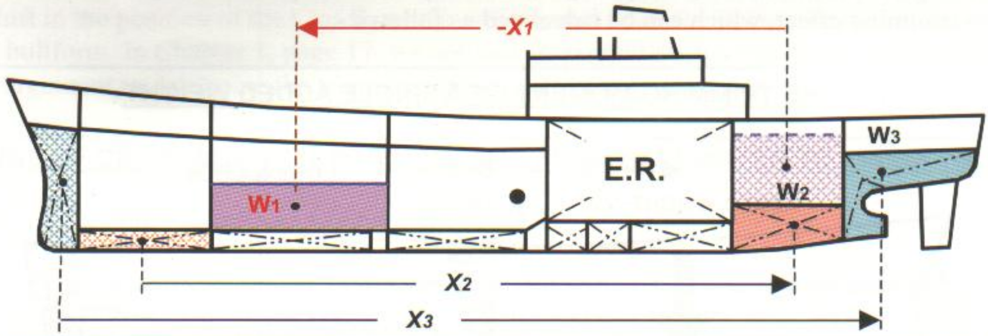
THE Σ (WEIGHTS) EQUALS THE LOADED DISPLACEMENT AND IS USED TO DETERMINE THE MEAN DRAFT 'd_M', LCB, LCF, AND MCTC FROM THE HYDROSTATIC DATA, WHICH ALSO LISTS THE lcg VALUES OF ALL THE TANK AND FULL CARGO COMPARTMENTS,

$$\text{TRIM} = \frac{\Sigma\,(\text{TRIMMING MOMENTS})}{100 \times \text{MCTC}_{(D.W.)}} \text{ M}$$

$$\text{AFT DRAFT } 'd_A' = d_M \pm \text{TRIM} \frac{LCF}{LBP} \text{ M} \quad \& \quad \text{FWD DRAFT } 'd_F' = d_M \mp \text{TRIM} \frac{LBP - LCF}{LBP} \text{ M}$$

SOME PRACTICAL CHANGE OF TRIM CALCULATIONS

TRIMMING MOMENT DUE TO SHIFTING WEIGHTS



WEIGHT	DISTANCE MOVED	TRIMMING MOMENT
W_1	$-X_1$	$- \; W_1 \times X_1$
W_2	X_2	$+ \; W_2 \times X_2$
W_3	X_3	$+ \; W_3 \times X_3$
-IVE VALUES INDICATE A TRIM BY THE HEAD		Σ (TRIMMING MOMENTS)

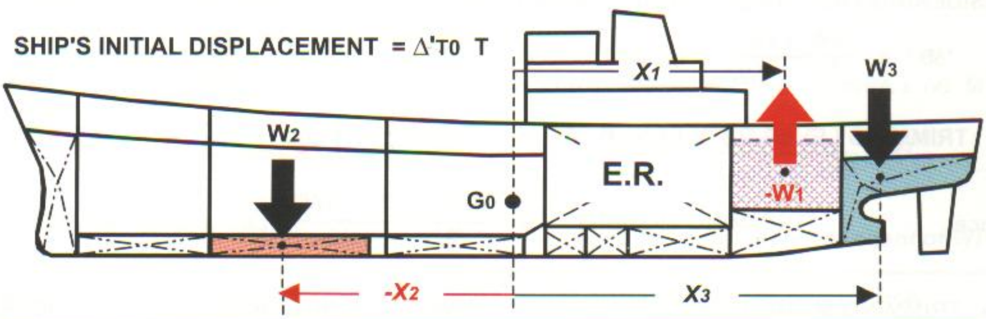
THE SUM OF THE INDIVIDUAL MOMENTS = DISPLACED WEIGHT x SHIFT IN C of G

So, THE SUM OF THE INDIVIDUAL MOMENTS = THE TRIMMING MOMENT T - M

THEREFORE :- THE CHANGE OF TRIM = $\dfrac{\Sigma \text{(TRIMMING MOMENTS)}}{100 \times \text{MCTC (D.W.)}}$ METRES

TRIMMING MOMENT DUE TO LOADING OR DISCHARGING WEIGHTS

SHIP'S INITIAL DISPLACEMENT = Δ'_{T0} T



WEIGHT	LEVER FROM INITIAL C of G	TRIMMING MOMENT
$-W_1$	X_1	$- \; W_1 \times X_1$
$+W_2$	$-X_2$	$- \; W_2 \times X_2$
$+W_3$	X_3	$+ \; W_3 \times X_3$
Σ (WEIGHT)		Σ (TRIMMING MOMENTS)

DISCHARGED WEIGHTS AND LEVERS FWD OF THE INITIAL C of G ARE NEGATIVE

NEW DISPLACEMENT = $\Delta'_{T0} \pm \Sigma$ (WEIGHT) T, & SHIFT IN THE C of G = $\dfrac{\Sigma \text{(MOMENTS)}}{\Delta_{T0} + \Sigma \text{(WEIGHT)}}$ M

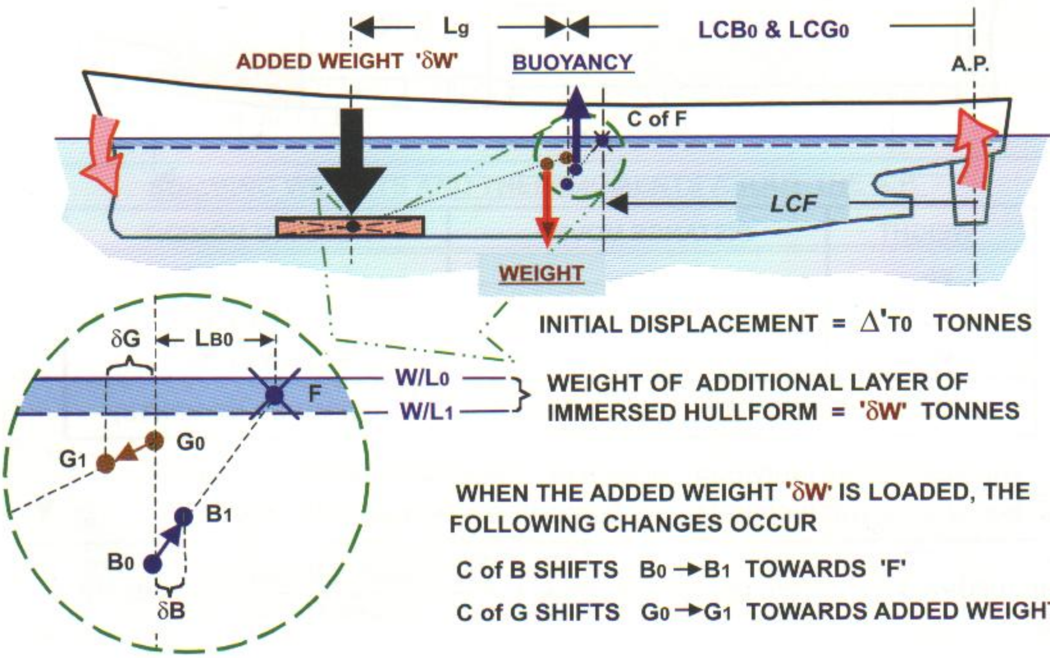
THE SHIFT IN THE C of G IS USED TO DETERMINE THE NEW LCG, WHILST THE HYDROSTATIC DATA WILL GIVE THE NEW MEAN DRAFT LCB , MCTC AND LCF FOR THE NEW DISPLACEMENT. THE CHANGE OF WEIGHT MAY BE +IVE OR -IVE, DEPENDING WHETHER MORE WEIGHT IS LOADED OR DISCHARGED

TRIM AT NEW DRAFT = $\dfrac{(\text{LCB} - \text{LCG}) \times [\Delta'_{T0} \pm \Sigma \text{(WEIGHT)}]}{100 \times \text{MCTC (D.W.)}}$ METRES

THE CHANGE OF TRIM DUE TO A SINGLE ADDED WEIGHT

When a single weight is loaded, both the weight distribution within the ship and its underwater hullform change, so the Centres of Buoyancy and Gravity will shift position. Each of these two shifts will have a trimming effect, which can be calculated as follows:-.

THE TRIMMING MOMENT OF A SINGLE ADDED WEIGHT



CONSIDERING THE LONGITUDINAL SHIFTS IN THE CENTRES OF BUOYANCY AND GRAVITY

$$'\delta B' = \frac{'\delta W' \times L_{B0}}{\Delta'_{T0} + '\delta W'} \text{ M AFT} \quad \& \quad '\delta G' = \frac{'\delta W' \times L_g}{\Delta'_{T0} + '\delta W'} \text{ M FWD}$$

So, TRIMMING LEVER BETWEEN $B_1 G_1 = \dfrac{'\delta W'}{\Delta'_{T0} + '\delta W'} \left[L_{B0} + L_g \right]$ METRES

Hence, TRIMMING MOMENT $= \dfrac{'\delta W' (\Delta'_{T0} + '\delta W')}{\Delta'_{T0} + '\delta W'} \left[L_{B0} + L_g \right]$ T - METRES

So, TRIMMING MOMENT DUE TO 'δW' $=$ 'δW' [DISTANCE FROM LCF] T - METRES

Hence, CHANGE OF TRIM DUE TO 'δW' $= \dfrac{'\delta W' \text{ [DISTANCE FROM LCF]}}{100 \times \text{MCTC}_{(D.W.)}}$ METRES

And INCREASE IN MEAN DRAFT $= \dfrac{'\delta W'}{\text{TPC}_{(D.W.)}}$ METRES

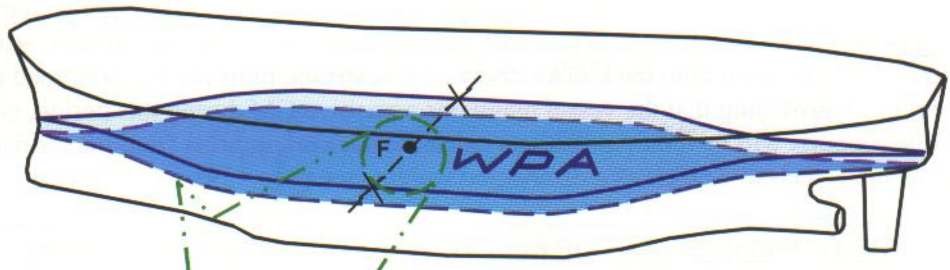
SO THE TRIMMING MOMENT DUE TO ADDING 'δW', IS THE MOMENT OF 'δW' ABOUT THE C of F. IF THE WEIGHT IS BEING DISCHARGED, RATHER THAN LOADED, THEN THE SHIFTS '$B_0 B_1$' AND '$G_0 G_1$' WOULD BE IN THE OPPOSITE DIRECTIONS, I.E. AWAY FROM THE C of F AND 'δW'. THE EQUATION WOULD BE THE SAME, BUT THE ACTION OF THE TRIMMING MOMENT WOULD BE REVERSED (I.E. IT WOULD BE BY THE STERN).

Trimming moments can be taken around any point along the ship's length and it is more convenient to measure them about the C of F if the draft change is small so that the LCF remains constant during the loading or discharging operation. If more than one weight is being loaded or discharged, then the total moment would be the sum of all the moments, taken about the Centre of Floatation.

THE CHANGE OF TRIM MOVING FROM SALT WATER TO FRESH WATER.

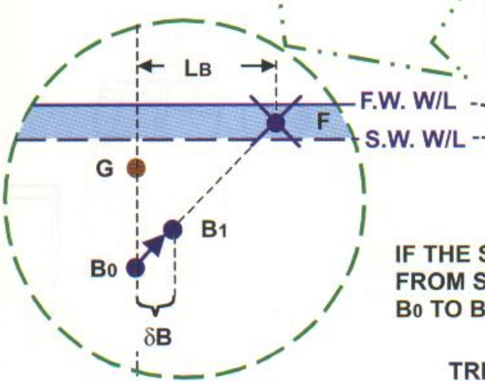
In this case, the underwater hullform changes due to the bodily sinkage of the Freshwater Allowance (the **FWA**) but the weight distribution remains constant. The resulting trimming moment is solely due to a shift in the position of the C of B, which will depend upon the volume of the additional immersed hullform. In Chapter 1, page 17, we derived an expression for this that is used in the following argument to determine the trimming effect of changing water density.

CHANGE OF DRAFT AND TRIM WHEN MOVING FROM SALT WATER TO FRESH WATER



S.W. DENSITY = 1.025 T/M^3, F.W. DENSITY = 1 T/M^3

SHIP'S DISPLACEMENT = Δ'_T TONNES



CHANGE OF DRAFT 'δd' = $\dfrac{\Delta'_T}{4000 \times TPC_{(SW)}}$ M

SEE CHAPTER 1, PAGE 15

IF THE S.W. DRAFT IS MAINTAINED AS THE SHIP MOVES FROM S.W. TO F.W., THEN THE C of B WOULD SHIFT FROM B_0 TO B_1, WHICH IS OUT OF VERTICAL LINE WITH G.

TRIMMING MOMENT = $\Delta'_T \times B_0 B_1$ T-M

THE WPA = $\dfrac{TPC_{SW}}{0.01025}$ M^3 & INCREASE IN IMMERSED VOLUME 'δV' = WPA x δd M^3

So INCREASE IN IMMERSED VOLUME 'δV' = $\dfrac{TPC_{(SW)} \times \Delta'_T}{0.01025 \times 4000 \times TPC_{(SW)}}$ M^3

Hence INCREASE IN IMMERSED VOLUME 'δV' = $\dfrac{\Delta'_T}{41}$ M^3 *and is independent of WPA*

AS THE DENSITY OF FRESH WATER IS 1.00 T/M^3, THEN THE WEIGHT OF THE EXTRA IMMERSED VOLUME IS ALSO EQUAL TO 'δV'

So, TRIMMING MOMENT = $\dfrac{\Delta'_T \times L_{B0}}{41}$ T- M

Hence, CHANGE OF TRIM = $\dfrac{\Delta'_T \times L_{B0} \times 1.025}{4100 \times MCTC_{(SW)}}$ M

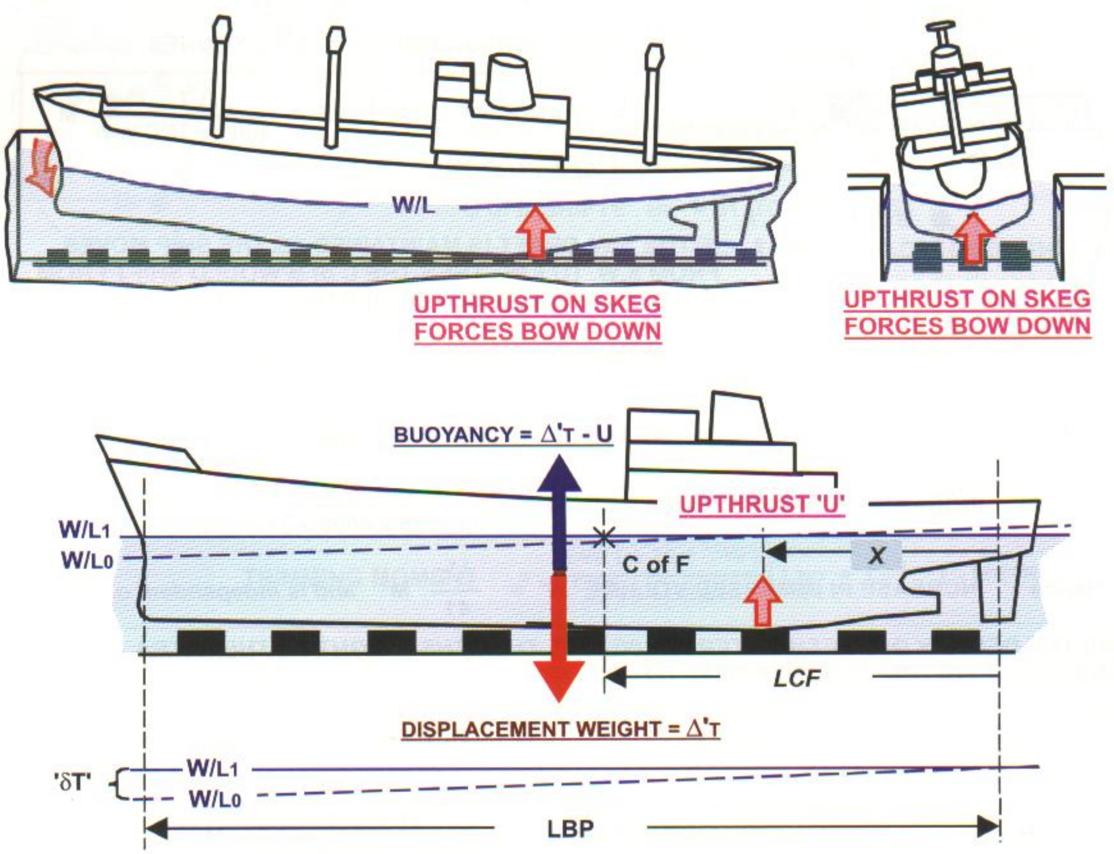
WHERE 'L_{B0}' IS THE DIFFERENCE BETWEEN THE VESSEL'S SALTWATER LCF AND LCG
NOTE THAT THE S.W. MCTC VALUE MUST BE CORRECTED FOR FRESHWATER

If we use the hydrostatic curves for M.V. Anonymous, on page 134, then at an 8 Metre even keel draft in salt water, the vessel has a displacement of 16800 T, a TPC of 23.2 T/cm, and an MCTC value of 197 T - Metres. The LCB is 4 Metres ahead of the LCF.
Putting these figures into the above equations, produces an increase in the mean draft of 18 cm and a trim of 8.3.cm by the stern, when the ship moves into fresh water so, although the trimming effect is not particularly large, it is significant when compared with the bodily sinkage.

TRIM AND STABILITY CONSIDERATIONS DURING DRYDOCKING

Drydocking is an operation in which the trim of a ship has a direct effect upon the changing transverse stability of the vessel as support of its weight is gradually transferred from actual buoyancy, acting through the Centre of Buoyancy, to upthrust acting directly on the ship's bottom. This effectively lowers the point on the ship's centreline, through which the upwards forces act or, in other words, the transverse KM value will gradually reduce to zero as the displaced buoyancy disappears. This means at some point the ship will lose positive transverse stability and before this happens, the vessel must be properly supported to prevent it from toppling over.
Ships generally enter the drydock with a slight stern trim so, initially, only the stern touches the blocks and there is the opportunity to make final adjustments before the hull is sat down on the blocks along its entire length. The ship is pivoted about the point of contact on the keel, as increasing upthrust forces the bow down onto the blocks. Most cargo carrying hulls are flat bottomed in the midships region so, providing that the vessel maintains a positive GM during this period, no additional side supports will be required. We can calculate the keel upthrust from the change of trim it produces, as it has the same effect as a discharged weight (see page 139).

THE TRIMMING EFFECT OF KEEL UPTHRUST DURING DRYDOCKING



AT ANY TIME AFTER TOUCHDOWN, THE EFFECT OF THE UPTHRUST 'U' ON THE TRIM, IS THE SAME AS IF A WEIGHT HAD BEEN DISCHARGED FROM THE POINT OF CONTACT ON THE HULL.

So, TRIMMING MOMENT OF 'U' = 'U' x (DISTANCE FROM LCF) T-M

Hence, $$\text{CHANGE OF TRIM } '\delta T' = \frac{'U'\,[LCF - X]}{100 \times MCTC_{(D.W.)}} \quad \text{METRES}$$

So, $$\text{UPTHRUST } 'U' = \frac{'\delta T'\,[100 \times MCTC_{(D.W.)}]}{LCF - X} \quad \text{TONNES}$$

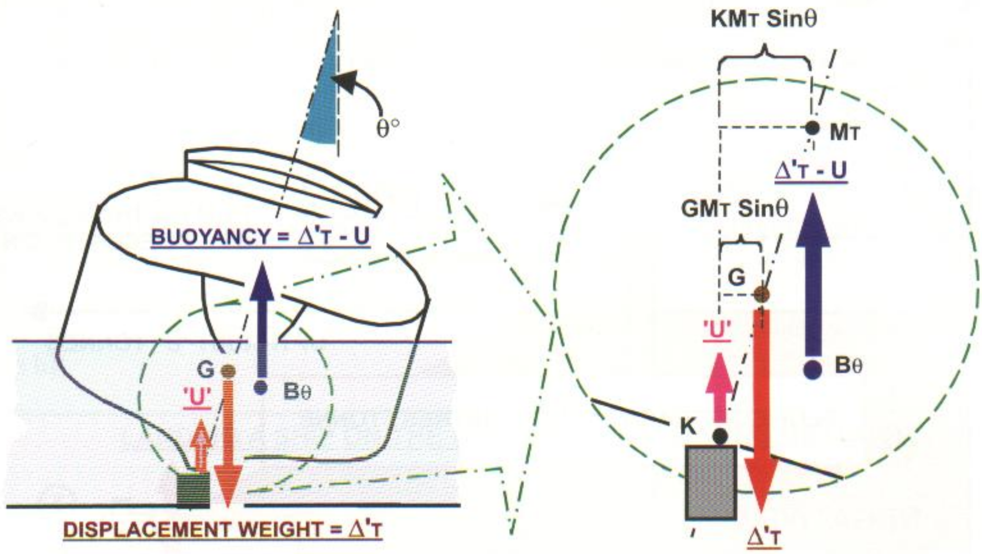
IF WE KNOW THE CHANGE OF TRIM REQUIRED TO LAND THE SHIP ON THE BLOCKS OVER ITS ENTIRE LENGTH, THEN WE CAN CALCULATE THE UPTHRUST AT THIS STAGE OF THE DOCKING

The Upthrust will usually have to be sufficient to force the ship's trim to even keel before the hull lands on the blocks along its entire length. However, some drydock floors slope downwards towards the dock gate. This is known as the dock **'Declivity'** and is usually expressed in terms of metres rise per 100 metres. The ship will therefore take the blocks fully before it reaches even keel if the dock has declivity and so:-

Change of Trim to land vessel = Free floating trim - (Declivity) LBP/100 Metres

As the water level falls, upwards support is progressively transferred from Buoyancy, acting through the Metacentre, to Upthrust acting upon the keel. This reduces the ship's GM_T as follows:-

THE LOSS OF GM_T DURING DRYDOCKING



THE SHIP DEVELOPS A SMALL ANGLE OF HEEL 'θ° ON TAKING THE BLOCKS

TO DETERMINE THE NET RIGHTING MOMENT, WE CAN TAKE MOMENTS ABOUT ANY POINT ON THE CENTRELINE, SO CONSIDERING MOMENTS ABOUT THE KEEL AT POINT 'K'

RIGHTING MOMENT = $(\Delta'_T - U) \times KM_T \sin\theta°$ & **CAPSIZING MOMENT** = $\Delta'_T \times KG \sin\theta°$

So, **NET RIGHTING MOMENT** = $\sin\theta° \, (KM_T \times \Delta'_T - KM_T \times U - \Delta'_T \times KG)$ T-M

And **NET RIGHTING LEVER** = $\dfrac{\text{NET RIGHTING MOMENT}}{\text{DISPLACEMENT } '\Delta_T'}$ METRES

Hence **NET RIGHTING LEVER** = $\sin\theta° \left(KM_T - KM_T \dfrac{U}{\Delta'_T} - KG \right)$ METRES

BUT (KM_T - KG) IS EQUAL TO THE INITIAL GM_T VALUE, PRIOR TO TAKING THE BLOCKS

So, **NET RIGHTING LEVER** = $\sin\theta° \left(GM_T - KM_T \dfrac{U}{\Delta'_T} \right)$ METRES

THE EFFECTIVE GM_T VALUE IS EQUAL TO THE NET RIGHTING LEVER DIVIDED BY Sin θ°

Hence, **LOSS OF GM_T** = $KM_T \dfrac{U}{\Delta'_T}$ METRES

And **NET RIGHTING MOMENT** = $\Delta'_T \sin\theta° \left(GM_T - KM_T \dfrac{U}{\Delta'_T} \right)$ T-M

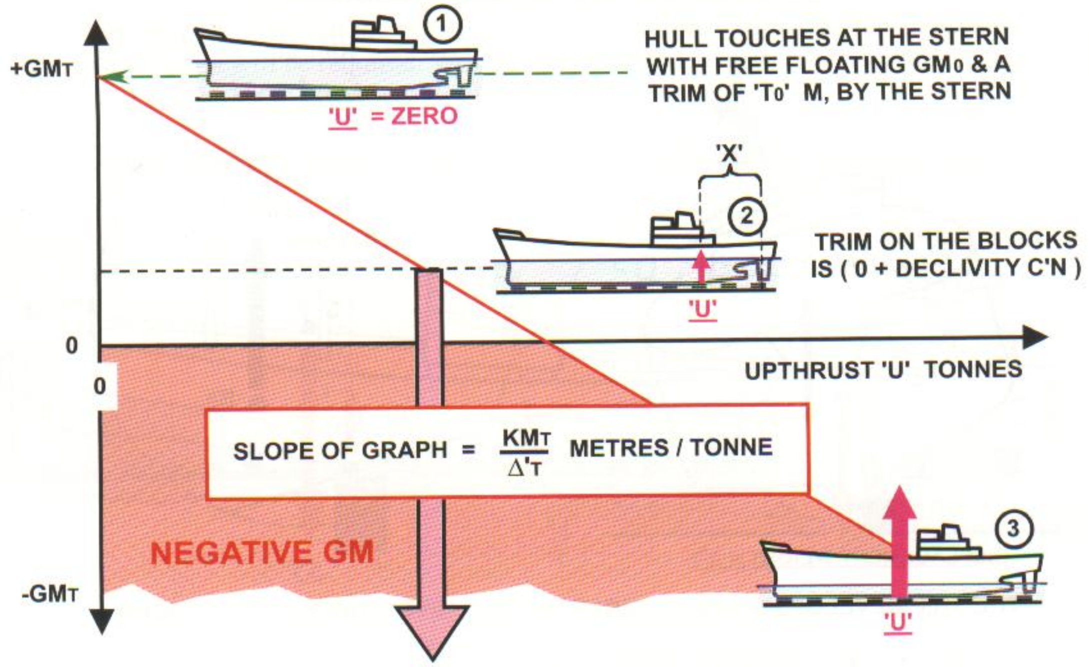
WHERE 'U' IS THE UPTHRUST ACTING UPON THE KEEL, Δ'_T IS THE SHIP'S FREE FLOATING DISPLACEMENT AND GM_T IS THE INITIAL FREE FLOATING GM VALUE.

THIS EQUATION IS KNOWN AS THE 'LOST BUOYANCY' METHOD

ASSESSING TRIM AND STABILITY REQUIREMENTS FOR DRYDOCK.

A scheduled drydock often produces a lot of conflicting demands upon the ship's staff and some of these will involve the trim and stability requirements. Routine inspection and repair work involving fuel, water and ballast tanks will require some tanks to be empty prior to docking, whilst the dockyard will put restrictions on the maximum acceptable draft and trim. Ship's officers must ensure that the vessel's stability is sufficient to allow safe docking, whilst working within these constraints. The previous page shows how the GM decreases at a constant rate with increasing upthrust on the keel and the easiest way to assess the changing stability is to plot a graph of GM against upthrust, as shown below:-

AN ACCEPTABLE LOSS OF GM DURING DRYDOCKING



$$\text{'U' ON TAKING THE BLOCKS} = \frac{(T_0 - \text{DECLIVITY} \times \text{LBP}/100)\,[100 \times \text{MCTC}_{(D.W.)}]}{\text{LCF} - X} \quad \text{TONNES}$$

$$\text{And} \quad \text{LOSS OF GM} = \text{'U'}\,\frac{KM_T}{\Delta'_T} \quad \text{METRES,}$$ ***WHICH MUST BE LESS THAN GMo***

SO THE FREE FLOATING GMo WHICH WILL MAINTAIN POSITIVE STABILITY FOR A GIVEN DOCKING TRIM, IS GIVEN BY:-

$$\text{THE FREE FLOATING } GM_0 \geq \frac{KM_T\,(T_0 - \text{DECLIVITY} \times \text{LBP}/100)\,[100 \times \text{MCTC}_{(D.W.)}]}{\Delta'_T\,(\text{LCF} - X)} \quad M$$

ALTERNATIVELY, THE DOCKING ACCEPTABLE TRIM FOR A GIVEN GM 0, IS GIVEN BY;-

$$\text{THE DOCKING TRIM 'T}_0\text{'} \leq \frac{\Delta'_T\,(\text{LCF} - X)\,GM_0}{KM_T\,[100 \times \text{MCTC}_{(D.W.)}]} + \text{DECLIVITY} \times \text{LBP}/100 \quad M$$

THE VESSEL WILL CONTINUE TO LOSE STABILITY AS 'U' INCREASES AFTER IT HAS TAKEN THE BLOCKS AT A RATE OF KM T / Δ'T WHERE 'U' IS GIVEN BY:-

$$\text{UPTHRUST 'U'} = \frac{\text{TPC}_{(D.W.)}}{\text{CHANGE OF MEAN DRAFT (cm)}} \quad \text{TONNES}$$

ALTERNATIVELY, THE DRAFT, FROM THE DOCK WATER LEVEL, CAN BE USED TO LOOK UP THE REMAINING BUOYANCY IN THE DISPLACEMENT / DRAFT TABLE

And THE UPTHRUST 'U' = FREE FLOATING DISPLACEMENT - BUOYANCY TONNES

ASSESSING TRIM AND STABILITY REQUIREMENTS FOR DRYDOCK (Cont).

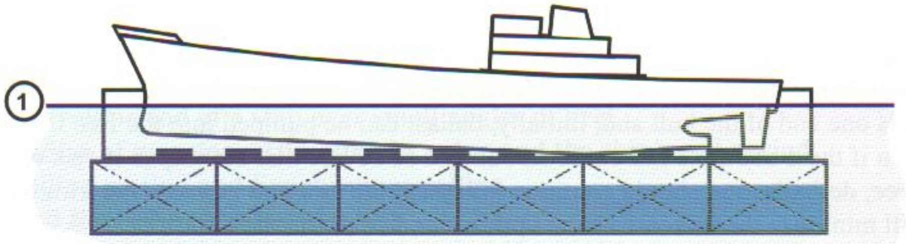
The ship can be docked safely, provided that it takes the blocks (stage 2 in the graph opposite) before its GM_T value goes negative. The continuing loss of GM after this point, is irrelevant if the hull is flat bottomed, but some vessels, such as fishing boats and tugs, are often built with a rise of floor or even an external keel plate (see Chapter 1, page 3). These require side support, in addition to the bilge blocks on the dockfloor, before they lose positive stability. Traditionally, drydock sides were 'terraced' and wooden shores were wedged against the hull side plating. However, modern docks built for this type of hull, use hydraulic side rams to stop the ship toppling over and slipping off the blocks as it becomes unstable. These docks may also have declivity as fishing trawler and tug hulls usually feature 'Rise of Keel' to produce a deep aft draft and ensure good propeller immersion. These ships are designed to float with a pronounced stern trim.

Undocking requires the vessel to meet the same criteria and this is usually achieved by ensuring that its condition on undocking is the same as when it went into the dock.

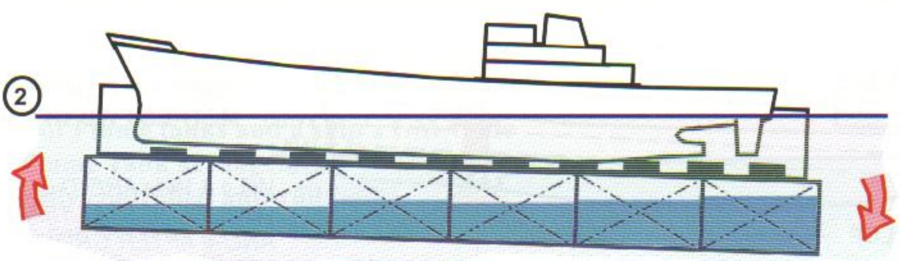
THE FLOATING DRYDOCK

When a ship docks in a floating drydock, as soon as the stern touches down, the dock is ballasted to match the vessel's trim so that the full hull length is landed on the blocks before lifting the ship any further. This means that, providing the free floating GM is positive, there should be no danger of a flat bottomed hull losing stability before it is properly supported by the bilge blocks. **It is important, however, that the dock, itself, maintains positive stability as it continues to lift the ship clear of the water.** During the operation, the dock must be able to retain sufficient ballast in its bottom tanks to ensure this, so there is a limit upon the weight of the vessel to be docked. There will also be limitations upon the trim that the dock can handle, as its ballast system is used to level the ship up during the lift.

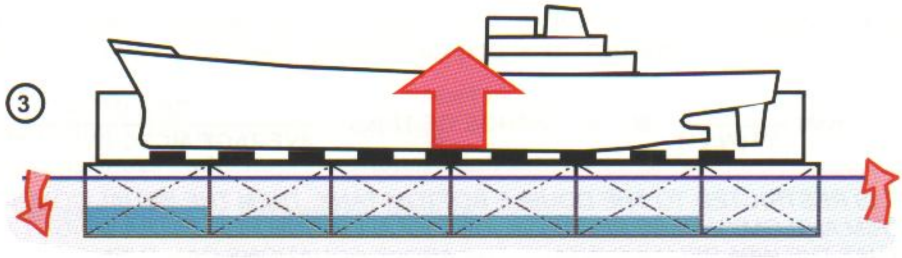
LANDING A SHIP ON THE BLOCKS OF A FLOATING DRYDOCK



THE DOCK IS BEING DE-BALLASTED AND RISES UP TO TOUCH THE STERN OF THE SHIP



THE BALLAST IN THE DOCK IS ADJUSTED TO MATCH THE TRIM OF THE SHIP AND SO LAND IT FULLY



DE-BALLASTING OF THE DOCK CONTINUES TO LIFT THE VESSEL AND LEVEL IT TO NEAR EVEN KEEL

BEACHING A DAMAGED VESSEL

If a ship is holed by collision in coastal waters, it may be possible to avoid sinking by running the ship up onto a beach. Obviously, the degree of planning such action is very limited and the Master must make the best of whatever beach is available, if any. Ideally, the beach should be even and preferably shelve at a relatively shallow gradient. Many commercial ships have large drafts and so are going to touch bottom quite a long way offshore, particularly if they are flooding, so the Master cannot rely on just the visual appearance of the shoreline. The chart must be examined as well and off-lying isolated shoal patches should be avoided if possible.
The ship is most likely to touch bottom on the bow first and is in danger of ahead power causing the vessel to swing around the point of contact. This will drive the ship broadside onto the beach where it is much more likely to be rolled over by wave action. Ahead power should be reduced just prior to grounding and then, after grounding, the ship must be ballasted down on the seabed along its entire length as soon as possible. This is even more important if there is a significant rise and fall of tide. Beaching on a falling tide is a drydocking situation and the ship must be landed on the seabed before it loses stability. If the tide is rising, ballasting should continue to prevent the undamaged part of the hull becoming free floating with the rising level of water, as this will again make the ship vulnerable to losing stability or being swept sideways onto the beach. The ship, however, must be landed sufficiently close inshore so that high water does not cause further flooding due to the upper deck and openings becoming awash.

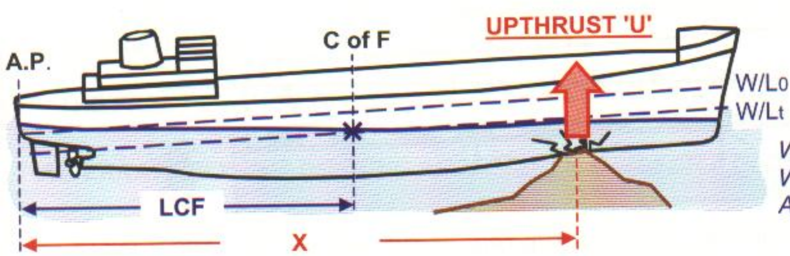
THE CONSEQUENCES OF STRANDING

When a ship grounds on a single point, usually in the forward region of the hull, there is risk of :-

1) The ship being swung broadside to the waves or wind and then the seas breaking it up.
2) The ship losing stability and/ or breaking its back on a falling tide due to an increasing upthrust acting on the hull at the point of contact

If the hull appears to have remained intact or flooding is limited to the double bottom tanks and the tide is falling, attempts should be made to re-float immediately after grounding. Going astern, combined with ballasting operations directed at reducing the draft at the point of contact, may break the vessel free. Changing the ship's trim is the quickest way of achieving this if the vessel is grounded towards one end of the hull and, initially, ballast can be pumped into the free floating end if necessary. Even if these attempts fail or the flooding is so severe that the ship is at risk of sinking if it does float free, de-ballasting should continue to lighten the hull and increase the trim. Both of these actions will minimise the increase of the upthrust and stress on the hull.

TRIM AND STABILITY CHANGES WHEN STRANDED DURING A FALLING TIDE



POINT OF CONTACT IS 'X' METRES FWD OF THE A.P.

W/L0 & W/Lt ARE THE MEAN WATERLINES, AT TIMES 'ZERO' AND 't' AFTER GROUNDING

IF THERE IS NO FLOODING FALL IN TIDE = THE CHANGE OF MEAN DRAFT $W/L_0 - W/L_t$

And UPHRUST 'U' = DISPLACEMENT CHANGE '$\Delta'T(W/L_0) - \Delta'T(W/L_t)$'

$$\text{LOSS OF } GM_T = KM_T \frac{'U'}{\Delta'T(W/L_0)} \text{ M} \quad \& \quad \text{CHANGE OF TRIM} = \frac{'U'\ (LCF - X)}{\text{AVERAGE MCTC } (W/L_0 \,/\, W/L_t)} \text{ cm}$$

IF FLOODING IS RESTRICTED TO THE DOUBLE BOTTOM TANK, THEN THE INITIAL CONDITIONS FOR 'W/L0', TRIM AND GMT CAN BE RECALCULATED BY ALLOWING FOR EXTRA WEIGHT AND ASSUMING THAT THE WPA REMAINS INTACT. THE ABOVE EQUATIONS CAN THEN BE APPLIED, USING THESE RE-CALCULATED INITIAL VALUES AND THE SHIP'S INTACT HYDROSTATIC DATA

CHAPTER 7

A SHIP'S MOTION IN A SEAWAY AND ANTI-ROLL MEASURES

SUMMARY

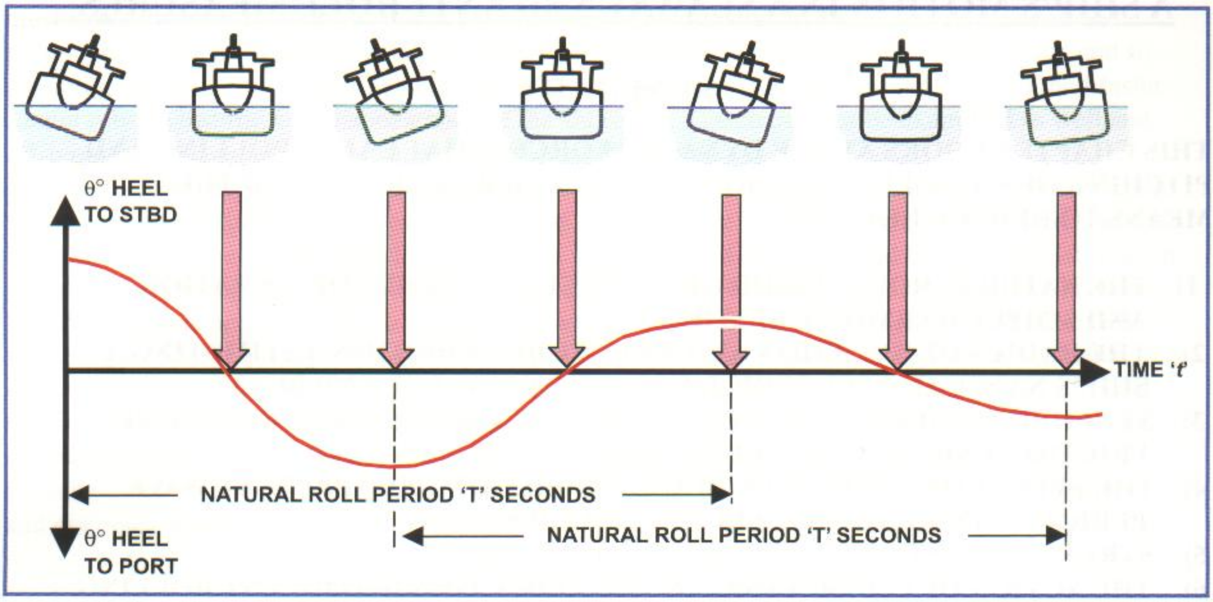
THIS CHAPTER LOOKS AT THE DYNAMIC FORCES THAT CAUSE ROLLING AND PITCHING OF A VESSEL, THE STRESSES INDUCED BY ROLLING AND THE MEANS TO REDUCE ROLLING

1) **THE NATURAL ROLL PERIOD OF A VESSEL, ITS RADIUS OF GYRATION AND SIMPLE HARMONIC MOTION**
2) **THE RADIUS OF GYRATION AND WEIGHT DISTRIBUTION. ESTIMATING A SHIP'S NATURAL ROLL PERIOD FROM ITS GM VALUE AND BEAM.**
3) **SYNCHRONISED ROLLING OF A VESSEL AND THE DAMPING EFFECT OF FRICTION ENHANCED BY BILGE KEELS.**
4) **THE EFFECT OF A SHIP'S SPEED AND COURSE ON THE APPARENT WAVE PERIOD. MANAGING A VESSEL IN A ROUGH SEAWAY**
5) **STRESSES IN A SHIP'S HULL INDUCED BY ROLLING**
6) **THE ACTION OF FLUME TANKS IN THE REDUCTION OF VIOLENT ROLLING.**
7) **THE RESPONSE OF A GYROSCOPE TO ROLLING TORQUE**
8) **THE FIN STABILISER AS AN ACTIVE ANTI-ROLL DEVICE**
9) **A SHIP'S PITCHING BEHAVIOUR**
10) **ROLL INDUCED OR PARAMETRIC ROLLING**

CONTENTS

THE NATURAL ROLL PERIOD OF A SHIP

A single disturbing force will make a ship rock from side to side as shown in the diagram below

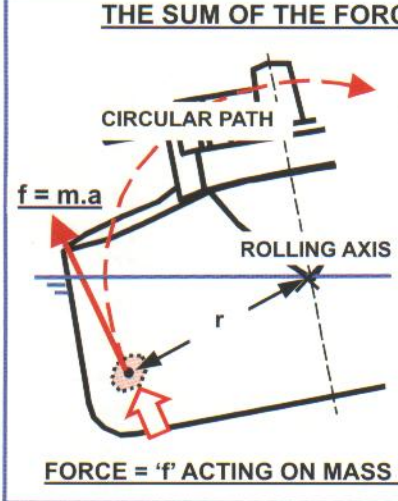


Although friction gradually reduces the maximum angle of roll, the time interval between successive rolls remains approximately constant. This natural roll period is a measure of how quickly the ship responds to a disturbing force and, consequently, is important in determining the extent to which a ship will roll when subjected to regular wave action.

THE MASS MOMENT OF INERTIA OF A ROTATING OBJECT

The Righting Moment caused by heeling a ship over is the sum of the moments that act upon every part of the ship's mass and individual moments cause forces to accelerate all parts of the ship along a circular path centred on the rolling axis. We can apply Newton's Second Law of Motion (Force = Mass x Acceleration) to the motion of one such part of the ship

THE SUM OF THE FORCES ACTING UPON ALL PARTS OF A ROLLING VESSEL



AT THE ONSET OF THE RETURN ROLL, ANGULAR ACCELERATION '$d^2\theta/dt^2$' RADIANS/SEC^2, RELATES TO PARTICLE ACCELERATION

$$a = \frac{d^2\theta}{dt^2}r \quad \& \text{ so} \quad f = \delta m \frac{d^2\theta}{dt^2}r$$

THE INDIVIDUAL SMALL RIGHTING MOMENT 'δ' R.M. WHICH PRODUCES THE FORCE 'f' ACTING UPON 'δm' IS GIVEN BY:-

$$\delta \text{ R.M.} = f.r \quad \& \text{ so} \quad \delta \text{ R.M.} = \delta m \frac{d^2\theta}{dt^2}r^2$$

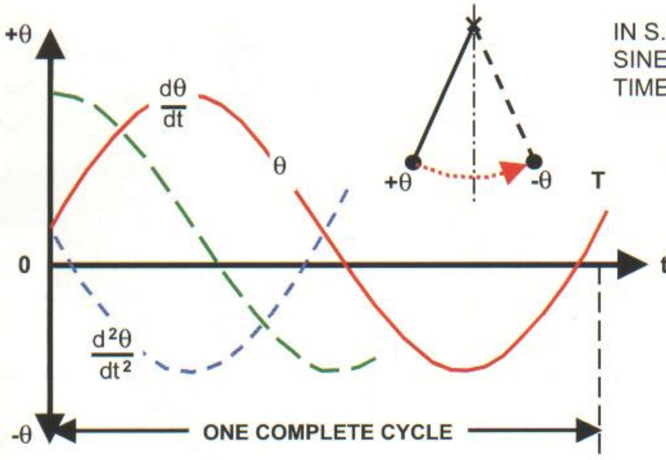
$$\text{SO THE TOTAL R.M.} = \frac{d^2\theta}{dt^2}\Sigma\ \delta m_1 r_1^2 + \text{------} + \delta m_n r_n^2$$

The sum of all the small masses times their rolling radius squared, (δmr^2), is known as the ship's Moment of Inertia. It is increased by distributing the ship's mass further away from the rolling axis. For a given Righting Moment, the angular acceleration and, hence, the quickness of the roll, is reduced if the Moment of Inertia is increased. 'Winging out' weight (i.e. stowing weight as far outboard as possible) is a technique that can moderate the violence of a ship's roll.

SIMPLE HARMONIC MOTION AND THE NATURAL ROLL PERIOD OF A SHIP

Simple Harmonic Motion (S.H.M.) is the freely swinging, or vibrating, response of any object to a single disturbing impulse. 'Free' means that no friction is involved, so no object on earth can follow S.H.M. but many situations, such as the swing of a pendulum, are close approximations of it. S.H.M. obeys basic mathematical rules that allow the motion's frequency to be calculated as follows:-

SIMPLE HARMONIC MOTION (S.H.M.)



IN S.H.M., THE OSCILLATIONS PRODUCE A SINE CURVE RELATIVE TO TIME, SO AT ANY TIME t'

$$\theta_t = \theta_{(MAX)} \sin \omega t$$

$$\& \quad \frac{d\theta}{dt} = \theta_{(MAX)}\, \omega \cos \omega t$$

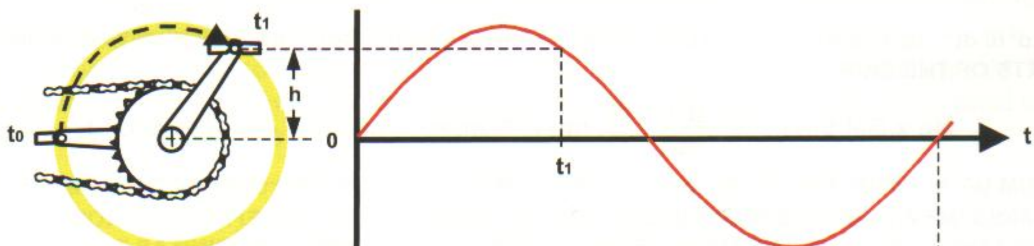
$$\& \quad \frac{d^2\theta}{dt^2} = -\,\theta_{(MAX)}\, \omega^2 \sin \omega t$$

$$\text{so} \quad \frac{d^2\theta}{dt^2} = -\,\omega^2 \theta$$

AT ANY INSTANT OF TIME

Note that rate of swing and the rate of change of swing both produce sinusoidal curves.
The term 'ω' (Omega) is the frequency of the oscillation in radians/second and can be thought as the rotational rate of an imaginary wheel driven by a crank moving up and down with the same frequency of the above pendulum's swing.

THE EQUIVALENT S.H.M. OF THE CIRCULAR ROTATION OF A BICYCLE PEDAL



$\omega = \frac{\theta}{t}$ RADIANS / SEC

THE TIME PERIOD 'T' OF ONE CYCLE, IS GIVEN BY $T = \frac{2\pi}{\omega}$ RADIANS

ANY FORM OF SIMPLE HARMONIC MOTION CAN BE CONVERTED MATHEMATICALLY TO CONSTANT SPEED CIRCULAR MOTION AND VICE VERSA. THESE TWO TYPES OF MOVEMENT ARE OPPOSITE SIDES OF THE SAME COIN, SO

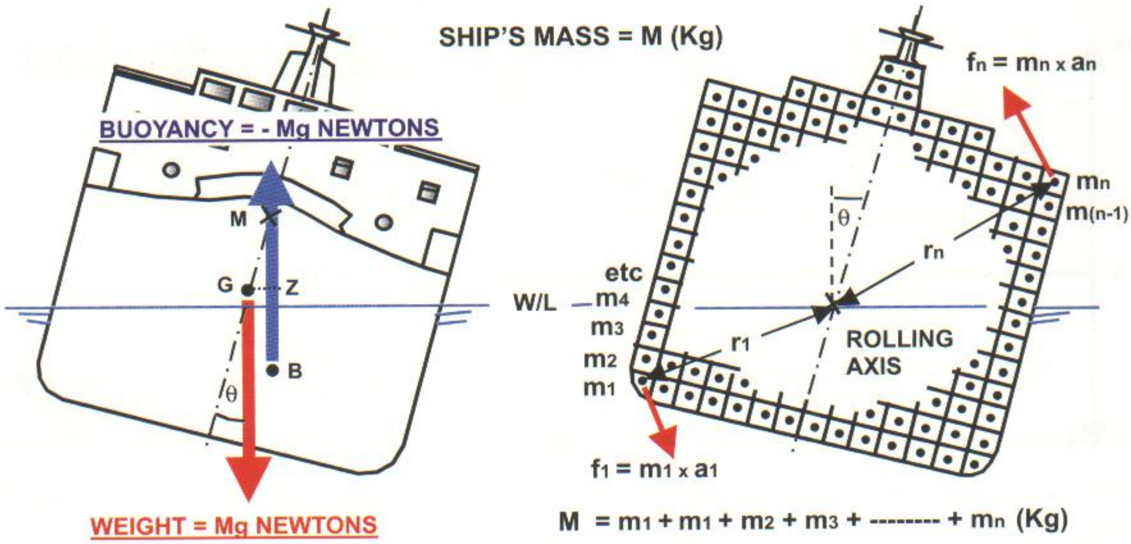
$$\text{IF} \quad \frac{d^2\theta}{dt^2} = -\,\omega^2\theta \qquad \text{THEN} \quad \frac{d^2\theta}{dt^2} = -\left(\frac{2\pi^2}{T^2}\right)\theta$$

IF WE CAN SHOW THAT THE NATURAL ROLLING PERIOD OF A SHIP OBEYS AN EQUATION IN THE FORM OF;-

$$\frac{d^2\theta}{dt^2} = (\text{A CONSTANT})\ \theta$$

THEN THE ROLL PERIOD **'T'** = $\dfrac{2\pi}{\sqrt{\text{CONSTANT}}}$ SECONDS

We have seen that if a ship is heeled over $\theta°$ by an external force which is then removed, the Righting Moment, R.M. forcing the ship back towards the upright must equal the sum of all the moments acting upon each individual part of the ship's mass. The effect of these moments is to produce an angular acceleration, $d^2\theta/dt^2$, which must be directly proportional to θ for the ship's motion to be Simple Harmonic Motion.



$$M = m_1 + m_1 + m_2 + m_3 + \text{--------} + m_n \text{ (Kg)}$$

$$\text{R.M.} = Mg \times GM \sin\theta \quad \& \quad \text{R.M.} = \sum(f_1 r_1 + f_2 r_2 + f_3 r_3 + \text{--------} + f_n r_n)$$

THE FORCE 'f', ACTING UPON AN INDIVIDUAL PART OF THE SHIP'S MASS, CAUSES THAT MASS 'm' TO ACCELERATE AT 'a' m/s^2 ALONG A CIRCULAR PATH AND, AS $f = ma$, SO;-

$$Mg \times GM \sin\theta = \sum(m_1 a_1 r_1 + m_2 a_2 r_2 + m_3 a_3 r_3 + \text{--------} + m_n a_n r_n)$$

THE LINEAR ACCELERATION 'a' ALONG A CIRCULAR PATH 'r' IS GIVEN BY $a = \dfrac{d^2\theta r}{dt^2}$,

WHERE $d^2\theta/dt^2$ IS THE ANGULAR ACCELERATION IN RADIANS / SEC_ AND IS CONSTANT FOR ALL PARTS OF THE SHIP

$$Mg \times GM \sin\theta = \frac{d^2\theta}{dt^2} \sum(m_1 r_1^2 + m_2 r_2^2 + m_3 r_3^2 + \text{--------} + m_n r_n^2)$$

IF WE SUM UP ALL THE INDIVIDUAL mr^2 VALUES AND THEN DIVIDE THIS SUM BY THE SHIP'S TOTAL MASS 'M', WE WILL OBTAIN A VALUE FOR 'R^2' WHERE 'R' IS THE EFFECTIVE ROOT MEAN DISTANCE OF THE SHIP'S MASS FROM THE ROLLING AXIS AND IS KNOWN AS THE '*SHIP'S RADIUS OF GYRATION*'

MOMENT OF INERTIA $MR^2 = \sum(m_1 r_1^2 + m_2 r_2^2 + m_3 r_3^2 + \text{--------} + m_n r_n^2)$

AND SO $\quad Mg \times GM \sin\theta = \dfrac{d^2\theta}{dt^2} MR^2$ *AND IF $\theta \le 10°$, THEN $\sin\theta = \theta$*

THEN $\quad Mg \times GM\,\theta = \dfrac{d^2\theta}{dt^2} MR^2$ AND $\dfrac{d^2\theta}{dt^2} = \dfrac{g \times GM}{R^2}\,\theta$

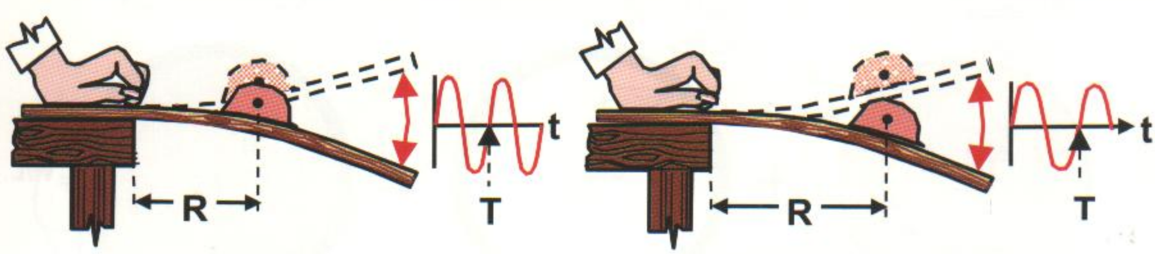
THIS EQUATION NOW OBEYS THE RULES OF SIMPLE HARMONIC MOTION SO, PROVIDING THAT WE ONLY CONSIDER ROLLING OVER A RANGE OF HEEL WITHIN 10° OF THE UPRIGHT, THEN

A SHIP'S NATURAL ROLL PERIOD 'T' $= 2\pi\sqrt{\dfrac{R^2}{g \times GM}}$ SECONDS

PROVIDED THAT THE ANGLE OF HEEL INDUCED IN THE ROLLING IS LESS THAN 10°

THE S.H.M. ROLL PERIOD OF A SHIP'S ROLL (Cont.)

The previous page showed that a ship's response to a single heeling force approximates Simple Harmonic Motion, provided that the range of heel angle remains within about 10º of the upright, i.e. the ship's GM value remains approximately constant. Furthermore, the natural roll period is a function of the GM and the root mean square of the distances of all the separate masses from the rolling axis. Changing this effective mean radius of the ship's mass distribution, known as the Radius of Gyration, can be demonstrated in following simple experiment with a flexible ruler and a lump of plasticine.

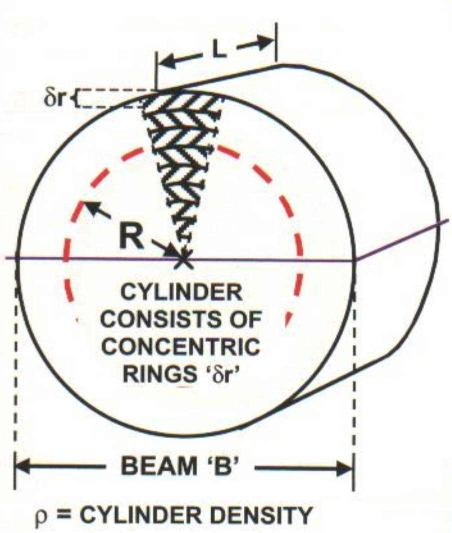


TIME PERIOD 'T' INCREASES AS DISTANCE 'R' INCREASES

The distance 'R' is the radius of gyration and the stiffness of the ruler is the equivalent to the ship's GM. Repeating the experiment with a more rigid ruler increases the frequency of vibration for any given position of the plasticine, just as an increased GM speeds up the rolling motion of a ship for a given weight distribution.

ESTIMATING A VALUE FOR A SHIP'S RADIUS OF GYRATION

Calculating a value for 'R' by summing up all the separate δmr^2 values of a ship would be a very lengthy process, requiring a knowledge of weight distribution to a level of detail not normally available to ship's officers. In any case, the roll period is not usually thought to be of primary importance when considering the operational requirements of loading a ship, the main priorities being stability, bending moments and accessibility of cargo.
It can, however, be useful to estimate the roll period 'T' in terms of the ship's beam and GM. To see how such an equation is derived, we can consider a floating solid cylinder with a draft equal to its radius, so that the rolling axis coincides with the centre of its circular cross-section.



MASS OF EACH CYLINDRICAL LAYER 'm' IS GIVEN BY:-

$$m = 2\pi rL\rho\ \delta r$$

$$\text{so } \sum (m_1 r_1^2 + m_2 r_2^2 + \text{----} + m_n r_n^2) = 2\pi rL\rho \int_0^{B/2} r^3\,\delta r$$

$$\text{so } \sum (m_1 r_1^2 + m_2 r_2^2 + \text{----} + m_n r_n^2) = 2\pi rL\rho \frac{B^4}{64}$$

$$\& \sum (m_1 r_1^2 + m_2 r_2^2 + \text{----} + m_n r_n^2) = MR^2$$

$$\text{whilst TOTAL MASS 'M' of cylinder} = \pi L\rho \frac{B^2}{4}$$

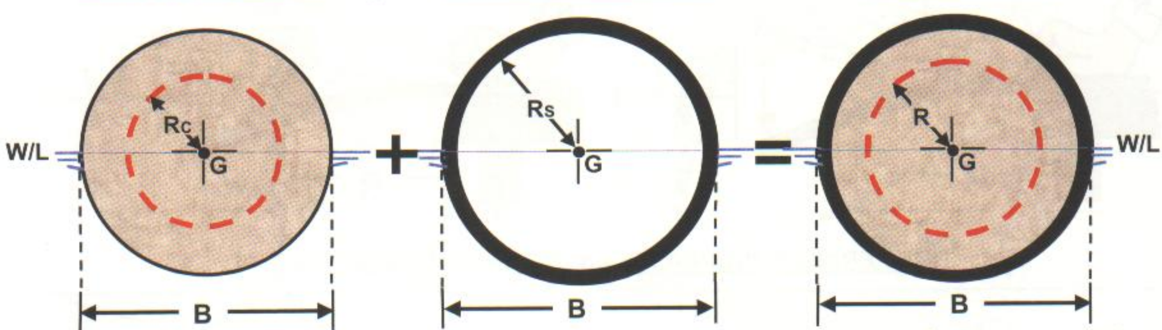
$$\text{so MOMENT OF INERTIA 'MR}^2\text{'} = M\frac{B^2}{8}$$

$$\boxed{\text{so RADIUS OF GYRATION 'R'} = \sqrt{\frac{B^2}{8}} = 0.35B}$$

ESTIMATING A VALUE FOR A SHIP'S RADIUS OF GYRATION (Cont.)

On the previous page, we determined the radius of gyration for a solid cylinder, rolling about its centre of gravity. We can consider this as a homogenous cargo so the next step is to surround this with a relatively thin steel shell, which could represent the ship's hull. If we add weight at the extreme distance from the rolling axis, we can expect to increase the Moment of Inertia and the Radius of Gyration. Typically, for an average cargo ship, the shell plating and stiffeners are about 1/5 of the ship's loaded displacement so we can use a ratio of 1:4 for the masses of the hull and cargo when summing up the moments of inertia.

CARGO CYLINDER + CYLINDRICAL SHELL = COMBINATION



$$McRc^2 = \frac{4}{5}\ M\,(0.35B)^2 \qquad MsRs^2 = \frac{1}{5}\ M\,(0.50B)^2 \qquad MR^2 = MB\left(\frac{4(0.35)^2 + 0.5^2}{5}\right)$$

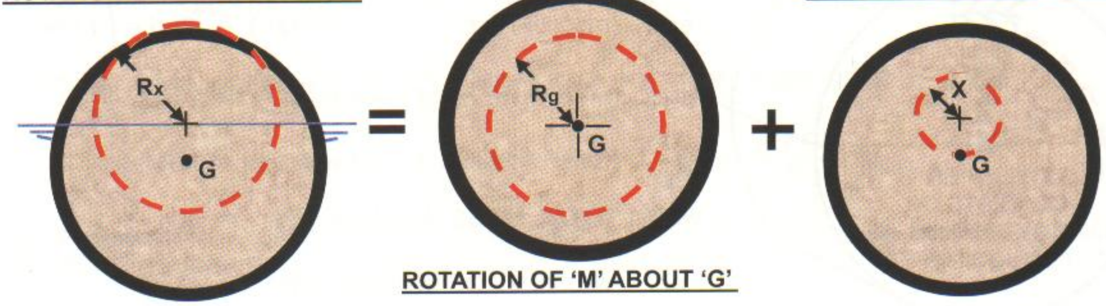
$$\boxed{\text{so RADIUS OF GYRATION 'R'} = B\sqrt{\frac{4(0.35)^2 + 0.5^2}{5}} \quad \text{\& HENCE } R = 0.38\,B}$$

When the loaded cylinder floats at the draft shown above, its axis of rotation passes through its centre of gravity, so all points of mass remain at constant distance from the rolling axis as the cylinder rolls. A laden ship usually floats at a draft deeper than its KG value so its rolling motion is a combination of each individual point mass rotating about the C of G, which itself is rotating about the rolling axis. The total Moment of Inertia and, consequently, the Radius of Gyration, 'R' are increased by this complexity.

We can approximate the ship to such a cylinder of equal width by assuming that the midships region with weight outside the cylindrical limits compensates for the reducing beam at the ends.

THE MOMENT OF INERTIA OF THE BODY ABOUT ANY AXIS = THE MOMENT OF INERTIA ABOUT ITS C OF G + THE SECOND MOMENT OF ITS MASS ABOUT THE AXIS

ROTATION OF 'M' ABOUT AXIS | **ROTATION OF 'G' ABOUT AXIS**



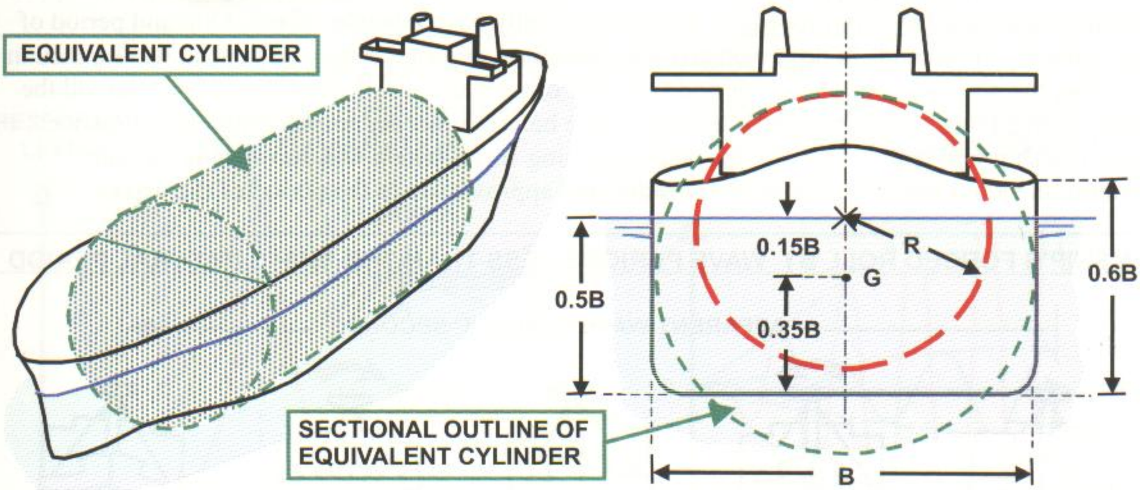
$$MRx^2 = MRg^2 + Mx^2$$

$$\boxed{\text{so RADIUS OF GYRATION 'Rx'} = \sqrt{Rg^2 + x^2}}$$

ESTIMATING A VALUE FOR A SHIP'S RADIUS OF GYRATION (Cont.)

The following diagram shows how the mass distribution of a typical large loaded commercial cargo vessel can be equated to that of an imaginary cylinder of equal width.

APPROXIMATING THE SHIP'S MASS TO A HOLLOW CYLINDER



A cylinder of the same displacement would be shorter than the ship, due to its greater sectional area, but this should not affect the undamped rolling behaviour, which is independent of length.

USING THE ABOVE PROPORTIONS, RADIUS OF GYRATION 'R' $= B\sqrt{(0.38)^2 + (0.15)^2}$

SO 'R' IS APPROXIMATELY EQUAL TO 0.41B

If we now return to the S.H.M. equation for the natural roll period 'T', we can obtain a simple approximate estimate for 'T', in terms of the ship's beam and GM value.

$$T = 2\pi\sqrt{\frac{R}{9.81 \times GM}} \qquad \text{so} \qquad T = 0.82\,\frac{B}{\sqrt{GM}} \text{ SECONDS}$$

A SHIP'S NATURAL ROLL PERIOD 'T' $\cong 0.8\,\dfrac{B}{\sqrt{GM}}$ SECONDS

WHERE BEAM 'B' AND GM ARE MEASURED IN METRES

The above equation is based upon the following assumptions

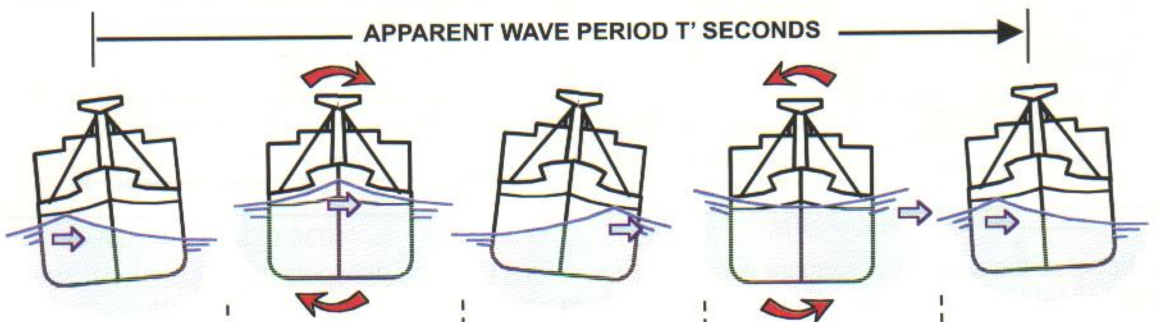
1) The proportions of Beam, Draft and KG are approximately those shown in the above diagram.
2) About 80% of the ship's total displacement consists of cargo filling most of the ship's hull and evenly distributed across the vessel's entire beam.
3) The hull is predominately parallel sided.
4) There is no account made for damping, due to friction against the ship's underwater hull, as it rolls.

These are reasonable assumptions when considering a fully laden cargo ship or pre-1983 tanker but they may become less appropriate when applied to some other types of ship. The I.M.O. gives a more complex formula in its code of Intact Stability, which takes account of some of the variables such as the ratio of Draft to Beam, but it also will only approximate the Natural Roll period. Basically, concentrations of weight inboard, (which can be achieved by very fine lined hulls), reduce the roll period for a given beam and, hence, produce a quicker rolling motion.

A SHIP'S ROLLING MOTION AND SYNCHRONOUS ROLLING

If we use the equation, derived on the previous page, for approximating the natural undamped roll period for two different vessels with a typical GM of 0.6 metres (a 20,000 T DWT ship of 20m beam and a smaller 2000 T DWT ship of 10m beam) then we can obtain the following estimates:-
Undamped Natural Roll Period for a GM = 0.6m is about 13 seconds and 27 seconds for vessels of 10m and 20m beam respectively.
Sea waves generated by wind speeds of 50 knots typically have a height of about 8m and period of approximately 10 seconds in deep water. If the above two ships are subjected to such waves, then in each case, the waves have a shorter period than that of the natural roll, so at the ends of the roll the vessel is forced into the return roll by the next wave before it reaches its maximum potential heel angle. The ship will tend to roll into the next oncoming wave, which will cause quite violent acceleration values but the roll will be restricted and approximate to the period of the waves.

A SHIP'S FORCED ROLL BY WAVE PERIODS LESS THAN THE SHIP'S NATURAL PERIOD



t = 0 sec	t = 1/4 T' sec	t = 1/2 T' sec	t = 3/4 T' sec	t = T' sec
θ = MAX.(STBD)	θ = ZERO	θ = MAX.(PORT)	θ = ZERO	θ = MAX.(STBD)
$\frac{d\theta}{dt}$ = ZERO	$\frac{d\theta}{dt}$ = MAX.(PORT)	$\frac{d\theta}{dt}$ = ZERO	$\frac{d\theta}{dt}$ = MAX.(STBD)	$\frac{d\theta}{dt}$ = ZERO
$\frac{d^2\theta}{dt^2}$ = MAX.(PORT)	$\frac{d^2\theta}{dt^2}$ = ZERO	$\frac{d^2\theta}{dt^2}$ = MAX.(STBD)	$\frac{d^2\theta}{dt^2}$ = ZERO	$\frac{d^2\theta}{dt^2}$ = MAX.(PORT)

θ = ANGLE OF HEEL, $\frac{d\theta}{dt}$ = ANGULAR VELOCITY, & $\frac{d^2\theta}{dt^2}$ = ANGULAR ACCELERATION

Apparent wave period is the true wave period combined with the ship's course and speed and when its period is longer than the natural roll period, the vessel will roll more leisurely away from each crest and so tend to remain perpendicular to the wave profile. If, however, the wave period coincides with the ship's natural roll period, then maximum angle of heel and the violence of the roll will progressively increase with each successive roll as each wave reinforces the ship's natural roll cycle. This is 'Synchronised Rolling' and is an example of resonance. It greatly increases the extent of the ship's rolling motion and might, in exceptional circumstances, heel the ship beyond the point from which it can expect to recover.

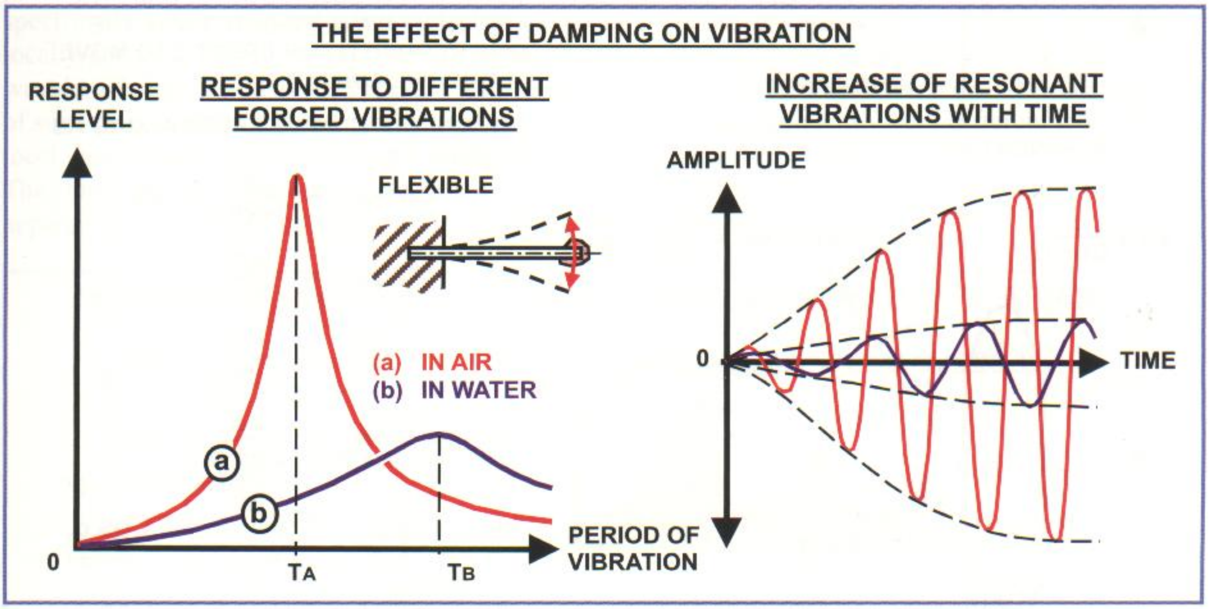
FRICTION AND THE ROLLING MOTION OF THE SHIP

Pure Simple Harmonic Motion is frictionless, but, of course, friction is always present and will damp the rolling of the ship by dissipating energy as heat. Friction has the following two effects:-

1) Friction reduces the angular velocity, $d\theta/dt$, so the roll is slowed down and its period increased.
2) Friction is directly proportional to $d\theta/dt$, so the rate at which it dissipates energy also increases as rolling motion builds up. A degree of rolling is reached where friction is dissipating energy at the same rate as each successive wave is inputting fresh energy. For steady sea conditions, friction will limit the extent to which synchronised rolling can build up.

THE EFFECT OF FRICTIONAL DAMPING ON RESONANCE

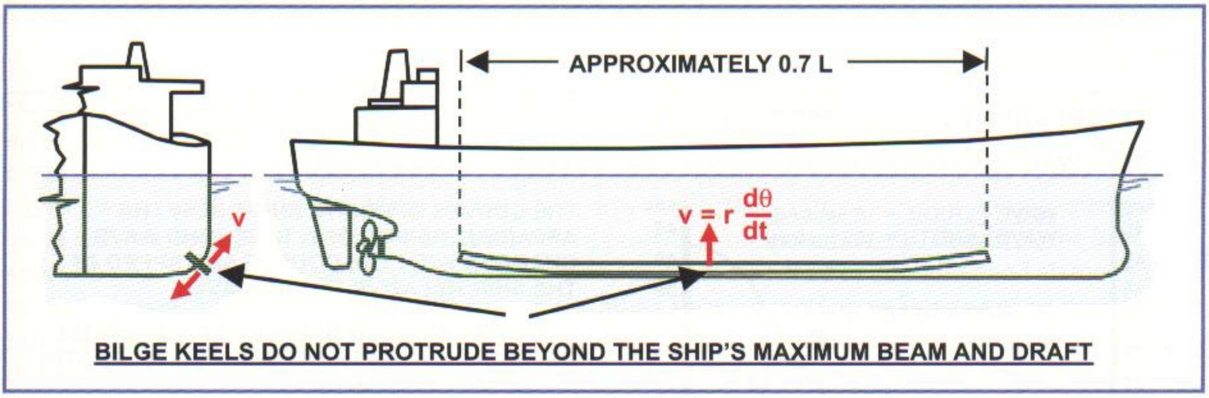
Friction is a means of limiting the rolling motion of the ship and can be used to make a ship more seaworthy and comfortable in a rough sea. This can be easily demonstrated by repeating the simple vibrating ruler experiment in a basin of water. When fully submerged, the weighted ruler's response to being vibrated is much more sluggish than in air, the period of oscillations is increased and they die away much more quickly than in air.



In the above diagram, 'Ta' and ' Tb' are natural resonant periods for a vibrating ruler in air and water respectively. The degree of damping underwater is much greater than that in air.

DAMPING A SHIP'S ROLLING MOTION WITH BILGE KEELS

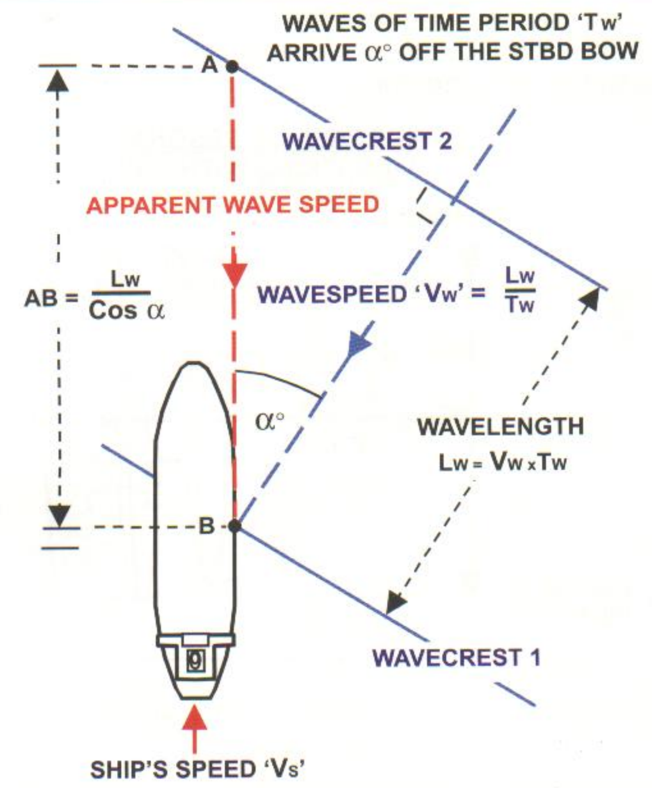
The friction acting against the motion of the smooth bottom of a rolling hull can be enhanced by fitting a shallow fore and aft plate to the turn of each bilge in the parallel body region of the hull. They are usually profiled to match the water flow along the hull so offer very little extra resistance to the ship moving ahead through the water, but they do produce considerable turbulence when rotated about the rolling axis. They are very effective anti-roll devices for the cost involved in fitting them, as they produce a degree of damping (up to 70% of the rolling energy is thought to be dissipated through them) far greater than their size would suggest.



Friction and damping increase with the speed at which the bilge keels move broadside through the water which depends not only on the angular roll velocity, $d\theta/dt$, but also on the radius, r , of the bilge keels' circular path, so even the relatively slow roll of a large vessel can be effectively damped because of the large value of 'r' involved.

A SHIP'S SPEED AND COURSE AND THE APPARENT WAVE PERIOD

A ship's motion will often include rolling when subjected to waves coming from a direction other than full on the beam. The following diagram illustrates how the apparent wave period 'Tw' experienced by the ship is altered by the ship's course and speed relative to that of the wave.



WAVECREST 1 IS CREATING A HEELING MOMENT AT POINT B ON THE SHIP. THE APPARENT WAVE PERIOD 'T w' IS THE TIME REQUIRED FOR CREST 2 TO MOVE ALONG THE LINE AB

THE APPARENT WAVE SPEED ALONG AB IS GIVEN BY:-

$$V'_{W\,(AB)} = \frac{V_W}{\cos\alpha} + V_S$$

$$\& \quad V'_{W\,(AB)} = \frac{AB}{T'_W}$$

$$\text{so} \quad \frac{AB}{T'_W} = \frac{V_W}{\cos\alpha} + V_S$$

$$\text{so} \quad \frac{L_W}{T'_W \cos\alpha} = \frac{V_W}{\cos\alpha} + V_S$$

$$\text{so} \quad \frac{L_W}{T'_W} = V_W + V_S \cos\alpha$$

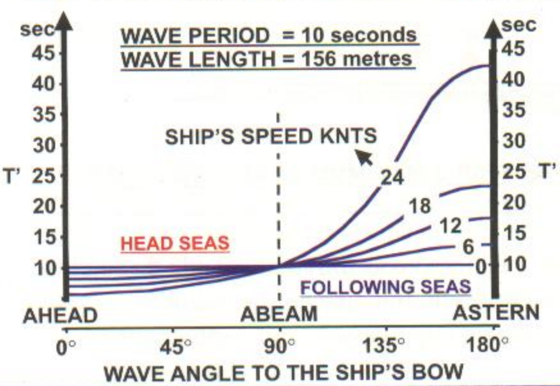
$$\text{so} \quad \frac{T_W}{T'_W} L_W = V_W + V_S \cos\alpha$$

$$\boxed{\text{i.e.} \quad T'_W = \left(\frac{V_W}{V_S\cos\alpha + V_W}\right) T_W}$$

When the angle 'α' between the ship's course and the wave direction is less than 90° (i.e head seas), the apparent wave period is reduced, but when 'α' is greater than 90°, cos α becomes negative, so the wave period increases with following seas. The 20,000T DWT ship with a 27 second Natural Roll Period, being subjected to 10 second waves, is only in danger of synchronous rolling when moving at speeds in excess of about 20 knots in a quarterly sea. This is when the waves approach the ship from between about 20° and 60° of the stern. Such quarterly seas are renowned for producing an unpleasant corkscrew type of motion that includes considerable rolling. In these circumstances, if the rolling becomes excessive, then either the ship's course or speed can be altered to shift the apparent wave period away from the synchronous condition.

APPARENT WAVE PERIOD AND THE SHIP'S RELATIVE COURSE AND SPEED

DEEP WATER WAVE SPEED V s = 1.56 x (WAVE PERIOD IN SECONDS) M/S



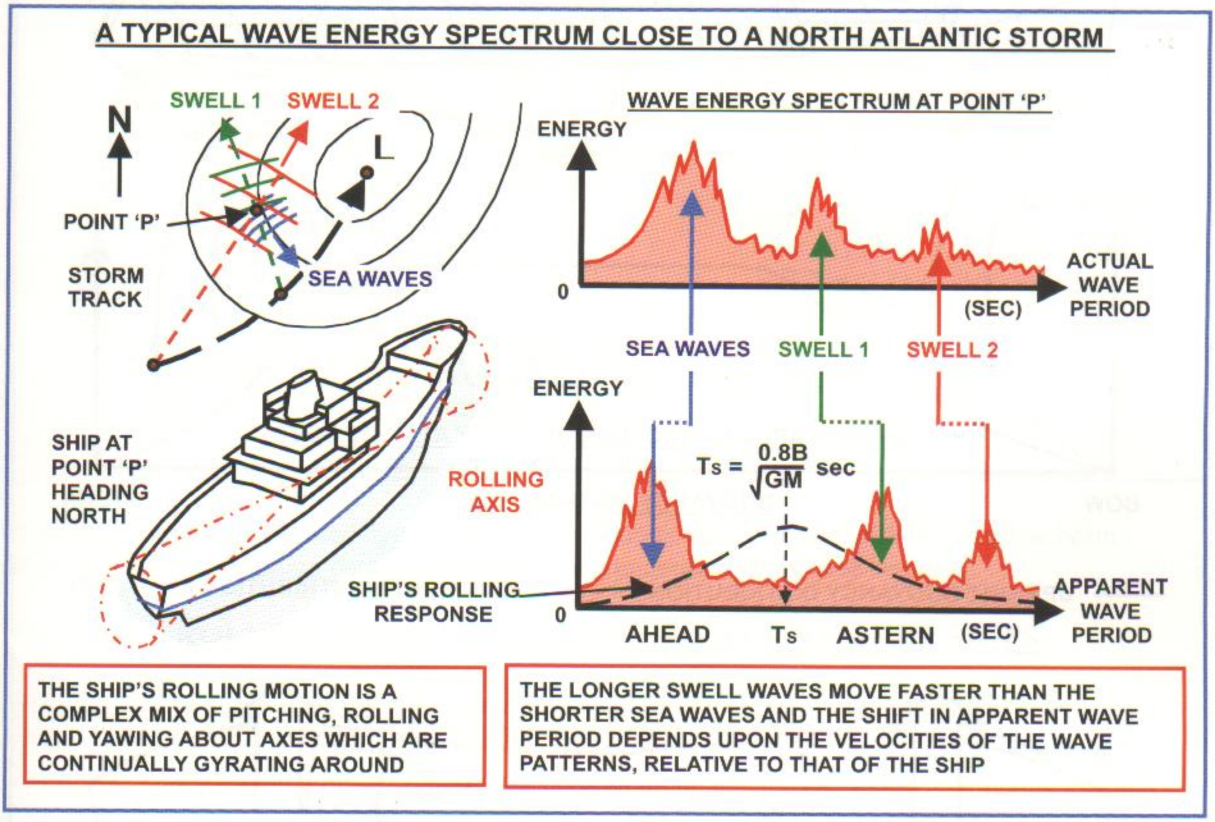
THE CURVES OPPOSITE SHOW HOW THE APPARENT PERIOD OF A 10 SECOND WAVE CHANGES WITH THE COURSE AND SPEED OF THE SHIP, RELATIVE TO THE WAVES

THE WAVE SPEED IN DEEP WATER (I.E. WATER DEPTHS GREATER THAN HALF A WAVELENGTH) IS 15.6 M/S OR 31 KNTS

THE APPARENT WAVE PERIOD IS MOST SENSITIVE TO CHANGES IN HEADING WHEN THE SHIP IS RUNNING BEFORE THE SEAS WITH A SPEED APPROACHING THAT OF THE WAVE

THE SHIP'S MOTION IN A SEAWAY

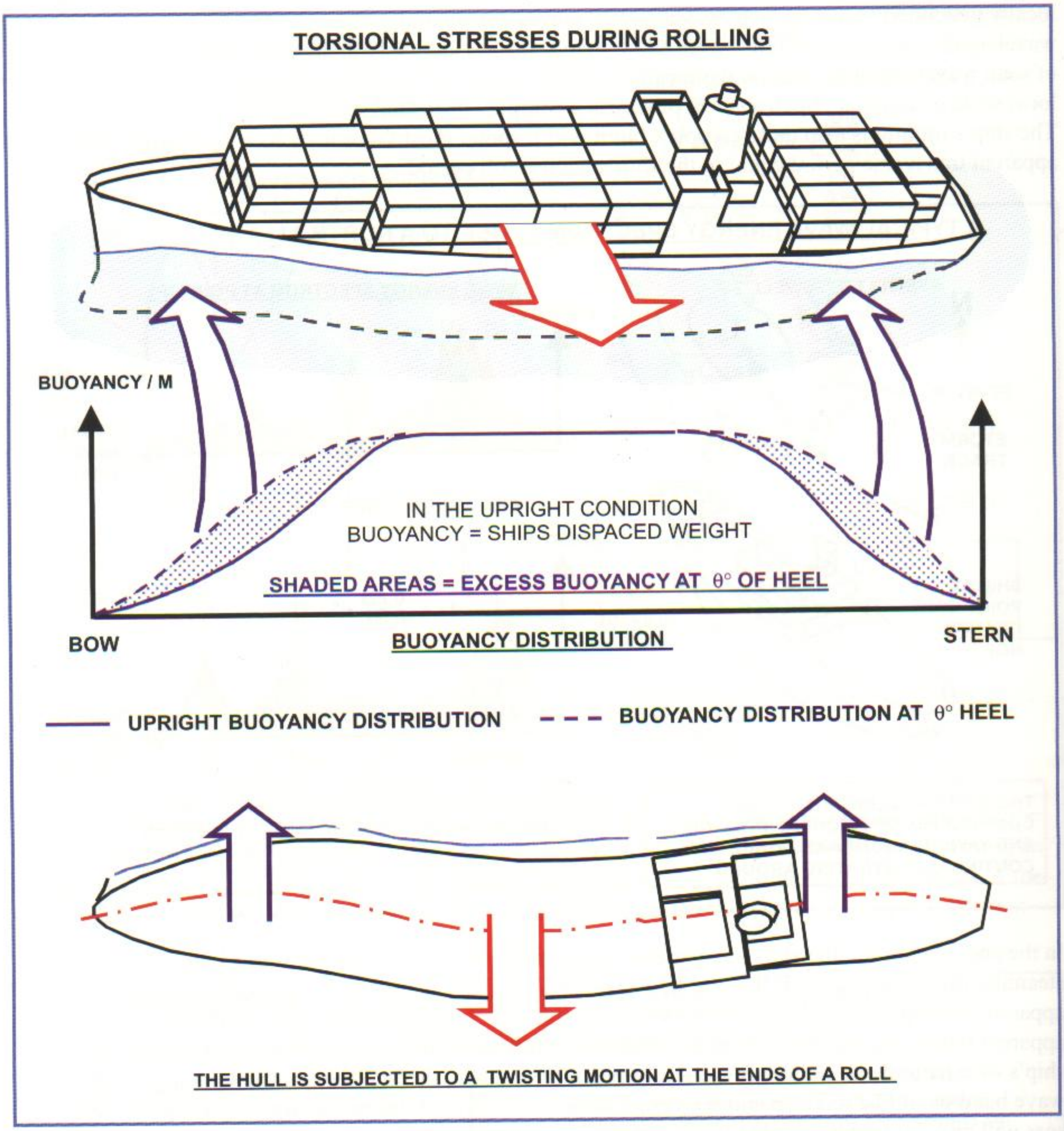
As wind blows over the open sea, after a period of some hours it will have generated waves of a predominant height, length and period, the values of which depend upon the average wind speed. The sea cannot respond instantly to change, so any variation of the wind speed or direction will superimpose another set of waves on top of the existing ones. There may also be underlying swell waves, produced some time previously at a distant location. A single well-developed storm can give rise to a sea, which includes a broad range of wave periods of varying direction and intensity. This spectrum will include swell waves, which may have travelled over considerable distances, as well as locally generated 'wind' or 'sea' waves. Swell waves are generally characterised by a long wavelength and period as long waves carry energy further and faster than short ones. The direction of such waves depends only on the location of their origin and is frequently very different from the local wind conditions. Such a storm produces a complex disturbed sea.
The ship's motion is also quite complex but it will respond most vigorously to wave energy with an apparent transverse period equal to the ship's natural roll period.



In the above diagram, the ship is experiencing quarterly swells (i.e. from abaft the beam) whilst steaming into strong seas off the port bow. The vessel's northwards course and speed will extend the apparent wave period of the following swells whilst reducing the period of the head seas, so the apparent transverse wave spectrum is stretched to produce relatively low energy in the region of the ship's own natural roll period. If the ship were to slow down, the width of this low rolling energy wave band would be reduced and the vessel would probably roll more heavily. However, the head seas will also cause the vessel to pitch quite severely and this may lead to pounding and slamming at the bow as well as increasing the resistance on the hull's forward motion, so it may not be possible to maintain the ahead speed without risk of damage. If the rolling becomes severe as the ship slows down, this can be moderated by altering course to put the bow into the most significant wave direction which, in this situation, is probably the NW'ly wind waves.
Sometimes, particularly in the hours of darkness, it can be quite difficult for the officer on watch to determine the wave pattern of such a confused heavy sea and using the ship's radar with low clutter settings can help identify predominant wave orientations. The radar tends to pick up the longer underlying swell waves that can be obscured by rough wind generated seas.

TORSIONAL STRESSES INDUCED BY A SHIP'S ROLLING MOTION

In Chapter 2, we looked into how the buoyancy distribution changes with angles of heel of the hull. In particular, flare increases buoyancy at the fore and aft ends as a vessel heels over, which will create bending stresses as this extra buoyancy, acting at the bow and stern, lifts the hull upwards. However, another effect of this feature of a ship shaped hull, is that as a ship reaches the ends of a roll, the restoring forces increase at the bow and stern regions more quickly than midships., which subjects the hull to twisting, or torsional, stresses.



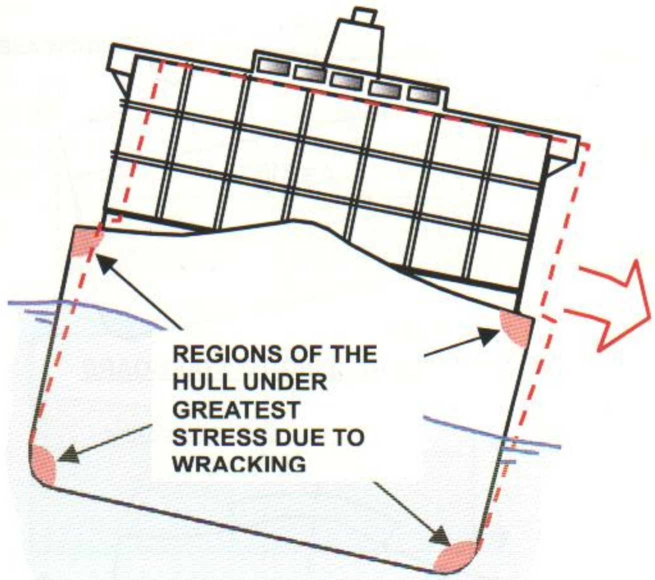
Continuous longitudinal structure of the hull must be sufficiently strong to withstand the torsional stresses and transmit the excessive righting forces at the ends of the hull to the midships region to minimise the twisting at the ends of a roll.

Over the years, ships' hatch openings have increased in area to allow more direct crane access to the underlying cargo hold spaces. The subsequent reduction in continuous fore and aft deck plating must be compensated for. Heavy box girders running along the length of the ship can be incorporated into the ship's structure. Another method is to build a substantial double hull, using the spaces created as wing tanks.

WRACKING STRESSES INDUCED BY A SHIP'S ROLLING MOTION

The righting moment and restoring forces produced by heeling a ship over are caused by the change of buoyancy distribution and, consequently, act initially on the submerged hull. During the rolling motion, the ship's structure must be capable of transmitting these forces to the parts of the vessel above the waterline, which will tend to continue their rotation beyond the end of the roll until the restoring forces have been effectively transmitted. This produces stresses, which act to distort the box section of the hull and are known as Wracking stresses.

WRACKING DUE TO HEAVY ROLLING



THE MASS OF SHIP ABOVE THE WATERLINE CONTINUES TO SWING BEYOND THE END OF THE ROLL UNTIL THE RESTORING FORCES HAVE BEEN EFFECTIVELY TRANSMITTED FROM THE SUBMERGED HULL STRUCTURE TO REST OF THE VESSEL.
DISTORTION OF THE HULL IS GREATEST AT THE DECK MARGINS AND THE TURN OF THE BILGE

Wracking stresses are resisted by re-inforcing the corners most affected with ties, such as beam knees or deep web framing between the frames and underdeck transverse beams.

ROLLING INDUCED STRESSES AND CONTAINER SHIPS

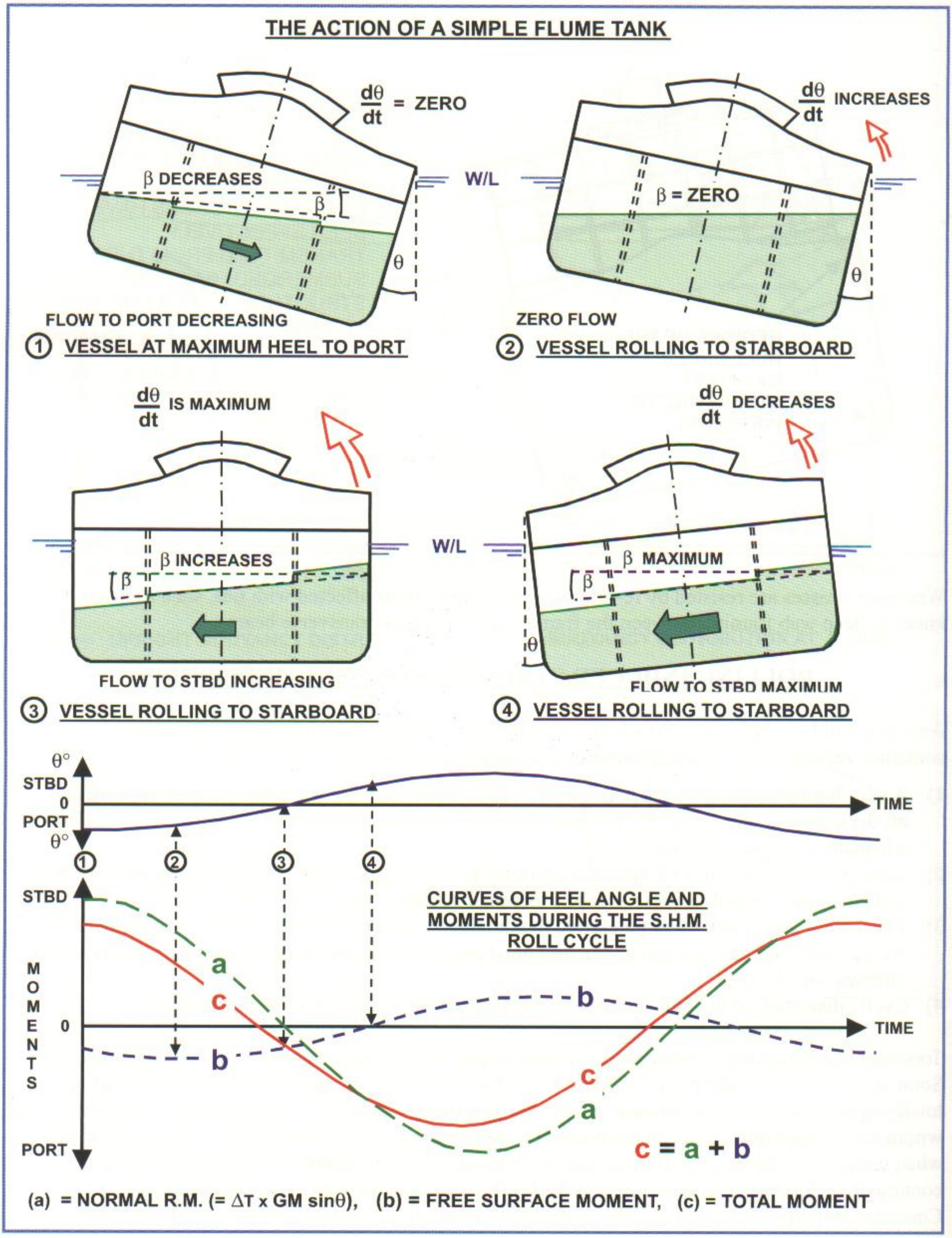
Any ship will be subjected to Wracking and Torsional Stress when rolling heavily. However, container vessels are particularly prone to such stresses, for the following reasons:-

1) A very high proportion of the upper deck is taken up by hatchways, so the continuous fore and aft deck area is quite small for the size of ship. This can lead to severe torsional stresses if not adequately compensated for.
2) Container ships are built for speed and so tend to be fine lined with considerable flare at the bow and stern. Such a hull is more prone to torsional stresses than a more full bodied ship.
3) The weight and height of cargo carried in containers, supported by the hatches, is considerable. At the ends of a roll, the hull's structure must force this weight back to upright, so the wracking stresses will be high.
4) Cyclic distortion of the hull can loosen the securing arrangements of deck containers.

Torsional and Wracking stresses occur together to produce twisting distortion at the ends of the roll. Some distortion must take place in order for the forces involved to spread over the entire ship. A totally rigid structure would fracture. However, the extent of distortion must be small to remain within the elastic limits of the material used to build the ship. Furthermore, these stresses are cyclic when induced by the vessel rolling so structural failure through fatigue is a danger if the ship is continually subjected to stresses close to the limits of what its structure can withstand.
Container ship operators, in particular, must be aware of the dangers of over-stressing the ship.

THE ACTION OF A FLUME TANK AS AN ANTI-ROLL DEVICE

Flumes are full width tanks, designed to be kept slack and so produce a free surface moment as the liquid within the tank moves from side to side with the ship's roll. The flow of the liquid is, however, restricted by baffles built into the tank's structure, so the mass transfer of liquid lags behind the changing angle of heel. When the tank is working correctly, this delayed transfer of weight acts against the ship's righting moment and reduces the extent to which the ship heels over beyond the upright on the return roll. The following diagrams illustrate the Flume action.

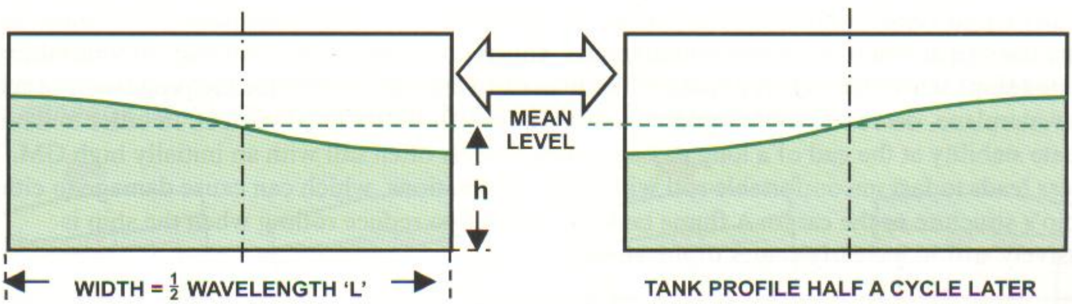


THE ACTION OF A FLUME TANK AS AN ANTI-ROLL DEVICE (Cont.)

The key feature of the flume's effectiveness in reducing a ship's roll, is that liquid continues to flow into the tank's low side after the maximum angle of heel has been reached and the ship has started the return roll. This delay in the transfer of fluid weight produces a reduced righting moment which reaches its peak value before the ends of the roll, so the momentum of the ship is reduced to diminish the maximum heel angle reached on the return roll. **The free surface flow in the tank lags behind the ship's roll**. The extent of this delay depends upon the ship's roll period, relative to the natural oscillation period of the fluid across the tank.

In considering 'Free Surface Effects', we have, so far, simply assumed that the liquid level in a slack tank remains horizontal whilst the ship rolls from side to side. This is reasonable when considering the transfer of weight if the ship is gradually heeled over but it ignores the dynamics of the continual to and fro fluid motion, which occurs in the tank when the ship is rolling. The changing liquid surface is actually a 'gravity wave' with a wavelength equal to twice the width of the tank, and is the result of oscillating flow in a relatively shallow depth of fluid. The theory of shallow liquid waves is beyond the intended scope of this book but for shallow waves the relationship between wave speed, length and period can be shown as follows:-

THE WAVE 'SLOP' ACROSS THE FREE SURFACE OF A RECTANGULAR TANK



IF 'h' IS LESS THAN 1/4L, THEN WAVE PERIOD 'T' APPROXIMATES TO $= \frac{L}{\sqrt{gh}}$ SECONDS

WHERE 'h' IS MEASURED IN METRES AND 'g' IS THE ACCELERATION DUE TO GRAVITY = 9.81 M / S²

IF WE MATCH THE NATURAL OSCILLATION PERIOD OF A SLACK TANK OF FULL BEAM WIDTH B, WITH THAT OF THE SHIP, THEN WE CAN MAKE THE FOLLOWING APPROXIMATIONS:-

NATURAL PERIOD 'T' OF THE TANK $\approx \frac{2B}{\sqrt{9.81h}}$ & NATURAL ROLL PERIOD 'T' OF THE SHIP $\approx \frac{0.8B}{\sqrt{GM}}$

$$\text{So} \quad \frac{0.64}{\sqrt{h}} \approx \frac{0.8}{\sqrt{GM}}$$

$$\frac{0.64^2}{h} \approx \frac{0.64}{GM}$$

$$\text{So} \quad h \approx 0.64\ GM$$

WHERE 'h' IS THE DEPTH OF FLUID IN THE SLACK TANK

The resonating 'slop' of a free surface in a tank can be easily experienced by sliding backwards and forwards in a bath tub, whilst taking a bath. The most water is transferred when the wave has a length equal to twice that of the bath or tank and the wave oscillates faster if the water depth is increased. The mathematical theory is relatively involved but we can say that the influence of the tank bottom, which slows down the flow, is greater at shallower depths.

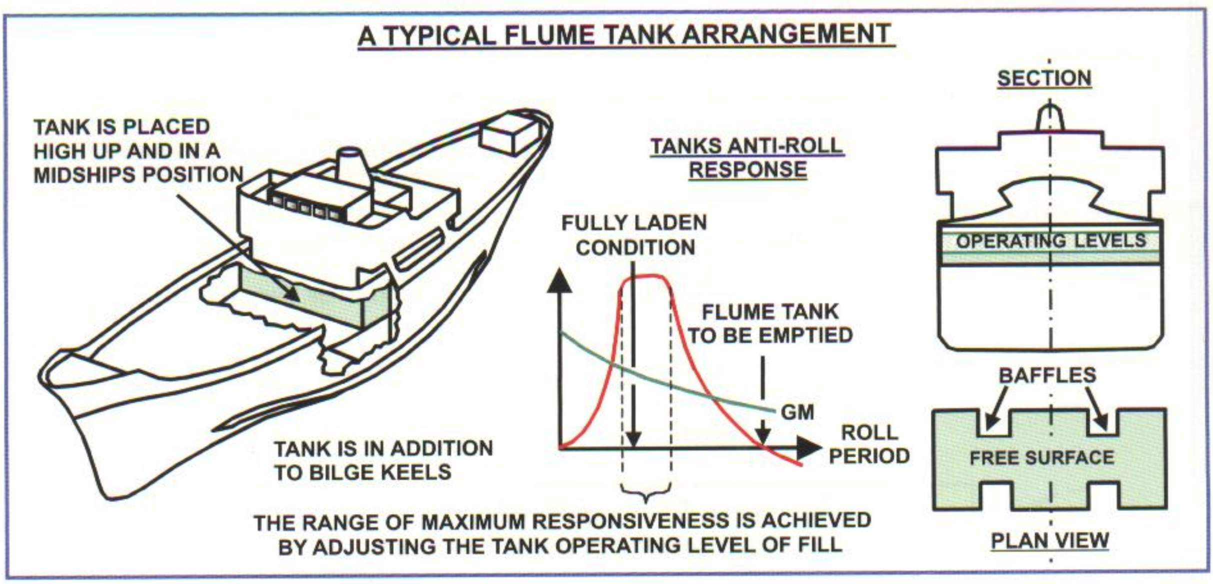
A flume tank will be most effective if the ship is rolling at the resonant frequency of the tank and the resulting oscillating free surface wave lags behind the roll of the ship by the greatest extent. This can be up to 90° if the degree of baffling is correctly designed for the ship's normal range of roll period.

THE ACTION OF A FLUME TANK AS AN ANTI-ROLL DEVICE (Cont.)

A flume tank is passive as it relies upon the roll to produce sufficient pressure gradient (angle β in the diagrams) to overcome the baffles' resistance to the flow of liquid. Such tanks are usually designed to have different operating levels, where each level is tuned to respond to a particular range of GM values. This means that the tank is most effective at resisting the onset of synchronous rolling when the ship's motion is likely to be most severe. The baffles' resistance to flow increases with the flow's velocity, so at faster rolls, the amount of liquid transferred and the tank's effectiveness are reduced. Conversely, if the roll is very slow, the resistance to flow is low and the time lag of the tank behind the roll is diminished, which again reduces the tank's effectiveness. At very slow roll periods, the liquid has time to catch up with the ship's motion and simply becomes a normal free surface effect, which reduces the ship's stability. **As a normal mixed sea can cause a ship to roll at a variety of periods, it is important to include a flume tank's Free Surface Effect when calculating the vessel's fluid GM value.** There will be a minimum GM value below which the Flume Tank cannot be used if the minimum stability criteria are to be met. At this point, the tank should either be fully emptied or pressed up and the pumping arrangements should be such that either operation can be carried out quickly and effectively.

A TYPICAL FLUME TANK ARRANGEMENT

Many dry cargo vessels suffer from a gradual loss of stability on long ocean passages as water and fuel are consumed from the double bottom tanks. This reduction of GM is often most noticeable on fine lined ships where the extent of parallel body at the waterline, and hence the proportion of full beam waterplane, decreases more rapidly with draft, than for fuller lined vessels. In order to have adequate stability at the end of a long passage, such vessels often sail with an initially high GM. This stiffness leads to fast uncomfortable roll with high accelerations, which can cause damage to either the ship's structure or the cargo. A flume tank can be used to reduce rolling when the ship is excessively stiff in the early stages of the voyage.

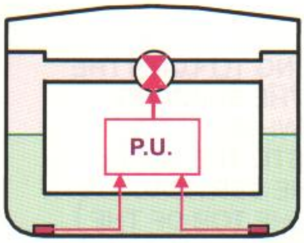


Ideally, the flume tank is located relatively high in the ship so, when in use, its liquid weight raises the ship's KG and so reduces the GM value. This will further moderate the excessive stiffness which the flume is fitted to counteract. It also allows for a reduction in top weight as the operating level is reduced with the gradual decrease of GM over the passage as bottom weight is lost by fuel consumption from the double bottom tanks. A midships position allows for a full beam width tank which can be used without any unwanted trim effects. The extent of baffling is relatively small as the tank is to be effective against a fast roll. This effectiveness will decrease as the ship becomes more tender. The shipbuilder should supply guidance regarding the operating levels of the flume tank over the range of stability conditions that it is designed to be used at.

PNEUMATICALLY CONTROLLED ANTI-ROLL TANKS

The simple flume tank only works effectively against a limited roll period. A more effective type of anti-roll tank ensures that the transfer of ballast water always occurs at the end of the roll, regardless of its period. This can be achieved by building a 'U' shaped tank into the hull and controlling the flow of water between the two sides of the tank with a system of pneumatically operated valves and pressure sensors.

A PNEUMATICALLY CONTROLLED ANTI-ROLL TANK

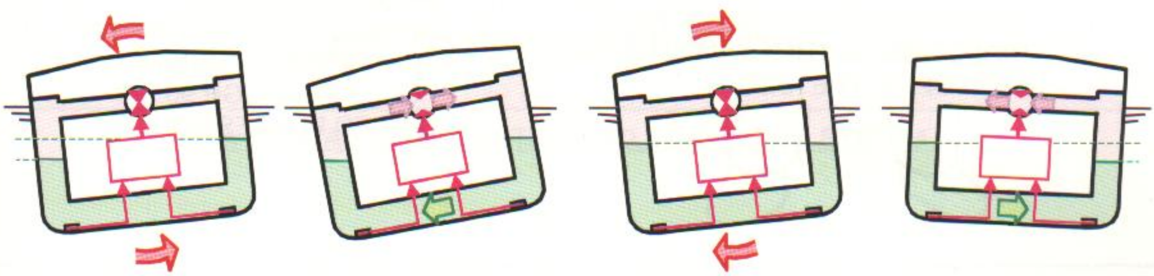


THE TANK IS UNIFORM IN CROSS SECTION

= COMPRESSED AIR
= WATER BALLAST
= PRESSURE SENSOR
P.U. = PROCESSOR UNIT
= OPEN AIR VALVE
= CLOSED AIR VALVE

THE PROCESSOR UNIT REGISTERS BOTH THE PRESSURE DIFFERENCE BETWEEN THE TWO TANKS AND THE RATE OF CHANGE OF PRESSURE DIFFERENCE. THE AIR VALVE IS ONLY OPENED IF THERE IS A PRESSURE DIFFERENCE AND THE RATE OF CHANGE OF PRESSURE DIFFERENCE IS ZERO

THE WATER CAN ONLY FLOW FROM ONE SIDE OF THE TANK TO THE OTHER IF THE AIR VALVE OPENS TO ALLOW THE AIR PRESSURE TO EQUALISE. THE AIR IS PRESSURISED TO MINIMISE THE SEAPAGE OF WATER ACROSS THE TANK BEFORE THE VALVE OPENS



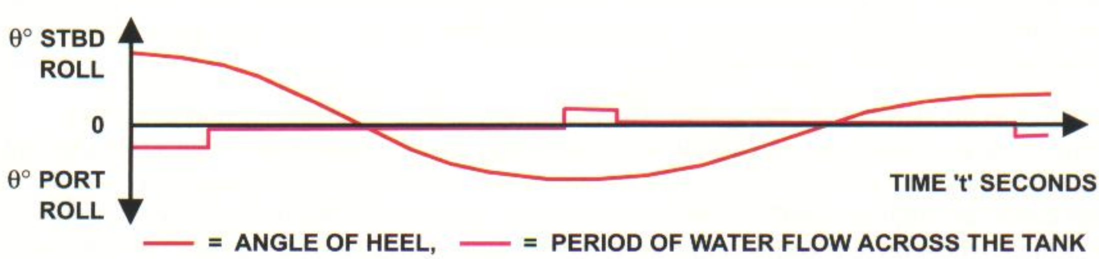
A - VESSEL ROLLING   B - THE END OF THE ROLL   C - THE RETURN ROLL   D - THE END OF THE RETURN ROLL

A - A PRESSURE DIFFERENCE EXISTS BUT IT IS STLL INCREASING SO THE VALVE REMAINS CLOSED

B - THE PRESSURE DIFFERENCE REACHES MAXIMUM, SO THE VALVE OPENS, ALLOWING WATER TO FLOW

C - PRESSURE DIFFENCE BECOMES ZERO, SO THE VALVE CLOSES AND WATER FLOW IS STOPPED

D - THE VALVE OPENS AS PRESSURE DIFFERENCE REACHES A MAXIMUM IN THE OPPOSITE DIRECTION.

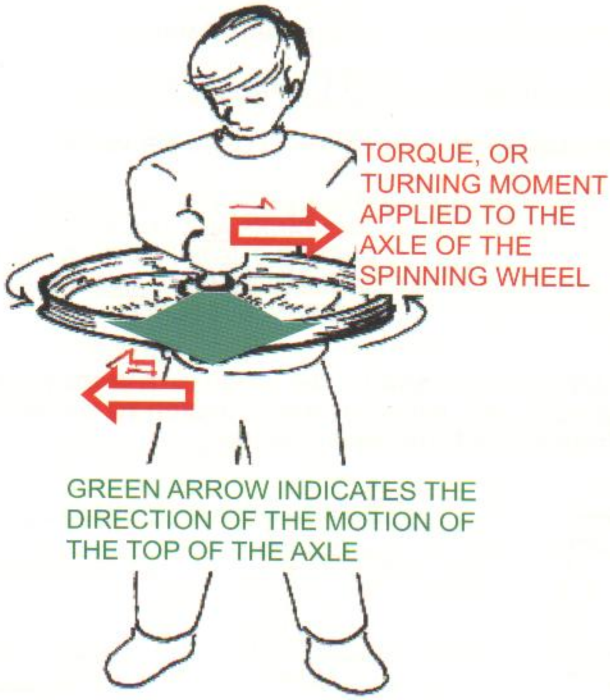


The flow of water across the tank remains passive, i.e. it relies only on gravity, but the processor unit and the valve operation require a small amount of power, which can be backed up by emergency power provision. A mid height level of water in the tank allows the greatest flow of water to occur. The air pressure must be sufficient to make the resistance of further compression of the trapped air sufficient to effectively prevent seapage of water across the tank, prior to the air valve opening. The water flow always acts to limit the extent of the return roll.

ACTIVE ANTI-ROLL DEVICES, – GYROSCOPIC STABILISATION

A spinning gyroscope possesses angular momentum and will behave such as to conserve this if it is subjected to turning force. It achieves this by transposing the applied torque through 90°, a phenomenom called 'Precession'. It is not within the scope of this book to prove the physics of this behaviour but it can easily be demonstrated by holding a bicycle wheel in one's hands and then attempting to tilt it whilst the wheel is spinning.

DEMONSTRATION OF GYROSCOPIC PRECESSION



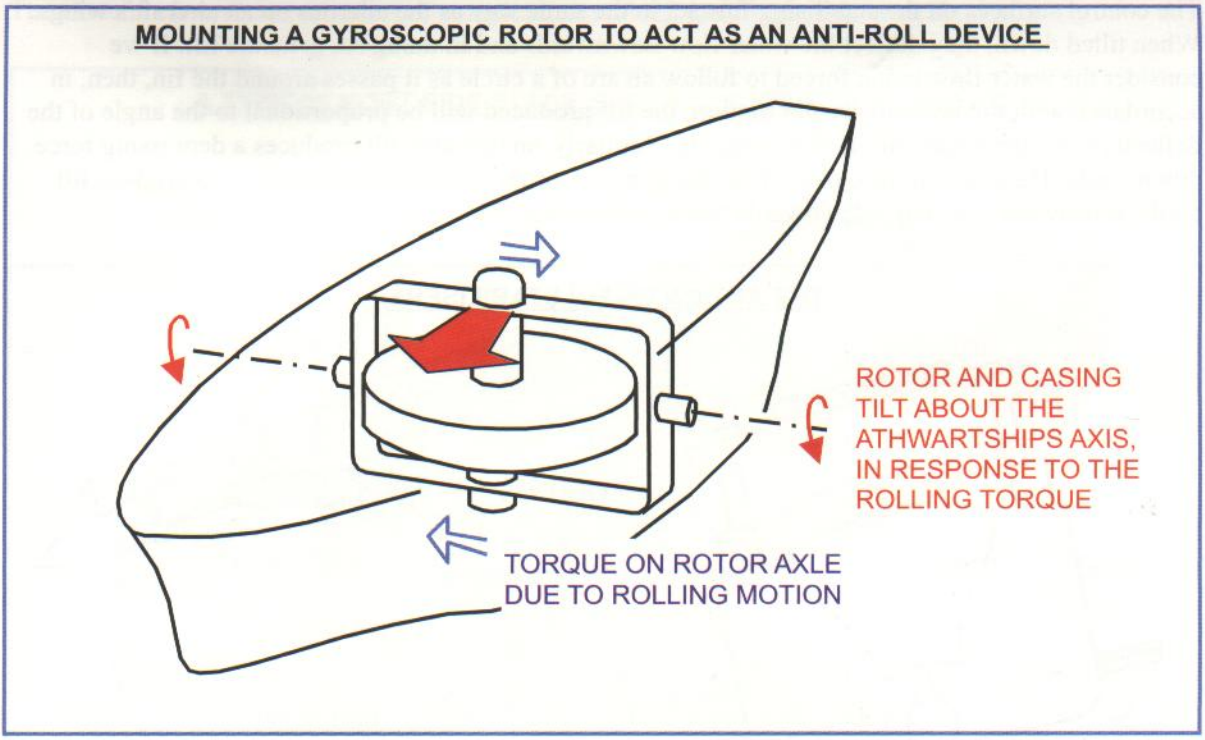
ATTEMPTING TO TURN THE AXLE OF THE SPINNING BICYCLE WHEEL SIDEWAYSTO THE LEFT ABOUT THE VERTICAL CAUSES THE TOP OF THE AXLE TO ROTATE ABOUT THE HORIZONTAL, AWAY FROM THE MAN. IF THE WHEEL IS SPUN THE OPPOSITE DIRECTION, THE TOP OF THE AXLE WOULD MOVE TOWARDS HIM. THIS PROPERTY OF THE SPINNING GYROSCOPE TO TRANSPOSE A TURNING FORCE THROUGH 90°, IS CALLED PRECESSION.

The extent to which the gyroscope resists the turning torque is increased if its angular momentum is increased, either by spinning the wheel faster or increasing its peripheral weight.

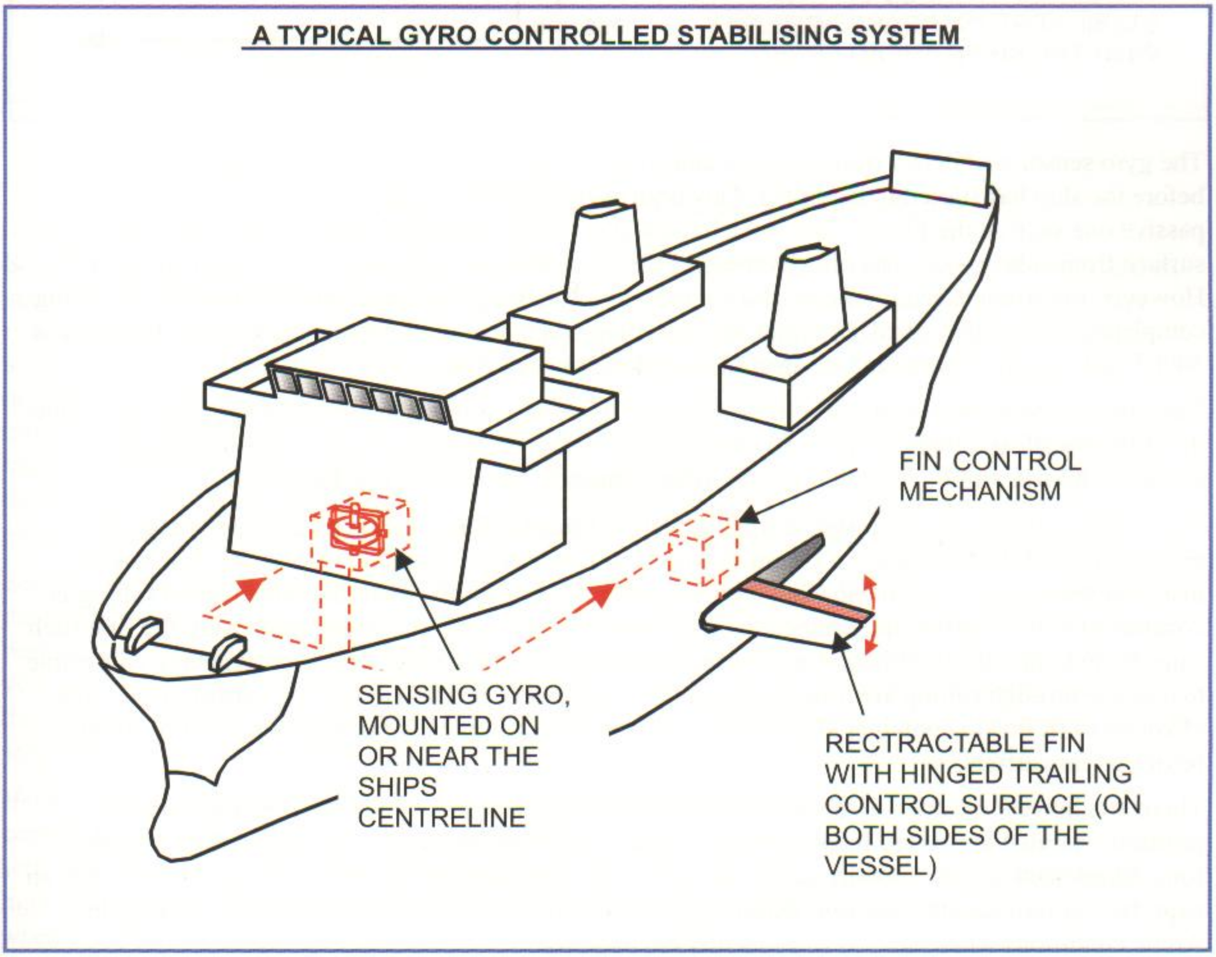
The principle of gyroscopic precession was applied directly as an anti-roll device to the 52000T Italian transatlantic liner, the 'Conte di Savoia' which was built in 1932. Three very large enclosed gyroscopes were mounted along the ship's centreline with the spinning axles aligned in the vertical. The gyroscope casings were then gimballed horizontally in the athwartships plane so, as the rolling motion attempted to rotate the gyro axles from side to side about a fore and aft axis, the motion was transposed to a fore and aft rotation about an athwartships axis, much in the same way as the bicycle wheel behaves in the above demonstration.

The system was not particularly successful. The gyroscopic rotors, their casings and gimballed mountings were very heavy so reduced the ship's carrying capacity. Furthermore, the structures took up a considerable amount of space, which could otherwise have been used for passenger accommodation or cargo space. The power requirements to spin the rotors at an effective speed was also very high, whilst there is considerable potential for damage caused by the gyroscopic motion becoming unstable if subjected to severe rolling and pitching motion.

All these factors make direct gyroscopic stabilisation unsuitable for ships and the experiment has not been repeated. However, a small gyroscope, mounted in a similar fashion, can be used as a sensor of angular acceleration due to rolling. The onset of rolling applies a transverse torque, which precession transposes to a vertical tilt of the spinning axis. This tilt angle can be used to provide electrical signals, which are then transmitted to work the control surfaces of stabilising fins. These retractable foils project outboard from the midships region of the hull bottom.

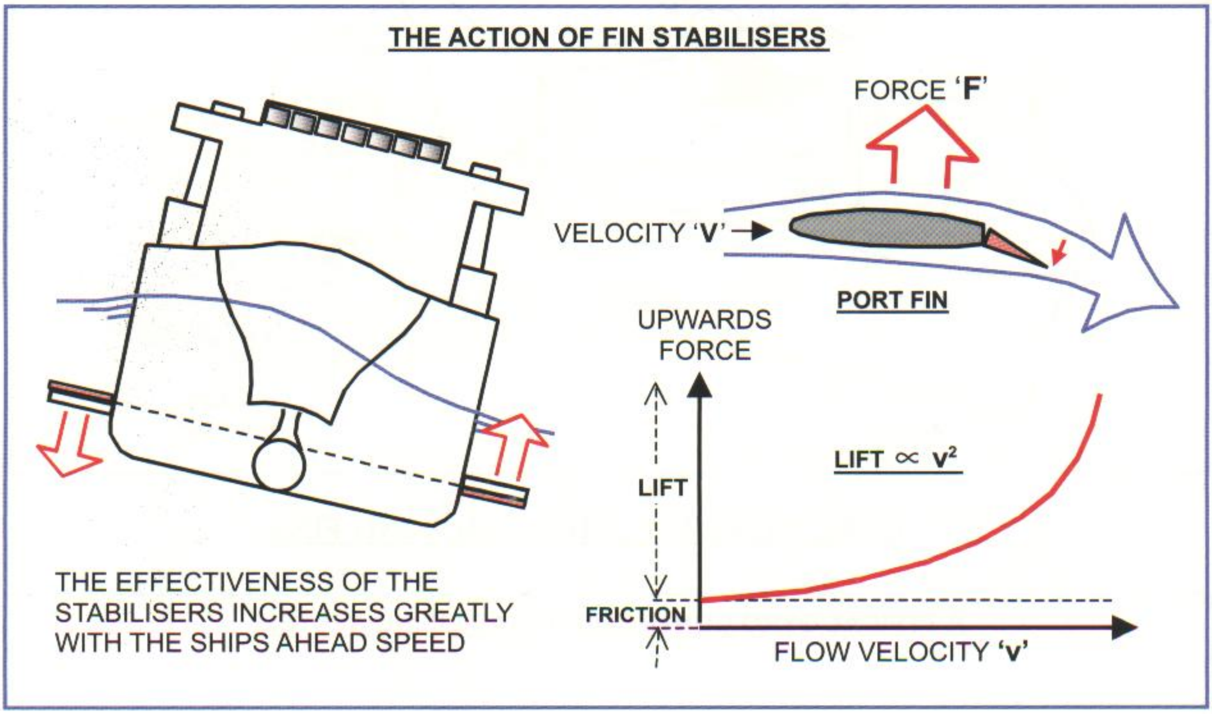


GYRO CONTROLLED STABILISING FINS



GYRO CONTROLLED STABILISING FINS (Cont.)

The control surfaces on the stabilising fins act in the same way as the ailerons on an aircraft's wings. When tilted down, they deflect the water flow downwards and in doing so, generate lift. If we consider the water flow being forced to follow an arc of a circle as it passes around the fin, then, in accordance with the laws of circular motion, the lift produced will be proportional to the angle of the deflection and the square of the flow velocity. Similarly, an upwards tilt produces a depressing force downwards. The controlling signals from the gyro sensor move the control surfaces to produce lift on the downward side and a downwards force on the other side



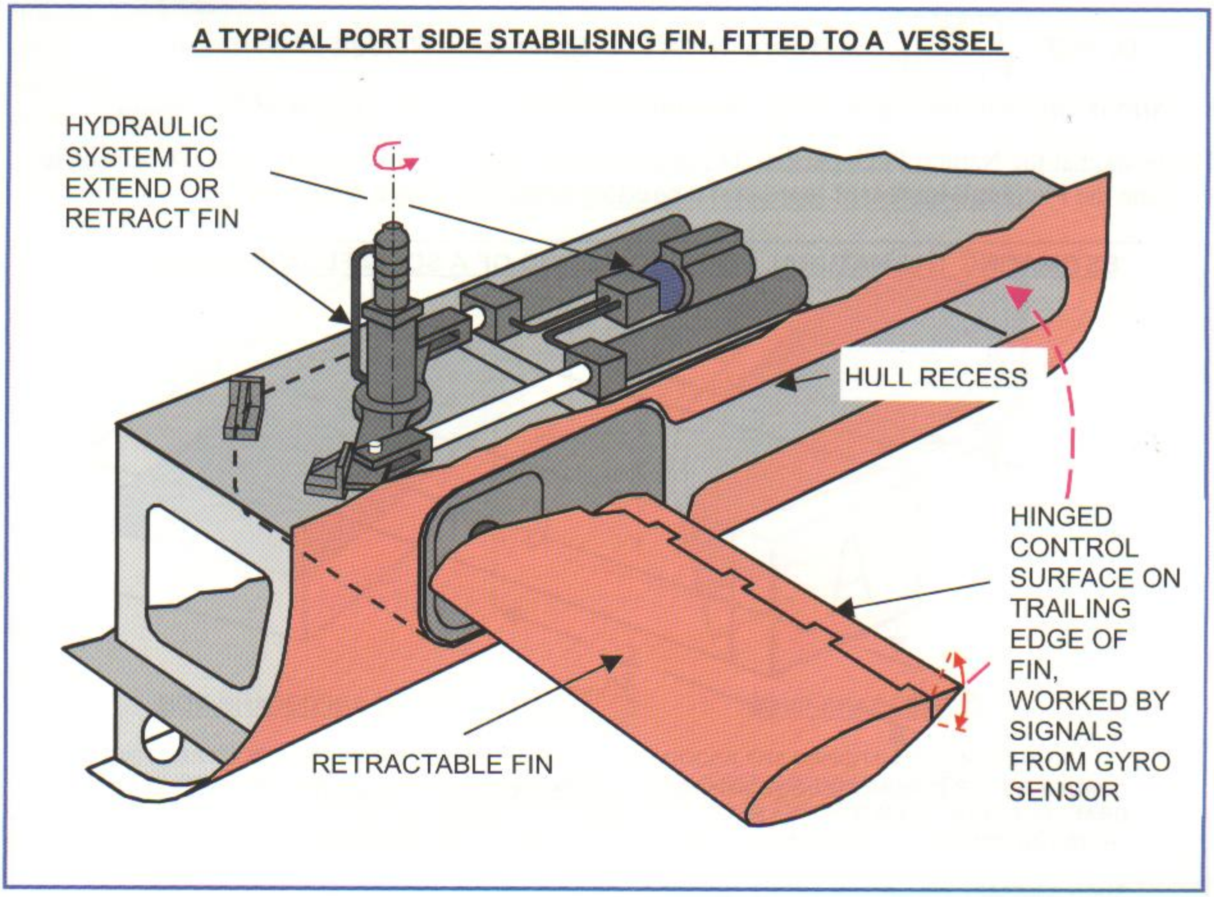
The gyro sensor will detect rolling torque and so move the fin control surfaces to oppose a roll before the ship has started to heel over. This is an important advantage of an active system over a passive one such as the Flume Tank, which must allow some rolling in order to move the liquid surface from side to side. The effectiveness of the fins increases with their surface area and span. However, the strength requirements and complexity of making fins large enough to eliminate rolling completely is such that usually smaller fins are fitted and a degree of rolling is accepted. Typically, a 3000T roll on-roll off vehicle ferry would have 4 metre span fins.

The fins are fitted amidships to avoid causing any trimming moment and must be retractable to allow the ship to berth and manoeuvre in confined waters. If the fin fails on one side, the remaining operating fin can still be used, though the effectiveness of the system will be halved.

Active anti-roll devices can respond equally well to forced rolling motion over a wide range of periods and stabilising fins are a common type of active anti-roll system in use today. The cost of installing them into a ship means that they are generally fitted only when the reduction of rolling is considered to be of prime importance in the operation of a vessel. Passenger ships were the first such ships to be built with stabilising fins, though other ships carrying cargo that is particularly vulnerable to damage through rolling are now being similarly equipped. It can also now be considered as cost effective to fit fins to container ships and vessels carrying complete vehicles, such as ro-ro freight ferries or car carriers.

There are, however, some vessels which require anti-roll measures whilst holding a stationary position. Stabilising fins are inappropriate in these conditions as there is little or no flow over the foils. Ships such as dive support vessels must rely upon versions of the anti-roll tank, as described on page 162, to provide stabilisation. Stabilising fins would also be unsuitable for ships working in Arctic conditions where sea ice is frequently encountered.

The sketch below shows the level of complexity involved in fitting a ship with fin stabilisers.



ROLLING MOTION – A SUMMARY

Severe rolling motion can be a serious hazard to the ship's structure and its cargo. Deck cargo and its securing arrangements are particularly vulnerable to the high accelerations associated with violent rolling. The ship's officers should be alert to the dangers of synchronised rolling and take action to avoid it as much as possible. There is not usually much scope for altering the ship's natural roll period, as most of the displaced weight of a vessel often consists of the cargo which is not easily transferable at sea, so altering the ship's course and speed is the quickest and most effective action to take. Anti-rolling devices, such as flume tanks and stabilising fins, can also be used to advantage, if the ship is so equipped. Active devices are more effective over a wider range of roll period than passive systems but can actually increase the rolling if the controlling mechanisms malfunction.

Severe rolling motion can contribute to disaster. In 1992 a large tanker, the 'Braer', went aground off the southernmost point of the Shetland Islands as a result of losing its main propulsion. This occurred because a stow of pipes on deck broke loose in heavy weather and damaged the ready use fuel tank vent, which subsequently allowed seawater into the fuel tank. The ship was driven ashore and lost though this may have been avoided if the vessel had been handled better to reduce the severity of the seas being taken over the deck (and, of course. if the pipe stow had been better secured).

Heavy rolling, is not the only means through which deck cargo can break loose but it has certainly contributed to such occurrences, particularly in the number of deck container stows which have been seriously damaged in recent years. Some of these incidents are almost certainly due to excessively high stows and a lack of appreciation of the forces to which the securing arrangements are subjected when the ship is rolling heavily.

THE PITCHING CHARACTERISTICS OF A SHIP

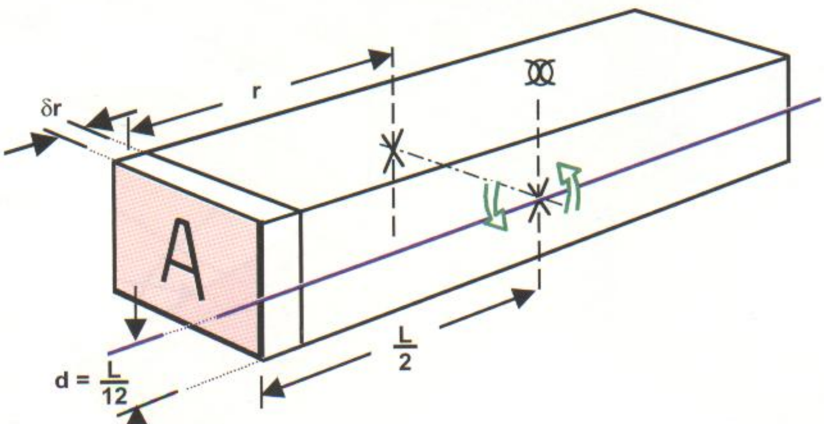
To understand the pitching behaviour of a ship, we must first look at the natural pitching period of the hull. Page 152 showed how the natural roll period 'T_R', of a ship is given by:-

$$T_R = 2\pi\sqrt{\frac{R_T^{\,2}}{g \times GM_T}}$$ ***Where 'R_T' = The Transverse Radius of Gyration***

and is determined by Rolling Moment of Inertia 'I' = '$R_T^{\,2}$' x The ship's Mass

It follows that the Natural Pitch period, 'T_P', must be given by a similar equation. This can be used to estimate the Pitching Moment of Inertia for a box-shaped hull, as shown below:-

ESTIMATING THE NATURAL PITCHING PERIOD OF A SOLID FLOATING BOX



CONSIDER A SOLID HOMOGENOUS WOODEN BOX-SHAPED VESSEL, AS SHOWN ABOVE. THE DRAFT IS 1/12 TH OF WATERLINE LENGTH 'L', WHICH IS TYPICAL OF MOST LADEN VESSELS 'ρ' IS THE DENSITY OF WOOD IN T/M^3 AND δr IS THE LENGTH OF EACH TRANSVERSE SLICE

SUCH A VESSEL WILL PITCH ABOUT THE MIDSHIPS AXIS OF THE WATERPLANE AND WE CAN CALCULATE THE MOMENT OF INERTIA OF EACH TRANSVERSE SLICE (SECTIONAL AREA 'A') ABOUT THE PITCHING AXIS AS FOLLOWS:-

PITCHING MOMENT OF INERTIA OF EACH TRANSVERSE SLICE = $\rho A\, \delta r\, r^2$ T-M^2

AND THE TOTAL PITCHING MOMENT OF INERTIA IS THE SUM OF ALL THESE MOMENTS

SO TOTAL PITCHING MOMENT OF INERTIA = $\int_{-0.5L}^{+0.5L} \rho A\, r^2\, dr$ T-M^2

SO TOTAL PITCHING MOMENT OF INERTIA = $\frac{\rho A L^3}{12}$ T-M^2

BUT TOTAL MASS = $\rho A L^3$ & MOMENT OF INERTIA = $MR_L^{\,2}$ T-M^2

SO LONGITUDINAL RADIUS OF GYRATION 'R_L' = $\frac{L}{\sqrt{12}}$ METRES

IF THE $BM_L = \frac{L^2}{12d}$ & IF $d = \frac{L}{12}$ THEN BM_L = L , AND SO $GM_L \approx$ LENGTH 'L'

NOW T_P, THE NATURAL PITCHING PERIOD, $= 2\pi\sqrt{\frac{R_L^{\,2}}{g \times GM_L}}$ SEC

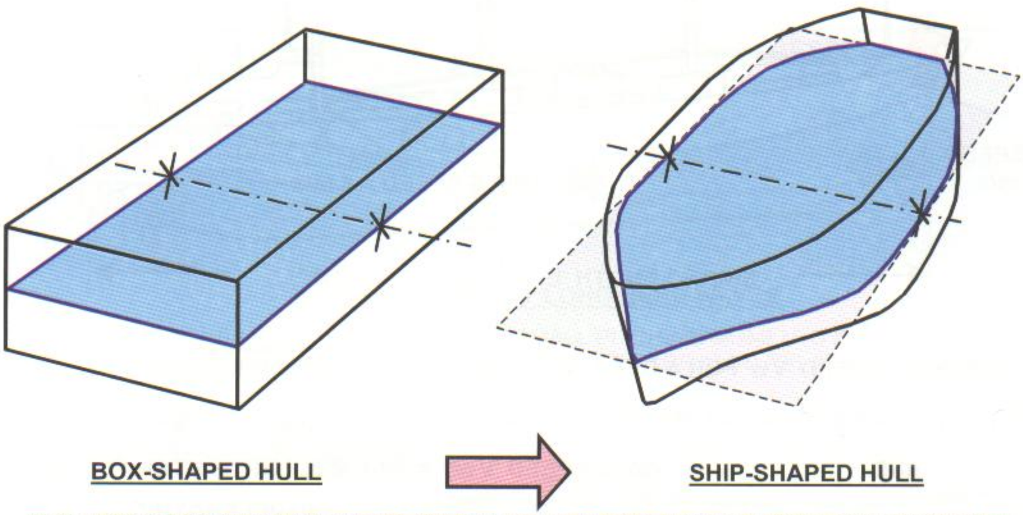
SO T_P, THE NATURAL PITCHING PERIOD, $\approx 2\pi\sqrt{\frac{L^2}{9.81 \times L}}$ SEC

HENCE T_P, THE NATURAL PITCHING PERIOD, $\approx 0.5\sqrt{\text{LENGTH 'L'}}$ SEC

THE PITCHING PERIOD OF A SHIP-SHAPED HULL

On first appearances, a solid box-shaped wooden floating block is not a good approximation of a ship-shaped hull. However, if we look at a typical commercial hull in a loaded condition, we can see that the equation for the natural pitching period, derived on the previous page, is a reasonable approximation to apply to real vessels. The weight of a full load of cargo can be considered to be homogeneously distributed along the hull and the effects of differences in hull shape on the pitching period largely cancel each other out.

COMPARING A BOX-SHAPED VESSEL TO A SHIP-SHAPED HULL



THE LOSS OF WEIGHT AT THE ENDS OF THE HULL REDUCES THE LONGITUDINAL RADIUS OF GYRATION, WHICH WILL REDUCE THE NATURAL PITCHING PERIOD. HOWEVER, THE BM L WILL ALSO DECREASE DUE TO THE LOSS OF WATERPLANE AREA IN THE BOW AND STERN REGIONS. THIS WILL TEND TO INCREASE THE NATURAL PITCHING PERIOD, SO THE TWO EFFECTS WILL LARGELY CANCEL EACH OTHER OUT.

If we consider a typical cargo ship with an LBP of 144 metres, a Beam of 20 metres, a Loaded Draft of 12 metres, and a GM_T of 0.81 metres, then its natural roll and pitching periods (**T_R** & **T_P** respectively) can be estimated as follows;-

$$T_R = 0.82 \frac{\text{Beam}}{\sqrt{GM_T}} \quad \textit{(See Page 152)} \quad \text{so } T_R = 0.82 \frac{20}{\sqrt{0.81}} = 18 \text{ SECONDS}$$

$$\& \ T_P = 0.5 \sqrt{\text{Length}} \quad \textit{(See Page 167)} \quad \text{so } T_P = 0.5 \sqrt{144} = 6 \text{ SECONDS}$$

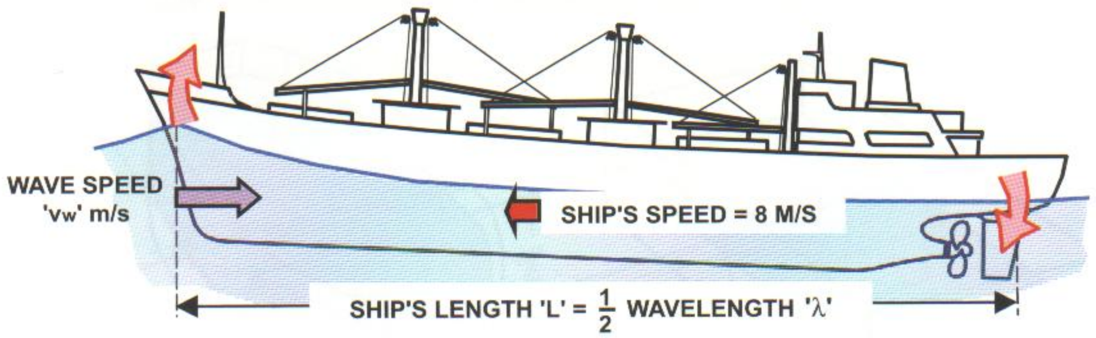
The values above are only approximate estimations but they illustrate the point that the greater stiffness of the hull's resistance to pitching, when compared to it rolling, produces a much quicker response to pitching than rolling. It should also be noted that a ship's ballast condition often requires a considerable proportion of ballast weight in the fore and aft peak tanks at the bow and stern, whilst the midships cargo spaces are empty. This will increase the 'Radius of Gyration' and so also increase the pitching period as well as concentrating weight in the ends of the hull where the pitching motion causes the greatest accelerations and local stresses.

Pitching behaviour must be looked at in a different light from rolling as it occurs in the direction that the ship is moving. Rolling motion is, at best, an inconvenience and there is a long history of efforts made to limit it to acceptable levels. Pitching, however, is essential to allow a ship to ride easily over head seas. If a ship's bow does not rise sufficiently to oncoming waves, it will ship heavy seas over the foredeck, which can cause severe damage, particularly if the ship is loaded with deck cargo, such as stows of containers or timber. Even if the deck is clear, hatches can be exposed to excessive forces of heavy seas crashing over them. The hull needs to have a pitching period shorter than that of the oncoming waves in a head sea in order to ride easily over the changing wave profile in such conditions. The apparent wave period is determined by both the wavelength and the ship's speed.

THE PITCHING OF A VESSEL IN A SEAWAY

Chapter 1, page 20, shows well-established equations that establish the relationships between wave height, wavelength, wave period and wave speed for sea waves in deep water. These can be used to calculate the apparent wave period for different wavelengths at different ship's speed so we can look at their effect upon the ship's pitching motion.

THE PITCHING OF A 144M SHIP STEAMING AT 16 KTS INTO HEAD SEAS



THE WAVE SPEED 'Vw' FOR A DEEP WATER WAVE = $\sqrt{1.56 \times \text{WAVELENGTH } \lambda}$

IF THE SHIP ABOVE HAS A LENGTH OF 144 M, THEN WAVELENGTH 'λ' IS 288 M

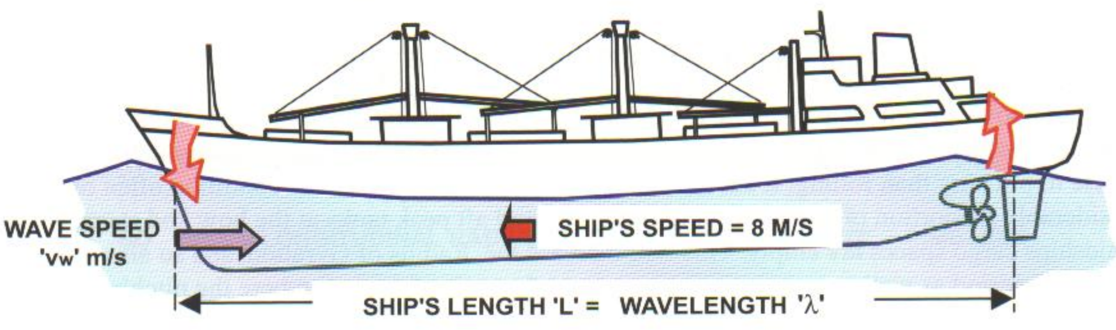
AND WAVE SPEED 'Vw' = 21.2 M/S (OR 42 KTS)

NOW, THE APPARENT PERIOD 'Tw' OF ANY WAVE = $\dfrac{\text{WAVELENGTH } \lambda}{\text{APPARENT WAVE SPEED } v_w}$

WHERE THE APPARENT WAVE SPEED IS A COMBINATION OF THE SHIP'S SPEED AND THE SPEED OF THE WAVE, SO FOR THE SHIP STEAMING AT 8 M/S

IN THE ABOVE SITUATION, 'Tw' = 9.7 SECONDS

THE PREVIOUS PAGE ESTIMATED THE SHIP'S PITCHING PERIOD TO BE 6 SECONDS SO IT SHOULD HAVE NO DIFFICULTY IN RIDING THE LONG HEAD WAVES SHOWN ABOVE. HOWEVER, CONSIDER THE SHIP'S RESPONSE TO SHORTER WAVES



WAVELENGTH NOW IS 144 M, SO WAVE SPEED = 15.0 M/S

SO NOW, APPARENT WAVE PERIOD = 6.2 SECONDS

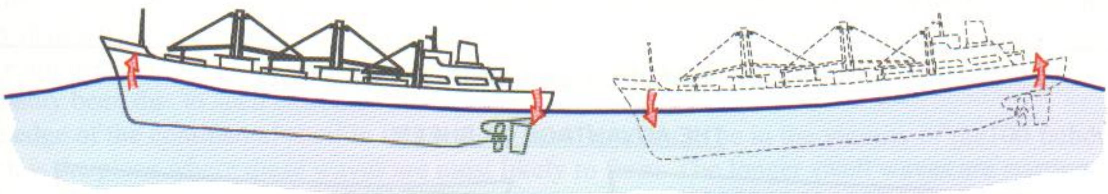
THE SHIP'S NATURAL PITCHING PERIOD REMAINS = 6 SECONDS

THE SHIP IS BEING PITCHED BOW DOWN BY THE WAVE AT THE STERN AND MAY NOW STRUGGLE TO LIFT THE BOW UP SUFFICIENTLY TO RIDE OVER THE NEXT ONCOMING WAVECREST. IN THESE CIRCUMSTANCES, THE SHIP MAY START TO TAKE SEAS OVER THE BOW, WHICH WILL GET HEAVIER IF THE APPARENT WAVELENGTH IS SHORTENED

STRESSES ASSOCIATED WITH THE PITCHING OF A VESSEL IN A SEAWAY

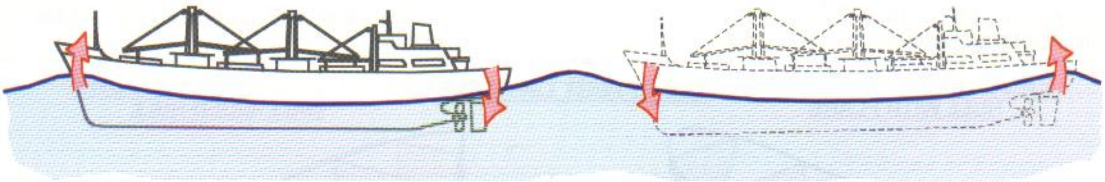
We have seen that when a ship is subjected to long head seas, it will ride easily over them and, in doing so, will ***profile*** the waveform. This leads to the hull rotating through relatively large angles of pitch whilst, conversely, being subjected to relatively small pitching moments. This is because the hull remains approximately parallel to the wave profile so the actual fore and aft distribution of buoyancy is not changing greatly. However, as the wavelength of head seas becomes shorter, the vessel has less time to respond, so it tends to drive through the seas more. The variation of pitch angle becomes less but the actual pitching moments increase because the waterline, relative to the hull, is oscillating more as waves move along the length of the vessel, which remains predominately horizontal. This situation will increase stresses, particularly at the bow and stern.

PITCHING MOTION AND ITS ASSOCIATED STRESSES



VESSEL 'PROFILING' LONG HEAD WAVES

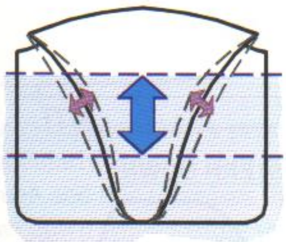
VESSEL PITCHES EASILY TO RELATIVELY SMALL PITCHING MOMENTS SO CHANGES IN THE WATERPLANE AND BUOYANCY DISTRIBUTION ARE ALSO RELATIVELY SMALL



VESSEL 'DRIVING THROUGH' SHORT HEAD WAVES

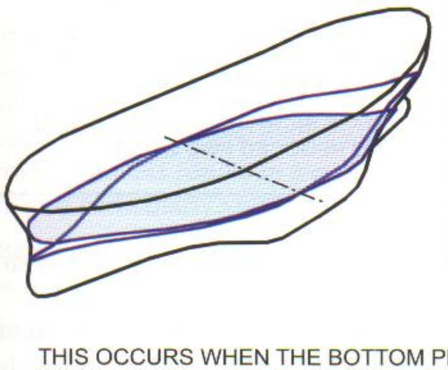
VESSEL CANNOT RESPOND FULLY TO THE RELATIVELY LARGE CHANGES IN PITCHING MOMENTS SO CHANGES IN THE WATERPLANE AND BUOYANCY DISTRIBUTION ARE CONSIDERABLE

STRESSES INDUCED BY DRIVING THROUGH SHORT HEAD SEAS



PANTING AT THE BOW AND STERN

THE CYCLIC RISE AND FALL OF THE WATERLINE AT THE BOW AND STERN CREATES AN ALTERNATING PRESSURE CHANGE AGAINST THE HULL. THIS CAUSES THE BOW AND STERN PLATING TO FLEX IN AND OUT AND CAN LEAD TO FATIGUE. FRAME SPACING AT THE ENDS OF THE HULL IS REDUCED AND ADDITIONAL LONGITUDINAL STRINGERS ARE BUILT INTO THE STRUCTURE TO RESIST PANTING



POUNDING

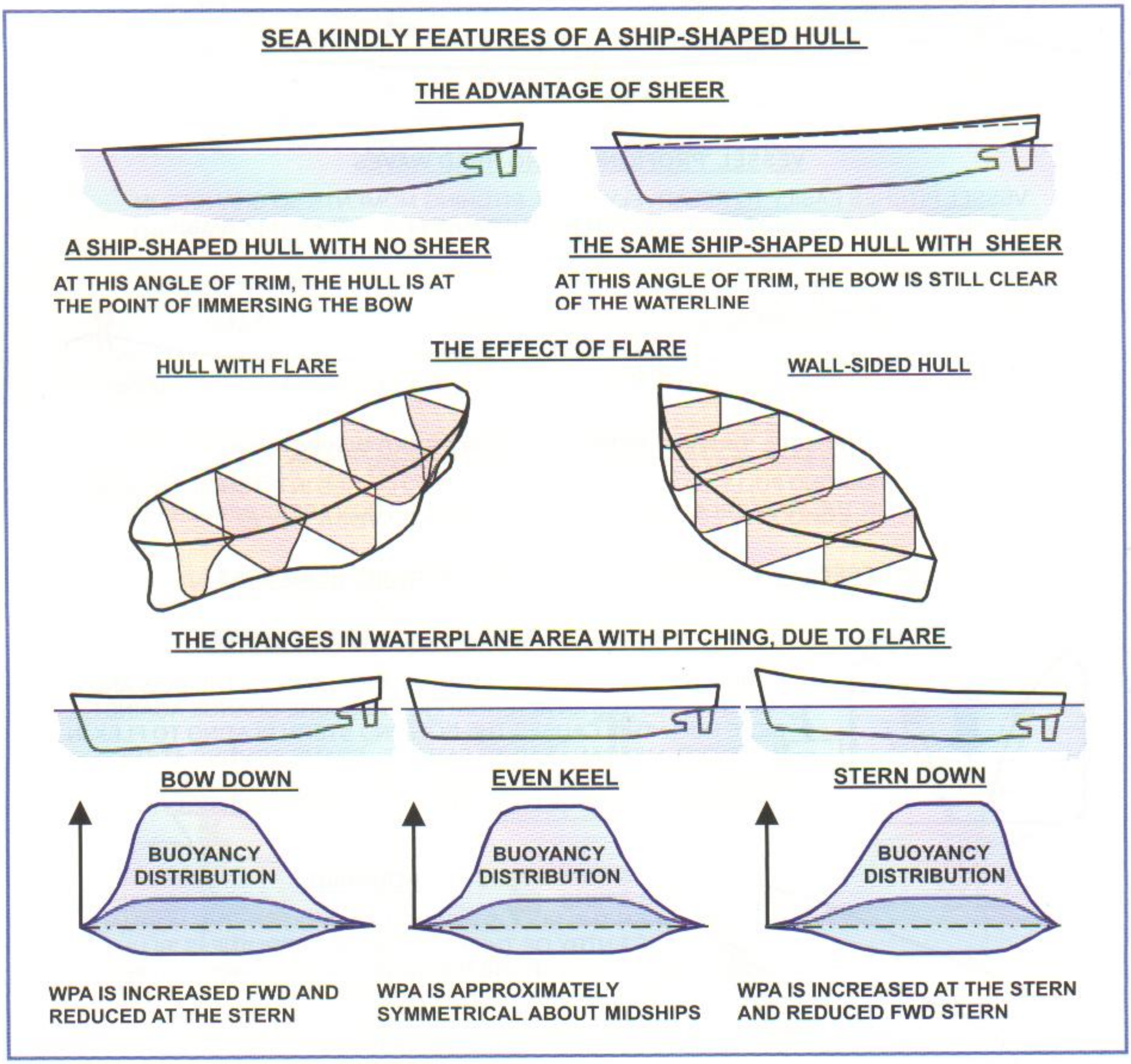
WHEN THE HULL IS PITCHED HEAD DOWN, THE NORMAL FINE ENTRY OF THE FORWARD WATERPLANE IS 'BLUNTED' BY THE WATERLINE RISING UP THE BOW, PARTICULARLY IF THE HULL HAS GENEROUS FLARE. THIS IS A MUCH LESS SUITABLE SHAPE FOR DISPLACING WATER AHEAD OF THE VESSEL AND, IF THE SHIP IS BEING DRIVEN HARD, WATER WILL NOT BE ABLE TO MOVE OUT OF THE WAY FAST ENOUGH. THE SHIP WILL SLOW DOWN, SHUDDER AND SHAKE, ALMOST AS IF IT HAS RUN INTO A SOLID WALL

SLAMMING

THIS OCCURS WHEN THE BOTTOM PLATES AT THE BOW AND STERN ARE LIFTED OUT OF THE WATER AND THEN RE-IMMERSED TOO RAPIDLY FOR THE WATER TO MOVE OUT OF THE WAY. SLAMMING TENDS TO 'CORRUGATE' BOTTOM PLATING AT THE FORE AND AFT ENDS.

HULL FEATURES THAT INFLUENCE A SHIP'S PITCHING

Tapering the fore and aft ends of the waterline is essential to produce an easily driven hull, but a simple wall-sided vessel will tend to drive through waves rather than ride over them, when encountering head seas, particularly if the apparent wave period is about the same or shorter than the hull's natural pitching period. If the hull is finely tapered and built to withstand this, then it will allow the vessel to maintain high speeds in relatively heavy head seas. Several large warships, such as the German Second World War battleship 'Bismark', were built with almost wall-sided finely tapered hulls for this reason. However, this is not a suitable design for a commercial cargo ship because such a hull has a low cargo carrying capacity for its length and the forward deck would be exposed to an unacceptable amount of battering by the seas sweeping over it.
Sea-riding capability in a high block coefficient cargo carrying hull, is achieved by incorporating sheer, which increases the bow and stern freeboard, and flare, which increases the buoyancy at the bow and stern whilst tending to deflect heavy spray outboard.

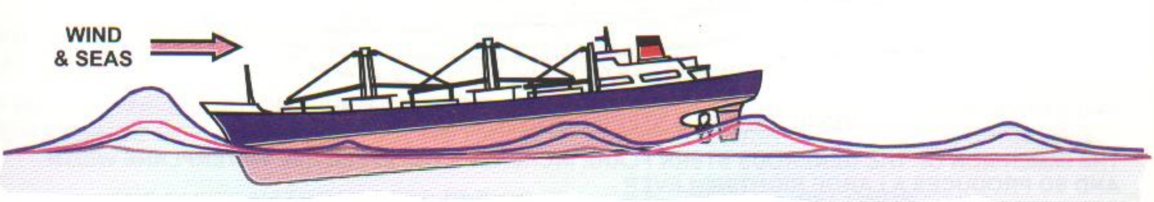


Flare enhances the increase in buoyancy at the fore and aft ends when they become immersed and so will lift the bow and stern over short head seas. However, if the ship is being driven too hard into heavy waves, the buoyancy will be increased at a rate too fast for the water to physically move out of the way. The flare of the bow (or stern) will smack into the water as if it has struck a solid surface and the entire hull will shudder and shake. **Fine lined ships with generous flare will be particularly prone to pounding, which should be avoided by reducing the ship's speed.**

MANAGING SITUATIONS PRODUCING EXCEPTIONAL HEAD SEAS

The previous page shows the extent to which a ship will ride over oncoming waves, when it is steaming into heavy head seas and how this is governed by the rate at which buoyancy is changing at the bow and the stern. The steepness of the wave slope is an important factor governing this, as well as the degree of flare, ship's speed into the wave and its responsiveness to longitudinal changes in buoyancy (i.e. its natural pitching period). Chapter 1, page 20, illustrates how the superimposition of different lengths of waves can produce exceptionally steep slopes on the occasional wave and some regions of the oceans are infamous for this effect. One such place is the south east coast of South Africa where Southern Ocean swell waves meet with waves produced by local Indian Ocean storms on the edge of the continental shelf. The combination of such waves creates the occasional very steep wave, preceeded by a deep trough, and westbound vessels have been severely damaged or sunk by encountering such a 'hole' as the following wave breaks over the ship's bows. I have, myself, seen the results of such an incident when, in 1973, the 12,000 GRT British cargo liner 'Ben Cruachan' was very distinctly bent by such a wave and had to be towed into Durban for major repair. Other ships have not been so lucky and several disappearances of vessels in this region over the years have probably been due to such extreme waves.
The edge of the continental shelf in this region (usually taken to be in the vicinity of the100 fathom line), is the place where these waves are most likely to form. The longer swell waves are starting to feel the bottom at water depths of about 200 metres so there is a steepening effect due to these waves slowing down and there is also the strong local southwest going Agulhas Current, which further fore-shortens the northeast moving swell. It is better to sail much closer inshore, where quite a lot of the wave energy has been dissipated or further out to sea in deep ocean water, when going west around the Southern tip of Africa.

ENCOUNTERING AN EXCEPTIONALLY DEEP TROUGH IN HEAVY HEAD SEAS



THE SHIP IS BEING PITCHED BOW DOWN INTO THE TROUGH AND MAY NOT BE ABLE TO RIDE OVER THE FOLLOWING EXCEPTIONALLY STEEP CREST

The exceptional wave produced in these circumstances occurs in seas that, overall, may not be particularly large. A ship can have been maintaining quite a fast speed prior to encountering the trough and this increases the force of the subsequent crest breaking over the bow. It is important to slow down when sailing in such an area in conditions of moderately rough seas, which can combine with the underlying swell to produce such waves.

For a given heavy head sea condition, the onset of slamming is quite distinctive and, in general, ships should not be driven so hard into head seas that it is a frequent occurrence. It puts undue stress on the hull, particularly up forward, and the resulting excessive vibration can cause damage elsewhere in the vessel. Many ships now have generous flare at the stern to provide a wide aft deck for deck cargo stowage and this results in a considerable expanse of nearly flat plating above the propeller cutaway, which is particularly vulnerable to slamming and pounding.

Altering course to put the seas to one side of the bow may allow speed to be maintained, providing the resulting rolling is acceptable, otherwise the ship simply has to slow down.

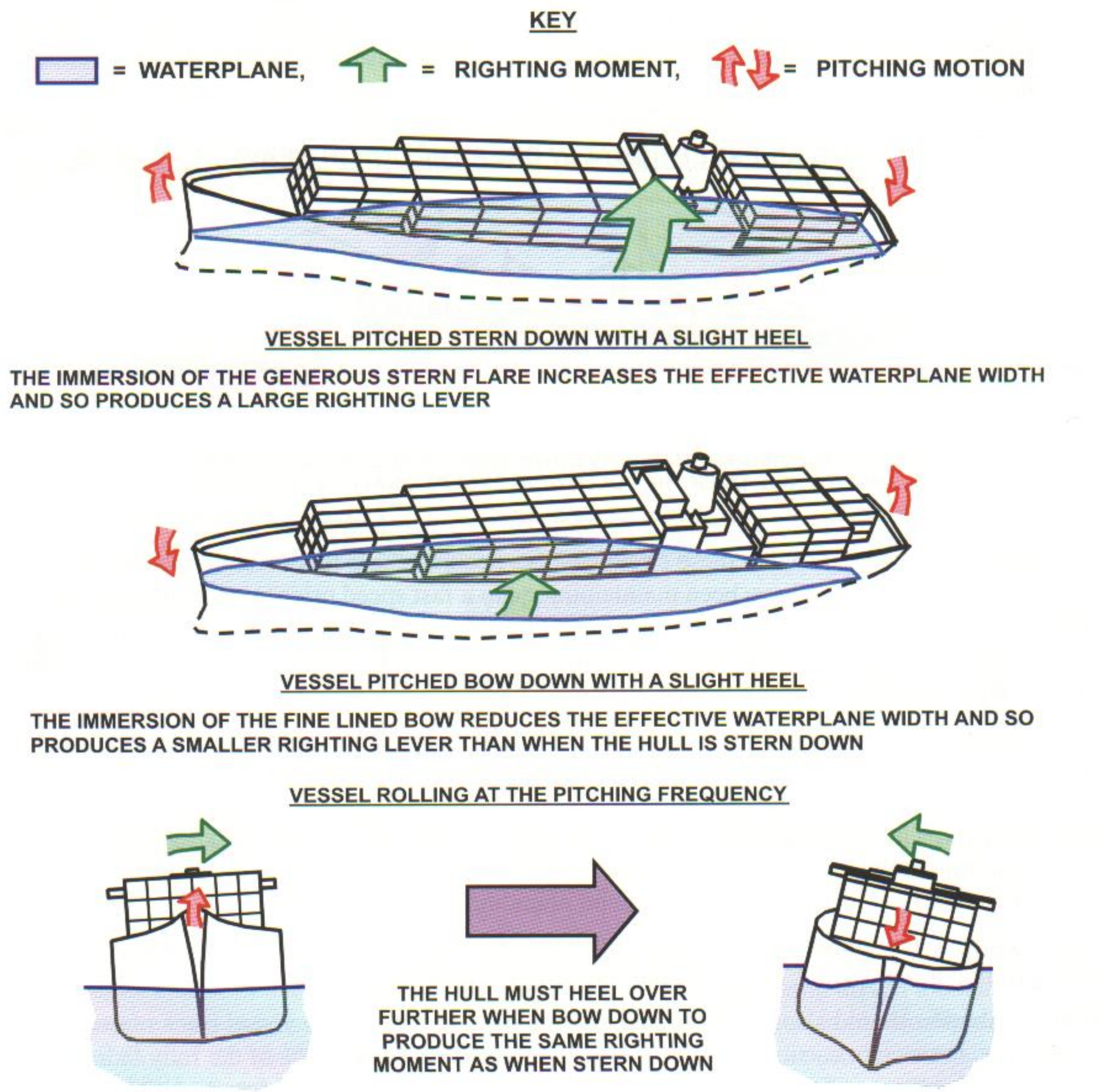
Ballast distribution should also be considered. Slamming is reduced if the vessel has a reasonable draft and, typically, the weight carried in ballast condition should be at least 40% of the fully laden deadweight, if heavy weather is expected. However, excessive weight in the fore and aft peak tanks will slow down the pitching response and so possibly increase the ship's vulnerability to pounding.

PITCH INDUCED OR PARAMETRIC ROLLING

The development of container ships, requiring large deck cargo capacities whilst having fast service speeds, has resulted in hullforms with fine lined bows combined with a wide stern in order to maximise the deck space available for container stows. This asymmetry between the bow and stern lines has produced the tendency for the ship's pitching motion to induce rolling, which has been very severe in several recent cases. The phenomenon is described in the paper ***'Parametric Roll: A Threat to large Container Ships'***, which was presented to the ***'Boxship 2001 Conference'*** by its author, Dirk Lehmann of 'Intering' (A manufacturer of stabilising systems).

This type of rolling has been called ***'Parametric Rolling'*** as it depends upon the parameters of the ship's displacement and righting lever and is most marked when the pitching period is either equal to or half that of the vessel's synchronous roll period. If the vessel has a slight heel due to windage or rudder action etc. then the effective immersed waterline beam and righting lever will increase as the hull pitches stern down. This will create a large restoring moment that will be unchecked if the hull then pitches bow down at the end of the return roll as its effective waterplane width and righting moment will be reduced. This cyclic variation in the righting lever occurs at the pitching frequency and can induce a rapid build up in rolling motion if its period is close to either the ship's natural roll period or half the vessel's natural roll period (See pages 149 to 152)

THE VARIATION IN RIGHTING LEVER WITH PITCH OF A FINE LINED, FULL STERN HULL



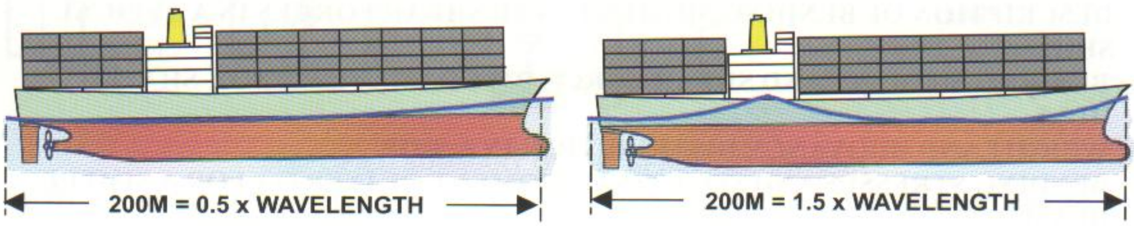
PITCH INDUCED OR PARAMETRIC ROLLING (Cont.)

When the ship is rolling at the pitching frequency, as shown on the previous page, one of the reversal points of the rolling motion will coincide with minimum stability whilst maximum stability will occur at the other reversal point. If the rolling is at twice the pitching frequency, then minimum stability will occur at both points of roll reversal. Both situations can lead to a severe build up in synchronous rolling and, as the angle of heel at the ends of the roll increases, so does the effect of the asymmetry between the bow and stern lines.

If we consider a typical 40,000 T deadweight container ship of 200 metres in length with a 30 metre beam, then we can look at the waves that will tend to cause the maximum pitching motion in the following diagram.

WAVE PROFILES LIKELY TO INDUCE THE MOST SEVERE PITCHING IN A 200 M LONG VESSEL

WAVE SPEED 'C' $= \sqrt{1.56 \times \text{WAVELENGTH}}$ **M/S & WAVE PERIOD 'T w'** $= \dfrac{\text{WAVELENGTH}}{\text{WAVE SPEED}}$ **SEC**



WAVE SPEED = 25 M/S, PERIOD = 16.0 S
IF THE VESSEL IS STOPPED IN THE WATER

WAVESPEED = 14.4 M/S, PERIOD = 9.2 S
IF THE VESSEL IS STOPPED IN THE WATER

THE WAVE PERIODS ABOVE WILL BE REDUCED IF THE VESSEL IS STEAMING INTO HEAD SEAS AND INCREASED IF IT IS RUNNING BEFORE FOLLOWING SEAS

IF WE USE THE EQUATION GIVEN ON PAGE 152 TO ESTIMATE THE SHIP'S NATURAL ROLL PERIOD FOR A RANGE OF GM VALUES, THEN WE CAN DETERMINE THE RISK OF PARAMETRIC ROLLING

THE SHIP'S NATURAL ROLL PERIOD 'Ts' $\approx 0.8 \dfrac{\text{BEAM}}{\sqrt{\text{GM}}}$ **SECONDS**

GM (METRES)	1	2	3
NATURAL ROLL PERIOD (SECONDS)	24.6	17.4	14.2

A SLOW MOVING OR DRIFTING SHIP WITH A 2 METRE GM WOULD BE PRONE TO PARAMETRIC ROLLING FROM WAVES IN THE REGION OF 2/3 OF THE VESSEL'S LENGTH AS THESE HAVE A PERIOD ABOUT HALF THAT OF THE NATURAL ROLL PERIOD. IF THE SHIP WAS RUNNING BEFORE SUCH WAVES AT FULL SPEED (AROUND 12 M/S) THEN THE APPARENT WAVE PERIOD WOULD BE CLOSE TO THE NATURAL ROLL PERIOD AND AGAIN PARAMETRIC ROLLING COULD OCCUR

Some of the worst incidents of parametric rolling have occurred to a ship steaming at reduced speed into heavy head seas. The violence of the rolling has been such that a considerable proportion of the deck stow of containers has been lost due to failure of the lashings.

There is little that the ship's crew can do to counter parametric rolling. If the ship is slow steaming into head seas to avoid pounding and slamming, then increasing speed is not an option and any course alteration is unlikely to change the pitching period significantly (See page 155). In any case the heeling effect of rudder action could exacerbate the situation.

Parametric rolling is only avoided by fitting the ship with an anti-roll tank stabilising system that acts as described on page 162 and so is effective at all speeds. The system must be reliable and capable of operating from the emergency electrical supply in the case of a 'dead ship'.

CHAPTER 8

SHEAR FORCES, BENDING MOMENTS AND LONGITUDINAL STRENGTH

SUMMARY

THIS CHAPTER DEFINES BENDING MOMENTS AND SHEAR FORCES, AND HOW THEY ARISE IN A FLOATING HULL. BENDING MOMENT CALCULATIONS FOR A BOX-SHAPED HULL ARE EXPLAINED WITH EXAMPLES AND THE METHODS FOR APPLYING THESE PRINCIPLES TO A SHIP-SHAPED HULL ARE OUTLINED.
THE CHAPTER THEN DEALS WITH THE PRINCIPLES OF BENDING STRESSES IN BEAMS AND APPLIES THIS TO A SHIP'S STRUCTURE.

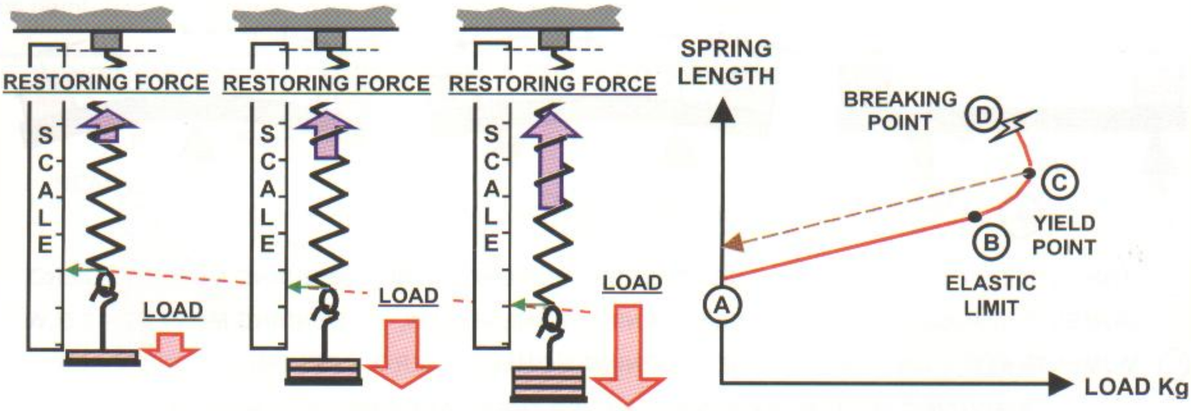
1) **ELASTICITY DESCRIBED**
2) **BENDING MOMENTS IN BEAMS.**
3) **DESCRIPTION OF BENDING MOMENTS AND SHEAR FORCES IN A TYPICAL SHIP.**
4) **BENDING MOMENT AND SHEAR FORCE DIAGRAMS FOR A BOX-SHAPED HULL.**
5) **WEIGHT AND BUOYANCY DISTRIBUTION IN A SHIP.**
6) **BENDING STRESSES AND SECTIONAL MOMENT OF INERTIA FOR A SIMPLE BEAM.**
7) **BENDING STRESS CALCULATIONS FOR COMPLEX SHAPED BEAMS.**
8) **BENDING STRESSES IN A SHIP AND POTENTIAL WEAKNESSES.**

CONTENTS

THE ELASTIC PROPERTIES OF SHIP BUILDING MATERIAL

The strength of a ship depends upon the elastic properties of its structural material. People usually associate the word 'elastic' with rubber bands rather than material such as steel. However, steel is elastic and this can be demonstrated by the simple secondary school physics experiment in which the stretch of a spring under varying loads is investigated.

THE ELASTIC BEHAVIOUR OF A LOADED STEEL SPRING



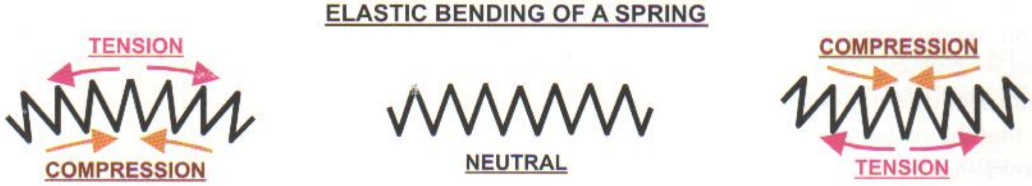
SIMPLIFIED SUMMARY OF THE SPRING BEHAVIOUR

'A' TO 'B' SPRING LENGTH INCREASES UNIFORMLY WITH ADDITIONAL LOAD AND RETURNS TO ITS ORIGINAL LENGTH WHEN THE LOAD IS REMOVED

'B' TO 'C' SPRING LENGTH INCREASES MORE EASILY WITH ADDITIONAL LOADS AND IS PERMANENTLY DEFORMED EVEN IF THE LOAD IS REMOVED LOAD

'C' TO 'D' SPRING LENGTH CONTINUES TO INCREASE, EVEN WHEN THE LOAD IS REDUCED

ELASTIC BENDING OF A SPRING



A material behaves elastically if it changes shape at a uniform rate in response to any applied force and returns to its original shape when such forces are removed. The molecular structure of the steel is such that the forces acting between atoms can be thought of as almost tiny springs themselves. At any given temperature, the atoms, though vibrating, maintain an average spacing between themselves. **Changing this spacing by force generates restoring forces that oppose further change in the molecular spacing.** When they are pushed closer together, repulsive forces are generated to push them further apart and when they are stretched further apart, increased attractive forces are produced to bring them back together. There is a limit to any such behaviour and if the spring is stretched beyond its ***elastic limit***, then the attractive forces between the atoms will no longer increase with further separation. If the load is now released, the spring will not return quite to its original length and will be weakened. In ***ductile*** materials, such as steel, further increases in load gradually weaken it more until the ***Yield Point*** is reached when the atoms continue to slide further apart, even if the load is removed. It is said to have become ***plastic*** and is near the point of failing.

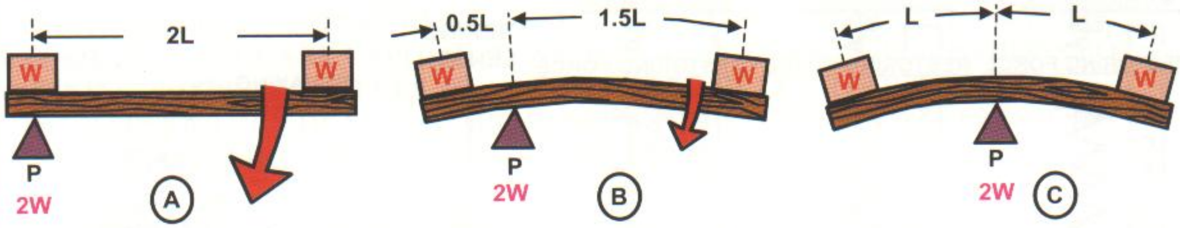
Ductile materials have a molecular structure that allows a localised stress to be spread out into the adjoining material. If we bend a steel spring, it will adopt a curved shape, rather than a distinct corner or kink, provided that the distortion remains within the spring's elastic limit. The ductile nature of the steel will tend to avoid stresses being concentrated at a single point and the spring can be flexed without losing its strength.

The strength of any structure can be maintained, providing the distorting forces applied to it do not produce stresses that exceed the material's elastic limit.

BENDING MOMENTS

If we consider a pivoted beam loaded with a weight at each end like a see-saw then, depending upon where the support is placed, the moments of the two weights will oppose each other to a greater or lesser extent. An unbalanced moment will cause the beam to turn or rotate, whereas balanced moments will cause the beam to bend. **When a beam is balanced then the bending moment, at any point along its length, is the moment about the point which is required to counter the turning moment.**

BENDING MOMENTS ACTING ON A BEAM WITH A MOVEABLE PIVOT

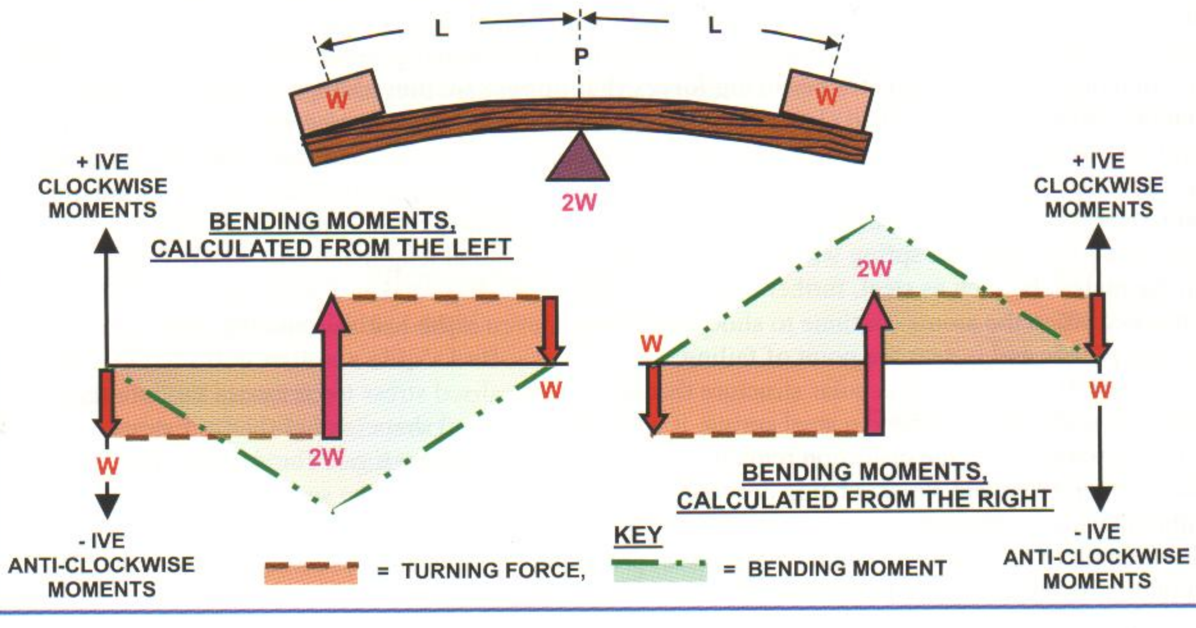


(A)	MOMENTS @ P = 2LW,	NET TURNING MOMENT = 2LW,	BENDING MOMENT = ZERO
(B)	MOMENTS @ P = 1.5LW - 0.5LW,	NET TURNING MOMENT = LW,	BENDING MOMENT = 0.5LW
(C)	MOMENTS @ P = LW - LW,	NET TURNING MOMENT = ZERO,	BENDING MOMENT = LW

A BENDING MOMENT EXISTS WHENEVER THERE ARE OPPOSING MOMENTS

If we consider situation 'C', in the above diagram, then there are two equal but opposite moments acting around the pivot. It does not alter the situation if we consider the clockwise moment to be the bending moment, countering an anti-clockwise turning moment, or vice versa. The beam will still be bent in the same way. However, when the bending moments are calculated, we must accumulate the area under the shear force curve from one end of the beam and, depending upon which end we choose to start from, we will obtain a positive or negative value for the bending moment. Providing the load distribution is obvious, it is relatively easy to determine which way the beam is bending. However bending moment calculations for complex structures, such as ships, are often computerised and it is important that the program writers ensure that the computerised answer clearly indicates whether the beam is bowed up or down in the middle. A program can be written for either possibility, depending on which end the bending moments are calculated from.

CLOCKWISE OR ANTI-CLOCKWISE BENDING MOMENTS



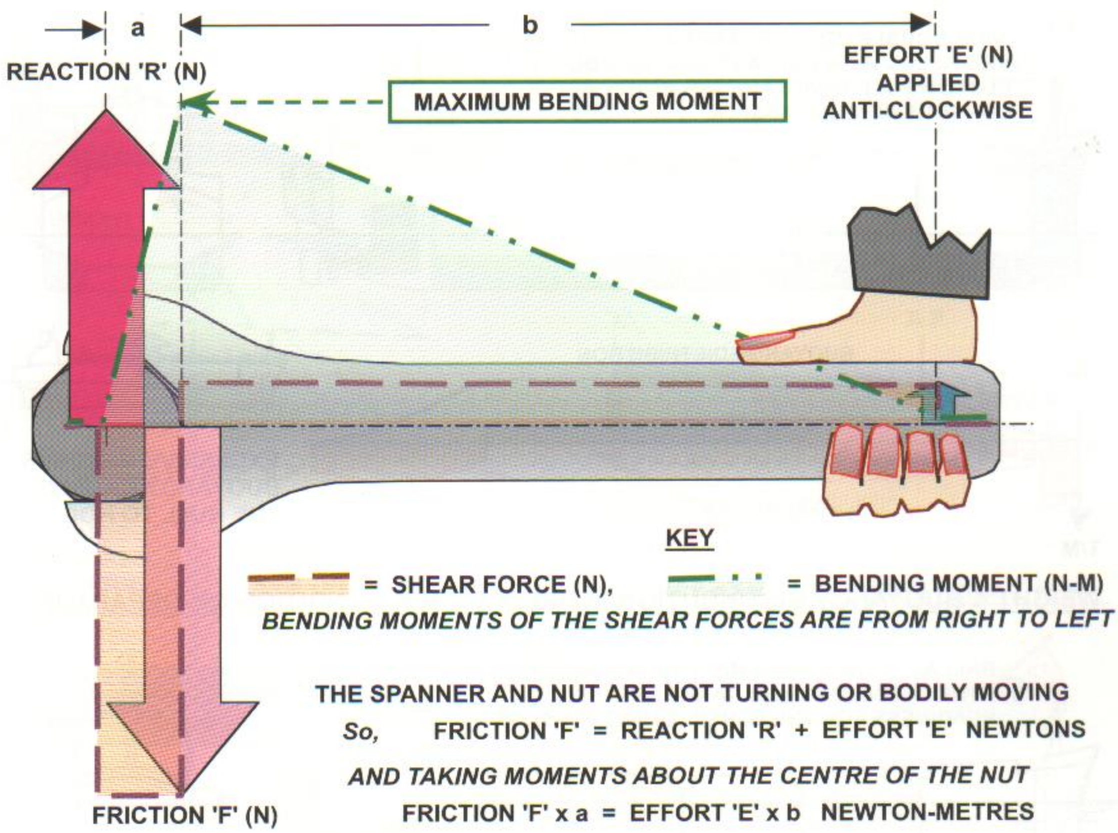
BENDING MOMENTS (Cont.)

We can see the effect of Bending Moments by considering the simple case of trying to loosen a tight nut with a spanner. The weight of the spanner can be considered insignificant, so there is no loading to take into account other than the following three point forces (measured in 'Newtons').

1) The effort being applied by hand to turn the spanner.
2) The frictional resistance in the thread of the nut.
3) The reaction of the bolted thread which the spanner head is being pulled against.

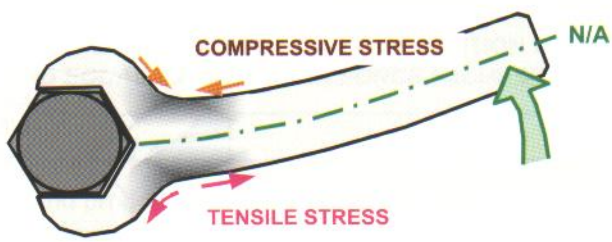
These forces are known as **Shear Forces** and are in balance until the nut actually starts to turn. Up to this point, their combined effect is to produce pure bending moments.

BENDING MOMENTS PRODUCED IN A SPANNER, PRIOR TO LOOSENING A TIGHT NUT



THE SHEAR FORCE REMAINS CONSTANT BETWEEN THE EFFORT AND THREAD FRICTION, SO THE BENDING MOMENT INCREASES UNIFORMLY TO REACH A MAXIMUM AT THE POINT OF CONTACT WITH THE THREAD, WHICH BECOMES THE POINT OF GREATEST STRESS

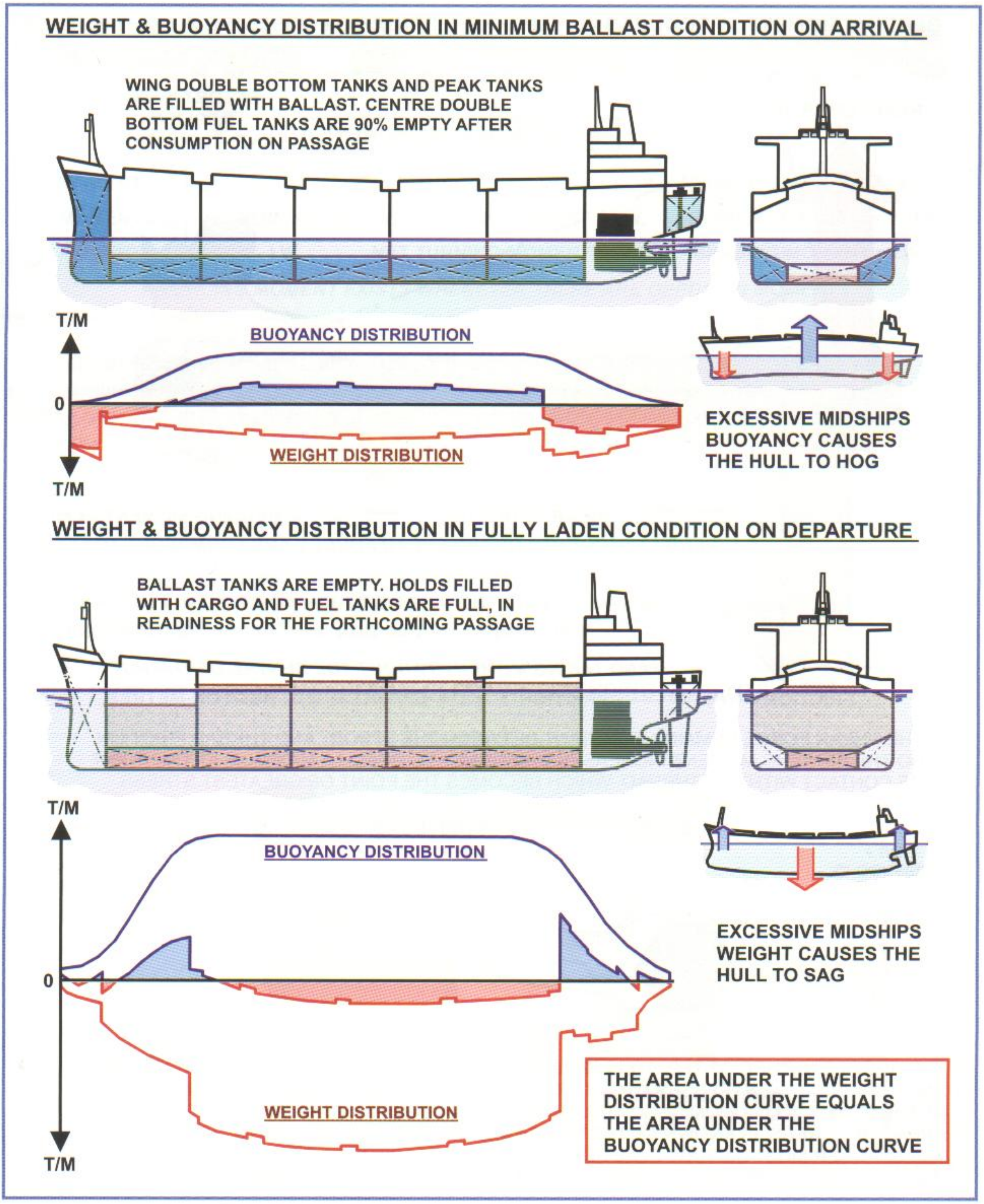
SPANNER BENDING UNDER LOAD



THE DUCTILE NATURE OF THE STEEL SPANNER TRANSPOSES SHEAR STRESS INTO TENSILE AND COMPRESSIVE STRESSES WHICH ARE GREATEST ALONG THE EDGES OF THE SPANNER AT THE POINT OF MAXIMUM BENDING MOMENT. THE MIDWAY AXIS, BETWEEN THESE TWO AREAS OF STRESS, IS KNOWN AS THE **'NEUTRAL AXIS 'N/A'** AND IS A LINE OF ZERO STRESS. THE SPANNER MUST DISTORT ELASTICALLY SO AS TO GENERATE RESTORING FORCES WITHIN ITS MOLECULAR STRUCTURE, WHICH ARE LARGE ENOUGH TO COUNTER FURTHER BENDING.

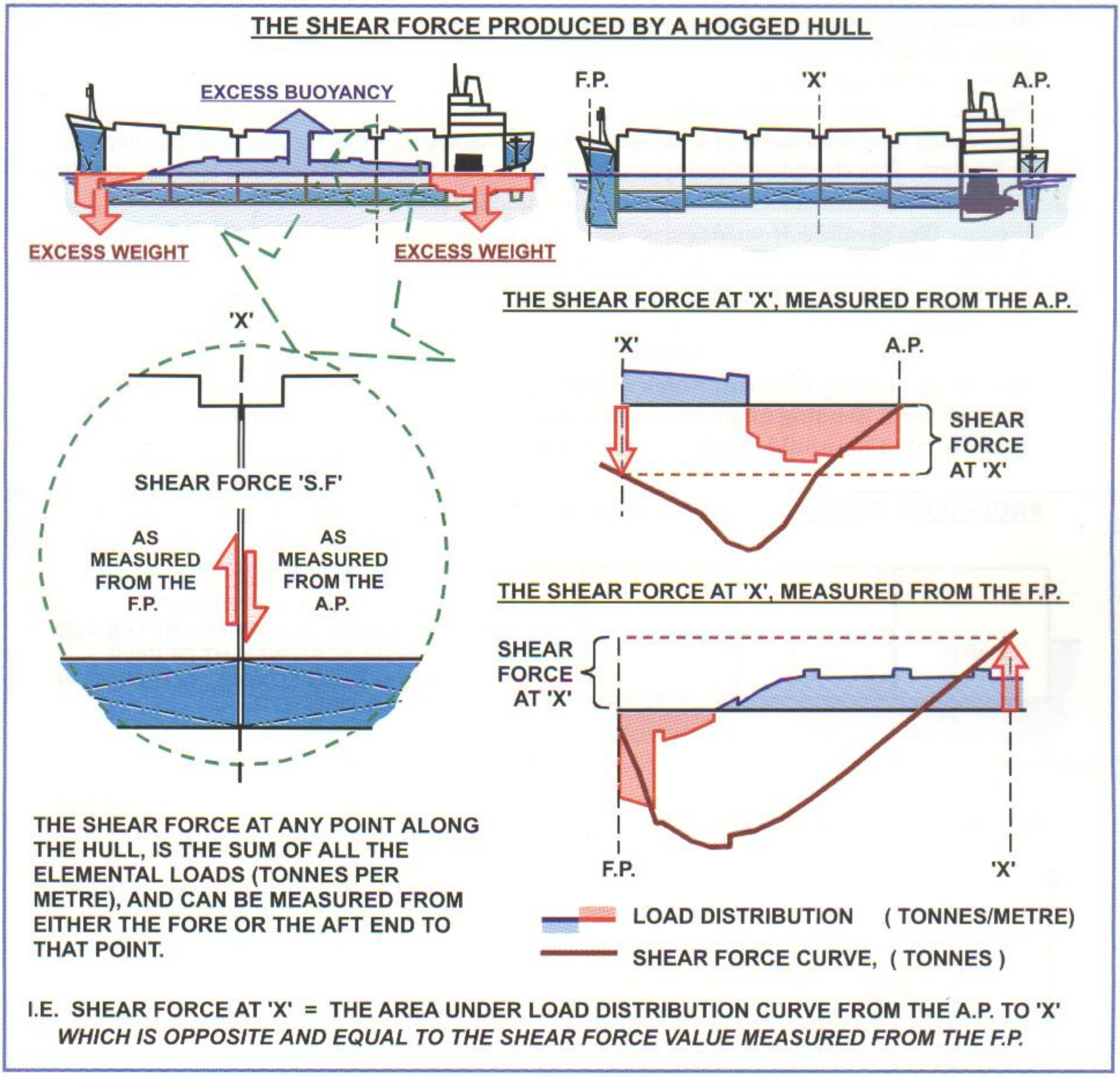
THE LONGITUDINAL BENDING OF A SHIP'S HULL

The total weight of any floating vessel must be supported by an equal and opposite upwards force of buoyancy, acting through the centre of buoyancy in a vertical line with the centre of gravity. However, the distribution of the separate weights along the length of the hull is very rarely exactly matched by the buoyancy distribution. If an excess of weight in the midships region of the hull is counteracted by excessive buoyancy at the fore and aft ends, the hull will tend to ***sag*** in the middle. The opposite situation will cause the bow and stern to droop, relative to the middle, and the hull is said to be ***'hogged'***. Most commercial cargo ships tend to sag slightly when fully loaded and hog when in the ballast condition.



SHEAR FORCES AND BENDING MOMENTS IN A SHIP'S HULL

The difference in the distribution of weight and buoyancy along the length of the hull, is known as the ***Loading Distribution*** and creates stresses which would be relieved if the various sections of the hull were free to float at different levels. If we look at a transverse section at any point along the hull length, then the accumulated load on one side of the section is known as the ***Shear Force***, because it is attempting to force the different hull sections to slide past each other. The hull is not bodily rising or falling at any point along its length, so the shear force of the accumulated load to one side of any point, will be balanced by an equal and opposite shear force due to the cumulative load on the other side of the point.



If we just considered the cumulative load between 'X' and the Aft Perpendicular, 'A.P.', then each elemental part of the load would produce a trimming moment about 'X'. However, the hull is not changing its trim, so these moments must be counteracted by the moments due to the load distribution between 'X' and the Forward Perpendicular, 'F.P.'. Either of these two equal but opposing moments can be considered to be the bending moment at 'X' and **will equal the sum of all the separate shear force values multiplied by their distances from 'X'. Hence, the bending moment at 'X' equals the area under the shear force curve measured between 'X' and either the A.P. or the F.P.**

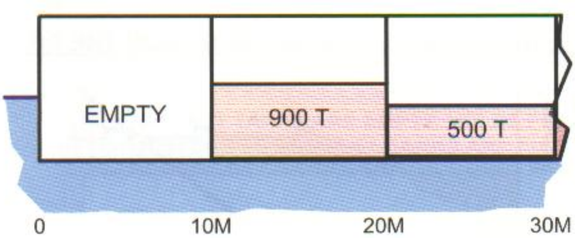
Shear forces are usually expressed in tonnes in keeping with normal practice in naval architecture.

BENDING MOMENT CALCULATIONS FOR A BOX-SHAPED HULL.

A floating ship's hull involves a complex interaction between the distributions of weight and its supporting buoyancy. Both these are usually expressed in tonnes per metre length. Carrying out a bending moment analysis for a real vessel by manual calculations would be quite an involved process but we can demonstrate the procedure by considering loading a box-shaped floating hull with a variety of uniform weight distributions over given sections of its length. This is often used as the basis for questions set in examinations for deck officers' certificates of competency. The next four pages show some worked examples which use the following method:-

1) Produce a ***Loading curve*** by plotting a scaled graph of the ***difference between weight and buoyancy distribution (expressed in tonnes / metre of length)*** along the length of the hull. The 'curve' should actually be a series of straight lines of constant loading over the different sections of the hull. The negative area of the curve will equal the positive area.
2) Produce the ***Shear Force curve*** by starting at one end of the hull and plotting the values of cumulative area ***(measured in tonnes)*** under the ***Loading curve*** at regular intervals along the hull length. This should be a series of straight lines over sections of constant loading, the slope of each section being determined by the value of loading in that section.
3) Produce the ***Bending Moment curve*** by again starting at the same end of the hull and plotting the values of cumulative area ***(measured in tonnes-metre)*** under the ***Shear Force curve*** at regular intervals along the hull length. This should only require calculating areas of different trapeziums, as the Shear Force curve is a series of straight lines. The 'best fit' smooth curve should then be drawn through the resulting points to indicate the value and position of **maximum bending moment, which will always occur where the shear force is passing through zero. Bending moments should be zero at the ends of the vessel, otherwise the load distribution would produce a net trimming moment**

PROCEDURE FOR CALCULATING BENDING MOMENTS FOR A BOX-SHAPED HULL



A 100 METRE LONG BARGE FLOATS AT EVEN KEEL WITH A LIGHTWEIGHT DISTRIBUTION OF 10 TONNES / METRE AND A TOTAL DISPLACED WEIGHT OF 7000 TONNES. CARGO IS LOADED AS SHOWN IN THE DIAGRAM OPPOSITE

LOADING CALCULATIONS

LOADING 0 - 10 M	= 70 - 10	= +60 T/M
LOADING 10 - 20 M	= 70 - (10 + 90)	= -30 T/M
LOADING 20 - 30 M	= 70 - (10 + 50)	= -10 T/M

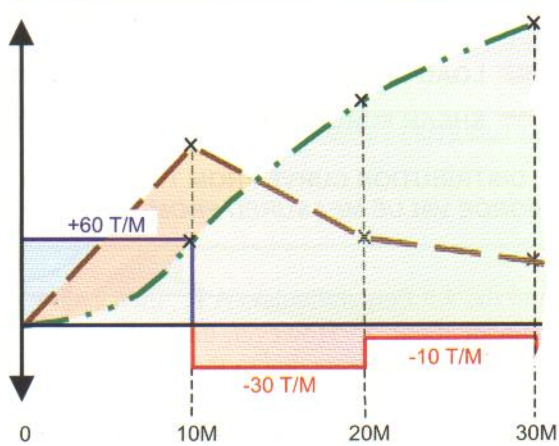
SHEAR FORCE CALCULATIONS

S.F. @ 10M	= 60 x 10	= +600 T
S.F. @ 20M	= 600 - 10 x 30	= +300 T
S.F. @ 30M	= 300 - 10 x 10	= +200 T

BENDING MOMENT CALCULATIONS

B.M. @ 10M	= 0.5(600 x 10)	= 3000 T-M
B.M. @ 20M	= 3000 + 10 x 450	= 7500 T-M
B.M. @ 30M	= 7500 + 10 x 250	= 10000 T-M

LOADING, S.F. AND B.M. CURVES, 0 TO 30 M

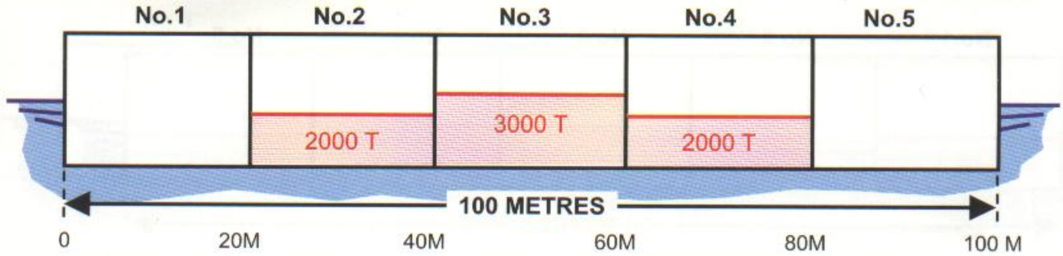


KEY

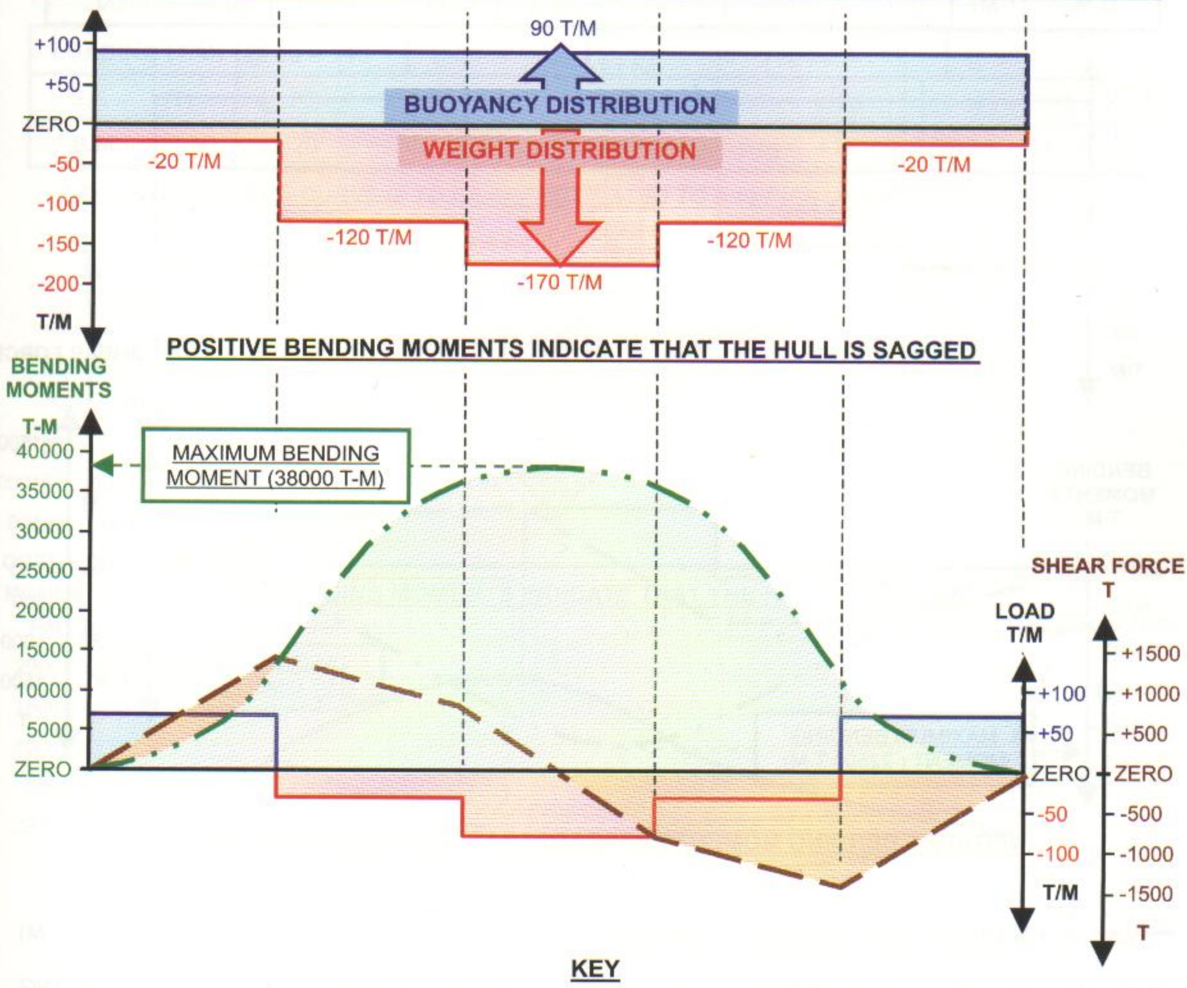
= LOADING (T/M) = SHEAR FORCE (T) = BENDING MOMENT (T-M)

(A) A BARGE LOADED AT EVEN KEEL WITH EMPTY END COMPARTMENTS

LIGHTSHIP WEIGHT OF 2000 T IS DISTRIBUTED UNIFORMLY OVER THE LENGTH AT 20T/M
CARGO WEIGHT OF 7000 T IS DISTRIBUTED AS SHOWN BY THE DIAGRAM BELOW



LOADING (T/M)	0-20 = +70	20-40 = -30	40-60 = -80	60-80 = -30	80-100 = +70
S.F. (T)	@ 20 = +1400	@ 40 = +800	@ 60 = -800	@ 80 = -1400	
B.M. (T-M)	@ 20 = +14000	@ 40 = +36000	@ 60 = +36000	@ 80 = +14000	



KEY

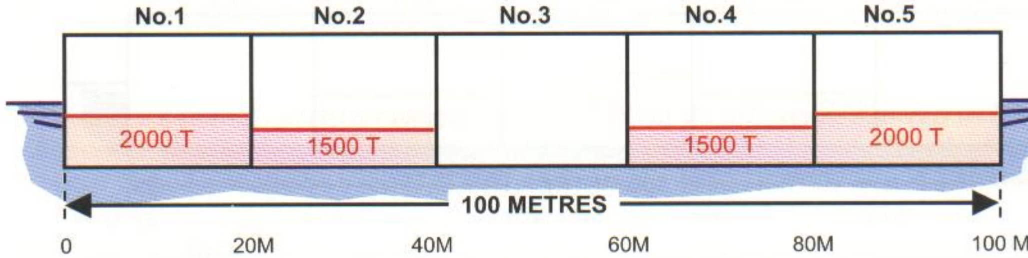
= LOADING (T/M) = SHEAR FORCE (T) = BENDING MOMENT (T-M)

LOADING IS THE DIFFERENCE BETWEEN BUOYANCY AND WEIGHT DISTRIBUTIONS
SHEAR FORCE (S.F.) IS THE CUMULATIVE AREA UNDER THE LOADING CURVE, FROM LEFT TO RIGHT
BENDING MOMENT IS THE CUMULATIVE AREA UNDER THE S.F. CURVE, FROM LEFT TO RIGHT

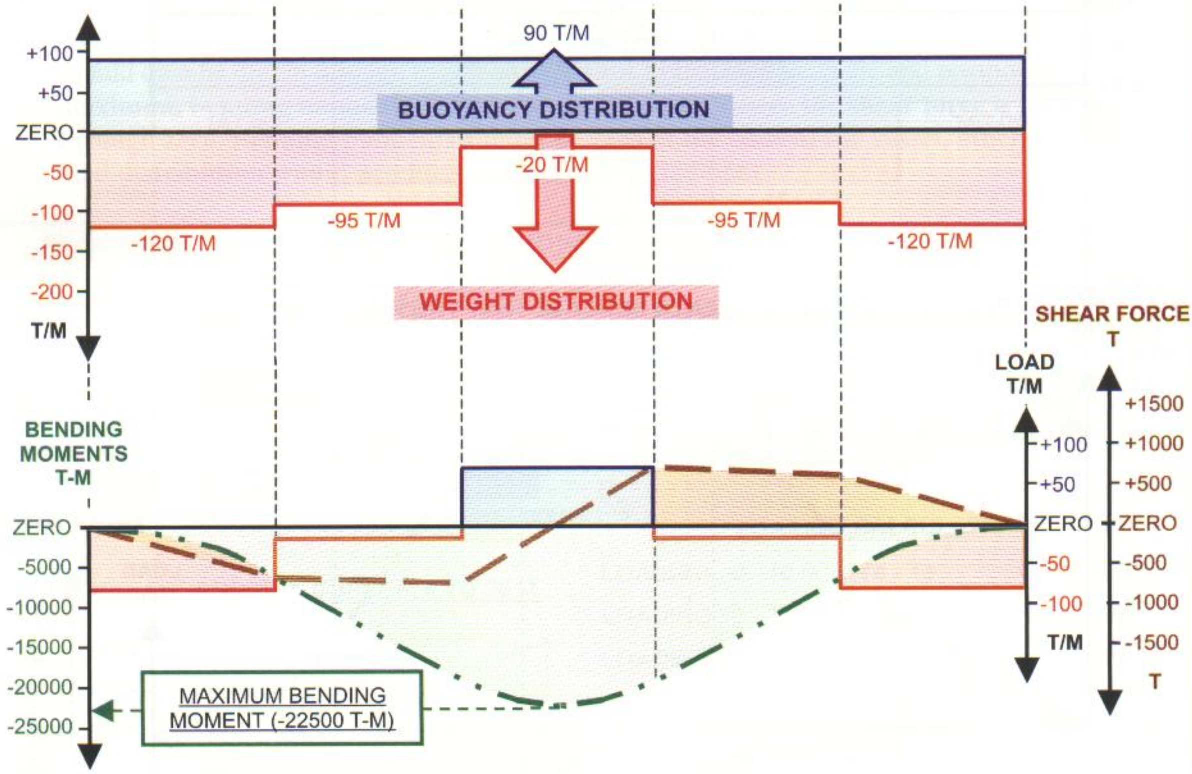
IF BUOYANCY IS POSITIVE AND WEIGHT IS NEGATIVE, ACCUMULATING AREAS UNDER THE LOADING AND SHEAR FORCE CURVES FROM THE SAME END, WILL PRODUCE POSITIVE SAGGING MOMENTS.

(B) A BARGE LOADED AT EVEN KEEL WITH EMPTY MIDSHIPS COMPARTMENT

LIGHTSHIP WEIGHT OF 2000 T IS DISTRIBUTED UNIFORMLY OVER THE LENGTH AT 20T/M
CARGO WEIGHT OF 7000 T IS DISTRIBUTED AS SHOWN BY THE DIAGRAM BELOW



LOADING (T/M)	0-20 = -30	20-40 = -5	40-60 = +70	60-80 = -5	80-100 = +30
S.F. (T)	@ 20 = -600	@ 40 = -700	@ 60 = +700	@ 80 = +600	
B.M. (T-M)	@ 20 = -6000	@ 40 = -19000	@ 60 = -19000	@ 80 = -6000	



NEGATIVE BENDING MOMENTS INDICATE THAT THE HULL IS HOGGED

KEY

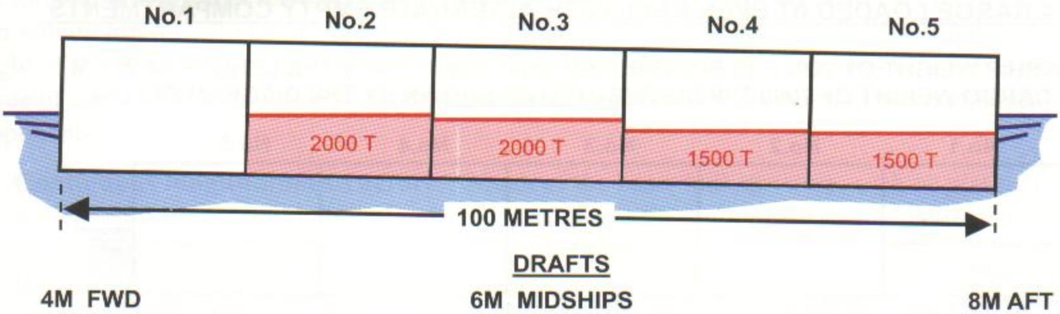
= LOADING (T/M) = SHEAR FORCE (T) = BENDING MOMENT (T-M)

IF BUOYANCY IS POSITIVE AND WEIGHT IS NEGATIVE, ACCUMULATING AREAS UNDER THE LOADING AND SHEAR FORCE CURVES FROM THE SAME END, WILL PRODUCE NEGATIVE HOGGING MOMENTS.

In both examples 'A' and 'B', the loading of the hull is symmetrical and so the maximum bending moment occurs exactly at the midships station. This is not the case, however, if we consider the next example where the hull is not at even keel. Here we must first calculate the buoyancy distribution at the bow and stern from the fore and aft draft ratio, in order to obtain the loading along the hull.

(C) A BARGE LOADED WITH A TRIM AND AN EMPTY END COMPARTMENT

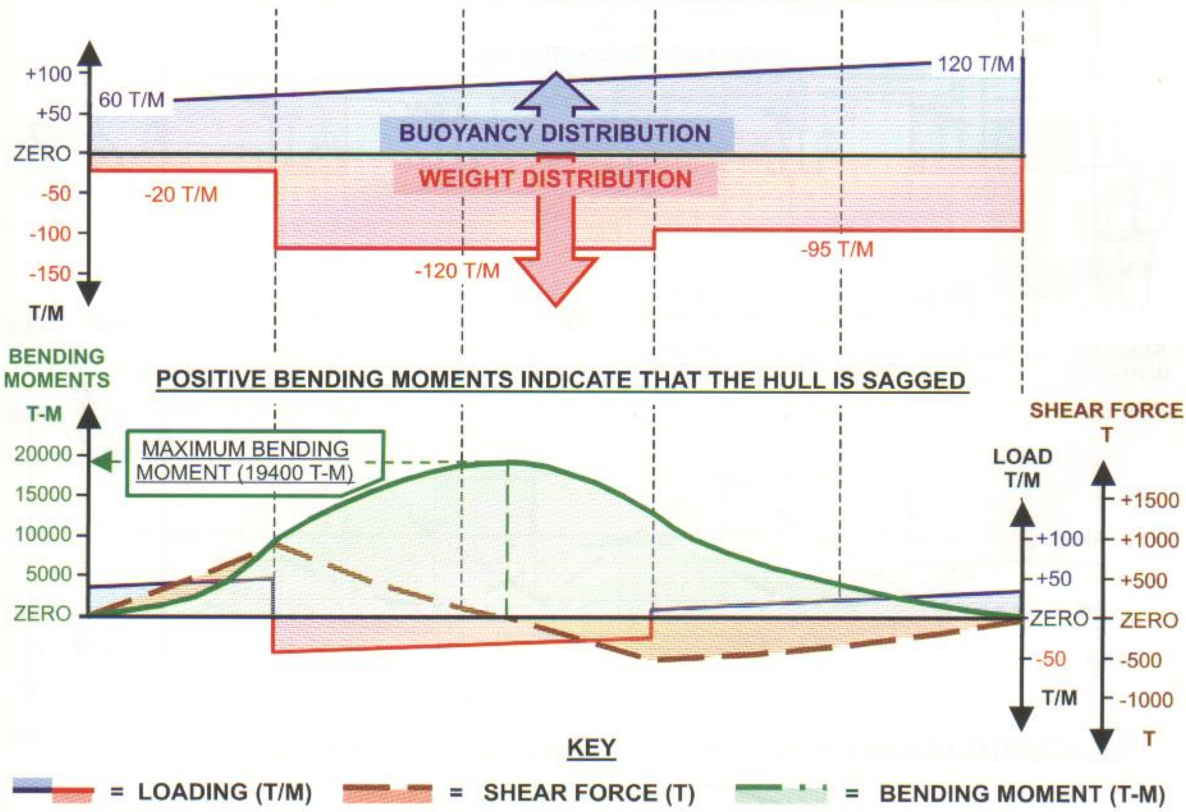
LIGHTSHIP WEIGHT OF 2000 T IS DISTRIBUTED UNIFORMLY OVER THE LENGTH AT 20T/M
CARGO WEIGHT OF 7000 T IS DISTRIBUTED AS SHOWN BY THE DIAGRAM BELOW



FOR A BOX-SHAPED HULL, THE BUOYANCY / METRE LENGTH AT ANY LONGITUDINAL STATION, WILL BE PROPORTIONAL TO THE DRAFT AT THAT POINT.

0	20M	40M	60M	80M	100 M
LOADING (T/M)	@ 10 = +46	@30 = -42	@ 50 = -30	@ 70 = +7	@ 90 = +19
S.F. (T)	@ 20 = +920	@ 40 = +80	@ 60 = -520	@ 80 = -380	
* B.M. (T-M)	@ 20 ≈ +9200	@ 40 ≈ +19200	@ 60 ≈ +13200	@ 80 ≈ +4200	

*** *THE SHEAR FORCE CURVE IS TO BE APPROXIMATED TO A SERIES OF STRAIGHT LINES***



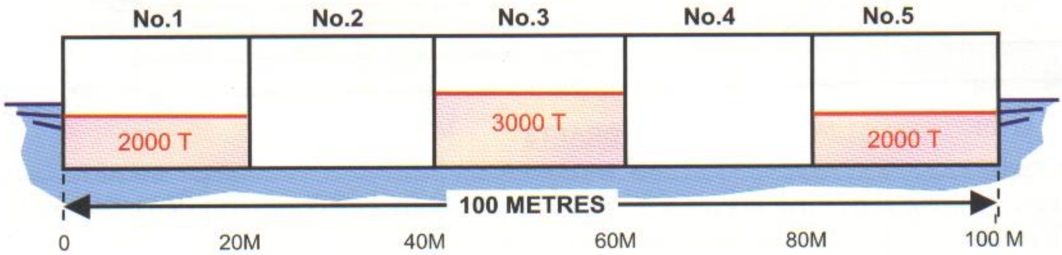
The example above shows that even with quite an extreme trim, the point of maximum bending moment still remains quite close to midships. Notice also that the resulting sagging moment is approximately half of the bending moment calculated for example 'A', in which a similar weight of cargo was loaded to put the barge on even keel by leaving both end compartments empty.

BENDING MOMENT CALCULATIONS FOR A BOX-SHAPED HULL.(Cont.)

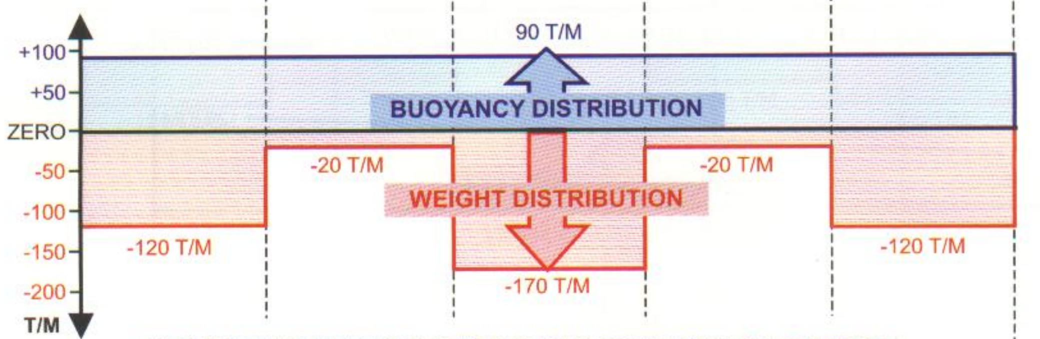
Alternate loaded and empty compartments can result in the bending moments reversing as we move along the hull. In this situation, peak values will occur at three points along the ship's length and though their magnitudes are relatively low, they are some distance from the midships region.

(D) A BARGE LOADED AT EVEN KEEL WITH ALTERNATE EMPTY COMPARTMENTS

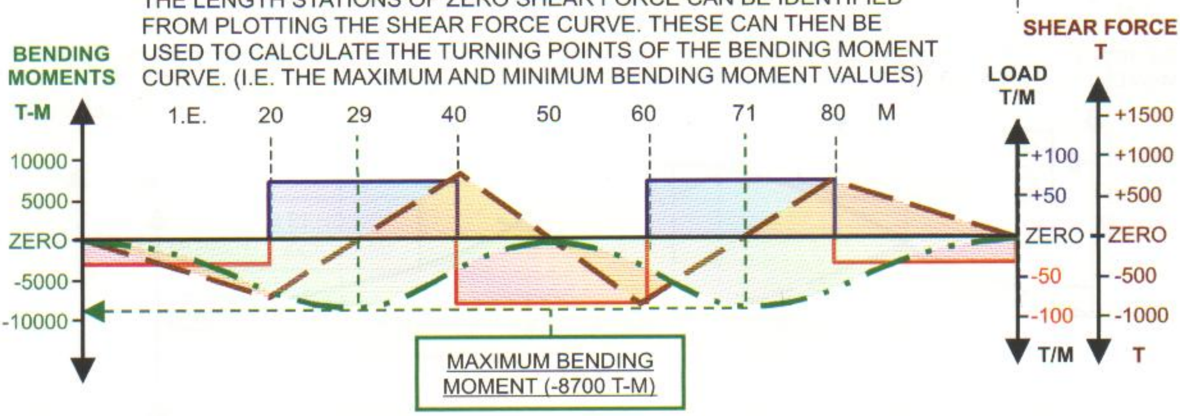
LIGHTSHIP WEIGHT OF 2000 T IS DISTRIBUTED UNIFORMLY OVER THE LENGTH AT 20T/M
CARGO WEIGHT OF 7000 T IS DISTRIBUTED AS SHOWN BY THE DIAGRAM BELOW



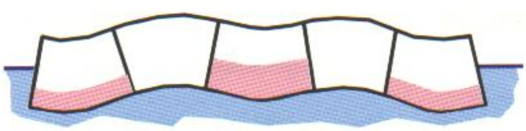
LOADING (T/M)	0-20 = -30	20-40 = +70	40-60 = -80	60-80 = +70	80-100 = -30
S.F. (T)	@ 20 = -600	@ 40 = +800	@ 60 = -800	@ 80 = +600	
B.M. (T-M)	@ 20 = -6000	@ 40 = -4300	@ 60 = -4000	@ 80 = -6000	



THE LENGTH STATIONS OF ZERO SHEAR FORCE CAN BE IDENTIFIED FROM PLOTTING THE SHEAR FORCE CURVE. THESE CAN THEN BE USED TO CALCULATE THE TURNING POINTS OF THE BENDING MOMENT CURVE. (I.E. THE MAXIMUM AND MINIMUM BENDING MOMENT VALUES)



THE NEGATIVE BENDING MOMENT INDICATES THAT THE HULL IS HOGGED IN TWO PLACES



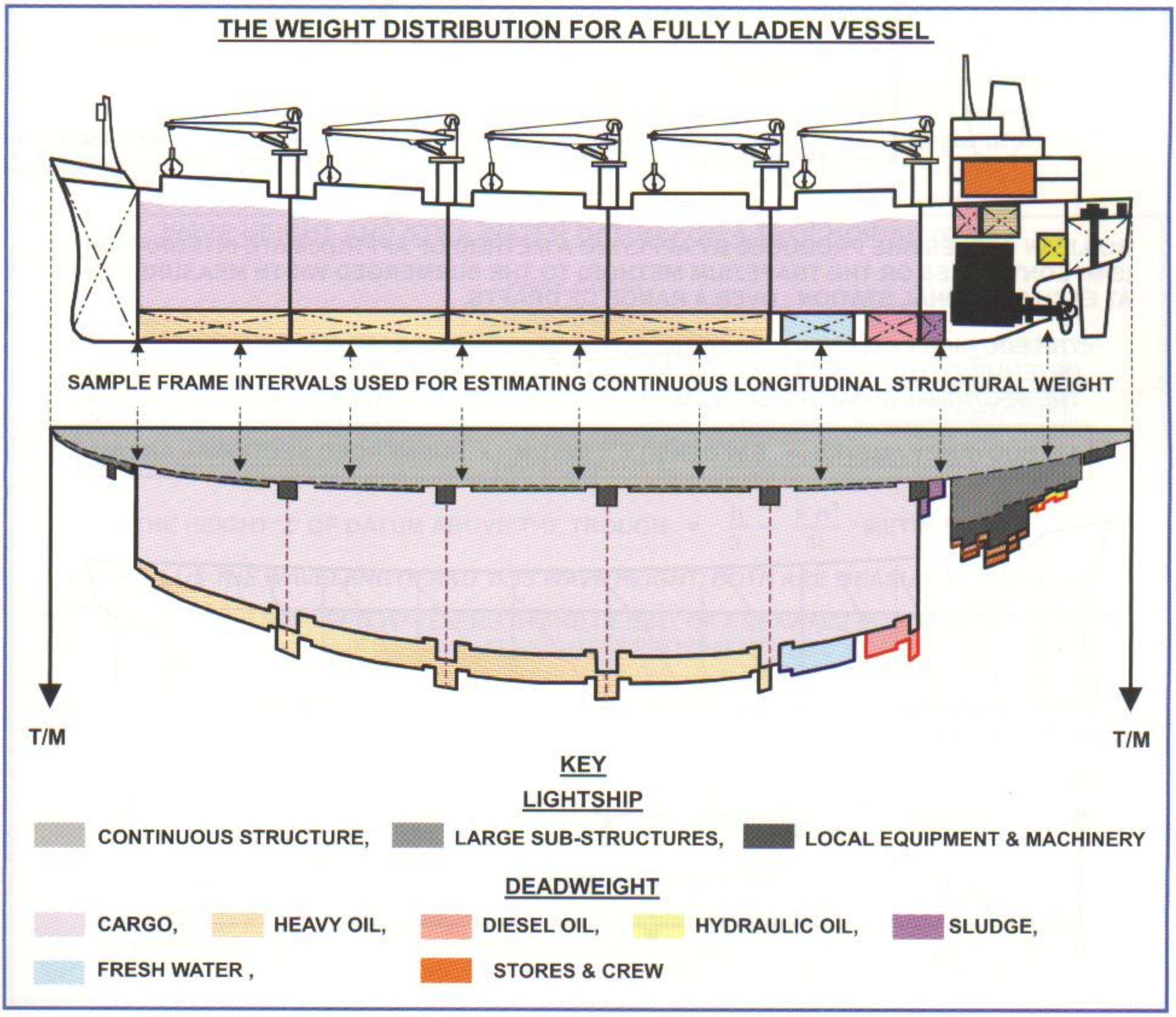
KEY

= LOADING (T/M) = SHEAR FORCE (T) = BENDING MOMENT (T-M)

THE WEIGHT DISTRIBUTION FOR A REAL SHIP

It is essential that a ship is built strong enough to withstand the bending moments that it is going to encounter when working within its designed operational conditions. Bending moment calculations, which require estimates of the weight and buoyancy distributions, must be carried out at the ship's design stage by the builder's naval architects. This will rely considerably upon estimates derived from data for similar ships that have been built previously. The deadweight distribution can be estimated from volumetric measurements of the cargo compartments, and of the fuel, ballast and water tanks. The weight of the lightship structure is more difficult and assessing this will invariably involve a certain degree of approximation, which is easier if the ship's structure is considered as three separate components.

1) **Continuous longitudinal structure.** Much of the basic hull structure, such as plating (including associated framing) and longitudinal beams, continue throughout the ship's length. The weight distribution is not constant but tends to only change gradually with the changing shape of the hull along its length. Weight distribution of this structure can be estimated by measuring the weight of one metre long transverse sections of the hull only, taken at suitable frame intervals, which should include changes of frame spacing that usually occur near to the bow and stern.
2) **Large substructures of significant length.** The weights of accommodation housing and superstructures should be calculated separately from the continuous hull structure, though the same approach of sample sections can be used.
3) **Single significant weights.** Weights of heavy items, such as the main engines, cargo handling equipment, masts etc. should be accounted for separately and spread over their particular lengths. These weights should include any local structural reinforcements, such as bed plates etc.

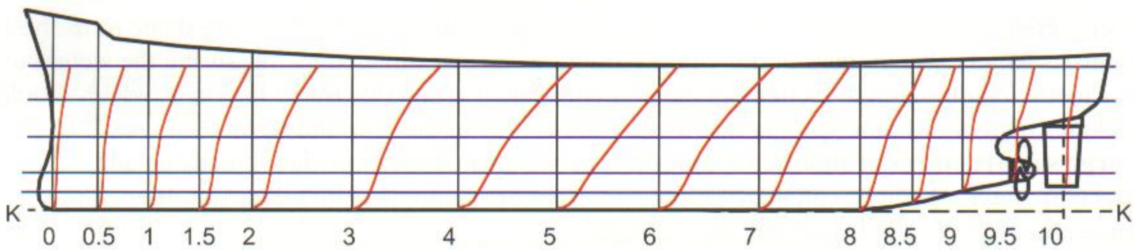


THE BUOYANCY DISTRIBUTION FOR A SHIP-SHAPED HULL

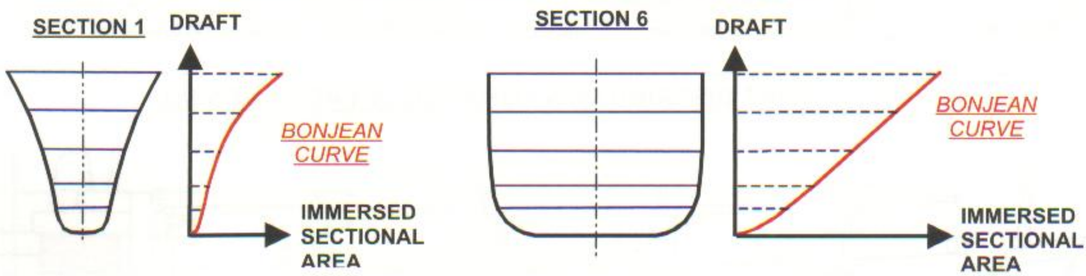
The longitudinal buoyancy distribution of a ship-shaped hull at different drafts and trim can be determined by superimposing the ship's waterline onto a profile of the vessel that includes curves of immersed sectional area /draft for every section station along the hull. These are known as ***Bonjean Curves*** and would be produced as part of the hullform analysis profile, described in Chapter 2.

Bonjean curves allow the calculation of buoyancy distribution for the ship floating in all situations, as any waterline, including wave profiles, can be superimposed onto the above diagram. This is important for determining the maximum bending moments due to wave action in a seaway.

LONGITUDINAL BUOYANCY DISTRIBUTION AND BONJEAN CURVES



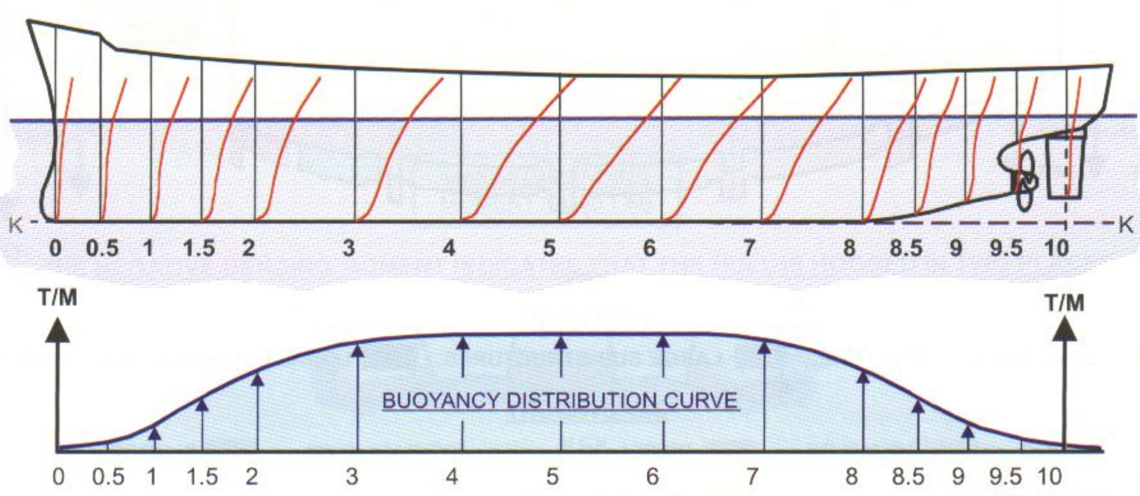
SAMPLE SECTIONS AND THEIR BONJEAN CURVES



BONJEAN CURVES ARE PRODUCED BY APPLYING A METHOD OF APPROXIMATE INTEGRATION (SIMPSON'S RULES OR THE TRAPEZIUM METHOD) TO THE SUBMERGED WIDTH MEASUREMENTS AT EACH SECTIONAL STATION , OVER A RANGE OF DRAFTS.

THE BUOYANCY DISTRIBUTION CURVE FOR A PARTICULAR DRAFT AND TRIM IS DETERMINED BY OVER-LAYING THE WATERLINE ONTO THE PROFILE AND READING OFF THE SECTIONAL SUBMERGED AREAS FROM THE BONJEAN CURVES.

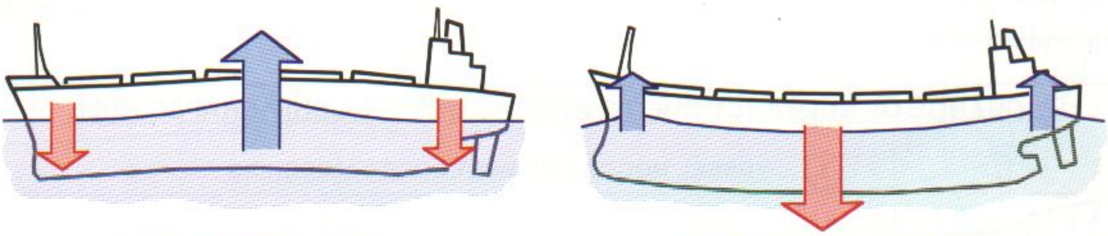
I.E. BUOYANCY / METRE = S.W DENSITY '1.025T/M^3' x SUBMERGED SECTIONAL AREA



BENDING MOMENTS DUE TO WAVE ACTION IN A SEAWAY

When a ship is steaming into head seas it may not ride easily over the waves as the wave length approaches the length of ship (See chapter 7, pages 169 and 170). This will result in the midships region being sometimes supported by a wave crest and then, at other times, being suspended over a wave trough as the wave profile moves along the ship from bow to stern. The changing buoyancy distribution produces a cycle of alternating hogging and sagging in the hull.

WAVE INDUCED BENDING MOMENTS



WAVE CREST MIDSHIPS

EXCESS BUOYANCY MIDSHIPS
HOGGING MOMENTS INCREASED

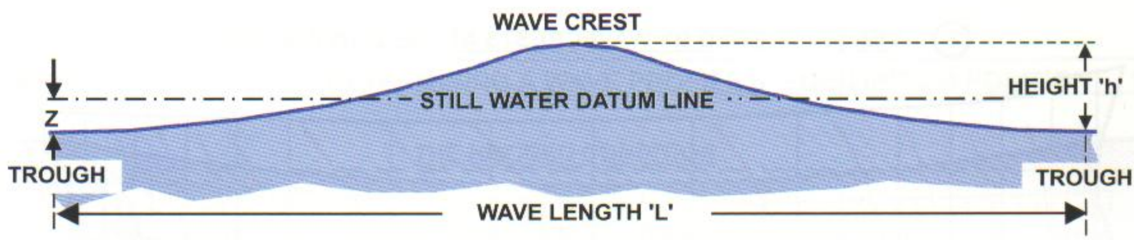
WAVE TROUGH MIDSHIPS

EXCESS BUOYANCY AT THE BOW AND STERN
SAGGING MOMENTS INCREASED

WHEN A VESSEL IS SUBJECTED TO HEAD OR STERN SEAS OF WAVELENGTHS SIMILAR TO THE SHIP'S OWN LENGTH, THEN THE PASSING OF WAVE CRESTS ALONG THE LENGTH OF THE HULL WILL INDUCE A CYCLE OF ALTERNATING HOGGING AND SAGGING. THIS WILL BE IN ADDITION TO ANY STILL WATER STRESSES DUE TO THE LOAD DISTRIBUTION

The buoyancy distribution of the above two situations can be determined by superimposing the wave profile onto the Bonjean diagram. In order to do this, we need to know the average water level of the wave, known as ***the still water datum line,*** and the height of the wave, relative to its length. Sea waves are approximately **trochoidal** in profile, which is characterised by long shallow troughs between sharper crests (See chapter 1, page 20).

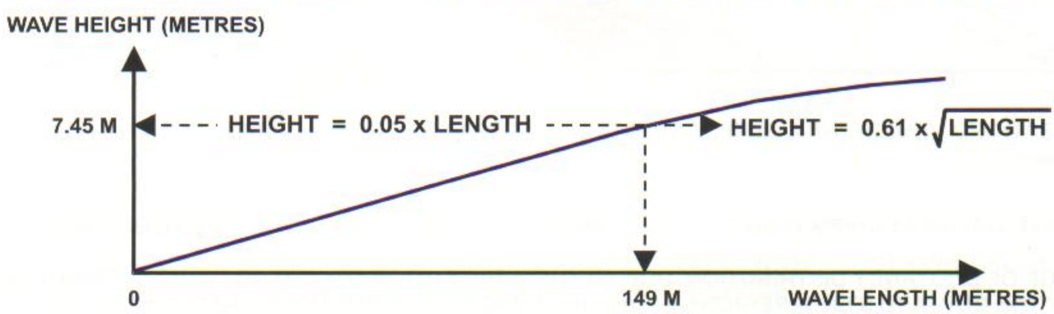
WAVE HEIGHT AND STILL WATER DATUM FOR A TROCHOIDAL WAVE



THE HEIGHT 'Z' OF DATUM ABOVE THE TROUGH $= \dfrac{h}{2} - \dfrac{\pi h^2}{4L}$ **METRES**

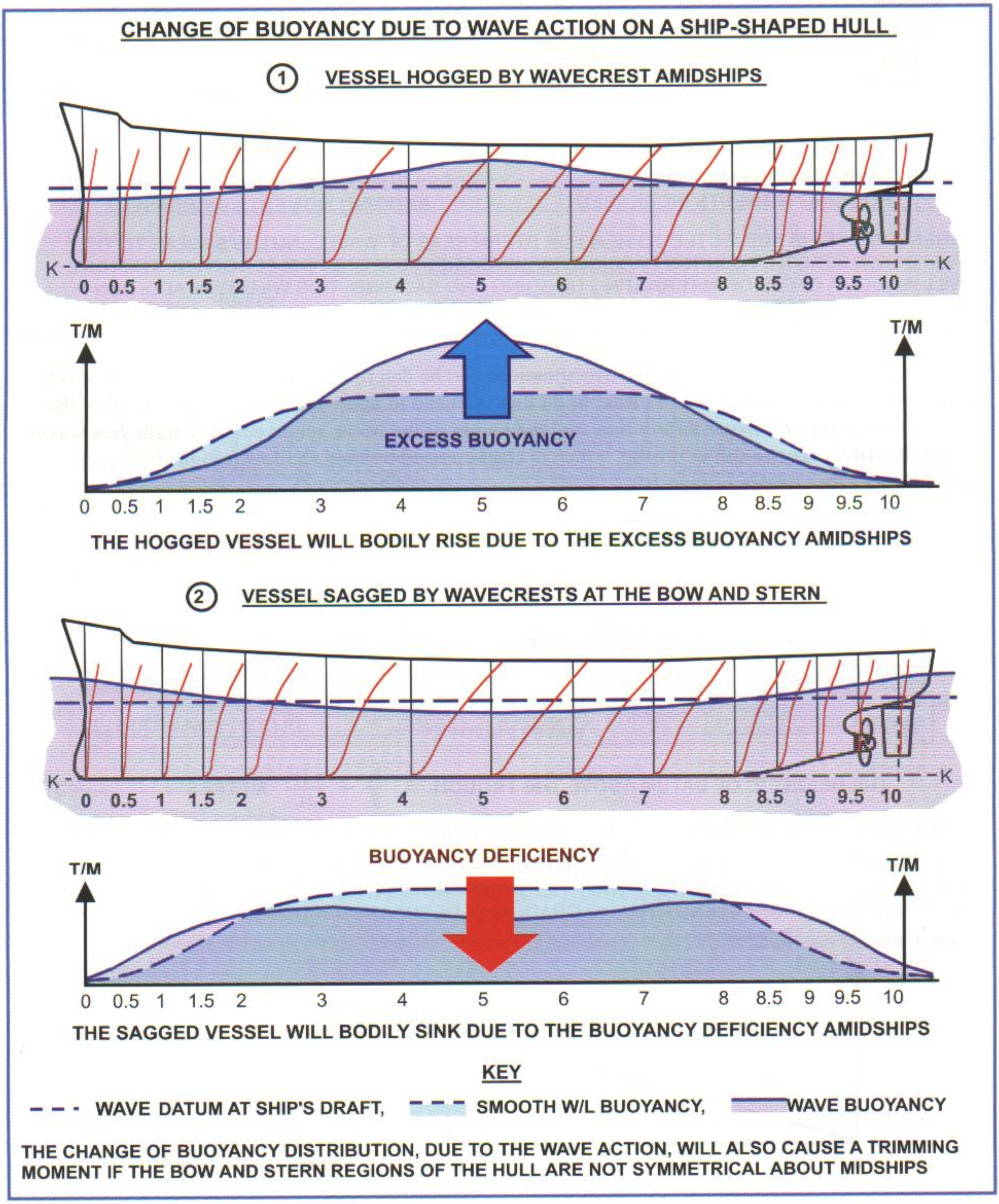
WHERE 'L' IS THE WAVELENGTH AND 'h' IS WAVE HEIGHT, BOTH ARE MEASURED IN METRES

THE RELATIONSHIP BETWEEN WAVELENGTH AND HEIGHT OF A SEAWAVE IS NOT SIMPLE. FOR WAVELENGTHS UP TO 150 METRES, WAVE HEIGHT APPROXIMATES TO 5% OF THE WAVELENGTH. HOWEVER, THE HEIGHT : LENGTH RATIO GRADUALLY DECREASES AT LONGER WAVELENGTHS



THE CHANGE OF BUOYANCY DISTRIBUTION DUE TO WAVE ACTION

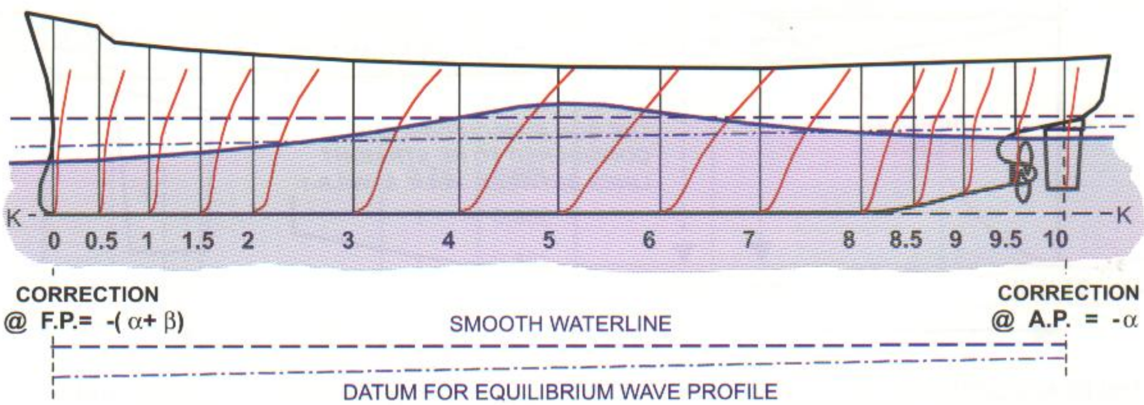
Wave induced bending moments are maximum when a ship is supported either at the ends or in the midships region by crests of waves which have a wavelength equal to that of the vessel's own waterline. If we superimpose such wave profiles onto the Bonjean curve diagram so that the wave datum line coincides with the ship's smooth waterline, then we will find that volumes of buoyancy transferred between the ends and midships regions of the hull, are not equal. The ship's hullform is much fuller in the midships region than it is at the bow and stern so when the vessel is hogged over the crest amidships, it will experience an excess of buoyancy that causes it to bodily rise. Conversely, when the bow and stern are supported by wavecrests, there is a deficiency of buoyancy amidships, so the ship will suffer bodily sinkage.



MUCKLE'S METHOD FOR FIXING A WAVE PROFILE WATERLINE

If the waterline of a ship-shaped hull changes from smooth water to that of being supported by a wave crest amidships, the trim alters and the hull bodily rises to reach a state of equilibrium in response to the change of buoyancy. We must place the wave profile at the equilibrium waterline in order to determine the buoyancy distribution and, hence, the maximum bending moments.

EQUILIBRIUM WATERLINE FOR A VESSEL HOGGED BY WAVECREST AMIDSHIPS



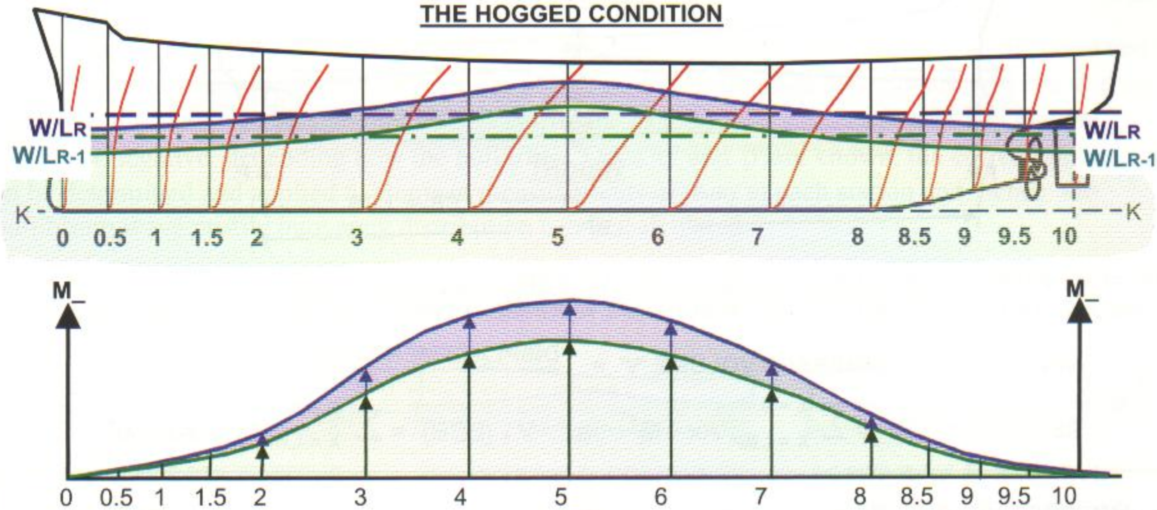
THE HULL HAS RISEN AND TRIMMED BY THE STERN, RELATIVE TO THE SMOOTH WATERLINE, TO REACH A STATE OF EQUILIBRIUM WHEN FLOATING TO THE WAVE PROFILE.

AFT DRAFT HAS DECREASED BY 'α' METRES WHILST FWD DRAFT DECREASE EQUALS ('α' + 'β')

Professor Muckle developed a procedure for determining the corrections 'α' and 'β' and, hence, fix the wave profile correctly for equilibrium. We must first overlay the wave profile onto the Bonjean curve diagram so that the wave datum coincides with the smooth waterline. This is the reference waterline, '**W/LR**'. Then, for the hogged situation above, we place a second identical waveform "**W/LR-1**, onto the diagram with its datum line set one metre below the reference waterline. The equilibrium datum will lie predominately between the datums of the two wave profiles, which are close enough together for the short intervening parts of the Bonjean curves to be considered as straight lines.

MUCKLE'S METHOD OF DETERMINING A WAVE PROFILE'S EQUILIBRIUM POSITION - 1

THE HOGGED CONDITION



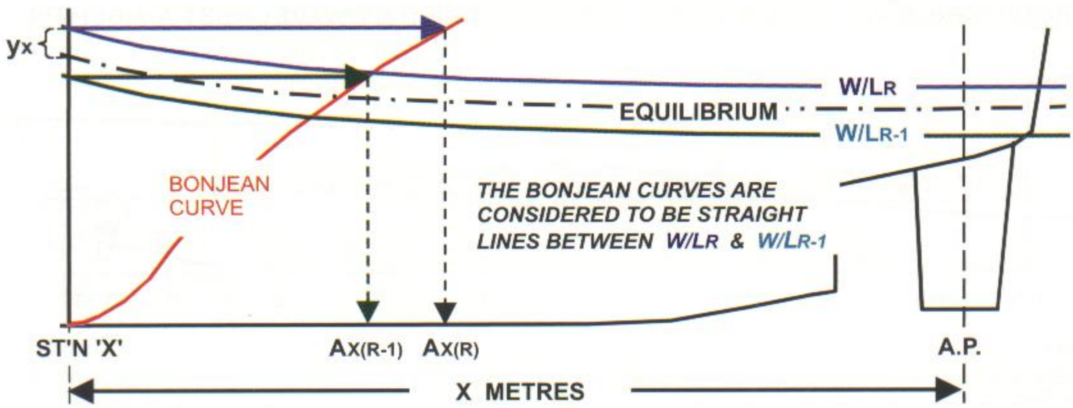
KEY

SUBMERGED SECTIONAL AREAS FOR WAVE PROFILE 'W/LR' WITH SMOOTH WATERLINE DATUM
SUBMERGED SECTIONAL AREAS FOR IDENTICAL WAVEFORM 'W/LR-1', 1 METRE BELOW 'W/LR'

MUCKLE'S METHOD FOR FIXING A WAVE PROFILE WATERLINE (Cont.)

The equilibrium wave profile should produce an immersed sectional area between the values given by the intersections of the plotted lines **'W/LR'** and **'W/LR-1'** with the Bonjean curves at each station.

MUCKLE'S METHOD OF DETERMINING A WAVE PROFILE'S EQUILIBRIUM POSITION - 2

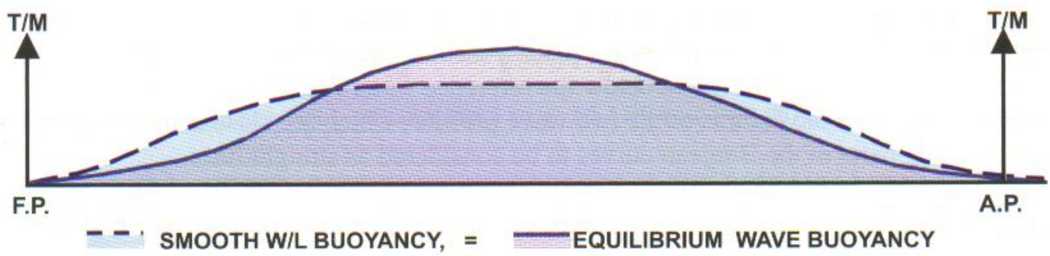


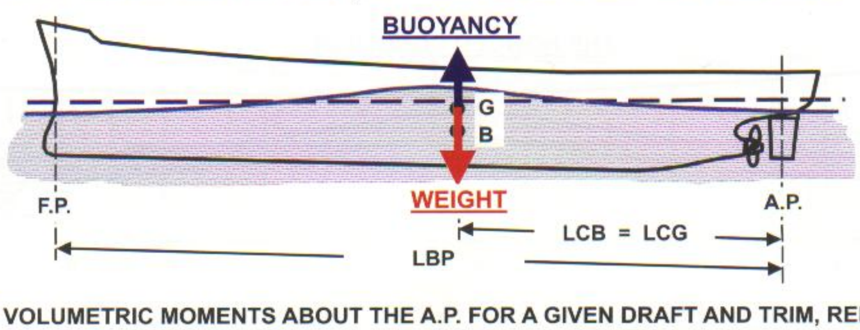
THE DRAFT CORRECTION ' yx' IS TO BE APPLIED TO ' AX(R) ', THE SUBMERGED SECTIONAL AREA, AT STATION 'X' FOR THE WAVE PROFILE , 'W/LR', BASED UPON THE SMOOTH WATERLINE

$$y_x = -\left(\alpha + \beta \frac{X}{LBP}\right) \quad \text{METRES}$$

$$\text{So CORRECTED SECTIONAL AREA } 'A_X' = A_{X(R)} - \left(\alpha + \beta \frac{X}{LBP}\right)\left[A_{X(R)} - A_{X(R-1)}\right] \quad M^2$$

WE MUST FIND TWO SIMULTANEOUS EQUATIONS TO DETERMINE THE VALUES OF THE CONSTANTS 'α' AND 'β'. __THESE ARE GIVEN BY THE FOLLOWING CONDITIONS__ :-





THE SUM OF VOLUMETRIC MOMENTS ABOUT THE A.P. FOR A GIVEN DRAFT AND TRIM, REMAINS CONSTANT AS THE LCB AND LCG ARE IN VERTICAL ALIGNMENT WHEN THE HULL IS IN EQUILIBRIUM

$$\text{Now} \quad \text{IMMERSED VOLUME } 'V' = \frac{\text{DISPLACEMENT WEIGHT } '\Delta_T'}{1.025} \quad M^3$$

$$\text{So} \quad V = \sum_{X=LBP}^{X=0} A_X \quad M^3 \quad \& \quad V \times (LCB) = \sum_{X=LBP}^{X=0} 'X' \times A_X \quad M^4$$

$$\text{So IMMERSED VOLUME } 'V' = C.I.\left\{\sum A_{(R)} - \alpha \sum\left[A_{(R)} - A_{(R-1)}\right] - \frac{\beta}{LBP}\sum X\left[A_{(R)} - A_{(R-1)}\right]\right\} \quad M^3$$

$$\text{And MOMENTS } 'V' \times LCB = C.I.\left\{\sum X A_{(R)} - \alpha \sum X\left[A_{(R)} - A_{(R-1)}\right] - \frac{\beta}{LBP}\sum X^2\left[A_{(R)} - A_{(R-1)}\right]\right\} \quad M^4$$

MUCKLE'S METHOD FOR FIXING A WAVE PROFILE WATERLINE (Cont.)

The values of 'V' and 'LCB' are known as they remain unchanged between the smooth water and equilibrium wave profile waterline so the two equations, shown on the previous page, can be solved be using one of the methods of approximation integration (such as the Trapezium method, shown below) to sum up the separate terms $A_{(R)}$, $[A_{(R)} - A_{(R-1)}]$, $X\,[A_{(R)} - A_{(R-1)}]$, $X\,A_{(R)}$, and $X^2\,[A_{(R)} - A_{(R-1)}]$ for the transverse sections at the Common Interval 'C.I.' of 0.1L along the ship's length.

DETERMINING THE VOLUMETRIC TERMS

STN	$A_{(R)}$	$A_{(R-1)}$	$[A_{(R)} - A_{(R-1)}]$	'M'*	X	$M[A_{(R)}]$	$M[A_{(R)} - A_{(R-1)}]$	$MX[A_{(R)} - A_{(R-1)}]$
0				0.25	L			
0.5				0.5	0.95L			
1				0.75	0.90L			
2				1	0.85L			
3				1	0.80L			
10				0.25	0			
**M = MULTIPLIER*				SUMS OF TERMS		$\sum[A_{(R)}]$	$\sum[A_{(R)} - A_{(R-1)}]$	$\sum X[A_{(R)} - A_{(R-1)}]$

DETERMINING THE VOLUMETRIC MOMENTS TERMS

STN	$A_{(R)}$	$A_{(R-1)}$	$[A_{(R)} - A_{(R-1)}]$	'M'*	X	$MX[A_{(R)}]$	$MX[A_{(R)} - A_{(R-1)}]$	$MX^2[A_{(R)} - A_{(R-1)}]$
0				0.25	L			
0.5				0.5	0.95L			
1				0.75	0.90L			
2				1	0.85L			
3				1	0.80L			
10				0.25	0			
**M = MULTIPLIER*				SUMS OF TERMS		$\sum x[A_{(R)}]$	$\sum x[A_{(R)} - A_{(R-1)}]$	$\sum x^2[A_{(R)} - A_{(R-1)}]$

The summation terms are substituted in the two equations at the top of the page and so the constants 'α' and 'β' are determined. **'W/LR'** and **'W/LR-1'** are placed on the assumption that the hull will rise and trim slightly by the stern but this may not be necessarily the case and the values of 'α' and 'β' may be positive or negative, depending how the hull trims. Once 'α' and 'β' are known, the correction factor $\mathbf{y_x}$ can be determined and applied to Bonjean measurements of $\mathbf{A_{(R)}}$ at each station and, hence, the equilibrium wave profile buoyancy distribution can be calculated.

DIFFERENT COMBINATIONS OF THE CONSTANTS 'α' & 'β'

'α' & 'β' ARE BOTH POSITIVE
EQUILIBRIUM DRAFT IS LESS THAN 'W/LR' ALONG THE ENTIRE LENGTH OF THE HULL

NEGATIVE 'α' & POSITIVE 'β'
STERN TRIM IS SUFFICIENT TO INCREASE THE EQUILIBRIUM DRAFT AT THE STERN

POSITIVE 'α' & NEGATIVE 'β'
THE HOGGED EQUILIBRIUM WATERLINE PRODUCES A HEAD TRIM

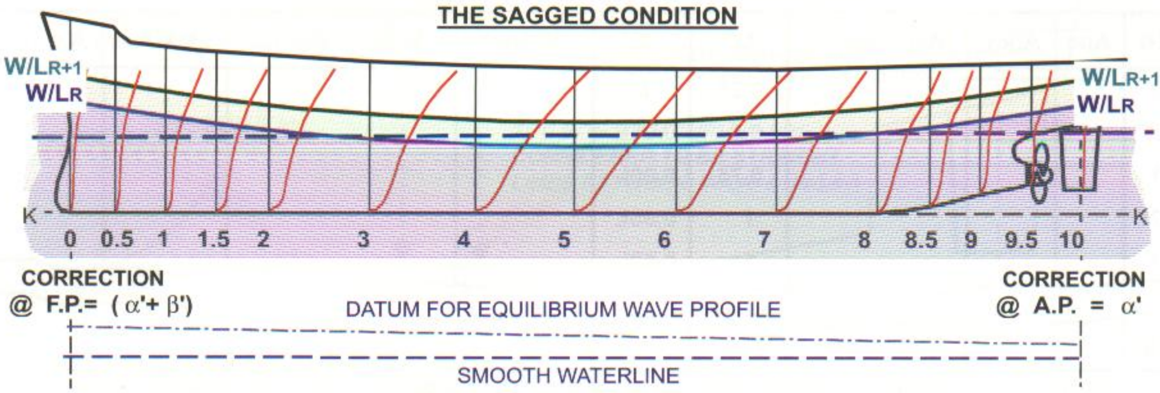
- - - - = REFERENCE WAVE PROFILE 'W/LR' **= EQUILIBRIUM WAVE PROFILE**

MUCKLE'S METHOD FOR FIXING A WAVE PROFILE WATERLINE (Cont.)

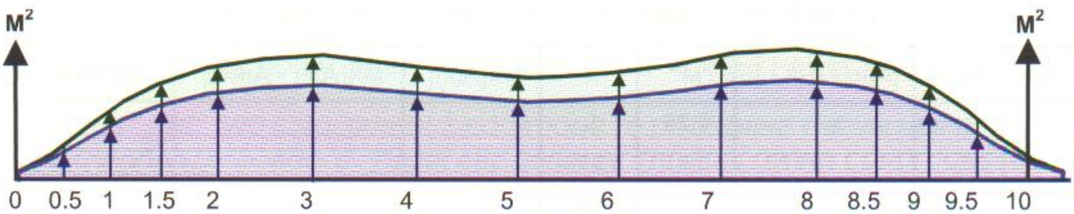
Muckle's process for determining the hogged equilibrium wave profile is used for the opposite effect when the midships region is suspended over a wave trough. The reference waterline **'W/LR'** is again superimposed onto the Bonjean so that its datum line coincides with the smooth waterline but, in this situation, the second wave profile **'W/LR+1'** is positioned one metre above.

MUCKLE'S METHOD OF DETERMINING A WAVE PROFILE'S EQUILIBRIUM POSITION - 3

THE SAGGED CONDITION



THE HULL HAS FALLEN AND TRIMMED BY THE HEAD, RELATIVE TO THE SMOOTH WATERLINE, TO REACH A STATE OF EQUILIBRIUM WHEN FLOATING TO THE WAVE PROFILE.



KEY

SUBMERGED SECTIONAL AREAS FOR WAVE PROFILE 'W/LR' WITH SMOOTH WATERLINE DATUM

SUBMERGED SECTIONAL AREAS FOR IDENTICAL WAVEFORM 'W/LR+1', 1 METRE ABOVE 'W/LR'

AFT DRAFT HAS INCREASED BY α' METRES WHILST FWD DRAFT INCREASE EQUALS (α' + β')

$$\textbf{STATION DRAFT CORRECTION 'yx'} = \left(\alpha' + \beta' \frac{X}{LBP}\right) \quad \textbf{METRES}$$

$$\textbf{So CORRECTED SECTIONAL AREA 'Ax'} = A_{X(R)} + \left(\alpha' + \beta' \frac{X}{LBP}\right)\left[A_{X(R)} - A_{X(R-1)}\right] \quad M^4$$

The two correction factors α' and β' will be different from the factors involved in the hogged condition. The simultaneous equations needed to find the values of these constants α' and β' are:-

$$\textbf{TOTAL IMMERSED VOLUME 'v'} = C.I.\left\{\sum A_{(R)} + \alpha \sum\left[A_{(R)} - A_{(R-1)}\right] + \frac{\beta}{LBP}\sum X\left[A_{(R)} - A_{(R-1)}\right]\right\} \quad M^3$$

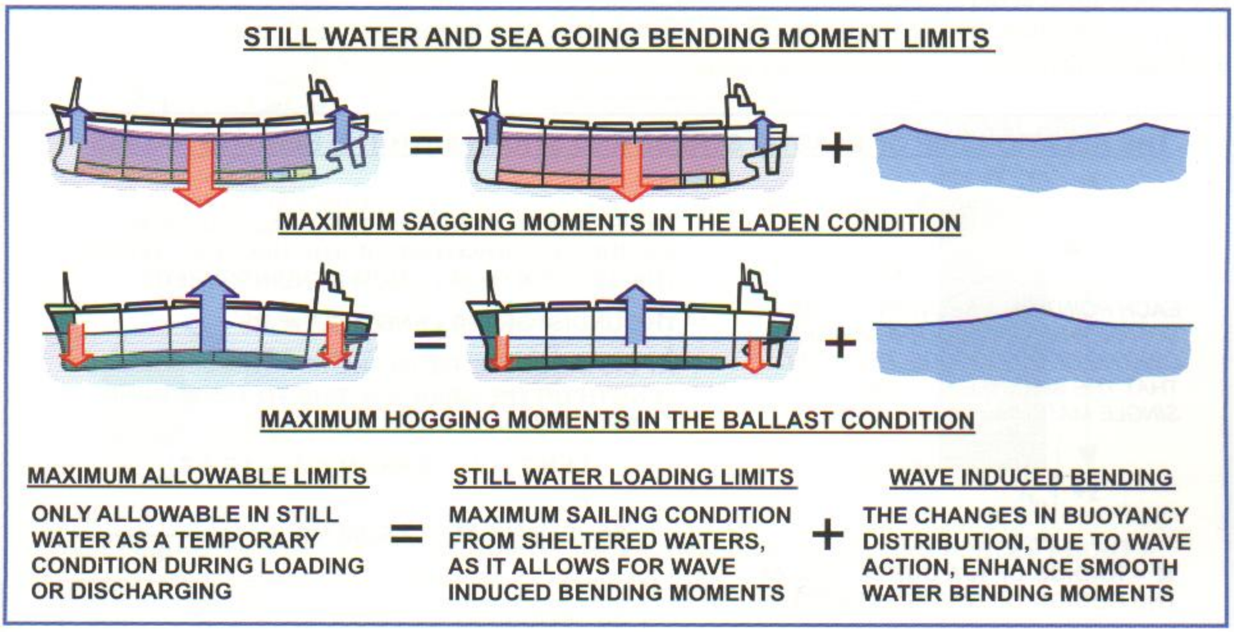
$$\textbf{VOLUME MOMENTS 'V' x LCB} = C.I.\left\{\sum X A_{(R)} + \alpha \sum X\left[A_{(R)} - A_{(R-1)}\right] + \frac{\beta}{LBP}\sum X^2\left[A_{(R)} - A_{(R-1)}\right]\right\} \quad M^4$$

Professor Muckle's method of fixing wave profile waterlines correctly at equilibrium may appear quite complex but this is because it involves processing a considerable amount of data. The actual principles involved are simple but, as with all the procedures involving hull analysis, the calculations must be carried out in a well ordered and disciplined way. This type of number processing is greatly facilitated by computerisation.

Once the equilibrium wave profile is found for the situations of maximum hog and sag, where the length of the wave coincides with the ship's waterline length, the respective buoyancy distributions can be calculated. This allows the design team to estimate the maximum hogging and sagging moments that a hull is likely to suffer in a seaway at any loaded condition.

A VESSEL'S BENDING MOMENT LIMITS.

Ships are designed to meet requirements that usually include its principal dimensions of length, draft and beam with the capability to carry a particular payload of cargo, fuel and stores. It is the job of the shipbuilder's naval architects to ensure that the vessel's structure is strong enough to withstand the bending moments caused by the load distributions of its normal operating range. The ship's builders should supply strength data that includes limits for the maximum hogging and sagging moments that the ship can be subjected to. Situations involving the greatest bending moments should be analysed to give this information. If we consider the bulk carrier shown on page 179 then the departure loaded condition with full cargo, fuel and water, would produce the greatest sagging moments that the hull can be expected to incur when it is subjected to the sagging half of the wave induced bending cycle. Conversely, the arrival ballast condition, after 90% consumption of fuel and water, will suffer the maximum hogging moments when subjected to the hogging half of the wave induced bending cycle. These maximum sagging and hogging moments will be the limits to which the ship should be subjected and the ship's officers must ensure that the weight distribution for any condition does not result in their being exceeded. On board bending moment calculations, however, are based upon smooth water conditions so stated sea going limits must be reduced by the degree of wave-induced bending that the Muckle's analysis indicates. During port loading and discharging operations, a ship can be subjected to the full bending moment limits provided that this is a temporary condition and that the bending moments are reduced to sea going limits before the ship sails into open waters.



Bending moment considerations have been a requirement for tankers and bulk carriers for a long time and, as a consequence of this, ship builders supply cargo procedures for such vessels that detail allowable loading/discharging sequences. Calculating machines have also been developed to assist any onboard calculations that would have to be carried out if a particular loading operation is not included in these procedures. These are now generally in the form of software programs for standard digital computers, but older vessels may still have the dedicated analogue computer, such as the 'Loadicator', with its separate dials representing each cargo and fuel compartment. (Even older systems relied upon mechanical calculating aids). Whatever system is provided with a vessel, it is important to ensure that bending moment limits are not exceeded at any time, by testing loading or discharge sequences, step by step, prior to carrying out the planned cargo operations.

The master or chief officer should be wary of any loaded state that is calculated to be very close to the maximum bending moments as the calculation's accuracy depends upon the accuracy of the weight distribution estimate. This is particularly so as bending moment calculations cannot be easily confirmed by onboard measurements or observations. This is unlike stability estimates, where the rolling and heeling behaviour of a vessel at least gives some indication of their accuracy.

BENDING STRESSES AND SECTIONAL MOMENT OF INERTIA.

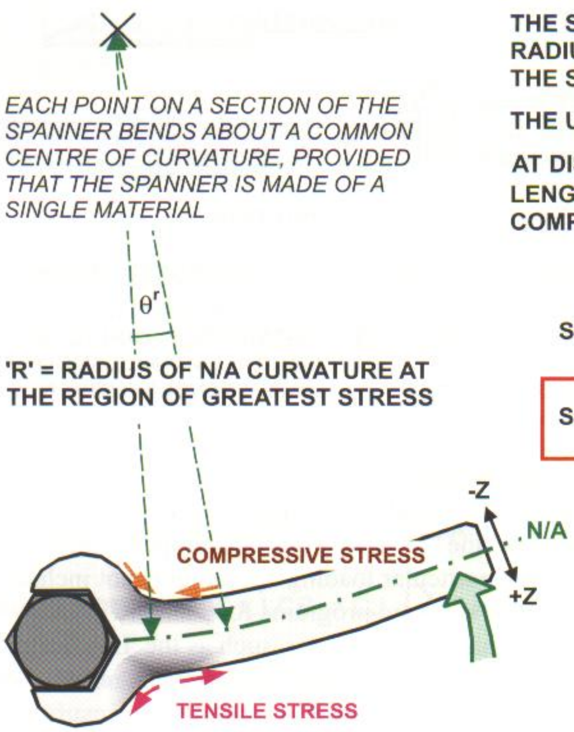
If we consider the ***elastic*** behaviour of a spring, then its change in length, either by being stretched or compressed, will be ***directly proportional*** to the force applied to it. However, the amount of distortion and the limiting elastic load will also depend upon the spring's material and its size. This is equally true when we consider the elastic behaviour of the steel spanner. We need a universal measure of a material's elastic properties that can be applied to structures of any dimensions. This is known as ***'Young's Modulus of Elasticity'***, or ***'E'*** and is defined as follows:-

> **YOUNG'S MODULUS OF ELASTICITY 'E' FOR A MATERIAL** = $\dfrac{\text{STRESS (N/M}^2\text{)}}{\text{STRAIN (M/M)}}$ **N/M^2 or MN/M^2**
>
> ***WHERE 'STRESS' IS THE FORCE (IN NEWTONS), ACTING UPON EACH SQUARE METRE OF SECTIONAL AREA OF THE STRUCTURE AND 'STRAIN' IS THE EXTENSION OR COMPRESSION OF THE STRUCTURE FOR EVERY METRE OF ITS UNDISTURBED LENGTH***

Notice that we have now precisely defined the term ***'Stress' as Force per unit area*** and that the ***force must be measured in the correct units of ' Newtons', 'KiloNewtons', or 'MegaNewtons'***. Each material has its own unique modulus value and a maximum stress for staying within its elastic limits, which must not be exceeded anywhere in a structure if it is not to be weakened or fail.

If we look again at the spanner under load, (shown on page 178), and assume that it is bending within its elastic limit, then we can relate the stress distribution to the 'tightness' of bending and the elastic modulus of its material, in the following way.

THE RELATIONSHIP OF ELASTIC STRESS AND STRAIN WITH THE BENDING RADIUS



THE SPANNER BENDS UNDER LOAD AND 'R' IS THE RADIUS OF CURVATURE OF THE NEUTRAL AXIS AT THE SECTION OF MAXIMUM BENDING WHERE

THE UNDISTORTED LENGTH 'L' = $R\theta^r$

AT DISTANCE '$\pm Z$' FROM THE N/A, THE DEFORMED LENGTH OF THE SPANNER, DUE TO TENSION OR COMPRESSION, IS GIVEN BY:-

$$\text{LENGTH AT } \pm Z \text{ FROM N/A} = (R \pm Z)\,\theta^r$$

$$\text{So STRAIN AT } +Z \text{ FROM N/A} = \frac{(R \pm Z)\,\theta^r - R\theta^r}{R\theta^r}$$

$$\boxed{\text{So STRAIN AT } \pm Z \text{ FROM N/A} = \pm \frac{Z}{R} \quad \text{M/M}}$$

STRAIN INCREASES IN PROPORTIONALLY WITH DISTANCE 'Z' FROM THE NEUTRAL AXIS AND WITH DECREASING BENDING RADIUS 'R'. I.E A TIGHTER CURVATURE OF BENDING INCREASES STRAIN

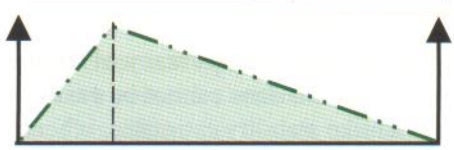
IF THE BEAM IS WITHIN ITS ELASTIC LIMIT

STRESS = 'E' x STRAIN

WHERE 'E' IS YOUNG'S ELASTIC MODULUS FOR THE MATERIAL OF THE SPANNER

$$\boxed{\text{STRESS AT } \pm Z \text{ FROM N/A} = \pm \text{'E'} \times \frac{Z}{R} \quad \text{N/M}^2}$$

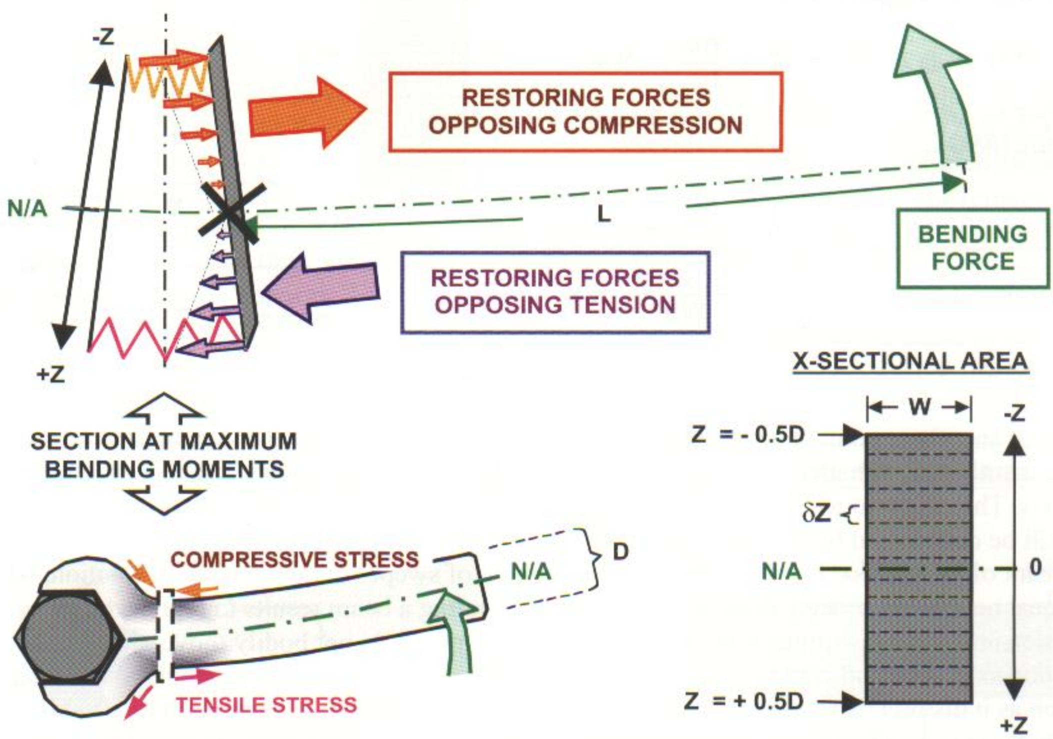
BENDING MOMENTS (NEWTON-METRES)



BENDING STRESSES AND SECTIONAL MOMENT OF INERTIA. (Cont.)

The forces that produce pure bending moments are acting perpendicular to a beam's neutral axis so there is no net compression or tension acting along the beam's length. The overall length and volume of the beam remains unchanged. **Consequently, the centre of cross sectional area must lie on the Neutral Axis as the material volume decrease due to compression on one side of it, equals the volume increase caused by tension to the other side.** The neutral axis of an asymmetrical section can be located by taking moments of area about a convenient axis, perpendicular to the loading and, hence, parallel to the neutral axis itself.
When a beam bends under load, it reaches a point of equilibrium where the turning moments of the restoring forces generated by compression and tension are balanced by the bending moment.

ELASTIC STRESS DISTRIBUTION AND BENDING MOMENTS



THE BENDING MOMENT 'BM' = THE SUM OF THE MOMENTS OF RESTORING FORCES ABOUT X

Where THE FORCE @ 'Z' = STRESS @ Z 'Sz' x SECTIONAL AREA STRIP 'WδZ'

So $$BM = \sum_{Z=-0.5D}^{Z=+0.5D} S_Z \times W\delta Z \times Z$$

But STRESS @ Z 'Sz' = $\frac{E}{R} \times Z$ *FOR THE BEAM TO BEND ELASTICALLY (PAGE 195)*

WHERE 'E' IS THE MODULE OF ELASTICITY AND 'R' IS THE BENDING RADIUS FOR THE N/A

So $$BM = \frac{E}{R}\sum_{Z=-0.5D}^{Z=+0.5D} Z^2 \times W\delta Z$$

Now $\sum_{Z=-0.5D}^{Z=+0.5D} Z^2 \times W\delta Z$ = *THE SECOND MOMENT OF THE SECTION AREA ABOUT THE N/A*

THIS IS KNOWN ALSO AS THE 'MOMENT OF INERTIA 'I_{NA}' OF THE MATERIAL SECTIONAL AREA

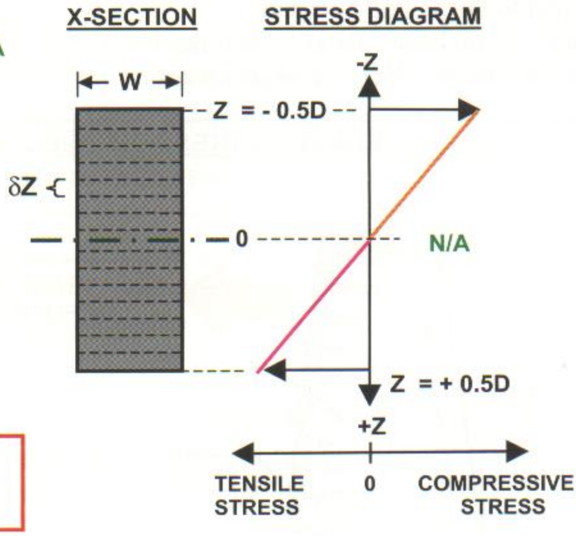
So $BM = \frac{E}{R} \times I_{NA}$ But $\frac{E}{R} = \frac{\text{STRESS @ Z}}{Z}$ *(SEE PAGE 195)*

So STRESS AT DISTANCE Z FROM THE N/A $\quad 'S_Z' = Z \times \frac{\text{BENDING MOMENT 'BM'}}{\text{MOMENT OF INERTIA '}I_{NA}\text{'}}$ N/M^2

BENDING STRESSES AND SECTIONAL MOMENT OF INERTIA. (Cont.)

The bending stresses of tension and compression that act upon the sectional material can be plotted across any section of a beam under load by using the equation derived on the previous page. In the case of the spanner, the sectional area is usually a simple narrow rectangle.

STRESS DIAGRAM FOR THE RECTANGULAR SECTION OF THE SPANNER



$$\text{STRESS @ Z 'Sz'} = Z \times \frac{BM}{I_{NA}} \quad N/M^2$$

THE MOMENT OF INERTIA OF A REGULAR RECTANGULAR SECTION IS GIVEN AS{-

$$I_{NA} = \frac{WD^3}{12} \quad M^4$$

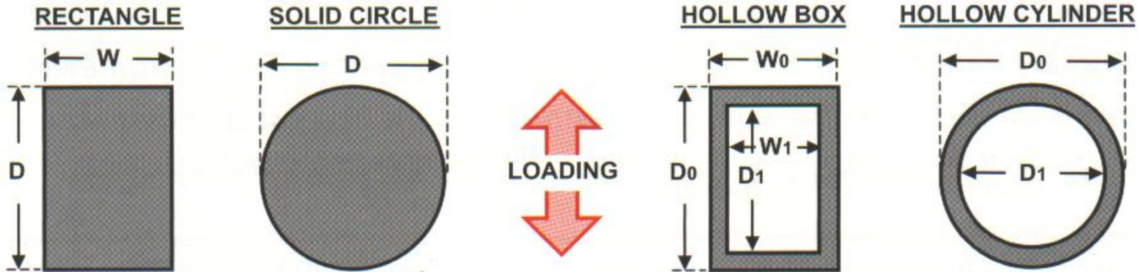
$$\boxed{\text{So STRESS @ Z 'Sz'} = Z \times \frac{12 \times BM}{WD^3} \quad N/M^2}$$

The stress at any given point on the spanner depends only upon the Bending Moment 'BM' at that point, the distribution of material in the cross sectional area and the distance of that point from the neutral axis. The extent to which a beam actually distorts for a given stress distribution, i.e. its bending radius, will be determined by the material's modulus of elasticity.

The Moment of Inertia 'I_{NA}' for the section, is the moment of swept volume / radian of rotation, when the sectional material is rotated about the neutral axis. Bending a beam results in regions of compression and tension. Although the molecules of the material do not bodily move, the volume changes due to tension and compression balance out, so an effective transfer of volume occurs across the section as it distorts by rotating in response to the bending moment. If the section 'I_{NA}' value is large, then there is a considerable amount of material distributed at some distance from the neutral axis. Restoring forces produced in such locations will have considerable leverage to oppose the bending moment hence the level of stress required to reach equilibrium will be lower than for sections with smaller moments of inertia value.

We have already encountered 'Moments of Inertia' for a sectional area, as a measure of resistance to volume transfer by rotation of the section, when regarding a ship's heeling and pitching behaviour (see pages 32 and 131). Elastic bending of a beam is simply another example of such a situation.

EQUATIONS FOR THE MOMENT OF INERTIA OF BASIC CROSS SECTIONAL AREAS



RECTANGLE: $I_{NA} = \frac{WD^3}{12}$

SOLID CIRCLE: $I_{NA} = \frac{\pi D^4}{64}$

HOLLOW BOX: $I_{NA} = \frac{1}{12}(W_0 D_0^3 - W_1 D_1^3)$

HOLLOW CYLINDER: $I_{NA} = \frac{\pi}{64}(D_0^4 - D_1^4)$

MOMENTS OF INERTIA FOR COMPEX GIRDER SECTIONS

Beams are most effectively reinforced against bending moments, by adding additional material as far away as possible from the neutral axis and hence, producing a large I_{NA}' value for their weight. This results in the standard '**T**' and '**H**' girders used in constructional engineering

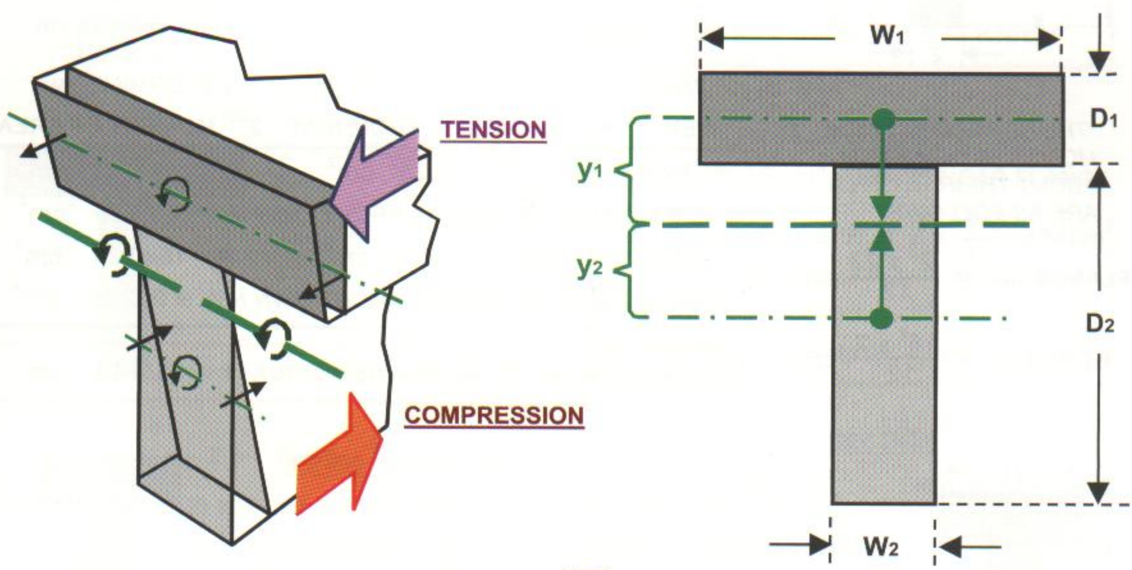
TYPICAL GIRDER SECTIONS

THE BEAM IS REINFORCED BY ADDING MATERIAL ON THE EDGES WHERE STRESS IS GREATEST

The sectional shapes shown above consist of more than one element, i.e. they are built up from an upright body with horizontal flanges added to the bottom and top edges. When such a complex girder is distorted by bending moments, there is a change of compression or tension across each element of the girder as well as the overall gradient of stress across the entire section. **The total Moment of Inertia for the whole section is the sum of each individual component's moment of inertia about its own neutral axis plus their second moment of area about the common neutral axis for the entire section.**

This is applying the ***Principle of Parallel Axes*** to the separate components of the girder. This states that the second moment of an area about any axis parallel to its neutral axis is equal to the moment about the neutral axis plus the second moment of the section's centre of area about the parallel axis.

APPLYING THE PRINCIPLE OF PARALLEL AXES TO A COMPLEX GIRDER SECTION



KEY

COMPONENT CENTRE OF AREA AND ITS INDIVIDUAL NEUTRAL AXIS

COMMON NEUTRAL AXIS FOR ENTIRE SECTIONAL AREA

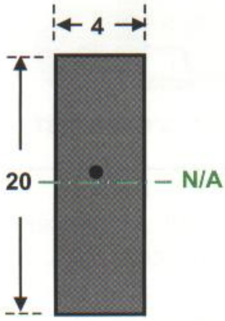
THE SECTIONAL AREA OF EACH COMPONENT PART OF THE GIRDER, ROTATES BOTH ABOUT ITS NEUTRAL AXIS AND THE COMMON CENTRE OF AREA FOR THE ENTIRE SECTION, SO:-

$$\text{MOMENT OF INERTIA FOR ENTIRE SECTION} = \frac{W_1 D_1^3}{12} + (W_1 D_1)\, y_1^2 + \frac{W_2 D_2^3}{12} + (W_2 D_2)\, y_2^2$$

COMPARING STRENGTH FOR DIFFERENT GIRDER SECTIONS

We can see the effectiveness of the reinforcement flanges on a beam, by comparing the Moment of Inertia of a simple rectangular beam to those of a 'T' beam and an 'I' beam with the same sectional area and depth of girder. In order to suit typical girder dimensions, we will use the units of centimetres and centimetre^2 for length and area.

MOMENTS OF INERTIA OF DIFFERENT GIRDER SECTIONS



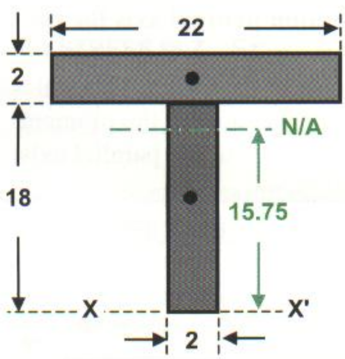
RECTANGULAR SECTION - 800 cm^2

THE SECTION HAS VERTICAL SYMMETRY, SO THE N/A IS AT MID HEIGHT
WE CAN USE THE STANDARD FORMULA FOR I_{NA} OF A RECTANGLE

I.E $\quad I_{NA} = \dfrac{WD^3}{12}$ cm^4

So $\quad I_{NA} = \dfrac{4 \times 20^3}{12}$ cm^4

So $\quad I_{NA} =$ 2666.67 cm^4 FOR A '▮' SECTION



'T' SECTION - 800 cm^2

THE SECTION HAS VERTICAL ASYMMETRY, SO THE N/A MUST BE LOCATED BY TAKING AREA MOMENTS ABOUT THE XX'

ITEM	AREA		LEVER FROM XX'		1ST MOMENT OF AREA
FLANGE	44 cm^2	x	19 cm	=	836 cm^3
BODY	36 cm^2	x	9 cm	=	324 cm^3
TOTAL	80 cm^2				1260 cm^3

So DISTANCE XX' FROM N/A $= \dfrac{1260}{80} = 15.75$ cm

TAKING THE 2ND MOMENTS OF AREA TO DETERMINE '$I_{N/A}$'

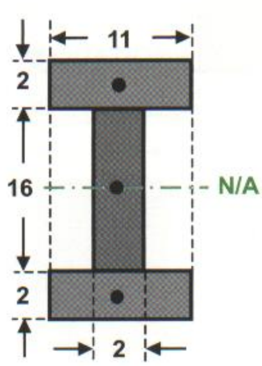
ITEM	AREA		(LEVER FROM N/A)^2		2ND MOMENT OF AREA
FLANGE	44 cm^2	x	3.25^2 cm^2	=	464.75 cm^4
ITEM MOMENT OF INERTIA ABOUT OWN N/A					*+ 14.25* cm^4
BODY	36 cm^2	x	6.75^2 cm^2	=	1640.25 cm^4
ITEM MOMENT OF INERTIA ABOUT OWN N/A					*+ 972.00* cm^4

TOTAL MOMENT OF INERTIA ABOUT N/A = 3151.25 cm^4

**THE SEPARATE ITEMS' MOMENTS OF INERTIA ABOUT THEIR OWN N/A, ARE AS FOLLOWS*

FLANGE $I_{NA'} = \dfrac{2^3 \times 22}{12}$ cm^4

BODY $I_{NA'} = \dfrac{18^3 \times 2}{12}$ cm^4



' I ' SECTION - 800 cm^2

THE SECTION HAS VERTICAL SYMMETRY, SO THE N/A IS AT MID-HEIGHT'
TAKING THE 2ND MOMENTS OF AREA TO DETERMINE '$I_{N/A}$'

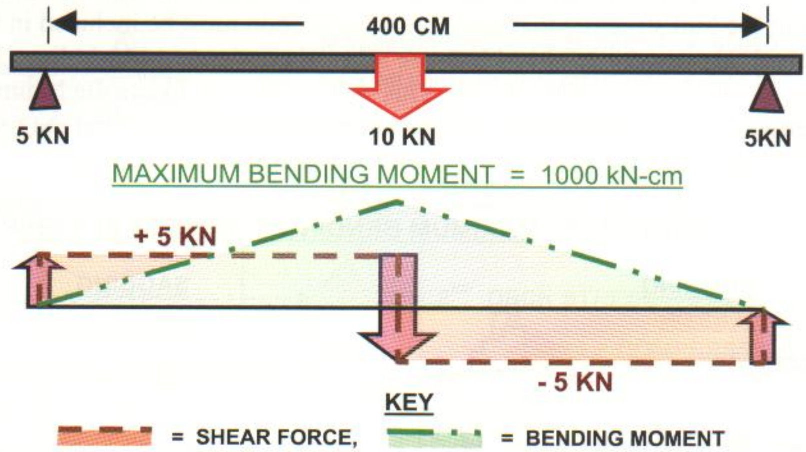
ITEM	AREA		(LEVER FROM N/A)^2		2ND MOMENT OF AREA
FLANGES (x2)	44 cm^2	x	9^2 cm^2	=	3564.00 cm^4
ITEM MOMENT OF INERTIA ABOUT OWN N/A					*+ 14.25* cm^4
BODY	32 cm^2	x	ZERO cm^2	=	ZERO cm^4
ITEM MOMENT OF INERTIA ABOUT OWN N/A					*+ 692.67* cm^4

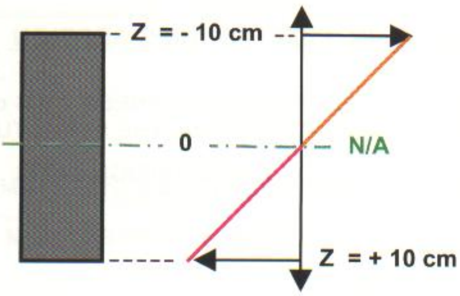
TOTAL MOMENT OF INERTIA ABOUT N/A = 4270.93 cm^4

COMPARING STRENGTH FOR DIFFERENT GIRDER SECTIONS (Cont.)

If girders of each of the three sections shown on the previous page are subjected to the same load and bending moments, we can compare the resulting stress diagrams to find their relative strengths. Lengths will be expressed in centimetres whilst the force of weight will be expressed in KiloNewtons.
Maximum strength is given by the greatest I_{NA} value, combined with a mid-height neutral axis

STRESS DIAGRAMS FOR DIFFERENT GIRDER SECTIONS UNDER THE SAME LOAD



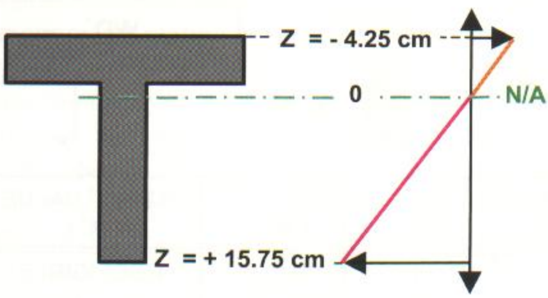


RECTANGULAR SECTION

I_{NA} = 2666.67 cm^4, N/A IS CENTRAL

Now STRESS @ 'Z' = $Z \times \frac{BM}{I_{NA}}$ KN/cm^2

So STRESS @ +10cm = +3.75 KN/cm^2
And STRESS @ -10cm = -3.75 KN/cm^2

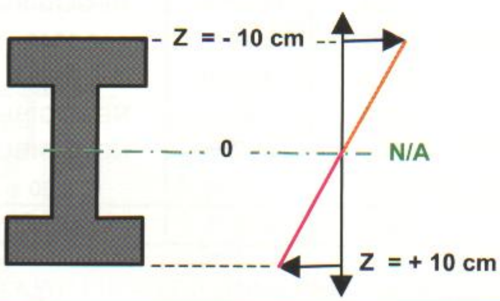


'T' SECTION

I_{NA} = 3151.25 cm^4, N/A IS NEAR FLANGE

Now STRESS @ 'Z' = $Z \times \frac{BM}{I_{NA}}$ KN/cm^2

So STRESS @ +15.75cm = +1.35 KN/cm^2
And STRESS @ -4.25cm = - 5.00 KN/cm^2



'I' SECTION

I_{NA} = 4270.93 cm^4, N/A IS CENTRAL

Now STRESS @ 'Z' = $Z \times \frac{BM}{I_{NA}}$ KN/cm^2

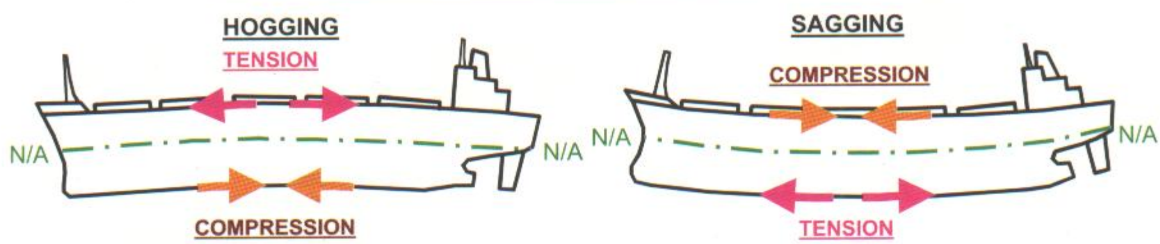
So STRESS @ +10cm = +2.34 KN/cm^2
And STRESS @ -10cm = - 2.34 KN/cm^2

THE WIDE 'T' GIRDER IS THE WEAKEST OF THE THREE, BECAUSE ALTHOUGH ADDING THE FLANGE INCREASES ITS MOMENT OF INERTIA RELATIVE TO THE RECTANGULAR SECTION, THE NEUTRAL AXIS IS MOVED UPWARDS SO MUCH THAT THE BOTTOM EDGE IS BEARING MOST OF THE STRESS.
EFFECTIVE FLANGES SHOULD BE MUCH NARROWER THAN THE DEPTH OF THE GIRDER BODY

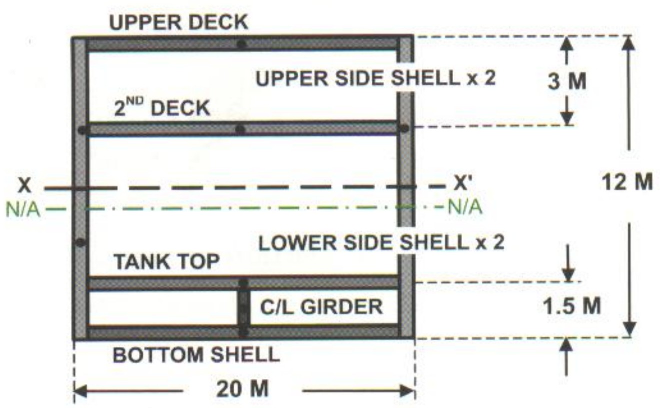
BENDING STRESS CALCULATIONS FOR SHIPS

A ship is a complex box girder in which the midships region of the upper deck and ship's bottom are subjected to the greatest stress when the hull bends under hogging and sagging moments.
The hull must be able to withstand the stresses created by the maximum bending moment limits that the ship has been designed to operate within. This requires determining the Moments of Inertia for transverse sections of the hull at significant points along its length, particularly in the midships region. The process is carried out by using basically the same method as is shown on page 199 for the complex girder sections, though there will be considerably more components to account for. All the longitudinal strength members of the hull structure that pass through a section must be included in the calculations. Their first and second moments of sectional area are taken about the mid-height line (XX') plus the moments of inertia about their own neutral axes. This will allow us to locate the Neutral Axis and determine the moment of Inertia about XX', which can then be corrected to find the true **'INA'** value by using the principle of parallel axes.

DISTRIBUTION OF MAXIMUM BENDING STRESSES IN A SHIP



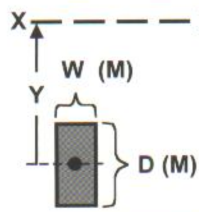
MOMENT OF INERTIA CALCULATION FOR A SIMPLIFIED MIDSHIPS SECTION



STEEL PLATE THICKNESS = 1.6 cm THROUGHOUT ALL THE STRUCTURE

XX' IS THE MID-HEIGHT AXIS
N/A IS THE NEUTRAL AXIS

'Y' INDICATES DISTANCES FROM XX'



$$'I' = \frac{WD^3}{12}\ cm^2\,M^2$$

ITEM	DIMENSIONS (M)	AREA 'A' (M^2)	LEVER XX' 'Y' (M)	A x Y (M^3)	A x Y^2 (M^4)	ITEM 'I' VALUE (M^4)
TANK TOP	20.0 x 0.016	0.320	- 4.50	- 1.440	6.4800	NEGLIGIBLE
BOTTOM SHELL	20.0 x 0.016	0.320	- 6.00	- 1.920	11.5200	NEGLIGIBLE
C/L GIRDER	1.5 x 0.016	0.024	- 5.25	- 0.126	0.6615	0.0045
L'R SIDE SHELL	12.0 x 0.016	0.192	- 3.00	- 0.576	1.7280	0.2880 x 2
UPPER DECK	20.0 x 0.016	0.320	+6.00	+ 1.920	11.5200	NEGLIGIBLE
2ND DECK	20.0 x 0.016	0.320	+3.00	+ 0.960	2.8800	NEGLIGIBLE
U'R SIDE SHELL	12.0 x 0.016	0.192	+3.00	+ 0.576	1.7280	0.2880 x 2
	TOTALS	1.688		- 0.606	36.5170	1.1565

VERTICAL DISTANCE FROM XX' TO N/A = $\frac{-0.606}{1.688}$ M So **THE N/A IS 0.36 M BELOW XX'**

SECTION MOMENT OF INERTIA ABOUT XX' = 36.5170 + 1.1565 = 37.6735 M^4

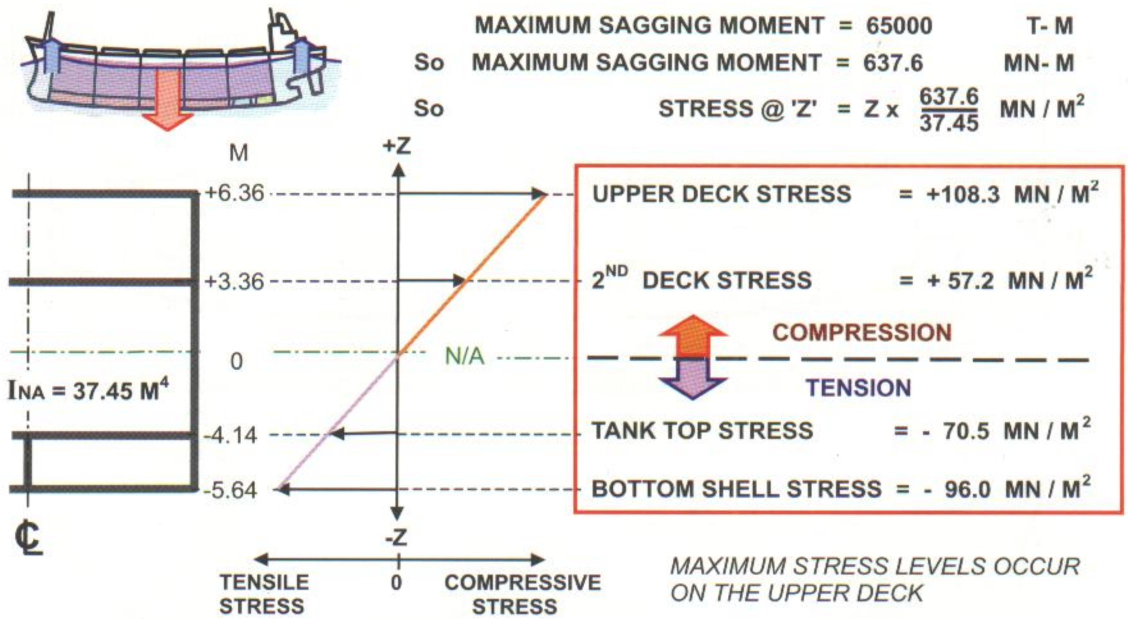
So MOMENT OF INERTIA ABOUT N/A = 37.6733 - (1.688) x $(0.36)^2$ M^4

MOMENT OF INERTIA ABOUT N/A = 37.4547 M^4 or approx. 37.45 M^4

BENDING STRESS CALCULATIONS FOR SHIPS (Cont.)

The maximum bending moment stress at a particular point along the ship's length is determined by applying the maximum bending moment to the sectional Moment of Inertia at that point and producing a stress diagram for the section. Bending Moments should be expressed in KiloNewton-Metres or MegaNewton-Metres, where 1 Tonne -metre is the equivalent to 9.81 KN-M and 1 MN is 1000 KN.

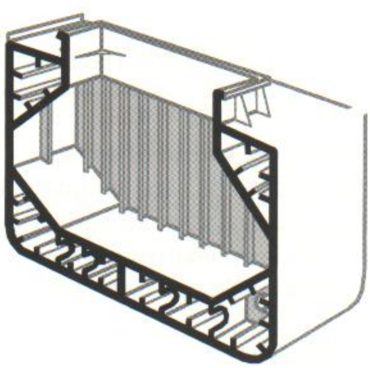
STRESS DIAGRAM FOR A SIMPLIFIED MIDSHIPS SECTION



ULTIMATE TENSILE STRESS FOR MILD STEEL ≈ 440 MN / M^2 OPTIMUM STRESS ≈ 110 MN / M^2

A hull with the maximum stress values shown above would have acceptable longitudinal strength, though the ultimate and optimum stress values for mild steel are approximate. There must be a generous safety margin between the maximum stresses experienced and the elastic limit. This allows a margin of error in the calculations and for steel wastage to occur over the life of the vessel. It is quite normal in most types of dry cargo ship, for the upper deck to be slightly more stressed than the bottom plating, as cargo weight bears directly onto the double bottom structure. A substantial double bottom will also provide reserve strength to compensate for any bottom damage if the ship goes aground. Actual midships sections of real ships, are considerably more complex than the example we have examined in these two pages. We would have to consider all the longitudinal stiffeners in the 'I_{NA}' calculations and some typical sections are shown below.

EXAMPLES OF TYPICAL MIDSHIPS SECTIONS

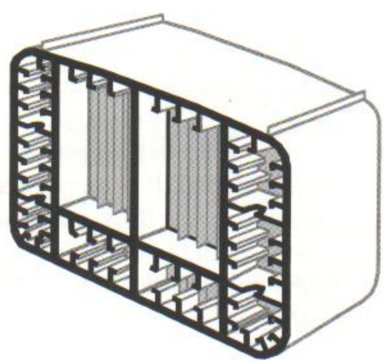


← **BULK CARRIER**

HOPPER SIDED CARGO HOLD WITH BALLAST TANKS

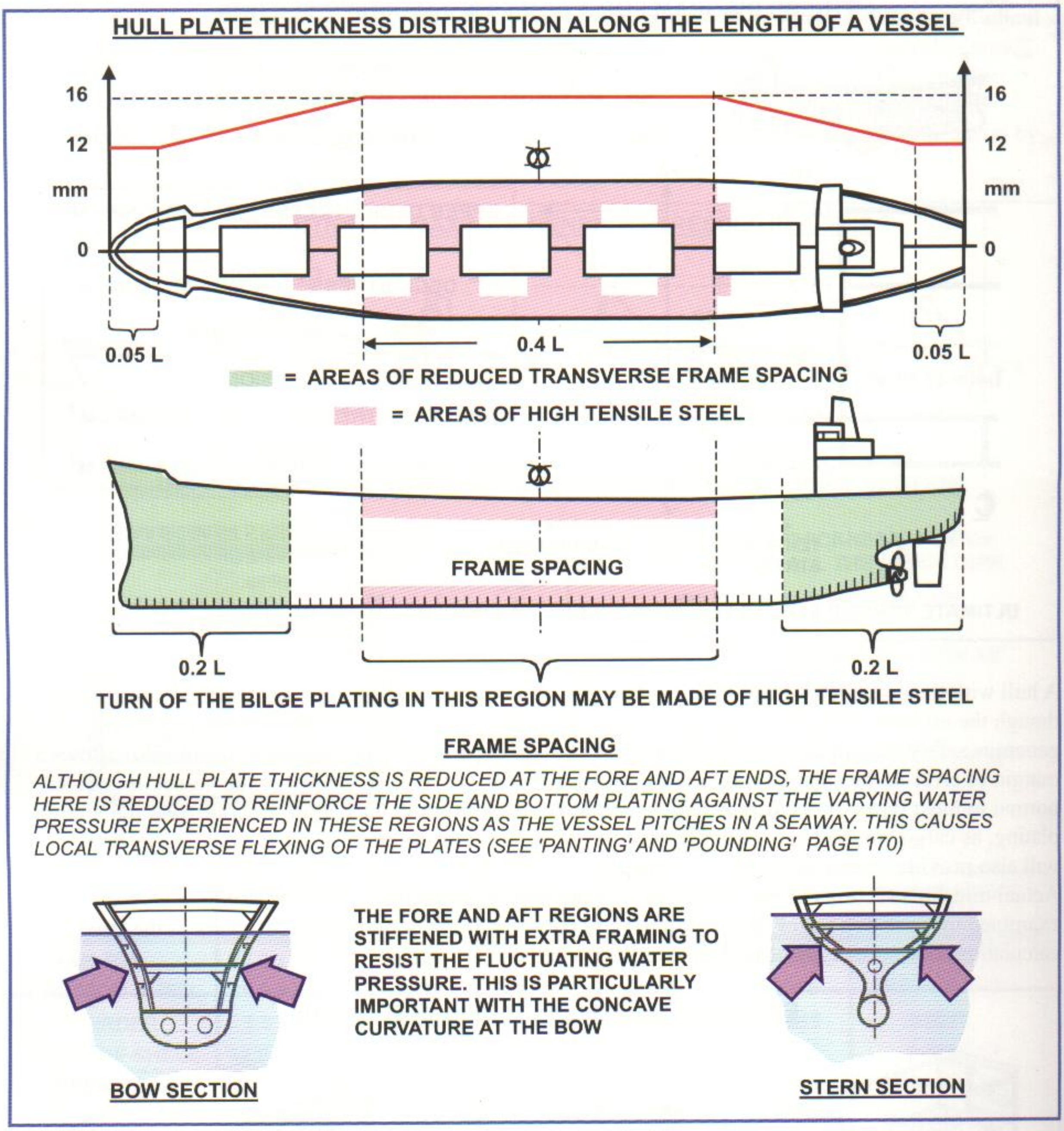
POST 1986 TANKER →

WING AND DOUBLE BOTTOM BALLAST TANKS SURROUND CARGO TANKS



STRESS DISTRIBUTION WITHIN THE SHIP'S HULL

Bending moments and their resulting stresses are greatest at the midships region of the hull in the upper deck and ship's bottom, whereas they are negligible at the bow and stern. Consequently, the thickness of the steel hull plating can be reduced near the fore and aft ends of the vessel. Regions of high stress are also more prone to cracking, so high tensile steel is often used to for the plating in these areas. This has the same modulus of elasticity as mild steel but its elastic limit is about 50% higher.

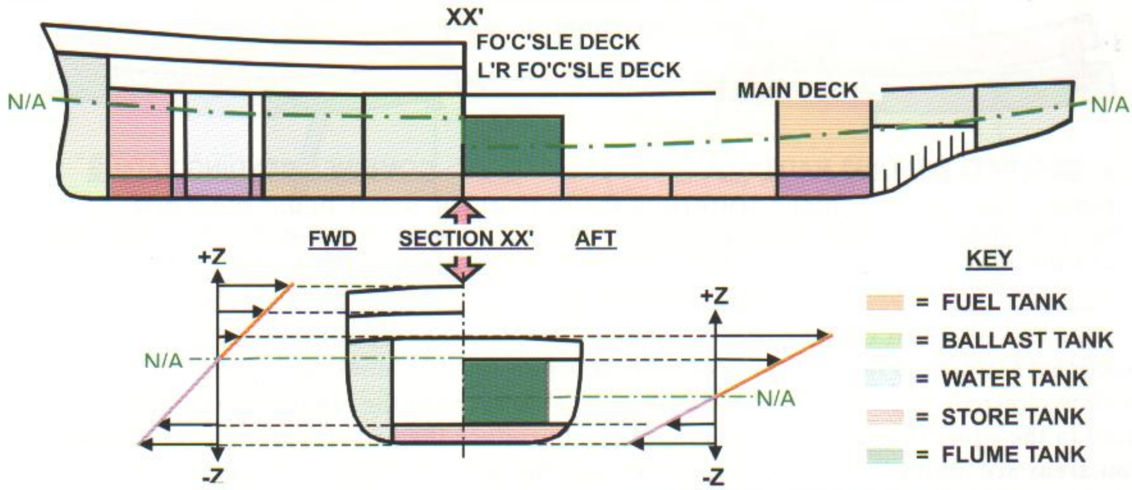


High tensile steel has better resistance to cracking than mild steel and also to the spread of a crack, once it has developed. It is more expensive than ordinary steel, so it only tends to be used sparingly. In the example above, the deck margins, turn of the bilge and corners of the hatchways are all built with high tension steel in the midships region of the hull. Up to the early 1970's, all welded vessels were built with the upper deck riveted onto the hull, as the row of rivets acted as a crack arrester but now there is sufficient confidence in the high grades of steel for this not be required.
The increased framing in the fore and aft ends is to resist transverse buckling of the hull, due to the stresses on the side and bottom plating caused by 'pounding', 'panting' and 'slamming' (See page 170).

STRESS DISTRIBUTION WITHIN THE SHIP'S HULL (Cont.)

The sectional height of a ship's hull is not usually constant throughout its length. Extra decks and superstructure extend over limited lengths of the vessel, whilst hatchways are cut into decks. These discontinuities would cause an instant change in the sectional stress distribution and create a potential weakness at the point of transition if this is not spread out into the surrounding structure. This is shown below in the case of an off-shore support vessel with a high fo'c'sle and low aft deck.
Bulwarks and railings are relatively light structures to provide protection to the crew on deck. They are fitted to the upper deck at the maximum distance from the N/A where the stress is greatest. Any crack that started in the bulwark would spread into the deck and hull if the bulwarks were continuously connected to the hull plating. This would be a serious weakness in the hull's strength so bulwarks and railings are fitted in short sections on vertical struts which effectively isolates them from the hull stresses as each section can move relative to its neighbouring sections.

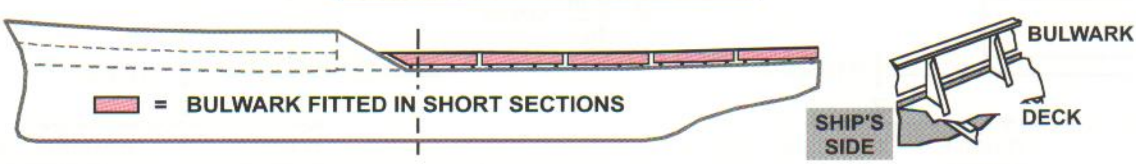
DISCONTINUITY IN STRENGTH IN THE CASE OF A HIGH FO'C'SLE AND LOW AFT DECK



ABRUPT DISCONTINUITY **DISCONTINUITY SPREAD OUT AND STRESS DISPERSED**

THE MOMENT OF INERTIA AFT OF SECTION XX' IS CONSIDERABLY LESS THAN THE FOR THE FWD PART OF THE HULL, SO STRESS LEVELS INCREASE IMMEDIATELY AFT OF IT, THE BEND RADIUS BECOMES TIGHTER AND THE NEUTRAL AXIS IS VERTICALLY SHIFTED. THE MIS- MATCH OF THE N/A AND STRESS LEVELS ACROSS XX' IS LIKELY TO LEAD TO FRACTURE.
THIS CONCENTRATION OF STRESS IS SPREAD OUT BY EXTENDING AND TAPERING THE FO'C'SLE SIDE PLATING INTO THE AFT HULL.
THE MIS-MATCH OF STRENGTH CHARACTERISTICS ACROSS XX' IS REDUCED BY DECREASING THE FWD SECTION MOMENT OF INERTIA. THIS COULD BE ACHIEVED BY DECREASING THE LOWER FO'C'SLE DECK THICKNESS OR INCREASING THICKNESS OF THE AFT MAIN DECK

BULWARKS AND SIDE RAILINGS

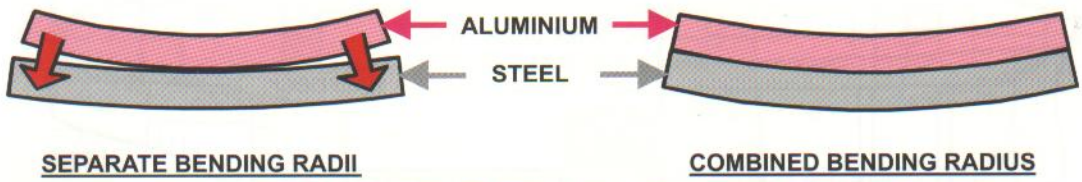


BULWARKS AND SIDE RAILINGS ARE FITTED ON THE TOP EDGES OF HULL PLATING AND AS SUCH WOULD BE UNDER MAXIMUM STRESS AND BE A POSSIBLE SOURCE OF WEAKNESS IF THEY WERE SIMPLY A DIRECT EXTENSION OF THE SHIP'S SIDE.
RAILINGS AND BULWARKS ARE NOT CONTINUOUSLY WELDED DIRECTLY TO THE DECK OR HULL ALONG THE SHIP'S LENGTH. INSTEAD, THEY ARE FITTED IN SECTIONS BY VERTICAL STRUTS SO THEY ARE DISCONNECTED FROM THE DECK EDGE STRESS AS THE SHORT SECTIONS CAN MOVE INDEPENDENTLY FROM EACH OTHER

COMPOSITE HULL STRUCTURE

Some ship designs use an aluminium alloy for deck housing in place of steel. This provides a weight saving and reduces the lightship height of centre of gravity (the 'KG' value). This is particularly beneficial for passenger ships as it can allow an extra accommodation deck to be added and so increase the vessel's earning capacity.
The bending stress calculations that we have considered so far are based upon a structure of material with a single elastic modulus value. Aluminium, however, has a modulus value of 7 MN/cm^2 compared with the 21 MN/m^2 value for steel, so the same stress will produce a strain in aluminium that is three times greater than that for steel. A given bending moment will produce a bending radius in an aluminium beam that is only one third of the value for a steel beam of the same cross sectional area, so joining the two materials produces a compromise bending curvature.

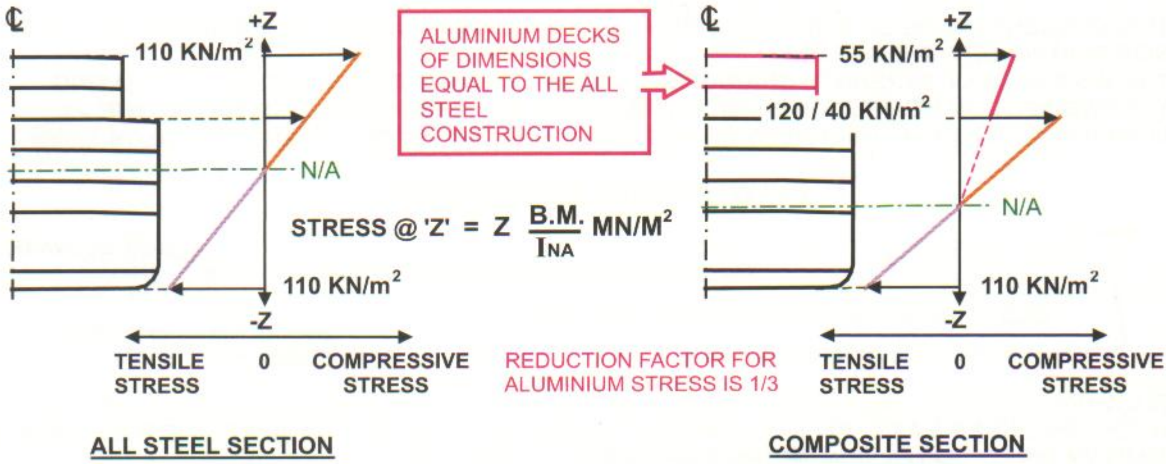
THE BENDING OF A COMPOSITE BEAM



JOINING THE TWO MATERIALS TOGETHER RESULTS IN THE STEEL BEING BENT INTO A TIGHTER RADIUS THAN IT WOULD ADOPT SEPARATELY, WHILST STRAIN IN THE ALUMINIUM IS REDUCED, SO STRESS IS TRANSFERRED FROM THE ALUMINIUM TO THE STEEL

The sectional Moment of Inertia for a composite structure and its neutral axis are determined in the way that is shown on page 201, **except that the sectional areas of the aluminium components are reduced to the equivalent areas of steel that would produce the same strain. The aluminium section areas are multiplied by the factor 0.33 (i.e. the ratio of 'E'steel : 'E'aluminium.).**
The stress at different distances from the neutral axis are calculated as before, **but the true sectional areas of aluminium must be taken into account, so their actual stress levels will be only 1/3 of the calculated values for the equivalent areas of steel.** Stress calculations for any composite structure will show that the steel is over-stressed, relative to the aluminium structure, and so the plate thickness of the upper steel decking must be greater than would be the case for an all steel construction.

STRESS DIAGRAMS FOR ALL STEEL AND COMPOSITE SECTIONS OF EQUAL DIMENSIONS



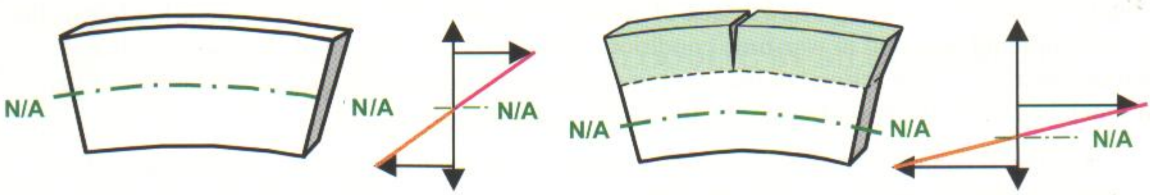
THE REPLACEMENT OF THE STEEL HOUSING BY ALUMINIUM REDUCES THE MOMENT OF INERTIA AND MOVES THE NEUTRAL AXIS CLOSER TO THE BOTTOM PLATING. THIS GREATLY INCREASES THE STRESS ON THE UPPERMOST STEEL DECK.

THIS DECK CAN BE RETURNED TO ITS ORIGINAL STRESS LEVEL BY INCREASING ITS THICKNESS

CRACKING AND OTHER SIGNS OF STRUCTURAL FAILURE

A crack creates a stress concentration that causes the crack to spread, so further intensifying the stress and increasing the rate at which the crack spreads. This process will eventually cause the structure to fracture. Cracks usually start at a point where a discontinuity in the structure has been poorly merged into the neighbouring structure. An example of this would be corners of hatchways or access cut-outs that have fillets of insufficient radius. A crack developing in the main hull structure is a serious problem that requires fairly immediate repair in a shipyard but minor cracking in deck house plating can be temporarily repaired by drilling a hole at the spreading end of the crack. This will reduce the stress concentration and act as a crack arrester (riveted seams used to limit the spread of cracking in older vessels). The hole can be sealed with a nut and bolt whilst the crack is partly ground out and filled by welding. This will not be as strong as the original structure so whatever the source of weakness is (typically a door cut-out with too tight a radius in the corners), it will lead to further cracking if the steelwork is not replaced with a more suitable shape at a later date.

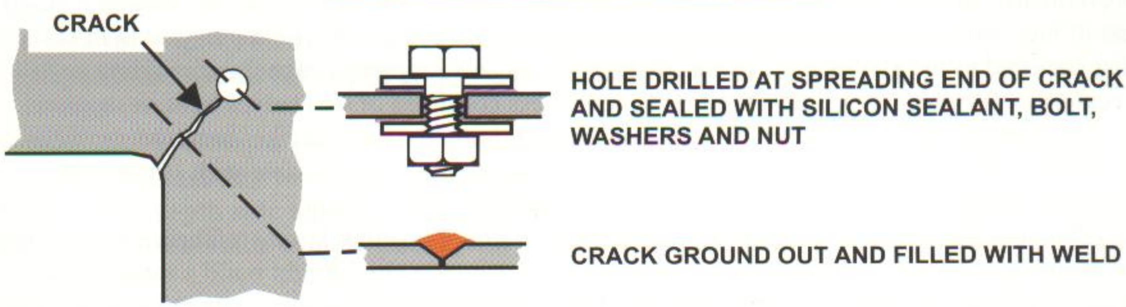
THE CHANGE OF STRESS DISTRIBUTION DUE TO CRACKING



INTACT PLATE AND STRESS DIAGRAM **PLATE AND STRESS DIAGRAM AFTER CRACKING**

AFTER CRACKING, THE GREEN SHADED AREA CONTRIBUTES NOTHING TO THE STRENGTH OF THE PLATE SO THE NEUTRAL AXIS MOVES DOWNWARDS AND STRESS IS GREATLY INCREASED AT THE TOP AND BOTTOM EDGES OF THE REMAINING CONNECTED MATERIAL. THIS WILL LEAD TO THE CRACK SPREADING FURTHER.

TEMPORARY REPAIR OF MINOR CRACKS



Aluminium structures must be isolated from the steel to avoid electrolytic corrosion between the two different metals so the riveted connection to the steel hull includes a layer of electrical insulating material, such as a tough thin layer of polythene. This insulating layer will break down over time, which leads the bottom edge of the aluminium plating being eaten away by corrosion.

A BRIEF NOTE ON SHIPBUILDING METHODS

Detailed descriptions of ship construction are outside the scope of this book but a general appreciation of shipbuilding techniques is useful as they influence the design of a vessel.

For hundreds of years, ships were built on the launching site (known as the 'stocks') from the bottom upwards. The keel and frames would be positioned and connected together then the hull would be built upon the resulting skeletal structure in lengths of wood planking, known as 'strakes'. This continued when iron and then steel replaced wood as the basic shipbuilding material. The strakes then consisted of lengths of overlapping plating joined together and to the frames by heat-shrunk rivets. The method allows for a vessel's hullshape to be formed and adjusted slightly during the building.

A BRIEF NOTE ON SHIPBUILDING METHODS (Cont.)

The problem, however, with the 'traditional' approach of constructing a vessel on the stocks from the keel upwards is that it is time consuming as all the fabrication occurs on site and so has to progress linearly. Work cannot start on the deck until the hull is completed. Furthermore, the technique of riveting increases the hull weight as plates must overlap and frames require flanges to rivet through when joining onto plating.

These disadvantages became critical during World War Two when the British needed to replace the large number of merchant ships being sunk by the Germans in their attempt to prevent trade with Britain. The U.S.A. had enormous industrial capability but relatively few shipyards so British naval architects designed standard vessels that could be built by non-specialised steel companies using prefabrication and welding methods. The most famous of these were the 'Liberty' cargo boats though they were followed by the 'Victory' ships and the 'T-2' tankers.

The 'Liberty' ship design was a great success in terms of allowing large numbers of vessels to be built quickly by workforces without previous shipbuilding experience, but structural strength was a problem, some vessels developing serious hull cracks in their first few months of service. It is difficult to say how much this was due to design flaws, lack of consistent good quality welding or poor loading as the ships were built and operated in war-time conditions. Hundreds of this type of ship were built and many were sunk by enemy action but the surviving vessels continued to be an important part of the world's merchant fleet for the twenty years that followed the end of the war.

At the end of the Second World War, shipyards generally returned to traditional riveting methods of shipbuilding and welding was limited to prefabricating larger sizes of plates for hull construction on the stocks. However, by the 1960's welding methods and their quality control had been developed sufficiently for increasing numbers of ships to have all-welded hulls, except for the joint between the uppermost strake (known as the 'sheer strake') and the deck edge. This was still riveted to act as a crack arrester in the region of the hull where bending stresses are greatest.

Incorporating a riveted seam long the full length of the ship made it difficult to prefabricate the hull so it was eventually replaced by the crack resistant high tensile steels, which allowed welding to be used in areas of high stress. The development of these steels was encouraged by the rapid growth in tanker size that occurred in the 1960's as there was distinct shortage of shipyards capable of building such large vessels in one piece. Several of the first supertankers consisted of two halves built in separate yards, often many miles apart and in different countries. Joining the hull together required a greater degree of precision than had been previously necessary but this became possible with automated techniques in design and plate cutting.
These techniques paved the way for ships to be built by welding together large prefabricated modules of hull, weighing several hundred tons, in much the same way as a child might build a model from a collection of plastic construction bricks, such as 'Lego'. Building time is greatly reduced as work can proceed simultaneously on different hull sections in enclosed protecting prefabrication halls. The modules generally include all the pipework and much of the ancillary equipment, such as engine room pumps, generators etc., so the vessel's 'fitting out' time is also reduced and nearly all merchant ships have been built in this manner since the 1970's.

Prefabrication with computerised control of plate cutting has tended to favour simple geometric shapes, so the generously radiused curves at breaks of superstructures that featured so much in ships built up to the 1960's have disappeared. The full rounded stern has also been replaced by the flat transom whilst decks now are usually built without 'camber', which increased a deck's strength as well as assisting drainage. More reliance is now placed on greater precision in calculating stresses and using the high grades of steel to prevent stress concentrations and ships are built with smaller ***scantlings*** (i.e. plate thickness and frame widths etc.) than was the case in the past.

The increased precision employed in shipbuilding and design has tended to reduce the margins allowed for error in construction and this can result in corrosion being more of a problem as the ship ages. Owners and ships' officers should appreciate that the economic circumstances that prevailed when a ship was built usually change and vessels often continue trading well beyond their original planned working life when their strength characteristics may become more suspect.

CHAPTER 9

THE CONSEQUENCES OF FLOODING THROUGH BILGING

SUMMARY

THIS CHAPTER OUTLINES THE WAYS OF CALCULATING THE EFFECTS OF ACCIDENTAL FLOODING UPON THE SHIP'S DRAFT, TRIM AND STABILITY.

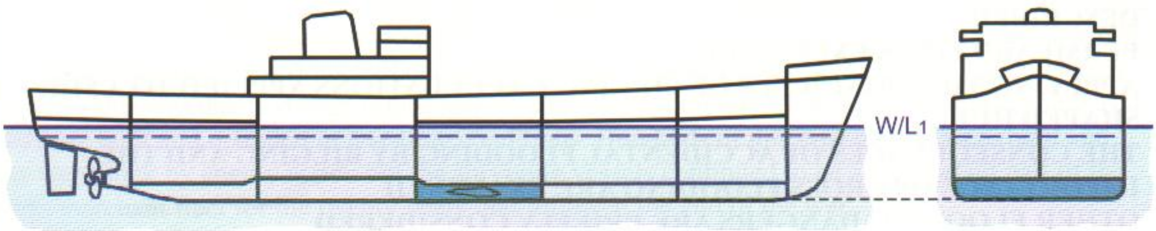
1) **THE APPROACHES OF 'ADDED WEIGHT' AND 'LOST BUOYANCY' ARE DESCRIBED AS ALTERNATIVE APPROACHES TO CALCULATING THE EFFECTS OF BILGING**
2) **STANDARD BILGING CALCULATIONS FOR BOX-SHAPED HULLS ARE DESCRIBED**
3) **PERMEABILITY IS EXPLAINED**
4) **APPLYING THE PRINCIPLES OF BILGING CALCULATIONS APPLIED TO A SHIP SHAPED HULL**
5) **THE CONSEQUENCES OF ACCIDENTAL FLOODING BY BILGING AND THE IMPORTANCE OF CROSS FLOODING ARE EXAMINED**
6) **OTHER FLOODING DANGERS ARE BRIEFLY CONSIDERED**

CONTENTS

BILGING OF A VESSEL

Bilging is the term given to the accidental flooding of a vessel through underwater damage by collision or stranding. If the hull is a single continuous compartment, then it will fill completely with the ingress of water and sink but all vessels of any significant size have a degree of internal subdivision that restricts the initial flooding to the damaged compartment only. This chapter is concerned with calculating the effects of bilging upon the ship's draft, trim and stability with regard to whether the ship can remain afloat after such an accident.
Bilging an empty double bottom tank is the simplest situation to assess the damaged condition, as the ingress of water will flood the tank completely. We can either consider this flooding to be 'added weight' or 'lost buoyancy'. Either approach can be used to calculate the vessel's damaged condition.

THE DAMAGED CONDITION OF A SHIP BILGED IN AN EMPTY MIDSHIPS DOUBLE BOTTOM TANK



VESSEL'S INTACT CONDITION, PRIOR TO BILGING

Δ'T = 10,000 T, DRAFT = 6.00 M, TPC = 18 T/cm, KG_0 = 6.50 M, KB_0 = 3.30 M, BM_0 = 4.10 M,

GM_0 = 0.90 M, FLOODED TANK CAPACITY = 450 M^3 @ Kg of 0.75 M

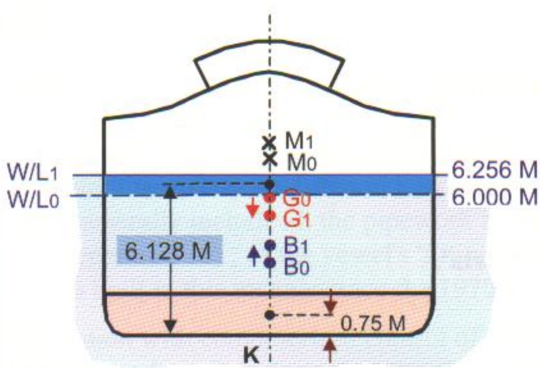
CONDITION AFTER BILGING BY THE 'ADDED WEIGHT' METHOD

WEIGHT OF SEA WATER FLOODING = TANK VOLUME x S.W. DENSITY = 450 x 1.025 T

And BODILY SINKAGE = $\frac{\text{WEIGHT OF WATER INGRESS}}{\text{TPC}} = \frac{450 \times 1.025}{18}$ cm

So BODILY SINKAGE = 25.6 cm & MEAN DRAFT = 6.256 M

WEIGHT INCREASE = 461.25 T @ Kg 0.750 M & BUOYANCY INCREASE = 461.25 T @ Kg 6.128 M



CHANGE IN TRANSVERSE STABILITY

KG IS REDUCED DUE TO ADDED BOTTOM WEIGHT

$$G_0G_1 = \frac{4.61.25}{10461.25} \times (6.500 - 0.750)\ \text{M}$$

So G_0G_1 = 0.254 M Downwards

KB INCREASES DUE TO INCREASE IN BUOYANCY

$$B_0B_1 = \frac{4.61.25}{10461.25} \times (6.128 - 3.300)\ \text{M}$$

So B_0B_1 = 0.125 M Upwards

ALTHOUGH WPA & ITS MOMENT OF INERTIA ARE ASSUMED TO REMAIN CONSTANT, THE 'BM' VALUE WILL DECREASE AS THE VOLUME OF DISPLACEMENT HAS INCREASED (SEE CHPTR 2)

I.e. DECREASE IN BM = $BM_0 - BM_0 \frac{V_0}{V_1}$ M, So DECREASE IN BM = $4.1(1 - \frac{10000}{10461.25}) = \underline{0.181\ \text{M}}$

THE TOTAL INCREASE IN GM = REDUCTION OF KG + INCREASE OF KB - REDUCTION OF BM

So TOTAL INCREASE IN GM = 0.254 + 0.125 - 0.181 = 0.198 METRES AFTER BILGING

So THE BILGED GM VALUE = 0.900 + 0.198 = 1.098 M FOR DISPLACEMENT OF 10461.25 T

BILGING OF A VESSEL (Cont.)

The previous page considered the ingress of seawater into the bilged double bottom as 'added weight' which was born by the hull bodily sinking to increase the buoyancy by the same amount. The alternative approach is to regard the bilged tank as open to the sea and therefore no longer contributing to the ship's buoyancy. This is the 'lost buoyancy' method, in which the displacement is unchanged and buoyancy is transferred from the bilged compartment to the layer of underwater hull created by the bodily sinkage.

THE DAMAGED CONDITION OF A SHIP BILGED IN AN EMPTY MIDSHIPS DOUBLE BOTTOM TANK (Cont.)

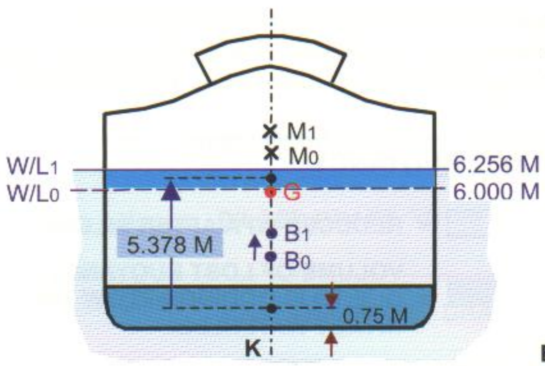
CONDITION AFTER BILGING BY THE 'LOST BUOYANCY' METHOD

WEIGHT OF SEA WATER FLOODING = TANK VOLUME x S.W. DENSITY = 450 x 1.025 T

And BODILY SINKAGE = $\dfrac{\text{WEIGHT OF WATER INGRESS}}{\text{TPC}} = \dfrac{450 \times 1.025}{18}$ **cm**

So BODILY SINKAGE = 25.6 cm & MEAN DRAFT = 6.256 M

WEIGHT & BUOYANCY REMAINS CONSTANT AT 10,000 T BUT THERE IS A SHIFT OF 461.25 T OF BUOYANT DISPLACEMENT FROM THE BILGED COMPARTMENT TO THE BODILY SINKAGE LAYER



THERE IS NO SHIFT IN THE C of G AS WEIGHT IS UNCHANGED AND KB IS INCREASED BY THE THE UPWARDS TRANSFER OF BUOYANCY

$$B_0B_1 = \frac{4.61.25}{10000} \times (6.128 - 0.750)\ \text{M}$$

So B_0B_1 = 0.248 M Upwards

THE BM VALUE REMAINS UNCHANGED AS THE DISPLACED VOLUME HAS NOT ALTERED, SO

$B_0B_1 = M_0M_1$ = 0.248 M Upwards

THE TOTAL INCREASE IN GM = INCREASE OF KM

So TOTAL INCREASE IN GM = 0.248 METRES AFTER BILGING

So THE BILGED GM VALUE = 0.900 + 0.248 = 1.148 M FOR DISPLACEMENT OF 10,000 T

The two methods of assessing the transverse stability of a damaged hull appear, at first glance, to produce different answers, as the GM values do not agree. However, we must remember that the true measure of stability is the **Righting Moment,** which is determined by the angle of heel, the upright GM value and the Displacement. The 'added weight' method is based upon the displacement increasing by the amount of the ingress of water whereas the 'lost buoyancy' approach is based upon the displacement remaining unchanged. If we compare the Righting Moments produced by the two methods, then we will find that they are in agreement, at least to within the limits of error in the hydrostatic data.

AT θ° OF HEEL, THE RIGHTING MOMENT = GM(UPRIGHT) x DISPACEMENT x Sin θ° T-M

SO CONSIDERING THE EXAMPLE OF THE VESSEL BILGED IN A MIDSHIPS DOUBLE BOTTOM TANK

RIGHTING MOMENT BY 'LOST BUOYANCY' = 1.148 x 10000 x Sin θ° = 11480 x Sin θ° T-M

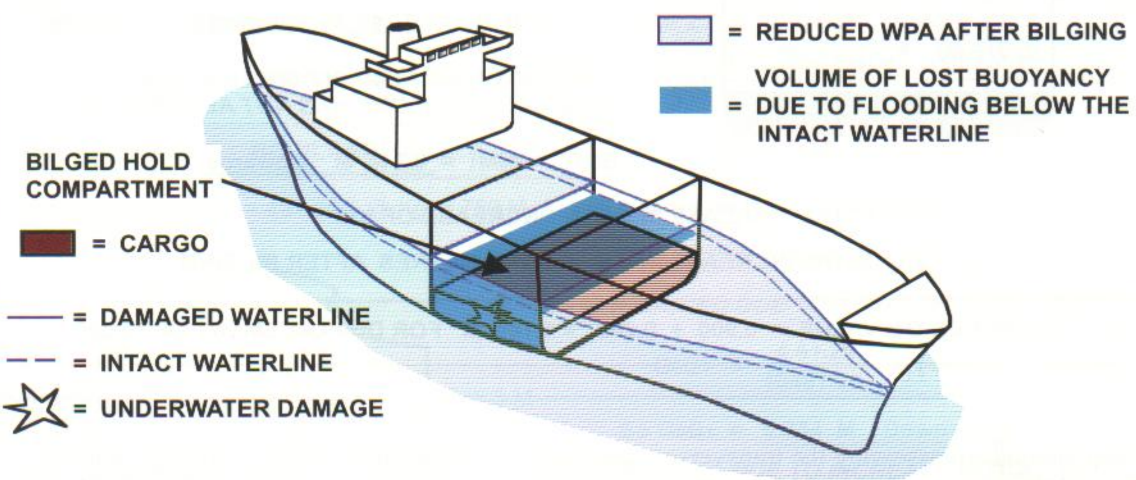
& RIGHTING MOMENT BY 'ADDED WEIGHT' = 1.098 x 10461 x Sin θ° = 11486 x Sin θ° T-M

BILGING OF A VESSEL (Cont.)

The 'added weight' method of assessing a bilged vessel's stability uses equations that may be more familiar to ship's officers who are regularly carrying out stability calculations for changing weight distribution within the ship. It is, however, more involved than the 'lost buoyancy' method and it is difficult to apply in circumstances where water floods into a compartment that is continuous above the vessel's intact waterline (as most cargo spaces do). In these situations, we would have to apply free surface effects of water ingress and it is not easy to determine the precise weight of water that floods into the space as this must eventually match the damaged waterline.
As a first approximation, we can assume that all the vacant space in the damaged compartment beneath the intact waterline floods. Then the trim, draft and heel will change to submerge the damaged space further, so there will be more progressive flooding until the waterline inside the flooded space matches that outside the hull. This additional flooding must then be added to our first estimate, so the calculation would have to be repeated in a re-iterative way. This makes the calculations complicated and, in any case, the amount of water inside the bilged space will continually change with the ship's pitching and rolling so the 'added weight' approach to estimating the ship's damaged condition is rather cumbersome and imprecise.

It is much more satisfactory to **consider the flooded space beneath the intact waterline as in the 'lost buoyancy' method.** The weight distribution throughout the ship remains the same as for the intact hull but the buoyancy and waterplane distribution change so draft and stability calculations for the damaged condition are based upon these changed values of WPA and underwater hull volume. Lost reserve buoyancy in the hull in the damaged compartment above the level of flood water is taken into account by applying bodily sinkage over the reduced waterplane. Only empty space in the damaged compartment can flood so the volume of any cargo must be excluded from the calculations.

BILGING CONSIDERED AS 'LOST BUOYANCY'



THE BILGING OF THE EMPTY HOLD RESULTS IN A LOSS OF BUOYANCY EQUAL TO THE WEIGHT OF WATER THAT FLOODS INTO THE EMPTY SPACE *BELOW THE INTACT WATERLINE*.

(ANY HOLD SPACE OCCUPIED BY CARGO IN THE DAMAGED COMPARTMENT WILL NOT BE FLOODED AND SO MUST NOT BE INCLUDED IN THE VOLUME OF LOST BUOYANCY).

THE BODILY SINKAGE OF THE DAMAGED HULL IS SPREAD OVER THE REDUCED WATERPLANE AREA (WPA), *SO THE SPACE IN THE DAMAGED COMPARTMENT ABOVE THE INTACT WATERLINE DOES NOT ACT AS RESERVE BUOYANCY*

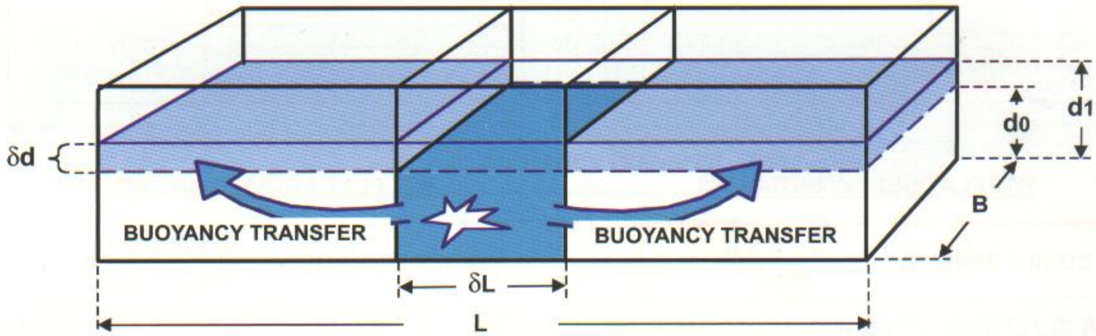
DAMAGED STABILITY AND TRIM CHARACTERISTICS ARE BASED UPON THIS *REDUCED WPA*

The 'lost buoyancy' method is the standard approach used by naval architects at the design stage of a ship, to predict a its damaged condition, both for transverse stability and longitudinal trim, if one or more of the vessel's various compartments are bilged at different states of loading.

BILGING A FULL WIDTH, FULL DEPTH EMPTY MIDSHIPS COMPARTMENT IN A BOX-SHAPED HULL

Considering the bilging of a box-shaped hull is a good way to understand the calculations for assessing the damaged draft, trim and stability by the 'lost buoyancy' method. Such examples are often used as questions in the Certificates of Competency examinations for Masters and Mates. One of the simplest cases is that of bilging a full width, full depth, midships empty compartment that has no trimming effect.

BILGING A EMPTY MIDSHIPS HOLD IN A BOX-SHAPED HULL



WHEN THE MIDSHIPS HOLD OF LENGTH 'X' IS BILGED, A BUOYANCY TRANSFER OCCURS AS THE BUOYANCY LOST UNDER THE UNDAMAGED WATERLINE IS COMPENSATED FOR BY BODILY SINKAGE OVER THE REDUCED WATERPLANE OF (L- δL) x B

So BODILY SINKAGE 'δd' $= \dfrac{\text{LOST BUOYANCY}}{\text{REDUCED WPA}} = \dfrac{L \times B \times d_0}{B\,(L - \delta L)}$ **M**

Hence BILGED DRAFT 'd_1' $= d_0 + \delta d$, **So 'd_1'** $= d_0 + \dfrac{L \times d_0}{(L - \delta L)}$ **M**

So THE BILGED DRAFT 'd_1' $= d_0 \dfrac{L}{(L - \delta L)}$ **METRES**

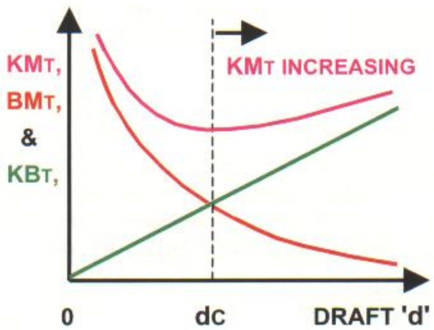
WHERE 'd0' = DRAFT FOR THE UNDAMAGED LENGTH 'L' AND 'X' = BILGED HOLD LENGTH

THE CHANGE OF DRAFT AND WATERPLANE AREA CAUSES BOTH THE KB_T AND BM_T VALUES TO CHANGE. THESE ARE CALCULATED BY THE EQUATION FOR KM_T OF A BOX-SHAPED HULL

I.e. AT THE BILGED DRAFT OF 'd_1' KB_T $= \dfrac{1}{2}d_1 + \dfrac{B^2}{12 d_1}$ **M** *(SEE CH'PT'R 2, PAGE 32)*

THE VESSEL'S DISPLACEMENT AND WEIGHT DISTRIBUTION ARE UNCHANGED BY THE BILGING, SO THE KG IS ALSO UNCHANGED. THE BILGED GM_T VALUE = KG - KB_T

THE INCREASE IN DRAFT RAISES THE KB VALUE BUT THE REDUCED WPA DECREASES THE BM VALUE. WHETHER THERE IS AN OVERALL INCREASE OR DECREASE IN THE KM VALUE AFTER BILGING, DEPENDS UPON THE RATIO OF THE BILGED DRAFT 'd0' TO THE VESSEL'S BEAM 'B'



KM_T AND, HENCE GM_T, INCREASE WHEN KB_T > BM_T

FOR A BOX-SHAPED HULL AT DRAFT 'd' AND BEAM 'B',

$$KB_T = \frac{1}{2}d \qquad \& \qquad BM_T = \frac{B^2}{12d}$$

So if $KB_T = BM_T$, THEN $\dfrac{1}{2}d_c = \dfrac{B^2}{12 d_c}$

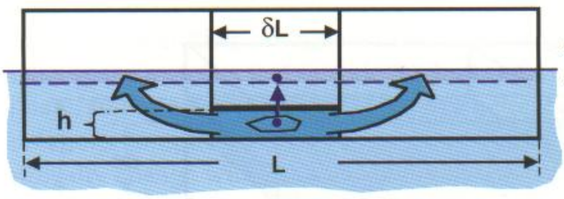
So $d_c = 0.41\,B$

GM_T IS INCREASED WHEN THE BILGED DRAFT > 0.41B

BILGING A FULL WIDTH EMPTY MIDSHIPS COMPARTMENT WITH A WATERTIGHT FLAT

A watertight flat in a bilged compartment (such as the double bottom tank top) can restrict flooding, depending upon its height relative to the final damaged draft. The calculation of the bodily sinkage, transverse stability and trim depends upon the particular circumstances of each case. Consider the case of the empty midships compartment shown below in a box-shaped hull of beam **'B'**.

BILGING A SPACE WITH A WATERTIGHT FLAT ABOVE OR BELOW INITIAL WATERLINE



W/T FLAT BELOW INITIAL W/L

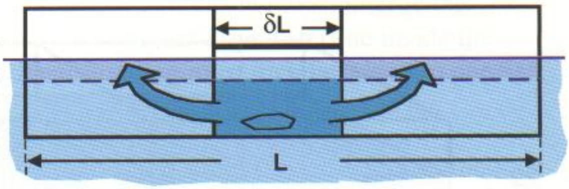
$$\text{BODILY SINKAGE} = \frac{\delta L \times \cancel{B} \times h}{\cancel{B} \times L} \text{ M}$$

WPA (B x L) & BM VALUE REMAINS CONSTANT. THE GM VALUE INCREASES AS KB MOVES UPWARDS WITH THE BUOYANCY TRANSFER

THE BILGED GM = (BILGED KB_T + BM_T) - KG

KB_T CAN BE FOUND BY SUBTRACTING THE MOMENT ABOUT THE KEEL OF THE BILGED VOLUME FROM THAT OF AN INTACT HULL AT DRAFT 'd1' AND DIVIDING THE ANSWER BY THE ACTUAL DISPLACED VOLUME

$$\text{BILGED } KB_T \text{ VALUE} = \frac{\cancel{B}\,(L d_1^2 - \delta L\, h^2)}{2\, L \cancel{B}\, d_0^2} \text{ M}$$



W/T FLAT ABOVE FINAL W/L

$$\text{BODILY SINKAGE} = \frac{\delta L \times \cancel{B} \times d_0}{\cancel{B} \times (L - \delta L)} \text{ M}$$

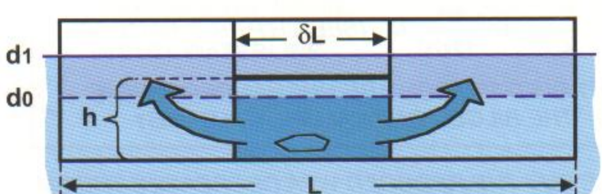
$$\text{So BILGED DRAFT} = d_0 \frac{L}{(L - \delta L)} \text{ M}$$

THE WPA (B x [L - δL]) IS REDUCED SO THE BM VALUE IS DECREASED BUT THE KB VALUE INCREASES AS THE C of B RISES DUE TO THE BUOYANCY TRANSFER (SEE PAGE 212)

THE BILGED GM = BILGED KB_T - KG

$$\text{BILGED } KB_T \text{ VALUE} = \frac{1}{2} d_1 + \frac{B^2}{12\, d_1} \text{ M}$$

WHEN THE WATERTIGHT FLAT IS ABOVE THE INITIAL UNDAMAGED WATERLINE, THERE WILL BE SOME SITUATIONS WHERE THE BILGED WATERLINE RISES ABOVE THE FLAT. IN THESE CIRCUMSTANCES, THE BODILY SINKAGE WILL OCCUR OVER THE REDUCED WATERPLANE AREA INITIALLY AND THEN FINALLY OVER THE FULL INTACT WATERPLANE



LOST BUOYANCY = $\delta L \times B \times d_0$ M^3

REDUCED WPA = $B \times (L - \delta L)$ M^2

$$\text{If 'h' IS GREATER THAN } \frac{\delta L \times \cancel{B} \times d_0}{\cancel{B} \times (L - \delta L)} \text{ M}$$

THEN W/T FLAT IS BELOW FINAL W/L

BUOYANCY LOST THROUGH BILGING = BUOYANCY REGAINED BY SINKAGE

i.e. $$\delta L \times B \times d_0 = B\,[\,(L - \delta L)(h - d_0) + L(d_1 - h)\,] \text{ M}^3$$

$$\text{So SINKAGE BEYOND FLAT 'd}_1 - h\text{'} = \frac{(\delta L \times d_0) - (L - \delta L)(h - d_0)}{L} \text{ M}$$

THE FINAL WPA (B x L) & BM VALUE REMAIN UNCHANGED BY THE BILGING. HOWEVER, THE KB INCREASES WITH THE UPWARD TRANSFER OF BUOYANCY, WHICH WILL INCREASE THE GM.

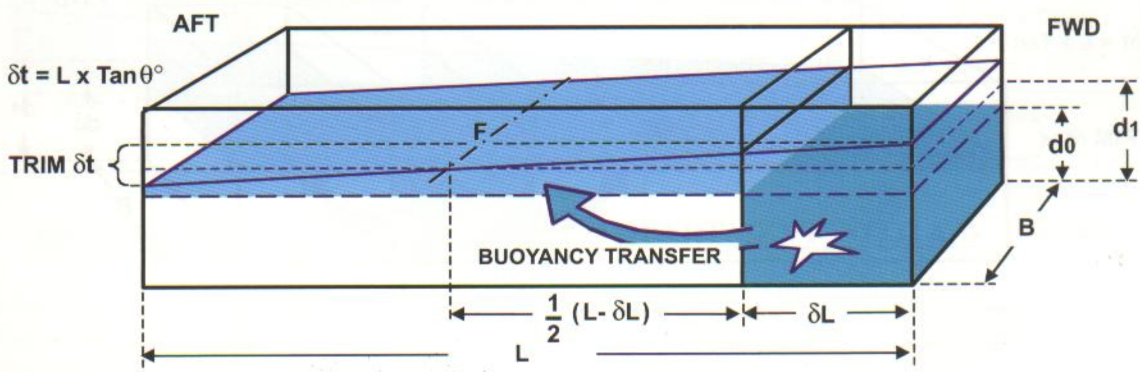
BY TAKING KEEL MOMENTS OF VOLUME OF LOST BUOYANCY FROM AN INTACT HULL AT 'd1'

$$\text{THE BILGED } KB_T \text{ VALUE} = \frac{\cancel{B}\,(L d_1^2 - \delta L\, d_0^2)}{2\, L \cancel{B}\, d_0^2} \text{ M}$$

BILGING A FULL WIDTH, FULL DEPTH EMPTY END COMPARTMENT

Most of a vessel's hull compartments are not exactly centred on the midships point and so will create a trimming moment when bilged. The longitudinal change in buoyancy distribution and waterplane area alters both the trim and GM_L value. The simplest example of this, is the case of bilging an end compartment in a box-shaped hull, as shown below.

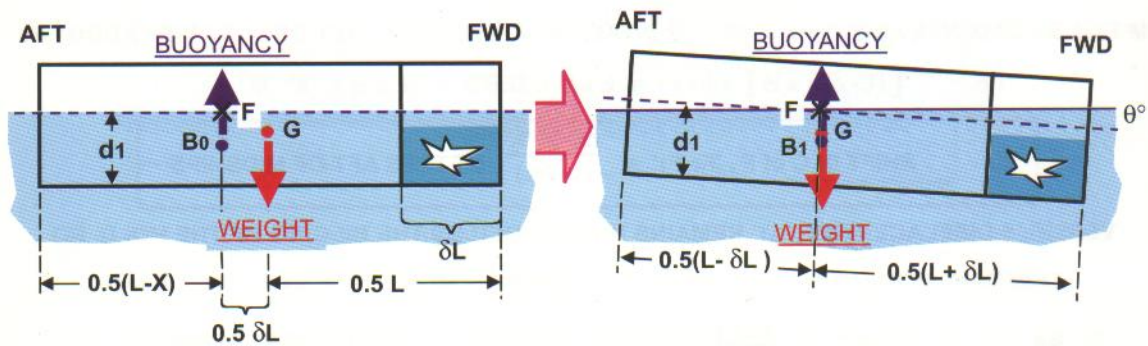
BILGING A EMPTY FWD END HOLD IN A BOX-SHAPED HULL



THE BODILY SINKAGE, MEAN DRAFT AND TRANSVERSE STABILITY ARE CALCULATED THE SAME WAY AS SHOWN IN THE PREVIOUS PAGE, I.E. AFTER A FWD HOLD OF LENGTH ' δL' IS BILGED

MEAN DRAFT 'd1' $= d_0 \dfrac{L}{(L - \delta L)}$ **METRES & KB_T** $= \dfrac{1}{2} d_1 + \dfrac{B^2}{12 d_1}$ **METRES**

THE BILGING, HOWEVER, HAS ALSO CAUSED SHIFTS IN THE CENTRES OF BUOYANCY AND FLOATATION ('F') AND SO CREATED A TRIMMING MOMENT BY THE HEAD



THE VESSEL'S DISPLACEMENT 'Δ_T' REMAINS UNCHANGED BY THE BILGING OF THE FWD HOLD

TRIMMING MOMENT $= \Delta_T \times 0.5\, \delta L$ **&** **TRIMMING MOMENT** $= \Delta_T \times GM_L \times \operatorname{Tan}\theta°$

So $\operatorname{Tan}\theta° = 0.5 \dfrac{\delta L}{GM_L}$ *WHERE 'GM_L' APPROXIMATES TO THE 'BM_L'*

Now $BM_L = \dfrac{(L - \delta L)^2}{12 d_1}$ *FOR A BOX SHAPED HULL (SEE CH'PT'R 6, PAGE 125)*

Hence $\operatorname{Tan}\theta° = \dfrac{6 \times \delta L \times d_1}{(L - \delta L)^2}$ *WHERE ' Sinθ° ' IS THE TRIM ANGLE BY THE HEAD*

THE HULL TRIMS ABOUT THE CENTRE OF FLOATATION 'F' WHICH IS THE WPA MIDPOINT

So THE DRAFT AFT $= d_1 - 0.5\,(L - \delta L)\operatorname{Tan}\theta°$

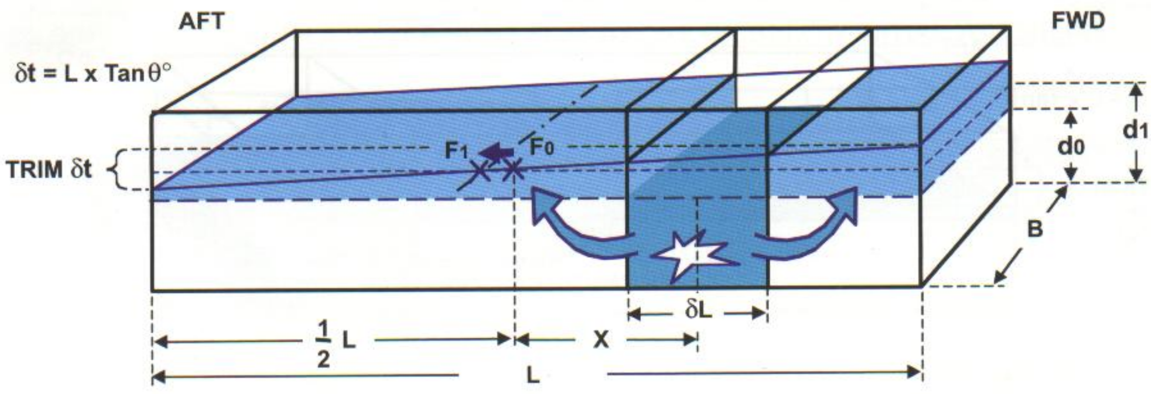
& THE DRAFT FWD $= d_1 + 0.5\,(L + \delta L)\operatorname{Tan}\theta°$

WHERE ' Sinθ° ' IS GIVEN AS ABOVE

BILGING ANY FULL WIDTH, FULL DEPTH EMPTY COMPARTMENT

We can now look at the more complex situation of bilging a full width hold that is not at the fore or aft end. Determining the shift in the Centres of Buoyancy and Floatation and the change in the GM_L value, require calculating Moments of Area and Inertia of the waterplane about the midships point.

BILGING ANY EMPTY HOLD IN A BOX-SHAPED HULL



BODILY SINKAGE, MEAN DRAFT AND TRANSVERSE STABILITY ARE CALCULATED AS BEFORE, I.E. AFTER A HOLD OF LENGTH 'δL' M, AND CENTRED 'X' M FROM MIDSHIPS, IS BILGED, THEN

$$\text{MEAN DRAFT } 'd_M' = d_0 \frac{L}{(L - \delta L)} \text{ METRES} \quad \& \quad KB_T = \frac{1}{2} d_1 + \frac{B^2}{12 d_1} \text{ METRES}$$

THE BILGING HAS ALSO CAUSED SHIFTS IN THE CENTRES OF BUOYANCY AND FLOATATION ('F'), EQUAL TO F_0F_1 AND SO CREATED A TRIM ANGLE OF θ° BY THE HEAD. THIS SHIFT IN 'F' CAN BE CALCULATED BY TAKING MOMENTS OF WPA ABOUT THE MIDSHIPS AXIS. I.E. ' F_0'

M'T OF BILGED WPA ABOUT F_0 = M'T OF INTACT WPA ABOUT F_0 - M'T OF LOST WPA ABOUT F_0

I.e. $$[(L - \delta L) \times B] \times F_0 F_1 = L \times B \times \text{ZERO} - \delta L \times B \times 'X' \quad M^3$$

$$\text{So} \quad \text{SHIFT IN C of F } 'F_0 F_1' = \frac{\delta L \times 'X'}{(L - \delta L)} \text{ METRES AFT OF MIDSHIPS}$$

VESSEL'S DISPLACEMENT 'Δ_T' REMAINS UNCHANGED BY THE BILGING OF THE HOLD, SO

$$\text{TRIMMING MOMENT} = \Delta_T \times F_0 F_1 \quad \& \quad \text{TRIMMING MOMENT} = \Delta_T \times GM_L \times \tan\theta°$$

So $$\tan\theta° = \frac{F_0 F_1}{GM_L}$$ *WHERE 'GM_L' APPROXIMATES TO THE 'BM_L'*

And $$BM_L = \frac{\text{MOMENT OF WPA INERTIA } 'I_L' \text{ ABOUT } F_1}{\text{DISPLACED VOLUME}} \quad M^4$$ *(SEE PAGE 131)*

'I_L' OF BILGED WPA ABOUT F_1 = 'I_L' OF INTACT WPA ABOUT F_1 - 'I_L' OF LOST WPA ABOUT F_1

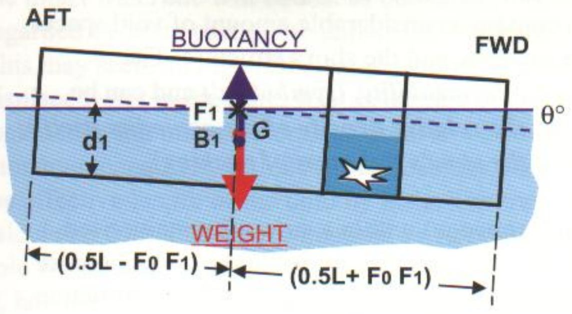
THE VALUES OF 'I_L' IN THE ABOVE EQUATION ARE CALCULATED BY APPLYING THE PRINCIPLE OF PARALLEL AXES, WHICH WE HAVE ALREADY ENCOUNTERED IN THE DIFFERENT CONTEXT OF BENDING STRESSES AND GIRDERS OF COMPLEX SECTIONAL AREAS. *(SEE PAGE 198)*

$$\text{So } 'I_L' \text{ OF BILGED WPA} = \frac{B \times L^3}{12} + B \times L \times (F_0 F_1)^2 - \left[\frac{B \times \delta L^3}{12} + B \times \delta L\,(X + F_0 F_1)^2\right] \quad M^4$$

$$\text{Hence} \quad BM_L = \frac{\cancel{B}\left[L^3/12 + L \times (F_0 F_1)^2 - \delta L^3/12 - \delta L\,(X + F_0 F_1)^2\right]}{\cancel{B} \times L \times d_0} \text{ METRES}$$

$$\text{So TRIM ANGLE } \tan\theta° = \frac{(F_0 F_1) \times L \times d_0}{L^3/12 + L \times (F_0 F_1)^2 - \delta L^3/12 - \delta L\,(X + F_0 F_1)^2} \quad \text{ABOUT } 'F_1'$$

BILGING ANY EMPTY HOLD IN A BOX-SHAPED HULL (Cont.)



THE TRIM ANGLE $\theta°$ HAS BEEN DETERMINED BY THE PROCESS SHOWN ON THE PREVIOUS PAGE, SO THE DRAFTS ARE AS FOLLOWS

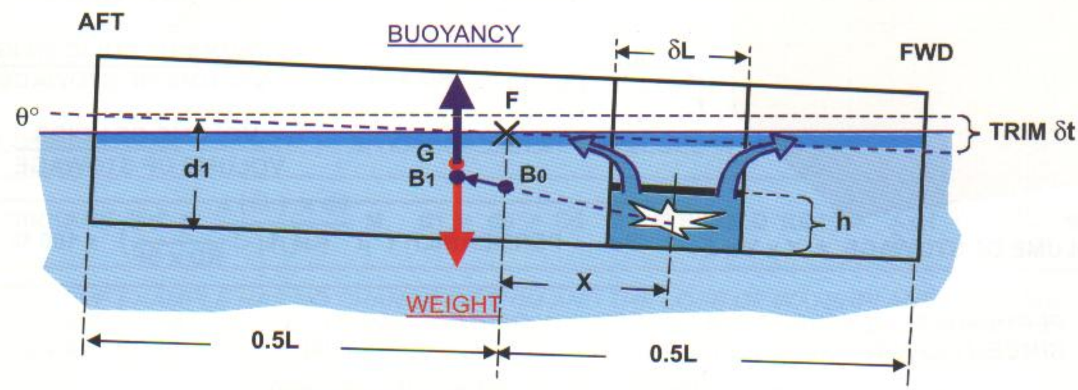
$$\text{AFT DRAFT} = d_1 - (0.5L - F_0F_1)\,\text{Tan}\theta°$$

$$\text{FWD DRAFT} = d_1 + (0.5L + F_0F_1)\,\text{Tan}\theta°$$

BILGING A FULL WIDTH, EMPTY COMPARTMENT WITH A W/T FLAT

In this situation, the waterplane area and, hence, GM_L value remain unchanged if the watertight flat is below the final trimmed waterline. We only need to calculate the shift in the Centre of Buoyancy to determine the trimming moment and the vessel will trim about the midships point.

BILGING ANY EMPTY COMPARTMENT WITH A W/T FLAT IN A BOX-SHAPED HULL



$$\text{BODILY SINKAGE} = \frac{\text{LOST BUOYANCY}}{\text{WATERPLANE AREA}}\ \text{M}, \quad \text{So} \quad \text{BODILY SINKAGE} = \frac{\cancel{B} \times \delta L \times h}{\cancel{B} \times L}\ \text{M}$$

$$\text{Hence AFTER BILGING THE SPACE OF LENGTH '}\delta L\text{', MEAN DRAFT '}d_1\text{'} = d_0 + h\,\frac{\delta L}{L}\ \text{M}$$

THE BILGING CAUSES THE C of B BOTH TO RISE (WHICH INCREASES THE GM_T) AND TO MOVE AFT. THE SHIFT AFT IS GIVEN BY DIVIDING THE MOMENT OF BUOYANCY TRANSFER BY THE DISPLACED VOLUME. THE BOX-SHAPED HULL, OF BEAM 'B', WILL TRIM ABOUT MIDSHIPS

$$\text{BILGED } KB_T \text{ VALUE} = \frac{\cancel{B}(Ld_1^2 - \delta L\,h^2)}{2L\cancel{B}\,d_0^2}\ \text{M}, \quad \text{AFT SHIFT '}B_0B_1\text{'} = \frac{X(\delta L \times \cancel{B} \times h)}{L \times \cancel{B} \times d_0}\ \text{M}$$

$$\text{TRIMMING MOMENT} = \Delta_T \times B_0B_1\ \text{T-M} \quad \& \quad \text{TRIMMING MOMENT} = \Delta_T \times GM_L \times \text{Tan}\theta°\ \text{T-M}$$

$$\text{So} \quad \text{Tan}\theta° = \frac{B_0B_1}{GM_L} \quad \& \quad GM_L \approx \frac{L^2}{12\,d_0}\ \text{METRES}$$

$$\text{Hence} \quad \text{Tan}\theta° = \frac{12\,B_0B_1 \times d_0}{L^2} \quad \& \quad \text{CHANGE OF TRIM '}\delta t\text{'} = L \times \text{Tan}\theta°\ \text{M}$$

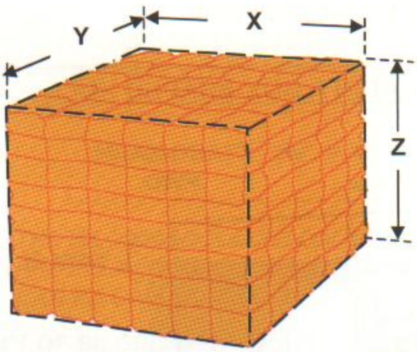
$$\text{So} \quad \text{FWD DRAFT} = d_0 + \frac{6\,B_0B_1 \times d_0}{L}\ \text{M} \quad \text{And} \quad \text{AFT DRAFT} = d_0 - \frac{6\,B_0B_1 \times d_0}{L}\ \text{M}$$

PERMEABILITY IN A BILGED COMPARTMENT LOADED WITH CARGO

Only empty space in a compartment is free to be flooded and, hence, when a hold, loaded with cargo, is bilged, only the void spaces within the cargo stow will fill with water. A general cargo stow of bags, bales, coils of cable, bundles of timber etc, can contain a considerable amount of void spaces between the individual pieces of the stow or between the stow and the ship's structure. The proportion of a cargo stow that is empty void space is its ***Permeability, (symbol 'μ')*** and can be calculated from the cargo's ***Relative Density 'R.D' (i.e. the relative density of the comodity itself)***, and its ***Stowage Factor 'S.F.' (which is the measure of the stow's tightness of packing).***

The void space due to loose packing (known as ***broken stowage***) within a stow of containerised cargo, is mainly sealed inside the containers themselves. Only the gaps between the containers are open to flooding and the total volume of these in any cargo hold is fixed by the ship's construction. Consequently, the compartment's permeability is independent of the cargo carried within closed containers though it will vary with the extent of open framed container cages (for carrying liquids in tanks) that are included in the stow.

RELATIVE DENSITY, STOWAGE FACTOR AND PERMEABILITY



VOLUME OF STOWAGE = X x Y x Z

$$\text{RELATIVE DENSITY 'R.D.'} = \frac{\text{TOTAL SOLID WEIGHT}}{\text{VOLUME OF SOLID WEIGHT}}$$

$$\text{STOWAGE FACTOR 'S.F.'} = \frac{\text{VOLUME OF STOWAGE}}{\text{TOTAL SOLID WEIGHT}}$$

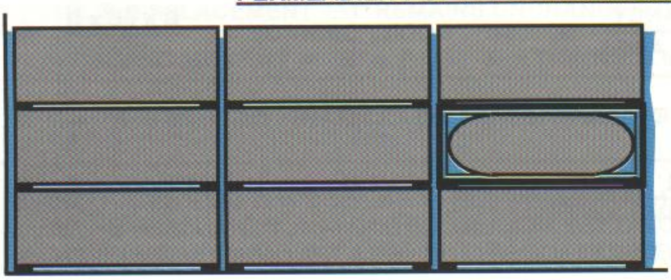
$$\text{So} \quad \frac{1}{\text{RD x SF}} = \frac{\text{VOLUME OF SOLID WEIGHT}}{\text{VOLUME OF STOWAGE}}$$

$$\text{Now} \quad \text{PERMEABILITY '}\mu\text{'} = \frac{\text{VOLUME OF VOIDS}}{\text{VOLUME OF STOWAGE}}$$

$$\boxed{\text{So} \quad \text{PERMEABILITY '}\mu\text{'} = \left(1 - \frac{1}{\text{RD x SF}}\right) \text{ x } 100\ \%}$$

PERMEABILTY MEASURES THE PROPORTION OF THE STOWAGE VOLUME THAT IS EMPTY SPACE. IT CAN BE EXPRESSED AS A PERCENTAGE OR A DECIMAL FACTOR, LESS THAN 1

PERMEABILITY OF A CONTAINER SHIP HOLD SPACE



SIDE VIEW OF PART OF STOW

END VIEW OF PART OF STOW

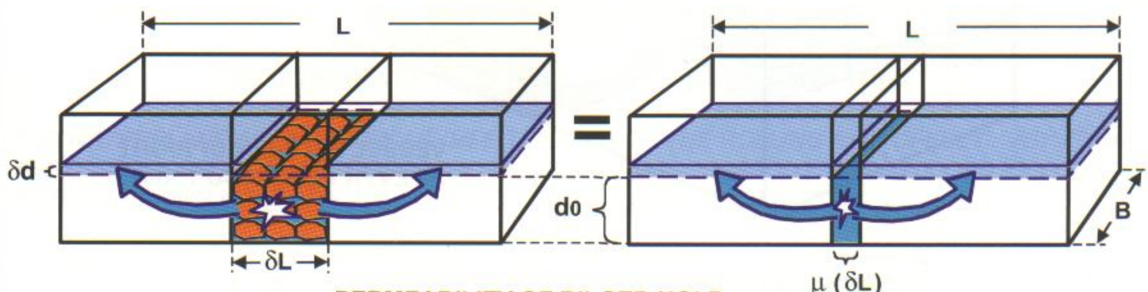
THE SHADED VOID SPACE IS OPEN TO AN INGRESS OF WATER IF THE COMPARTMENT IS BILGED, AND IS LARGELY PREDETERMINED BY THE BUILT IN SPACING BETWEEN THE CONTAINER CELL GUIDES. HOWEVER, IT WILL ALSO VARY IF OPEN CAGE CONTAINER FRAMES ARE INCLUDED IN THE STOW. DATA WHICH ALLOWS THE CALCULATION OF LOST BUOYANCY FOR EACH HOLD IN THE EVENT OF BILGING, SHOULD BE PREPARED AT THE DESIGN STAGE OF THE VESSEL'S CONSTRUCTION.

Determining the permeability of a cargo hold loaded with a mixture of freight is, in reality, quite an imprecise estimation. The actual void spaces will not necessarily be uniformly distributed within the hold, particularly in a mixed open stow of general cargo. Calculating the effects of bilging a partly loaded cargo hold will involve a considerable degree of approximation.

APPLYING PERMEABILITY 'μ' TO BILGING CALCULATIONS

If the permeability, 'μ' of a bilged loaded cargo hold is 20% then only 20% of the hold space beneath the intact waterline will be lost as buoyancy. The remaining 80% is filled with cargo which must be regarded as being part of the undamaged hull space and, as such, continues to provide buoyancy. This may seem strange if the cargo is more dense than water but as bodily sinkage occurs, it will displace water in the same way as the rest of the undamaged vessel and so its horizontal area must be included in the intact waterplane area. If we consider the bilging of a full width midships hold (as shown on page 212) with 20% permeability, then we can calculate the bodily sinkage by reducing the effective length of the hold by 80%. The transverse stability for the bilged vessel can also be calculated from this approximation, as floodwater amongst the cargo is free to move across the full hold width. *(Lost waterplane area is equivalent to free surface effects. See Chapter 4, page 76).*
If, however, we are considering the trim effect of bilging a loaded hold, then the reduced waterplane area should extend over the entire length of the space when calculating the GM_L value.

BILGING OF PART LOADED HOLDS, ALLOWING FOR PERMEABILITY

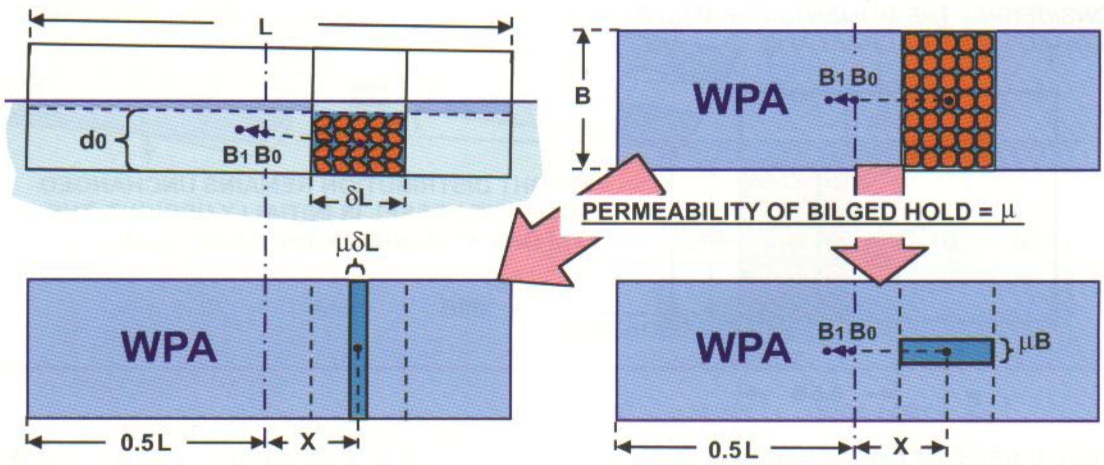


PERMEABILITY OF BILGED HOLD = μ

LOST BUOYANCY = $\mu(\delta L) \times B \times d_0$ M² & BILGED WPA = $B\,[L-\mu(\delta L)]$ M²

$$\text{So SINKAGE } '\delta d' = \frac{\mu(\delta L) \times d_0}{L-\mu(\delta L)} \text{ M}, \quad d_1 = d_0 + \delta d \text{ M} \quad \& \text{ BILGED } 'KB_T' = \frac{1}{2}d_1 + \frac{B^2}{12\,d_1} \text{ M}$$

THE ABOVE EXAMPLE HAS NO CHANGE OF TRIM AND THE LOSS OF WATERPLANE AREA HAS ONLY A NEGLIGIBLE EFFECT ON THE LONGITUDINAL TRIMMING CHARACTERISTICS. IF, HOWEVER, WE CONSIDER THE BILGING OF A LOADED FWD OR AFT HOLD, THEN WE MUST CONSIDER THE TRIM AS WELL AS THE TRANSVERSE STABILITY



TRANSVERSE STABILITY

LOST WPA IS ASSUMED TO BE 'B' BY 'μδL'

$$\text{BILGED } KB_T \text{ VALUE} = \frac{1}{2}d_1 + \frac{B^2}{12\,d_1} \text{ M}$$

LONGITUDINAL STABILITY

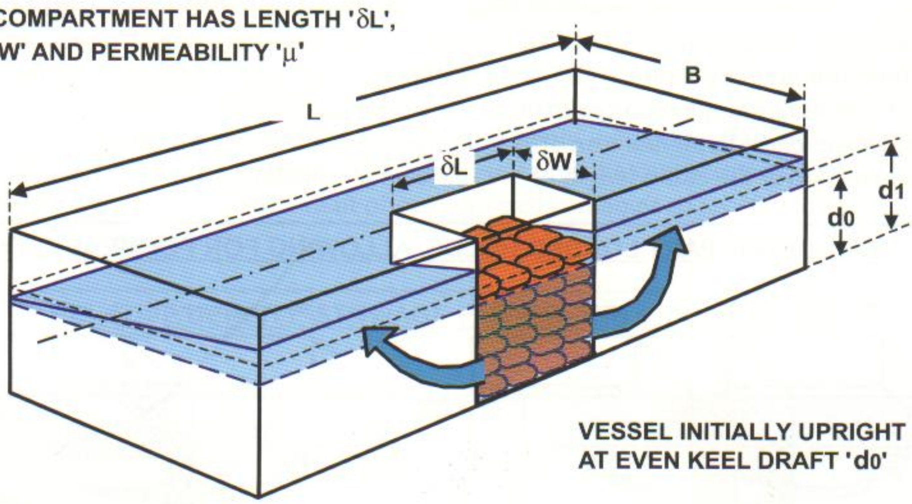
LOST WPA IS ASSUMED TO BE 'μB' BY 'δL'

THE BILGED GM_L VALUE AND, HENCE, THE TRIM ARE DETERMINED BY THE MOMENT OF INERTIA OF THE REDUCED WPA ABOUT 'B_1'

BILGING A FULL DEPTH SIDE HOLD IN A BOX-SHAPED HULL

In this situation, we must calculate the BM value of an asymmetrical waterplane and the list that results from bilging a side compartment. A midships side compartment is probably the most involved calculation that features in certificate of competency examinations for masters and mates.

BILGING A LOADED MIDSHIPS SIDE COMPARTMENT

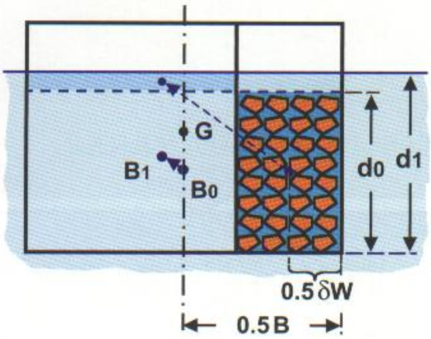


$$\boxed{\text{WATERPLANE AREA 'WPA'} = (B \times L) - \mu\,(\delta W \times \delta L)\ \mathrm{M}^2}$$

$$\text{BODILY SINKAGE} = \frac{\text{LOST BUOYANCY}}{\text{WATERPLANE AREA}}\ \mathrm{M},\quad \text{So BODILY SINKAGE} = \frac{\mu\,(\delta W \times \delta L \times d_0)}{(B \times L) - \mu\,(\delta W \times \delta L)}\ \mathrm{M}$$

$$\boxed{\text{Hence AFTER BILGING,}\qquad \text{MEAN DRAFT 'd}_1\text{'} = d_0 + \frac{\mu\,(\delta W \times \delta L \times d_0)}{(B \times L) - \mu\,(\delta W \times \delta L)}\ \mathrm{M}}$$

THE BILGED WATERPLANE AREA IS CONSTANT THROUGHOUT THE DEPTH OF THE IMMERSED HULL SO THE TRANSVERSE SHIFT IN THE CENTRE OF AREA IS THE SAME AS THAT FOR THE CENTRE OF BUOYANCY. WE CAN DETERMINE THIS SHIFT IN THE CENTRE OF BUOYANCY 'B' BY CONSIDERING THE MOMENTS CREATED BY SUBTRACTING THE BILGED COMPARTMENT FROM AN INTACT HULL AT DRAFT 'd 1'



WEIGHT DISTRIBUTION REMAINS UNCHANGED SO, IF THE VESSEL IS INITIALLY UPRIGHT, THE C of G 'G' REMAINS ON THE CENTRELINE

$$\text{TRANSVERSE SHIFT OF 'B'} = \frac{0.5\,(B - \delta W) \times \mu\,(\delta W \times \delta L \times \cancel{d_1})}{(L \times B \times \cancel{d_1}) - \mu\,(\delta W \times \delta L \times \cancel{d_1})}\ \mathrm{M}\ \ \&\ \ \text{RISE OF 'B'} = 0.5\,(d_1 - d_0)\ \mathrm{M}$$

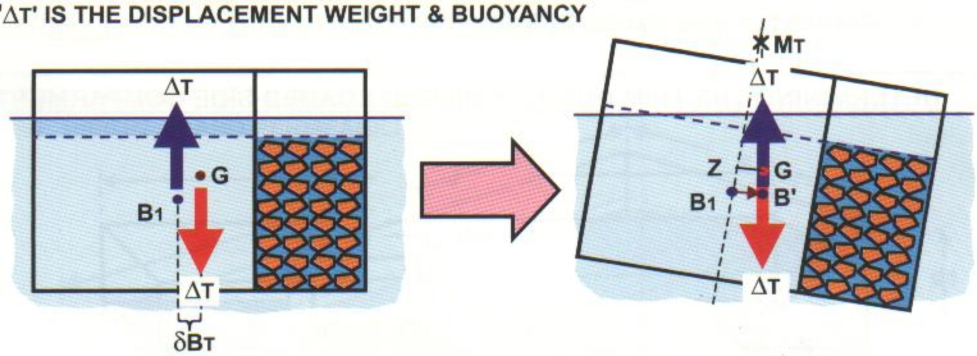
$$\text{So TRANSVERSE SHIFT OF C of B, '}\delta B_T\text{'} = \frac{(B - \delta W) \times \mu\,(\delta W \times \delta L)}{2\,[(L \times B) - \mu\,(\delta W \times \delta L)]}\ \mathrm{M}\quad \&\quad \text{'KB}_T\text{'} = \frac{1}{2} d_1\ \mathrm{M}$$

THE RESULTING CAPSIZING MOMENT = TRANSVERSE SHIFT IN THE C of B x DISPLACEMENT

BILGING A LOADED MIDSHIPS SIDE COMPARTMENT (Cont.)

VESSEL HEELS θ° IN RESPONSE TO C of B BEING OFFSET FROM THE CENTRELINE

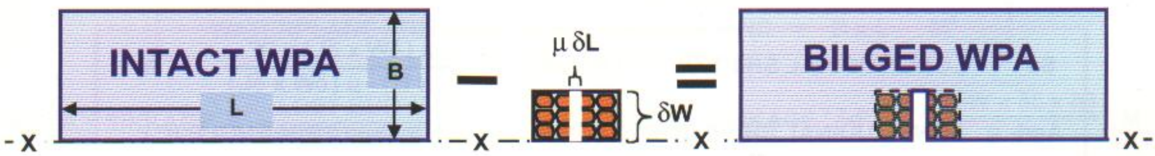
'Δ_T' IS THE DISPLACEMENT WEIGHT & BUOYANCY



CAPSIZING MOMENT = $\delta B_T \times \Delta_T$ & RIGHTING MOMENT = $GM_T \times \tan\theta° \times \Delta_T$

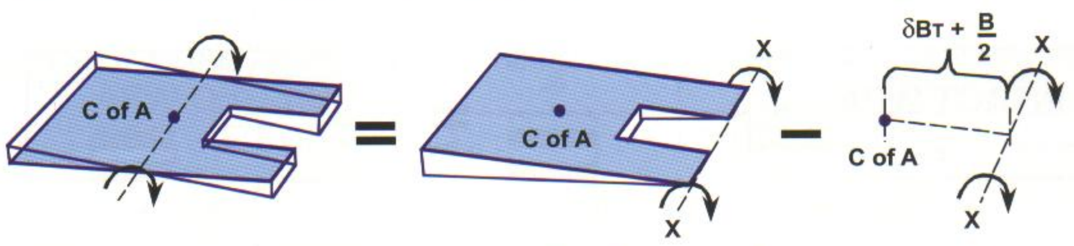
So TRANSVERSE SHIFT OF C of B, 'δB_T' = $GM_T \times \tan\theta°$

THE WATERPLANE AREA IS NOW ASYMMETRICAL WITH THE ROLLING AXIS OFFSET 'δB_T' FROM THE CENTRELINE. THE MOMENT OF INERTIA OF THE WPA 'I_T' ABOUT THIS AXIS CAN BE FOUND BY FIRST TAKING MOMENTS ABOUT THE BILGED WATERPLANE EDGE 'xx'. I.E.



$$I_{XX} = \frac{1}{3} L B^3 \ \ M^4 \qquad I_{XX} = \frac{1}{3} \mu \delta L \delta W^3 \ \ M^4 \qquad I_{XX} = \frac{1}{3}(L B^3 - \mu \delta L \delta W^3) \ \ M^4$$

NOW THAT WE HAVE FOUND THE MOMENT OF INERTIA 'I_{XX}' ABOUT THE BILGED WATERPLANE EDGE 'xx', WE CAN APPLY THE PRINCIPLE OF PARALLEL MOMENTS TO DETERMINE THE MOMENT OF INERTIA 'I_T' ABOUT THE ROLLING AXIS



M'T OF INERTIA ABOUT ROLLING AXIS 'I_T' $= \frac{1}{3}(L B^3 - \mu \delta L \delta W^3) - WPA\left(\delta B_T + \frac{B}{2}\right)^2$ **METRE^4**

And $BM_T = \dfrac{I_T}{\text{DISPLACED VOLUME}}$ **M**

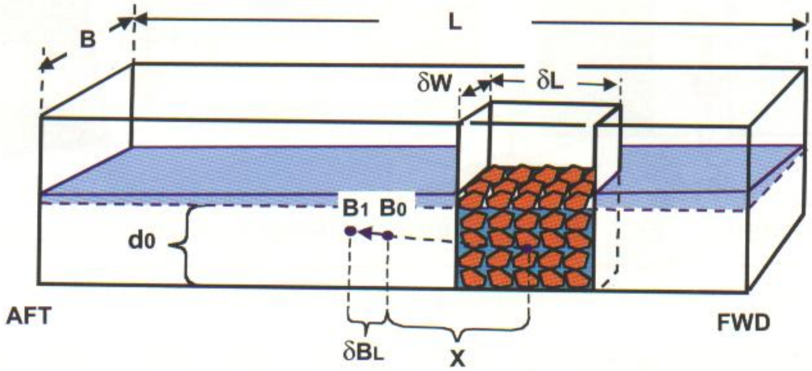
$$GM_L = KB + BM_T - KG, \quad \textit{Where} \quad BM_T = \frac{\frac{1}{3}(L B^3 - \mu \delta L \delta W^3) - WPA\left(\delta B_T + \frac{B}{2}\right)^2}{L \times B \times d_0} \ \ M$$

So LIST 'θ°' IS GIVEN BY:- $\tan\theta° = \dfrac{\text{TRANSVERSE SHIFT OF C of B, '}\delta B_T\text{'}}{KB + BM_T - KG}$

WHERE THE EQUATIONS FOR 'KB', WPA AND 'δB_T' ARE GIVEN ON THE PREVIOUS PAGE

The previous two pages outline how the transverse stability and list can be calculated for a bilged midships side compartment. If, however, the bilged space is towards the fore or aft end, there will be a trimming effect as well as the resulting list. The longitudinal BM value 'BML' approximates to the GML and is determined in much the same way as the transverse value. This then allows the trim angle to be calculated in the same way as the list.

DETERMINING THE TRIM DUE TO A BILGED LOADED SIDE COMPARMENT



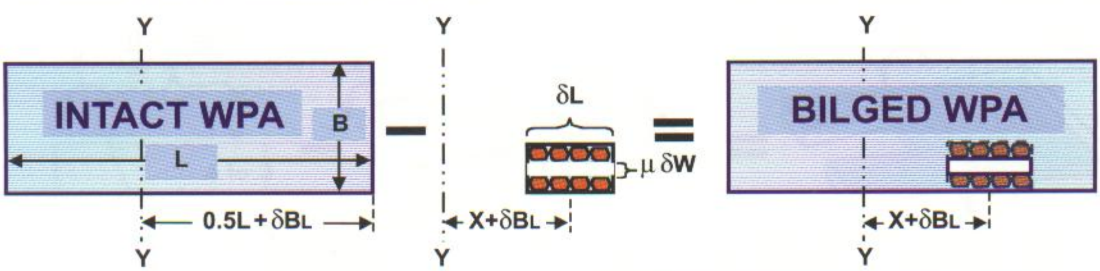
THE MEAN BILGED DRAFT, CHANGE OF TRANSVERSE STABILITY AND LIST ARE CALCULATED AS SHOWN ON THE PREVIOUS TWO PAGES. THE LONGITUDINAL SHIFT OF THE CENTRES OF WPA AREA (THE CENTRE OF FLOATATION) AND BUOYANCY IS FOUND AS FOLLOWS

So LONGITUDINAL SHIFT OF C of B, 'δB_L' $= \dfrac{\mu\,(\delta W \times \delta L) \times \text{'X'}}{(L \times B) - \mu\,(\delta W \times \delta L)}$ M

And LONGITUDINAL SHIFT OF C of B, 'δB_L' $= GM_L \times \tan(\text{TRIM ANGLE } \theta°)$

THE AREAS OF THE BILGED WATERPLANE AND THE LOST WPA DO NOT SHARE A COMMON TRANSVERSE EDGE SO IT MORE CONVENIENT TO TAKE MOMENTS OF INERTIA OF AREAS DIRECTLY ABOUT THE TRIMMING AXIS 'YY', WHICH IS 'δBL' METRES AFT OF MIDSHIPS

THE PRINCIPLE OF PARALLEL MOMENTS MUST BE APPLIED TO EACH RECTANGULAR AREA



$$\frac{B \times L^3}{12} + B \times L \times (\delta B_L)^2 - \left[\frac{\mu \times \delta W \times \delta L^3}{12} + \mu \times \delta W \times \delta L \times (X)^2\right] = \text{M'T OF INERTIA 'I}_{YY}\text{'} \quad M^4$$

THE LONGITUDINAL GM VALUE ≈ 'BML' *WHERE* $\text{'BM}_L\text{'} = \dfrac{I_Y}{\text{DISPLACED VOLUME}}$ M

Now TRIMMING MOMENT $= \Delta_T \times \delta B_L$ & TRIMMING MOMENT $= \Delta_T \times GM_L \times \tan\theta°$

So THE TRIM ANGLE 'θ°' ABOUT THE C of A, IS GIVEN BY:- $\tan\theta° = \dfrac{\delta B_L}{BM_L}$

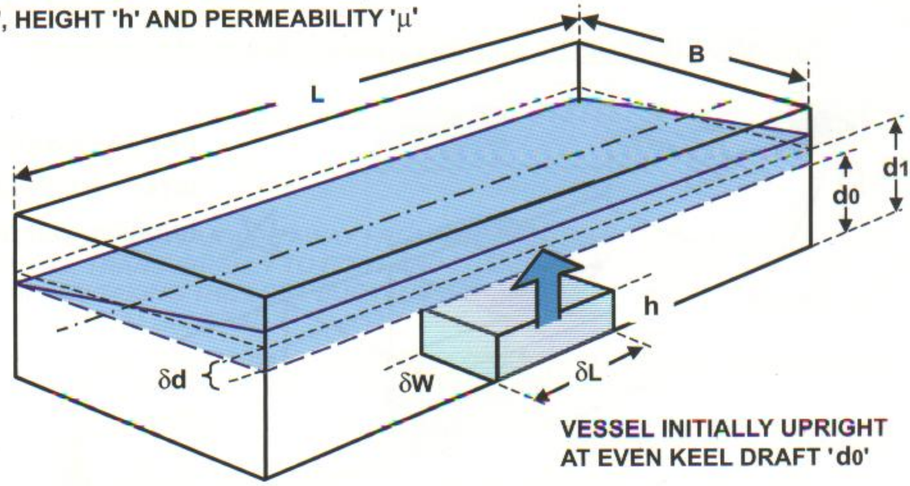
THE BILGED FORE AND AFT DRAFTS CAN THEN BE CALCULATED BY ROTATING THE BILGED WATERLINE, θ° ABOUT THE C of A,,WHICH IS 'δBL' METRES AFT OF THE MIDSHIPS POINT.

BILGING A MIDSHIPS SIDE COMPARMENT WITH A W/T FLAT

This situation is also sometimes set in examination questions. The transfer in buoyancy causes both a vertical and transverse shift in the Centre of Buoyancy, which results in the vessel developing a list. The waterplane area and BM_T remain unchanged by the bilging.

DETERMINING THE LIST FOR A BILGED SIDE COMPARTMENT WITH A W/T FLAT

BILGED COMPARTMENT HAS LENGTH 'δL', WIDTH 'δW', HEIGHT 'h' AND PERMEABILITY 'μ'



VESSEL INITIALLY UPRIGHT AT EVEN KEEL DRAFT 'd_0'

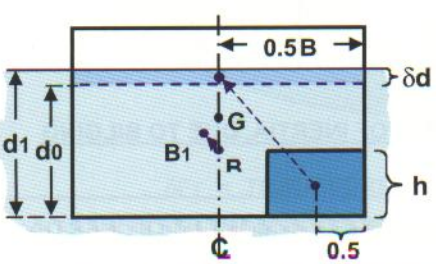
WATERPLANE AREA 'WPA' IS UNCHANGED AND = B x L M^2

BODILY SINKAGE $= \dfrac{\text{LOST BUOYANCY}}{\text{WATERPLANE AREA}}$ **M, So BODILY SINKAGE 'δd'** $= \dfrac{\mu\,(\delta W \times \delta L \times h)}{B \times L}$ **M**

Hence AFTER BILGING, MEAN DRAFT 'd_1' $= d_0 + \dfrac{\mu\,(\delta W \times \delta L \times h)}{B \times L}$ **M**

THE CENTRE OF WATERPLANE AREA (I.E. THE C of F) REMAINS ON THE CENTRLINE AMIDSHIPS AFTER BILGING BUT THE CENTRE OF BUOYANCY MOVES PARALLEL TO THE TRANSFER OF BUOYANCY.

DISPLACED VOLUME x (SHIFT IN C of B) = TRANSFERRED VOLUME x (DISTANCE TRANSFERRED)

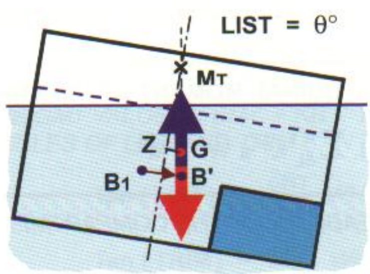


SHIFTS IN C of B

VERTICAL 'δB_V' $= \dfrac{\mu\,(\delta W\ \delta L\ h)\,[d_0 + 0.5(\delta d - h)]}{L\ B\ d_0}$

TRANSVERSE 'δB_T' $= \dfrac{\mu\,(\delta W\ \delta L\ h) \times 0.5\,(B - \delta W)}{L\ B\ d_0}$

THE GM_L INCREASES BY THE RISE OF THE C of B 'δB_V' AND 'δB_T' IS THE HEELING LEVER



$KM_T = KB + BM_T$ *WHERE* $BM_T = \dfrac{L \times B^2}{12\,d_0}$

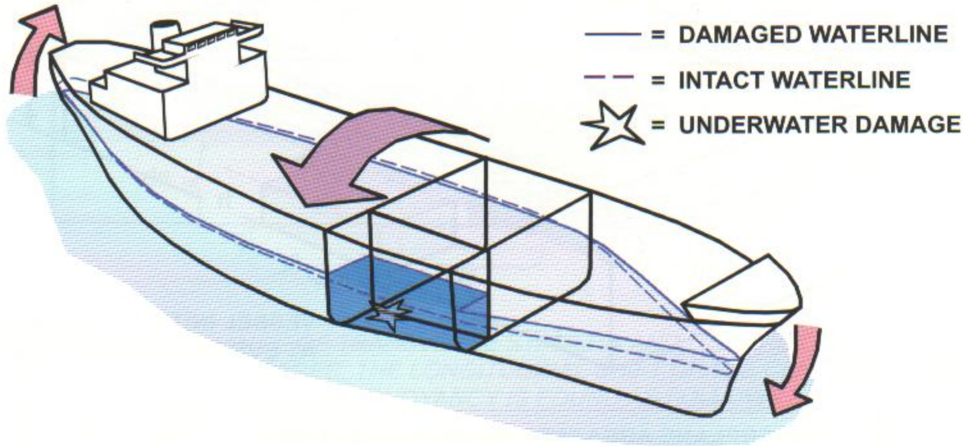
$GM_T = KM_T - KB$ & $KM_T = \dfrac{1}{2} d_0 + {}'\delta B_V{}' + \dfrac{L \times B^2}{12\,d_0}$

THE LIST '$\theta°$' IS GIVEN BY Tan $\theta°$ $= \dfrac{\delta B_T}{GM_T}$

BILGING CALCULATIONS FOR A SHIP-SHAPED HULL

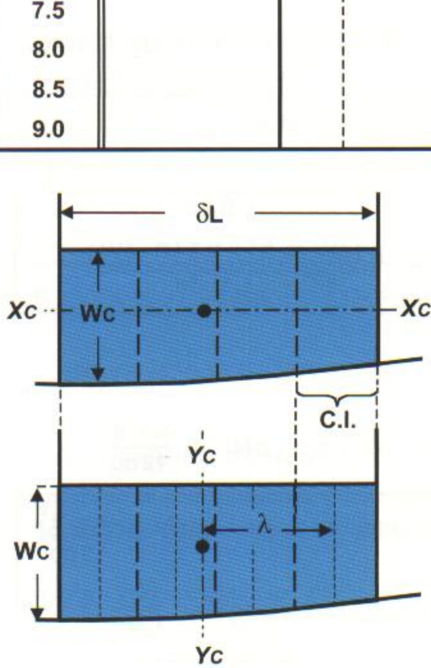
The previous pages have concentrated on the bilging of a box-shaped hull and all the possible situations that are likely to be used in examination questions. Bilging calculations for a real ship-shaped hull require a more thorough analysis of the hull shape and will include data for all the separate compartments when flooded to different drafts, such as volumes, positions of the centres of volume and centres of waterplane area. The transverse and longitudinal Moments of Inertia for the compartment waterplane areas are determined as shown in the following example. These values will also be required to calculate the compartments' free surface effects.

TRIM AND LIST DUE TO BILGING A FWD WING COMPARTMENT



COMPARTMENT TABLE OF DATA

DRAFT (M)	SUBMERGED VOLUME (M^3)	C of B ORDINATES Kb (M)	C of B ORDINATES FROM ℄ (M)	C of B ORDINATES FROM AP (M)	C of F ORDINATES FROM ℄ (M)	C of F ORDINATES FROM AP (M)	M'TS OF WPA INERTIA 'IXCXC' (M^4)	M'TS OF WPA INERTIA 'IYCYC' (M^4)
6.0								
6.5								
7.0								
7.5								
8.0								
8.5								
9.0								



LOST WPA MOMENTS OF INERTIA, DUE TO BILGING

COMPARTMENT TRANSVERSE MOMENT OF INERTIA

$$'I_{XcXc}' = \mu \, \frac{C.I.}{12} \sum_{\delta L}^{0} Wc^3 \qquad M^4$$

COMPARTMENT LONGITUDINAL MOMENT OF INERTIA

$$'I_{YcYc}' = \mu \, C.I. \sum_{\delta L}^{0} \lambda^2 \times Wc \qquad M^4$$

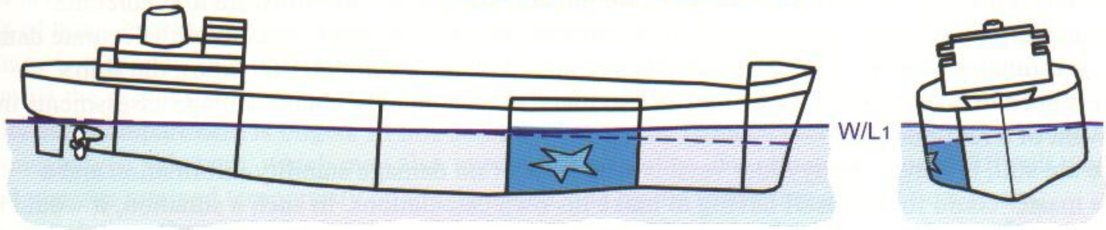
WHERE 'μ' IS THE PERMEABILITY OF THE COMPARTMENT

BILGING CALCULATIONS FOR A SHIP-SHAPED HULL (Cont.)

The damaged condition of the vessel after being bilged can be estimated from the compartment data, (shown on the previous page), in the following way:-

TRIM AND LIST DUE TO BILGING A FWD WING COMPARTMENT (Cont.)

$$\text{BODILY SINKAGE} = \frac{\text{LOST BUOYANCY THROUGH BILGING}}{\text{REDUCED WATERPLANE AREA}} \text{ M}$$



THE CENTRE OF BUOYANCY MOVES AWAY FROM THE CENTRE OF LOST BUOYANCY, SO IT MOVES UPWARDS, TRANSVERSELY TO PORT AND LONGITUDINALLY AFT. THESE SHIFTS CAN BE CALCULATED BY TAKING THE LOST BUOYANCY AWAY FROM THE INTACT HULL AT THE DAMAGED DRAFT, I.E. IN GENERAL :-

$$\text{SHIFT IN THE C of B} = \frac{\text{DISTANCE: CENTRE OF LOST BUOYANCY} \rightarrow \text{C of B INTACT HULL}}{\text{DISPLACED VOLUME}}$$

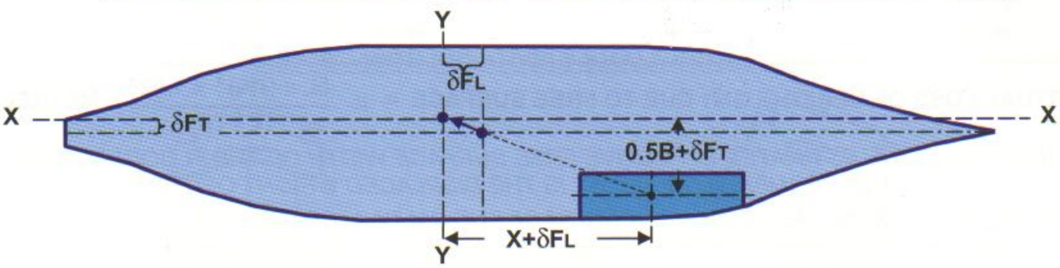
THIS EQUATION CAN BE APPLIED TO FIND THE VERTICAL, TRANSVERSE AND LONGITUDINAL SHIFTS IN THE C of B.
A SIMILAR PROCESS CAN BE USED TO DETERMINE THE TRANSVERSE AND LONGITUDINAL SHIFT IN THE CENTRE OF WATERPLANE AREA (I.E. THE 'C of F') AND, HENCE, LOCATE THE BILGED ROLLING AND TRIMMING AXES

$$\text{SHIFT IN THE C of F} = \frac{\text{DISTANCE: CENTRE OF LOST WPA} \rightarrow \text{C of F FOR INTACT WPA}}{\text{REDUCED WPA AFTER BILGING}}$$

THE TRANSVERSE AND LONGITUDINAL SHIFTS IN THE C of B ACT AS HEELING AND TRIMMING LEVERS ABOUT THE DAMAGED ROLLING AND TRIMMING AXES RESPECTIVELY WHILST THE RISE IN THE C OF B IS AN INCREASE IN THE KB VALUE AND SO AFFECTS THE GM_T VALUE.

THE BM_T AND BM_L MUST BE CALCULATED FOR DAMAGED WATERPLANE AREA BY TAKING MOMENTS OF WPA INERTIA ABOUT THE BILGED ROLLING AND TRIMMING AXES RESPECTIVELY

ROLLING AND TRIMMING AXES FOR THE REDUCED WPA AFTER BILGING



δF_T = TRANSVERSE SHIFT OF C of F FROM INTACT WPA ROLLING AXIS

δF_L = LONGITUDINAL SHIFT OF C of F FROM INTACT WPA TRIMMING AXIS

$$\text{BILGED } 'I_{XX}' = \left[\text{INTACT } 'I_{WPA(T)}' + \text{INTACT WPA} \times (\delta F_T{}^2) \right] - \left['I_{XcXc}' + \text{LOST WPA} \times (0.5B + \delta F_T{}^2) \right]$$

$$\text{BILGED } 'I_{YY}' = \left[\text{INTACT } 'I_{WPA(L)}' + \text{INTACT WPA} \times (\delta F_L{}^2) \right] - \left['I_{YcYc}' + \text{LOST WPA} \times (X + \delta F_L{}^2) \right]$$

$$BM_T = \frac{\text{BILGED } 'I_{XX}'}{\text{DISPLACED VOLUME}} \quad \& \quad BM_L = \frac{\text{BILGED } 'I_{YY}'}{\text{DISPLACED VOLUME}}$$

THE GM_T VALUE = (BM_T, + KB - KG) WHILST THE GM_L VALUE APPROXIMATES THE BM_L. THE ANGLES OF LIST AND TRIM CAN THEN BE CALCULATED AS FOLLOWS

$\text{Tan (LIST)} = \dfrac{\delta B_T}{GM_T}$ *WHERE δB_T IS THE TRANSVERSE SHIFT OF THE C of B*

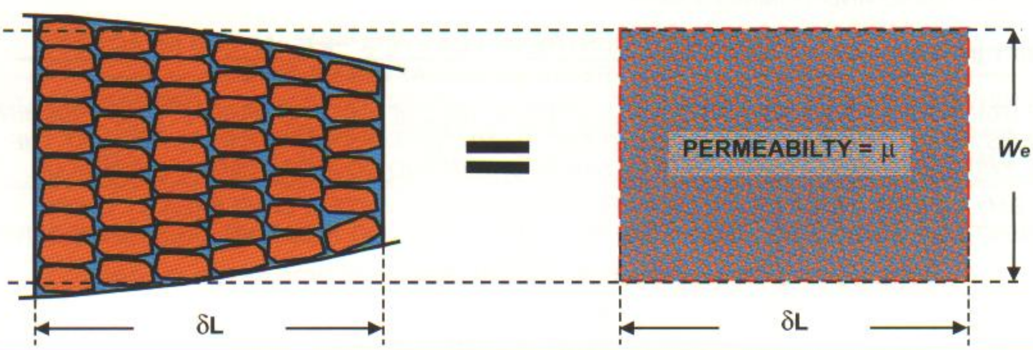
$\text{Tan (TRIM)} = \dfrac{\delta B_L}{GM_L}$ *WHERE δB_L IS THE LONGITUDINAL SHIFT OF THE C of B*

BILGING CALCULATIONS FOR A SHIP-SHAPED HULL (Cont.)

Every merchant ship has internal subdivision and damage assessment information increasingly included in the stability data provided by the shipbuilder. The range of actual loaded conditions, in terms of draft, trim, cargo distribution and the permeability of individual stows is considerable so the design team is likely to restrict bilging calculations to the standard loaded conditions contained in the approved stability book. This, however, should be sufficient for the ship's officers to gain a feel for the degree of damage that the ship is likely to be able to survive over its normal operating range of conditions. Furthermore, dedicated stability computers now contain the software to predict the consequences of bilging for any loaded state, provided that the program contains with accurate data, such as estimates of permeability. If a ship is supplied with such computer facilities, the ship's officers need to become familiar with its use so that they can quickly obtain damage assessments in the event of a real flooding.

There are still, however, many older vessels with little or no damage stability information and a ship's master could find himself having to make his own calculations. In such a situation, it would be quite reasonable to use the 'added weight' method to estimate the damaged draft, list and trim, provided that a good estimate of the weight of floodwater is made and free surface effects are allowed for the flooded compartment. Deck officers will then be able to use the normal ship's hydrostatic data, which they should have become familiar with in carrying out the routine stability calculations. Volumes and positions of centre of volume for all cargo and tank spaces are given in the ship's capacity plans. Free surface effects for dry cargo holds may have to be estimated by approximating their surface area to be rectangular and using the equation derived on page 73 of chapter 4, as shown below:-

THE ESTIMATED FREE SURFACE EFFECT OF THE FLOOD WATER IN A LOADED HOLD



$$\text{VIRTUAL LOSS OF UPRIGHT } GM_T \text{ DUE TO FREE SURFACE} = \mu \frac{\delta L \times (W_e)^3 \times 1.025}{12 \times \Delta T} \text{ METRES}$$

WHERE 'W_e' IS THE ESTIMATED EFFECTIVE FREE SURFACE WIDTH OF THE FLOODED HOLD OF LENGTH 'δL' AND PERMEABILITY 'μ'. 'ΔT' IS THE DISPLACED WEIGHT OF THE DAMAGED VESSEL AND 1.025 IS THE RELATIVE DENSITY OF THE SALTWATER INGRESS

THE EFFECTIVE WIDTH OF THE HOLD IS GREATER THAN SIMPLY THE AVERAGE VALUE AS FREE SURFACE EFFECT VARIES WITH THE CUBE OF WIDTH. IT CAN BE ESTIMATED BY EYE

The free surface effect of the floodwater on the ship's transverse stability when using the 'added weight' method is the equivalent to the loss of waterplane area in the 'lost buoyancy' approach. In both methods the cargo is assumed to remain undisturbed by the ingress of water. However, certain bulk cargoes, such as coal, are liable to shift if the moisture content exceeds a certain limit. (See Chapter 5, pages 111 and 112). This will create an additional capsizing moment that is also based upon the hold area's Moment of Inertia. It can be considered as an extra 'free surface effect' due to the cargo becoming fluid and is likely to be much more significant than that of the flood water as such cargoes are considerably more dense than water.

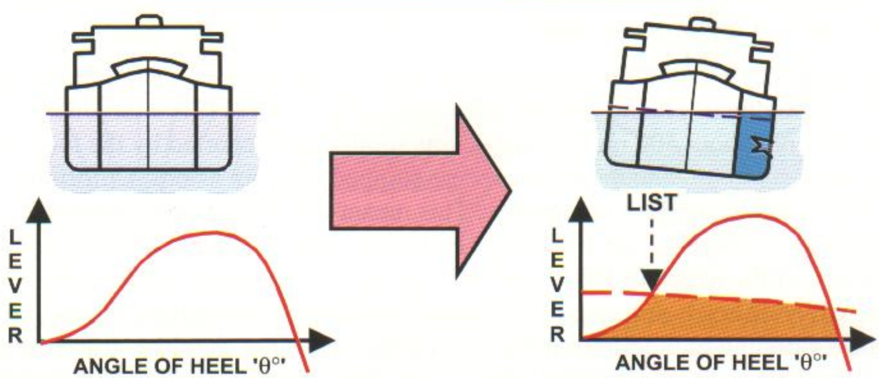
It is important to allow for any shift of cargo that is likely to occur due to the bilging of a cargo space when assessing, by any method, a ship's damaged stability and its survivability.

THE CONSEQUENCES OF ACCIDENTAL FLOODING BY BILGING

Obviously, the flooding of part of a ship's hull is a serious event but whether or not this necessarily leads to the vessel sinking, depends upon the ship's particular circumstances. Bilging the hull always reduces the reserve buoyancy with the bodily sinkage. This alone can lead to the vessel sinking and will reduce the range of positive stability. However, as page 212 shows, the ship's upright transverse stability may be actually enhanced by the flooding as the increase in KB can be greater than the reduction of the BM_T value. This is particularly so when the ship is deeply laden as the draft to beam ratio is relatively large.
Loss of transverse stability is not necessarily a problem but asymmetrical flooding due to side damage is a danger, as it will produce a list that may be fatal to the ship. The following diagrams examine the response of two ships at the same draft but of differing intact GM_T values when an empty side compartment is bilged. In this case, the list can be reduced relatively easily, by ballasting the undamaged side tank, which may also improve the stability. The situation is more difficult when flood water is retained temporarily on the damaged side by the restricting effect of cargo or accommodation. Once the list develops, it becomes progressively harder for the water to drain across the width of the vessel, so the list increases.

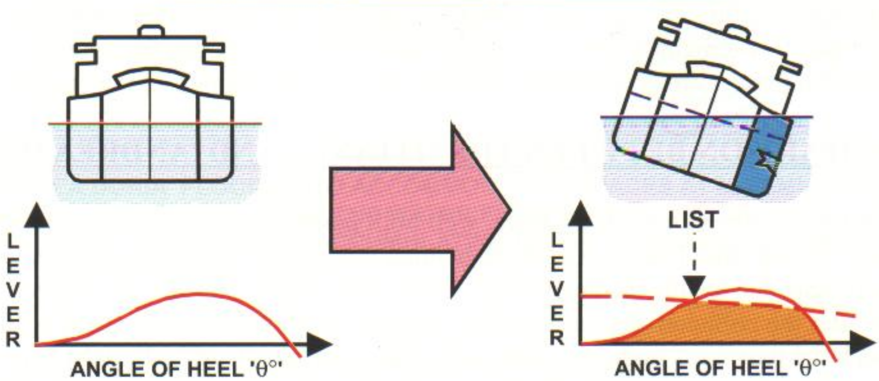
THE EFFECT OF BILGING AN EMPTY SIDE COMPARTMENT

SHIP 'A' - WITH RELATIVELY LARGE INITIAL GM $_T$



BODILY SINKAGE ON BILGING INCREASES THE ALREADY INITIALLY LARGE UPRIGHT GMT VALUE SO THE RESULTING LIST IS COMPARATIVELY SMALL AND THE VESSEL RETAINS AN ACCEPTABLE RANGE OF POSITIVE STABILITY

SHIP 'B' - WITH RELATIVELY SMALL INITIAL GM $_T$



BODILY SINKAGE ON BILGING INCREASES THE SMALL UPRIGHT GMT VALUE BUT THIS IS INSUFFICIENT TO PREVENT THE RESULTING LIST BEING COMPARATIVELY LARGE. THE RANGE OF POSITIVE STABILITY IS DANGEROUSLY SMALL AND THE SHIP IS LIABLE TO ROLL OVER

RAPID WATER TRANSFER BY CROSS FLOODING OR BALLASTING INTO THE STARBOARD SIDE COMPARTMENT WOULD REDUCE SHIP B'S LIST AND INCREASE ITS CHANCES OF SURVIVAL

Most bilging incidents occur either through collision or strandings. Collisions between two ships, in particular, tend to follow the pattern of the bow of one vessel striking the other in its side. Ships generally survive 'head on' damage to the forward end but are much more vulnerable to side damage when heavy listing often leads to the vessel sinking. The classic example of such a collision is the case of two passenger liners, the Italian 'Andrea Doria' and the Swedish 'Stockholm' which collided in the approaches to New York in 1956. The Andrea Doria, which rolled over and eventually sank, was hit in the starboard side by the bow of the Stockholm, which survived with a damaged bow and was able to proceed into port under its own power. Although 44 people died onboard the Italian liner, the loss of life was relatively low as it was carrying 1,134 passengers at the time. This was largely due to the 'Stockholm's' crew carrying out prompt rescue action.
One important feature of this accident was the fact that the 'Andrea Doria' almost immediately after the collision developed a severe starboard list that prevented the lifeboats on that side of the vessel from being launched. This occurred despite the ship having no longitudinal watertight bulkheads. Accommodation, however, is constructed with a lot of internal partitions (cabin and alleyway bulkheads) which, though not strictly watertight, will impede and restrict the free flow of water. If the ingress of water is partly trapped in the vicinity of the hole in the hull, the vessel will heel over towards the damaged side and flood unevenly, which further increases the list.
Effective cross flooding must prevent the accumulation of flood water on the damaged side of the hull by providing the water with a less restricted route so that it can drain downwards and then flow across the width of the ship at a similar rate to the rate of flooding.

Sinking due to progressive longitudinal flooding is less common after collision, though the loss of the British transatlantic liner 'Titanic' due to striking an iceberg in 1911, is a famous example of such a case. An iceberg, only about 500 metres ahead, was spotted and reported by the lookout. The wheel was put hard over to starboard and the engines were stopped then put astern. Unfortunately the ship *almost* managed to miss the iceberg completely. This meant that the collision impact was very slight and damage to the hull was relatively light but it extended along a considerable portion of its length. Since the discovery of the wreck in 1985, it is believed that the damage was probably limited to simply shearing off rivet heads and some local minor buckling of hull plating. It was, however, sufficient to allow flooding in the forward five compartments. If the ship had struck the iceberg at a less oblique angle, the energy of the impact would have been concentrated in a much shorter region of the bow. The damage here would have been very severe and there would have probably been a heavy loss of life amongst those in the forward region (which was the crew's accommodation) but the ship and most of the people onboard would have almost certainly survived. Contrary to popular myth, the 'Titanic' was a well-built ship with a substantial degree of subdivision within its hull. It could have survived flooding in four of the forward compartments but not five. Although the ship sank relatively swiftly, it remained more or less upright in calm conditions for over an hour after the collision. Tragically, there were only enough lifeboats to evacuate about half the people onboard.

A COMPARISON BETWEEN THE 'TITANIC' AND 'ANDREA DORIA'

The 'Titanic' and the 'Andrea Doria' were large, well built passenger ships that were relatively new at the time of their sinking (the 'Titanic' was on its maiden voyage). Both vessels sank as the result of collision in calm seas due to side damage to their hulls. The 'Titanic', unlike the 'Andrea Doria', stayed upright after the collision and remained so right up the point of actually sinking. Comparing the way in which the two ships sank, indicates some important points regarding the flow of water into the hull and why effective cross flooding seemed to have occurred on the 'Titanic' but not the 'Andrea Doria', though neither vessel incorporated purpose built cross flooding facilities into their design.

1) Damage to the 'Titanic' appears to have been in the forward hull, low down close to the keel. The compartments here consisted of store rooms, an alleyway space and, further aft, stoke hold and boiler room, which would be of significant width. All these spaces, with the exception of the store rooms right almost in the bow, would have allowed a relatively free flow of water across the bottom of the ship, so the spaces progressively flooded from the bottom upwards.

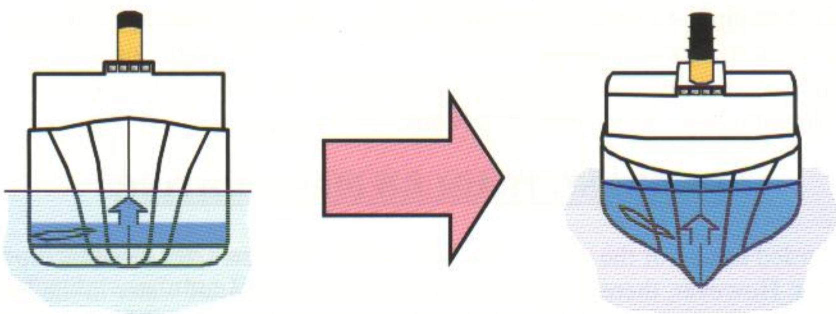
A COMPARISON BETWEEN THE 'TITANIC' AND 'ANDREA DORIA' (Cont.)

2) The 'Andrea Doria' was struck relatively high up in the passenger accommodation region amidships.(The colliding vessel did not have a bulbous bow so the top of its flared fo'c'sle was the first point of contact with the 'Andrea Doria'). Flooding would have occurred through the restricted accommodation spaces and spread from waterline level downwards so, in addition to causing a list, the trapped water relatively high up in the hull would not have increased the ship's stability to the extent that would have occurred in the case of the 'Titanic'.

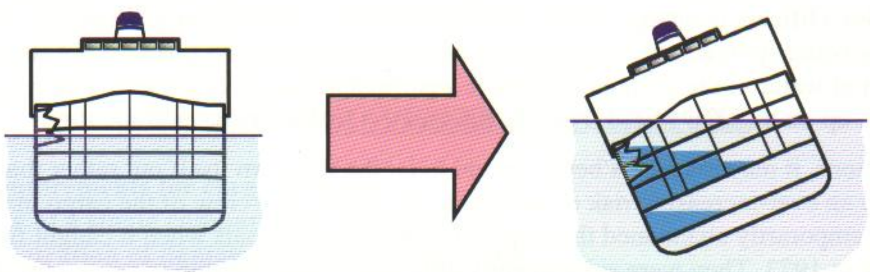
3) The 'Titanic' sank in just over two hours, whereas the 'Andrea Doria' took about twelve hours to actually sink, though it was a smaller ship (697 feet long, compared with the Titanic's length of 882 feet). This suggests that water actually flooded into the 'Titanic' at a much faster rate than in the case of the Italian liner.

Point number '3' demonstrates how a relatively small amount of flooding can have dramatic and severe consequences if it is trapped high in one side of the vessel. In 1914, the cargo and passenger liner 'Empress of Ireland' capsized and sank in 15 minutes after a collision with a Norwegian freighter in the Gulf of St Lawrence and over 1000 people onboard were drowned.

COMPARING THE SINKINGS OF THE TITANIC WITH THE ANDREA DORIA



THE 'TITANIC' SINKS BOW FIRST BY PROGRESSIVE FLOODING FROM FWD TO AFT. RELATIVELY FREE FLOW OF WATER ACROSS THE BOTTOMS OF EACH COMPARTMENT MAINTAIN THE VESSEL'S UPRIGHT CONDITION THROUGHOUT THE FLOODING



THE 'ANDREA DORIA' FLOODS INTO ACCOMMODATION SPACES AT MAXIMUM BEAM AMIDSHIPS. THE RESTRICTION OF THE WATER FLOW ACROSS THE WIDTH OF THE SHIP RESULTS IN FLOOD WATER ACCUMULATING ON THE DAMAGED SIDE, WHICH CAUSES THE SHIP TO ROLL OVER.

In the event of such collisions, it is important that, if possible, the ramming vessel stays imbedded in the hull of the stricken ship hit in the side to 'prop it up' for at least the time needed to evacuate it.

If cross flooding is not being effective at limiting the list of the side damaged ship, then ballasting the undamaged side should be considered. This, however, is not without risk, as floodwater may drain across to the undamaged side of the ship and suddenly produce an even bigger list in the other direction. However, if the pumps are of a sufficient capacity, the list may be reduced to an extent that allows the lifeboats to be launched. Keeping some list to the damaged side should provide some control over the floodwater and prevent the ship suddenly lurching over in the opposite direction.

FLOODING THROUGH HATCHWAYS, VENTS AND OTHER OPENINGS

Holing a ship under the waterline is not the only way in which a ship can be flooded and sink. Heavy seas breaking over the deck have stove in hatchways and flooded cargo holds. As mentioned in Chapter 7, the British bulk carrier 'Derbyshire' sank in a typhoon as a result of the sea breaking into a hatch on the fo'c'sle, which produced a head trim that then led to further progressive flooding.
The 'free trim' effect described in Chapter 3, pages 65 and 66, has lead to several offshore supply vessels foundering in heavy seas after waves had swept over the low aft deck and flooded into exposed engine room uptakes. More stringent design requirements have greatly reduced this particular danger but managing a ship well in heavy weather and ensuring that the vessel is well secured before encountering it, is essential to good seamanship.
Relatively small details can become extremely important when a vessel is in really rough seas. Ships have been wrecked due to flooding through fuel tank vents, which then resulted in the vessel's engines failing. The crew lose control of the ship, which then drifts beam onto the seas. Rolling can increase to violent levels, making repairs more difficult and the ship can be driven aground as happened to the tanker 'Braer' in 1992. (See Chapter 7, page 166)
Not all such flooding occurs in bad weather. The British ro-ro passenger ferry 'Herald of Free Enterprise' rolled over and sank in shallow water just twenty minutes after leaving the berth due to massive flooding onto the car deck through the bow door. Ferry vehicle decks are usually full-length enclosed spaces that extend over the entire width of the ship. If they do flood, there is a massive free surface effect, which is almost certain to capsize the ship. There are now much more stringent requirements regarding the construction and operation of watertight cargo doors in the ship's hull. There are also requirements that enclosed vehicle decks must be fitted with sufficient drainage to clear the deck of free moving water, whether it is due to flooding or fire fighting.

FLOODING THROUGH FIRE FIGHTING

Fire is rightly regarded as one of the greatest hazards at sea and water still remains the most effective means of fighting it. However, there is a problem with releasing a lot of water in the confines of a ship. It can accumulate on one side of the vessel and cause it to progressively heel over until it rolls right over. Fire fighting onboard a ship must always be carried out with this in mind. Excessive use of water can be just as dangerous as the fire itself and every effort must made to ensure that it drains overboard or down to holding tanks in the bottom of the hull at the same rate as it is being used in fighting the fire. Officers in charge of fire fighting must keep a particularly close track on the rate at which water is being sprayed out from fixed sprinkler systems and the spaces it is being directed in. Accumulation of water in the accommodation is a particular danger as it tends to be trapped relatively high up on the ship where it causes the greatest loss of transverse stability.

In some situations, it may even be a better option to let the fire burn itself out rather than continuing to fight it with an ever-increasing risk of capsizing the vessel. The crew of the cargo mail liner 'Good Hope Castle' temporarily abandoned the ship and took to the lifeboats when it caught fire in the South Atlantic in 1973. They were subsequently able to re-board the ship when the fire had finished and it was then towed into port. Even if the ship is totally burnt out, it is still afloat and is likely to be a safer haven than the lifeboats. There is a good chance that food and water will be available and the crew may be able to restore some basic services, such as emergency power to work the radio and some lighting.

Some of the most spectacular incidents of turning a ship over as a consequence of fire fighting have happened in port. The French liner 'Normandie' was being re-fitted as a troopship in New York harbour, when, in 1942, a fire broke out onboard. After twelve hours of the city fire brigade putting water into the fire, the ship rolled over onto its side and grounded, half submerged, on the harbour bottom. It was one of the biggest ships of its time and it remained on its side blocking the pier for the next eighteen months. At the time of the fire, there were 3000 workmen onboard and a considerable amount of fire fighting would have been necessary to evacuate the workforce (which was achieved with the loss of only one life). After achieving this, however, it would have better to have concentrated on preventing the fire spreading ashore and allowing the ship to burn out.

CHAPTER 10

THE 'SOLAS' SUBDIVISION AND DAMAGE STABILITY REQUIREMENTS FOR PASENGER SHIPS AND CARGO VESSELS AND THE 'MARPOL' TANKER SUBDIVISION REGULATIONS

SUMMARY

THIS CHAPTER OUTLINES SUBDIVISION AND DAMAGE STABILITY REGULATIONS FOR :-

1) PASSENGER SHIPS

2) CARGO VESSELS OF 100 METRES IN LENGTH OR MORE.

3) OIL TANKERS.

The contents of this chapter only cover those parts of the regulations that I consider to be mainly concerned with damage stability and trim. The text is not intended to explain every aspect of the rules, which can only be fully understood by studying the regulations themselves

CONTENTS

'SOLAS' PASSENGER SHIP REGULATIONS

'SOLAS' CARGO VESSEL REGULATIONS

'MARPOL' OIL TANKER REGULATIONS

SUBDIVISION REQUIREMENTS FOR PASSENGER SHIPS

There is a long history, since the sinking of the 'Titanic', of governments passing laws requiring passenger ships to meet certain minimum subdivision criteria to improve safety standards.
Part 'B' of Chapter II-1 of The 1974 'International Convention for the Safety of Life at Sea', or ***'SOLAS'*** contains the current regulations that are adopted by U.K. legislation as the 'Passenger Ship Construction Regulations of 1998'. (Statutory Instrument number 'S.I. 2514' & MSN 1698(M))

The ***'Bulkhead Deck'*** is the uppermost continuous deck below which the ship is watertight and the watertight divisions within the hull must extend up to at least this level. **'SOLAS' Regulations 4 to 7** require the longitudinal watertight compartments, beneath the 'Bulkhead Deck' should not normally exceed a prescribed ***'Permissible Length'***, which depends upon a compartment's position in the hull. A line, known as the ***'Margin Line',*** is drawn parallel to and 760 mm below the 'Bulkhead Deck'

The length of hull, in regard to these rules, is expressed as the ***Length of Subdivision***, or ***'Ls'*** and is the maximum moulded length that can be measured at any height beneath the bulkhead deck.

The Permissible Length is derived from the ***'Floodable Length'***, which is a compartment's maximum floodable length that will not immerse the 'Margin Line', when the ship is floating at its ***loaded draft***.
The calculations of 'Floodable Length' use prescribed permeability values for different compartment usage and assume that the ship remains upright when the compartment is flooded.
The rules also give two different equations (one relates to roll-on, roll-off passenger ferries whilst the other is for non ro-ro vessels) for average permeability values of machinery and non-machinery spaces. These can be used as an alternative to applying different permeabilities to each compartment.

The Permissible Length of a compartment is determined by applying a reduction factor, known as the ***'Factor of Subdivision'*** or ***'F'***, to the Floodable Length value, so, for a given position along the hull, the ***Permissible Length = 'F' x Floodable Length.***

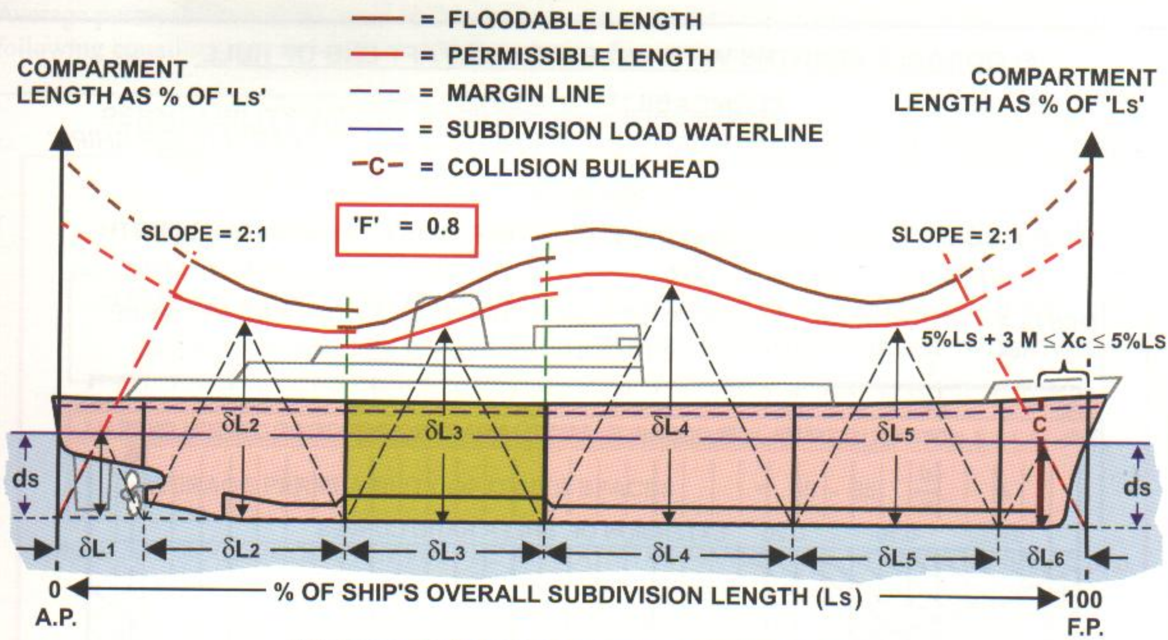
The Factor of Subdivision 'F' is determined by the size of the vessel, the enclosed volume beneath the Bulkhead Deck that is given to passenger accommodation, relative to cargo and machinery, and the total number of passengers carried. This complex process allows for the extent to which different ships carry a varying mix of passengers and freight. If a vessel is almost exclusively a passenger ship, then the 'F' factor will be greater than if it was mainly a cargo ship carrying relatively few passengers and, consequently, the permissible lengths of compartments will be less. This mix of passenger capacity to cargo carrying, is summed up by a ship's assigned ***'Criteria of Service'*** number which ranges from 23, for mainly cargo vessels, to 123, for predominately passenger carrying ships.
Compartments may exceed the permissible length, provided that the ***combined length of any two adjacent compartments is less than either twice the permissible length*** (for ships with an 'F' value less than 0.5) ***or the floodable length*** (for ships with an 'F' value greater than 0.5)***, which ever of these two values is the lesser when measured at the centre of the combined length.***

The MCA' booklet, 'Survey of Passenger Ships, vol. III' contains curves for determining floodable lengths at a particular station along the ship's length for a standard permeability of 60%, from a ship's particulars, such as length, block coefficient, sheer and freeboard to draft ratios.
Curves of 'Floodable' and 'Permissible' lengths are plotted onto the ship's profile. The distinctive '*w*' shape is a consequence of the combined trim and bodily sinkage being greatest around 30% of the ship's length from the bow and stern, so the floodable lengths are shortest in these regions. There will be a discontinuity in the curve where there is a change in permeability, such as the transition from cargo to machinery spaces. The curve effectively is bounded at both ends by a 2:1 sloping line as the floodable length values relate to the position of the compartment centres which must have finite lengths, extending equally fore and aft of their centre point

'SOLAS' Regulation 10 requires a ***collision bulkhead*** to be fitted between ***5% + 3 metres and 5% of the ship's length, aft of the fwd perpendicular***, and may be considered as the first subdivision bulkhead if the vessel is less than 100 metres in length. ***For ships of 100 metres or more in length***, the collision bulkhead is ***additional*** to the bulkhead division derived from these regulations

'SOLAS' Regulation 12 specifies that passenger ships of 76 metres in length or more must be built with a full-length double bottom. A reduced length of double bottom is allowed in shorter vessels.

PASSENGER SHIP FLOODABLE AND PERMISSIBLE COMPARTMENT LENGTHS



KEY TO SPACES UNDER THE FREEBOARD DECK

= MACHINERY SPACE (PERMEABILITY = 85%) = CARGO SPACES (PERMEABILITY = 63%)

(THESE PERMEABILITY VALUES WOULD BE TYPICAL OF A MAINLY CARGO CARRYING VESSEL)

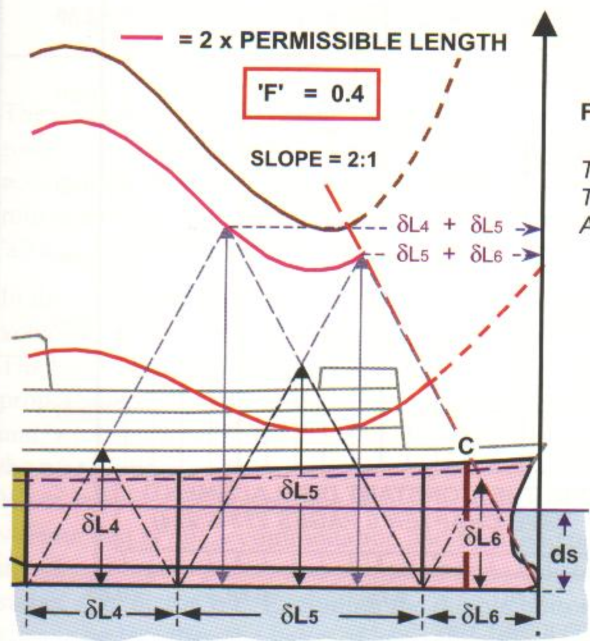
PERMEABILITIES FOR EACH TYPE OF SPACE ARE SPECIFIED BY THE REGULATIONS, OR BY DETERMINING AVERAGE VALUES FOR MACHINERY AND NON MACHINERY SPACES FROM THE APPROPRIATE EQUATIONS GIVEN THE RULES FOR RO-RO AND NON RO-RO VESSELS

PERMISSIBLE COMPARTMENT LENGTH = FLOODABLE LENGTH x SUBDIVISION FACTOR 'F'

THE VESSEL ABOVE HAS THE MAXIMUM PERMISSIBLE LENGTH FOR COMPARTMENTS 2,3,4 & 5

IF A VESSEL'S LENGTH OF SUBDIVISION IS 100 METRES OR MORE, THEN THE COLLISION BULKHEAD IS ADDITIONAL TO THE BULKHEAD SPACING DERIVED FROM THESE RULES

REQUIREMENTS WHEN A COMPARTMENT EXCEEDS THE PERMISSBLE LENGTH



VESSEL HAS A LOW VALUE OF 'F' (I.E. < 0.5)
SO, AT ANY POINT ALONG THE HULL

FLOODABLE LENGTH >2 x PERMISSIBLE LENGTH

THE LENGTH OF COMPARTMENT '5' EXCEEDS THE PERMISSIBLE LENGTH 'P_{L5}' FOR ITS POSITION ALONG THE HULL, SO ;-

LENGTH '$\delta L_4 + \delta L_5$' $\leq 2 \times P_{L45}$

& LENGTH '$\delta L_5 + \delta L_6$' $\leq 2 \times P_{L56}$

THE DIAGRAM OPPOSITE SHOWS MAXIMUM ALLOWABLE LENGTHS OF THE ADJOINING COMPARTMENTS '4' AND '6' FOR THAT PARTICULAR REGION OF THE HULL

= **ACCOMMODATION (PERMEABILITY 95%)**

NOTE THE INCLUSION OF THE PROTRUDING BULBOUS BOW IN THE SUBDIVISION LENGTH

A SAMPLE CURVE FOR DETERMINING FLOODABLE LENGTH

FLOODABLE LENGTHS WITH MID-STATION AT AFT END OF HULL

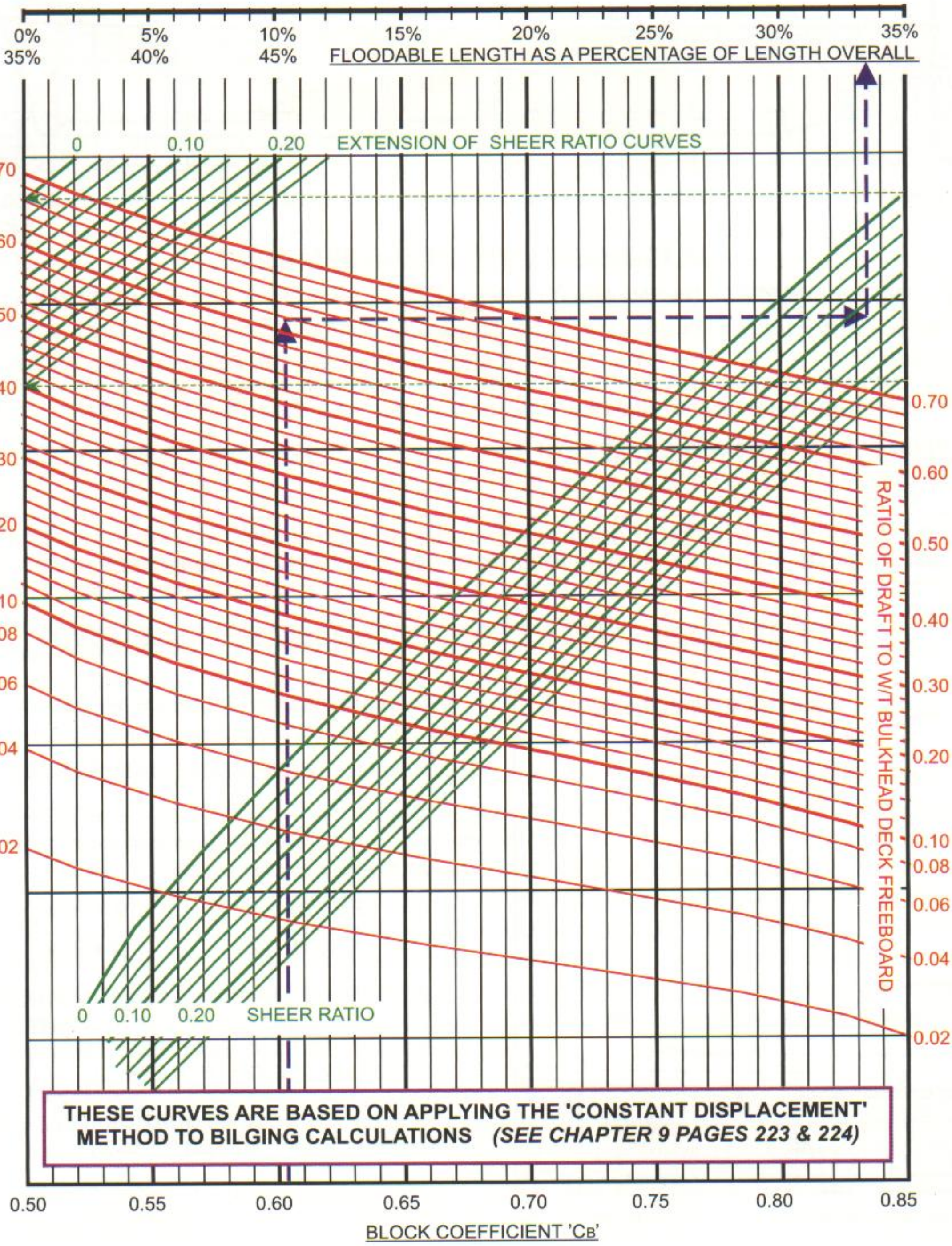
PERMEABILITY 'μ' = 60%

BASED UPON U.K. MARITIME AUTHORITY'S BOOKLET 'SURVEY OF PASSENGER SHIPS'

EXAMPLE

BLOCK COEFF'NT 'CB'	FREEBOARD RATIO	SHEER RATIO	FLOODABLE LENGTH
0.603	0.625	0.125	33.4% of LOA

THIS IS A HYPOTHETICAL LENGTH AS NO COMPARTMENT OF FINITE LENGTH CAN BE CENTRED AT THE AFT TERMINAL POSITION OF THE HULL



PERMEABILITY VALUES USED IN FLOODABLE LENGTH CALCULATIONS

Average permeability values, used in floodable length calculations, can be derived from the following equations, given by **'SOLAS' Regulation 5, paragraphs 2 to 4.**

PERMEABILITY VALUES FOR USE IN FLOODABLE LENGTH CALCULATIONS

FOR ALL PASSENGER VESSELS (SOLAS REGULATION 5 §2.1)

AVERAGE PERMEABILITY OF MACHINERY SPACES $= 85 + 10 \dfrac{(a - c)}{v} \%$

FOR PASSENGER SHIPS OTHER THAN RO-RO VESSELS (SOLAS REGULATION 5 §3)

AVERAGE PERMEABILITY OF OTHER SPACES $= 63 + 35 \dfrac{a}{v} \%$

FOR SHORT HAUL ROLL ON - ROLL OFF FERRIES (SOLAS REGULATION 5 §4.1)

AVERAGE PERMEABILITY OF OTHER SPACES $= 95 - 35 \dfrac{b}{v} \%$

Where 'a' = ACCOMMODATION VOLUME BENEATH THE MARGIN LINE, WITHIN THE SPACE

'b' = VOLUME OF THE SPACE BENEATH THE MARGIN LINE, EXCLUDING THE VOLUME OF ACCOMMODATION, DOUBLE BOTTOM TANKS AND PEAK TANKS

'c' = CARGO AND STORES VOLUME BENEATH THE MARGIN LINE, WITHIN THE MACHINERY SPACE

And 'v' = TOTAL VOLUME OF THE SPACE BENEATH THE MARGIN LINE.

ALTERNATIVELY, THE FOLLOWING TABLE CAN BE USED TO DETERMINE THE PERMEABILITIES OF THE INDIVIDUAL SPACES (SOLAS REGULATION 8 §2.3)

TYPE OF SPACE	PERMEABILITY
STORE ROOMS & CARGO SPACES	60%
MACHINERY SPACES	85%
VEHICULAR CARGO SPACES	85%
ACCOMMODATION SPACES	95%
VOID SPACES AND EMPTY TANKS	95%

ACCOMMODATION INCLUDES GALLEYS, MESSES AND SHOPS FOR CREW AND PASSENGERS

The permeability of a compartment occupied by machinery (engine rooms, steering flats, hydraulics rooms etc.) is considered to be 85% but this is modified by the presence of any cargo, stores or accommodation spaces within the compartment. The average permeability will be reduced if there is more cargo and stores volume than accommodation within the compartment (i.e 'c' is greater than 'a') and increased if the opposite is the case.

In the case of compartments forward or aft of the machinery spaces, the rules distinguish between vehicular roll on -roll off ferries and other types of passenger ships.
The basic permeability of these spaces in non ro-ro vessels is 63% but this is increased by the proportion of any accommodation within the total volume of these parts of the hull (the factors 'a' and 'v'). Permeability for these spaces can reach a maximum value of 98% if all the volume beneath the margin line is accommodation (i.e 'a' = 'c').
When considering these same spaces in ro-ro ferries, the rules consider the proportion of total volume that is *not* accommodation and the basic permeability for such compartments is 95%, but this is *reduced* by the proportion of total volume beneath the margin line that is occupied by cargo or stores. (i.e. factor 'b'). Excluding the double bottom and peak tanks from 'b', ensures that it will always be less than the value of 'c' and that the average permeabilities for these spaces will allow for some of the tank space to be empty (such as used for quick trimming and heeling during loading)

THE CRITERIA OF SERVICE NUMBER 'Cs' FOR PASSENGER SHIPS

The Factor of Subdivision, 'F' is never greater than '1' and **'SOLAS' Regulation 6, paragraph 2** gives two different equations relating 'F' to the length of the ship. One is for type 'A' vessels, which are predominately loaded with cargo and carrying only a small number of passengers in excess of 12 (the limit allowed to be carried on a pure cargo vessel). The other equation is for mainly passenger ships, which are known as type 'B or 'BB' vessels.
The vessel is assigned a Criteria of Service number 'Cs', which is used to interpolate between these two equations to determine the ship's own particular value of 'F'. The greater the number of passengers that the ship carries, the higher is the 'Cs' value and the lower is the value of factor 'F'.

Prior to the days of mass air travel, large numbers of people travelled across the world by passenger ships and many vessels were built with accommodation occupying a large proportion of the space available under the margin deck. 'Cs' was calculated solely on the basis of the proportions of these underdeck accommodation and machinery spaces, relative to the total volume enclosed beneath the margin line. The result was that ships carrying a large number of passengers had a relatively *high 'Cs' value*, which produced a *low subdivision factor* and, subsequently, demanded a *high degree of subdivision* to be built into the vessel. This satisfied the primary objective of the rules to provide minimum standards of damage survivability that *increase* with the number of passengers.
However, passenger ship design has changed significantly in response to the changing demands of the market and an increasing number of vessels concentrated passenger accommodation and facilities above the margin line. This resulted in ships having a relatively low 'Cs' value whilst still carrying a large number of passengers, so **'SOLAS' Regulation 6, paragraph 3.2** allows for a minimum volume, known as ***'P_1'***, beneath the margin line to be considered as accommodation volume in the calculation of 'Cs'. The 'P_1' value depends upon the ship's length and the number of passengers it is certified to carry. If the actual volume of accommodation beneath the margin line (denoted as 'P' in the Rules) exceeds a defined minimum, then the older form of calculating 'Cs' still applies.

CALCULATING THE CRITERIA OF SERVICE NUMBER 'Cs' FOR NON RO-RO SHIPS

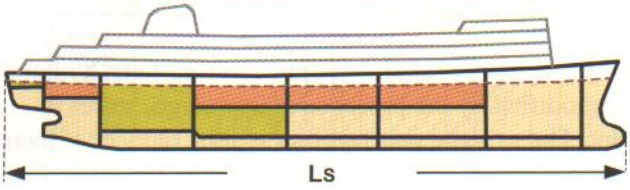
Where 'P' > 0,056 x N x Ls, VESSEL'S SERVICE CRITERIA 'Cs' = 72 $\left[\frac{M + 2P}{V}\right]$

'M', 'P' AND 'V' ARE VOLUME MEASUREMENTS OF SPACES BENEATH THE MARGIN LINE

'M' = MACHINERY SPACE, 'P' = ACCOMMODATION SPACE AND 'V' = TOTAL SPACE

'V' WILL INCLUDE CARGO AND STORES SPACES AS WELL AS 'M' AND 'P' MEASUREMENTS

VOLUMES BENEATH THE MARGIN LINE (SOLAS REGULATION 6 §3.2)



= 'M', MACHINERY SPACE

= 'P', ACCOMMODATION SPACE

= OTHER SPACES

-- = MARGIN LINE

'Cs' VALUES GREATER THAN 123 ARE TAKEN TO = 123 (MAINLY PASSENGER VESSELS)

'Cs' VALUES LESS THAN 23 ARE TAKEN TO = 23 (PREDOMINATELY CARGO SHIPS)

THE MINIMUM ALLOWED ACCOMMODATION VOLUME BENEATH THE MARGIN LINE FOR 'N' NUMBER OF CERTIFIED PASSENGERS, IS KNOWN AS 'P_1'

Where 'P' > 0.056 x N x Ls, VESSEL'S SERVICE CRITERIA 'Cs' = 72 $\left[\frac{M + 2P_1}{V + P_1 - P}\right]$

If 'P' $\leq$ 0.056 x Ls x N but > TOTAL ACCOMMODATION VOLUME WITHIN THE VESSEL

Then 'P_1' = 0.056 x Ls x N

If 'P' $\leq$ TOTAL ACCOMMODATION VOLUME WITHIN THE VESSEL

Then 'P_1' = 0.037 x Ls x N *or* TOTAL ACCOMMODATION VOLUME *WHICHEVER IS GREATER*

THE FACTOR OF SUBDIVISION 'F' FOR PASSENGER SHIPS

The equations, shown on the previous page, show that the volume of accommodation 'P' has twice the effect upon the 'Cs' value as the machinery volume, 'M'. The resulting 'Cs' value for a particular ship can then be used to interpolate between the Factors of Subdivision for type 'A' ('Cs' = 23) and type 'B' or 'BB' ('Cs' = 123) vessels, which are defined by equations that depend upon whether or not the ship is a ro-ro passenger ferry.

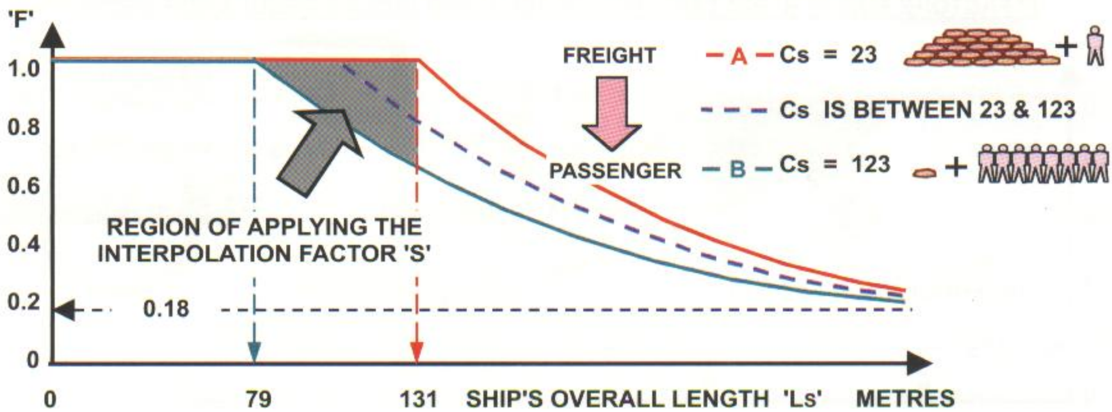
THE FACTOR OF SUBDIVISION 'F' FOR NON RO-RO PASSENGER SHIPS

FOR TYPE 'A' VESSELS, (PREDOMINATELY CARGO) $\quad 'F_A' = \dfrac{58.2}{L - 60} + 0.18$

FOR TYPE 'B' VESSELS, (PREDOMINATELY PASSENGER) $\quad 'F_B' = \dfrac{30.3}{L - 42} + 0.18$

WHERE 'L' IS SHIP'S OVERALL LENGTH (METRES) AND 'FA' & 'FB' HAVE A MAXIMUM VALUE OF 1

'F' FACTORS FOR 'A' & 'B' TYPE PASSENGER SHIPS (SOLAS REGULATION 6 §2.3)



A PARTICULAR VESSEL'S 'F' FACTOR IS DETERMINED BY USING ITS SERVICE CRITERIA 'Cs' TO INTERPOLATE BETWEEN 'FA' & 'FB'. THE INTERPOLATION EQUATIONS USED DEPENDS UPON WHETHER THE LENGTH IS GREATER THAN 131 M OR BETWEEN 79M AND 131 M

FOR VESSELS GREATER THAN 131 M IN LENGTH (SOLAS REGULATION 6 §4.1)

$$\text{FACTOR OF SUBDIVISION } 'F' = F_A - \left[\frac{(F_A - F_B)(C_S - 23)}{100} \right]$$

FOR VESSELS BETWEEN 79 M AND 131 M IN LENGTH (SOLAS REGULATION 6 §4.3)

$$\text{FACTOR OF SUBDIVISION } 'F' = 1 - \left[\frac{(1 - F_B)(C_S - S)}{123 - S} \right] \text{ BUT } 'F' \leq '1'$$

Where THE INTERPOLATION FACTOR 'S' $= \dfrac{3547 - 25L_S}{13}$ *FOR A SHIP'S LENGTH OF 'Ls'*

IF 'Cs' = 23, THEN 'F' = FA & IF 'Cs' = 123, THEN 'F' = FB

THERE ARE SOME EXCEPTIONAL CIRCUMSTANCES WHERE THE VALUE OF 'F' IS ALTERED FOR SPECIFIC LOCAL REGIONS OF THE HULL

LOCALISED VARIATIONS OF THE SUBDIVISION FACTOR (SOLAS REGULATION 6 §4.1 & 2)

1) IF A VESSEL HAS A 'Cs' VALUE EQUAL TO OR GREATER THAN 45 AND A VALUE OF 'F' THAT IS EQUAL TO OR LESS THAN 0.65 BUT GREATER THAN 0.5 THEN A VALUE OF 0.5 MUST BE APPLIED TO THE FORWARD MOST COMPARTMENT IN THE HULL.

2) IF A VESSEL HAS A VALUE OF 'F' LESS THAN 0.4, THEN A FACTOR OF 0.4 CAN BE USED FOR MACHINERY COMPARTMENTS IF IT IS IMPRACTICAL TO INSTALL MACHINERY IN A SHORTER SPACE

THE FACTOR OF SUBDIVISION 'F' FOR PASSENGER SHIPS (Cont.)

The Subdivision Factor for ro-ro passenger ferries is determined in the same manner as shown on the previous page, except that ships with the maximum 'Cs' value of 123 are defined as 'Type BB' vessels and are required to have a greater degree of subdivision than the older 'Type B' classification. The minimum volume allowance for accommodation beneath the margin line ***'P₁'***, is also defined slightly differently from the equations given on page 235, to reflect a different passenger market.

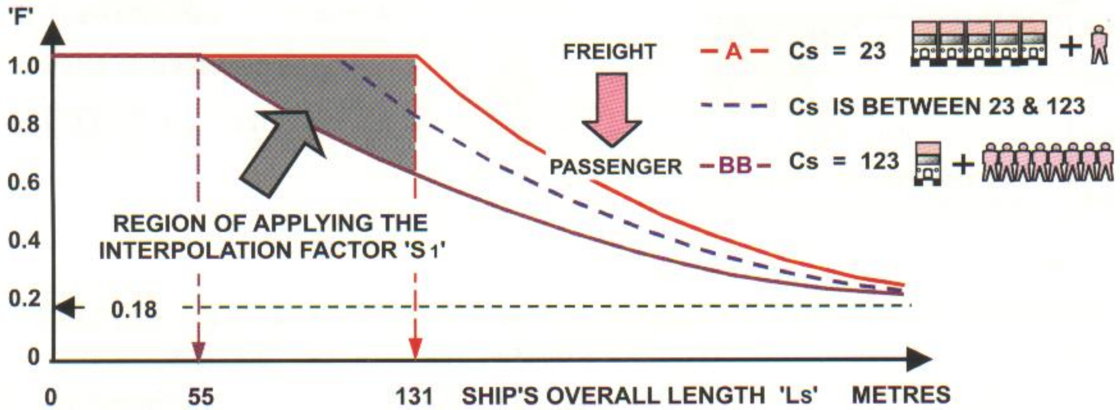
THE FACTOR OF SUBDIVISION 'F' FOR RO-RO PASSENGER FERRIES

FOR TYPE 'A' VESSELS, (PREDOMINATELY CARGO) $\quad 'F_A' = \dfrac{58.2}{L - 60} + 0.18$

FOR TYPE 'B' VESSELS, (PREDOMINATELY PASSENGER) $\quad 'F_{BB}' = \dfrac{17.6}{L - 33} + 0.20$

WHERE 'L' IS SHIP'S OVERALL LENGTH (METRES) AND 'F_A' & 'F_B' HAVE A MAXIMUM VALUE OF 1

'F' FACTORS FOR 'A' & 'BB' TYPE PASSENGER SHIPS (SOLAS REGULATION 6 §5.2)



THE INTERPOLATION FACTOR BETWEEN TYPES 'A' AND 'BB' WILL DIFFER FROM THE PREVIOUS PAGE AND WILL BE APPLIED TO SHIPS BETWEEN 55 M AND 131 M LONG

FOR VESSELS GREATER THAN 131 M IN LENGTH (SOLAS REGULATION 6 §5.2/3)

$$\text{FACTOR OF SUBDIVISION } 'F' = F_A - \left[\frac{(F_A - F_{BB})(C_S - 23)}{100} \right]$$

FOR VESSELS BETWEEN 55 M AND 131 M IN LENGTH (SOLAS REGULATION 6 §5.2/4)

$$\text{FACTOR OF SUBDIVISION } 'F' = 1 - \left[\frac{(1 - F_{BB})(C_S - S_1)}{123 - S_1} \right] \quad \underline{\textit{BUT}}\ 'F' \le '1'$$

Where **THE INTERPOLATION FACTOR 'S_1'** $= \dfrac{3712 - 25L_S}{19}$ *FOR A SHIP'S LENGTH 'L_S'*

IF 'C_S' = 23, THEN 'F' = F_A & IF 'C_S' = 123, THEN 'F' = F_{BB}

IN DETERMINING 'C_S', THE VALUE 'P_1', AS DESCRIBED ON PAGE 235, IS MODIFIED SLIGHTLY TO REFLECT THAT SHORT HAUL FERRIES CARRY UNBERTHED AND BERTHED PASSENGERS

P_1 FOR BERTHED PASSENGERS = 0.056 x N x Ls <u>or</u> 3.5 x N *WHICHEVER IS THE GREATER*

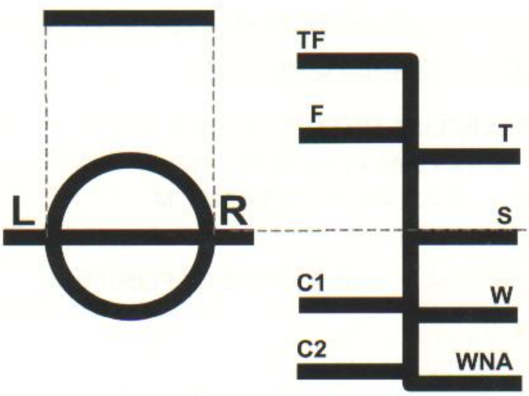
P_1 FOR UNBERTHED PASSENGERS = 3.5 x N *WHERE 'N' IS THE NUMBER OF PASSENGERS*

THE EXCEPTIONAL CIRCUMSTANCES WHERE THE VALUE OF 'F' IS ALTERED FOR SPECIFIC LOCAL REGIONS OF THE HULL (I,E. THE FORWARD MOST COMPARTMENT AND MACHINERY COMPARTMENTS) REMAINS AS FOR ALL OTHER TYPES OF PASSENGER SHIPS

SUBDIVISION LOAD LINES

A subdivision load line indicates the maximum draft to which a ship can be loaded and still comply with Regulations for its particular Criteria of Service Number. **'SOLAS' Regulation 13** requires this to be marked amidships with the letter 'C' on the port and starboard side of every passenger ship beneath the Tropical Mark.
Some passenger vessels are designed for flexible operating schedules in which they may carry varying mixes of passengers and cargo on different voyages and so will have two or more Critical Service Numbers assigned to them. Each 'Cs' value will have a corresponding Subdivision load line, marked on the ship's side and indexed C_1, C_2, C_3, etc (for passenger ships making International voyages) or C_A, C_B, C_C etc (for passenger ships engaged on Home Trade)
When operating at a higher 'Cs' value (i.e carrying a high proportion of passengers, relative to cargo), the ship will require a greater freeboard to meet the subdivision requirements and damage stability criteria than for when it is carrying predominately cargo.

PASSENGER SUBDIVISION LOAD LINE MARKS FOR INTERNATIONAL VOYAGES



MARKS ON STARBOARD SIDE

THE 'C1' MARK ALLOWS FOR A DEEPER DRAFT AND SO RELATES TO A SMALLER OPERATING VALUE OF 'Cs' THAN FOR THE 'C2' MARK

VESSEL SAILS AT A REDUCED PASSENGER CAPACITY WHEN OPERATING TO THE 'C1' MARK

THE 'WNA' MARK INDICATES THAT THIS VESSEL IS LESS THAN 15O METRES IN LENGTH

MINIMUM DAMAGE STABILITY CRITERIA FOR PASSENGER SHIPS

A passenger ship must meet certain minimum damage stability criteria, specified by **'SOLAS' Regulation 8**, in addition to complying with the subdivision requirements explained on the previous pages. The assumed extent of damage is specified by **Regulation 8, paragraph 4** as:-

Damaged Length is 3 + 0.03 x Ls metres, but not greater than 11 metres
Breadth penetration is 0.2 x maximum waterline beam
Vertical extent is unlimited

The length of damage must be progressively moved along the underwater length of the ship to identify all the possible combinations of flooded compartments. At the very least this must include all pairs of adjacent compartments to allow for the possibility of damage spreading across from one side of a transverse bulkhead to the other. However, ***for vessels with a Subdivision Factor 'F' of 0.33 or less, the length of damage should be increased, if necessary, to ensure that three adjacent compartment flooding is considered.*** **('SOLAS' Regulation 8, section 1.4)**

All horizontal divisions within a flooded length, (such as the tank top of the double bottom), must be assumed to be damaged as the vertical extent of damage is considered to be unlimited. However, longitudinal watertight bulkheads can be considered to remain intact if they are further inboard than 20% of the ship's maximum waterline beam. This will reduce the extent of flooding but produce a list that must meet limits specified in the required minimum stability criteria

The **U.K.regulations** require a passenger ship to meet one of two different sets of minimum damage stability criteria, depending upon whether the vessel was built before or after 29th April 1990. The post-1990 U.K. requirements have been incorporated into **'SOLAS'** as **Regulation 8, paragraph 2.3 Regulation 8-1** lays down a time- table for the older vessels to meet these higher standards.

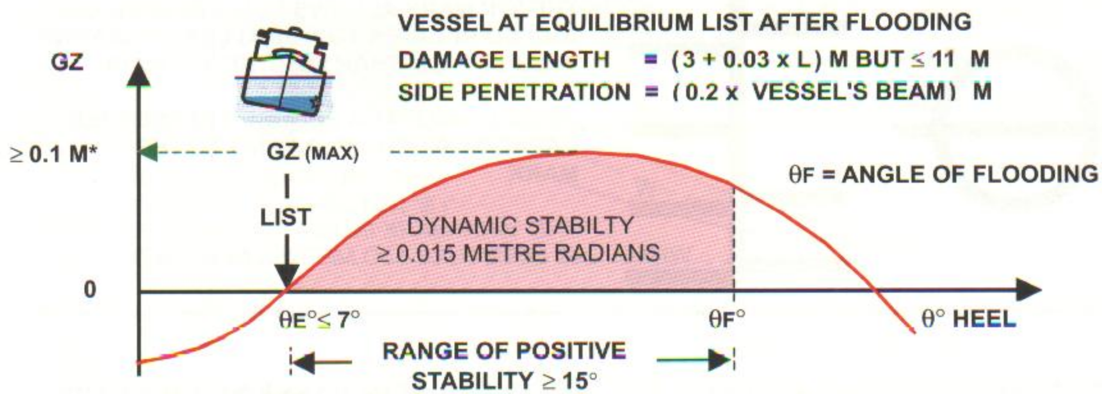
DAMAGE STABILITY CRITERIA FOR PASSENGER SHIPS, INCLUDING RO-RO VESSELS

U.K. VESSELS BUILT PRIOR TO 20TH APRIL 1990

1) AFTER REACHING EQUILIBRIUM THE DAMAGED LIST 'θ_L' SHOULD NOT EXCEED 7°
2) IF THE VESSEL REMAINS UPRIGHT, IT SHOULD HAVE A MINIMUM GM_T OF 0.05 METRES.
3) THE DAMAGED RANGE OF POSITIVE STABILITY SHOULD BE CONSIDERED SUFFICIENT BY THE PROPER AUTHORITIES
4) THE MARGIN LINE MUST NOT BE SUBMERGED IN THE EQUILIBRIUM CONDITION, NOR AT ANY INTERMEDIATE STAGE OF FLOODING IF THE VESSEL IS CARRYING VEHICLES ON THE BULKHEAD DECK.
   HOWEVER, THE MARGIN LINE OF OTHER VESSELS MAY BE IMMERSED AT INTERMEDIATE STAGES OF FLOODING IF THERE IS WATERTIGHT DIVISION ABOVE THE FREEBOARD DECK THAT WILL RESTRICT THE FLOW OF FLOOD WATER AND THE VESSEL DOES NOT HEEL OVER MORE THAN 20°

U.K. VESSELS BUILT ON OR AFTER 20TH APRIL 1990 (ADOPTED AS SOLAS REGULATION 8 §2.3)

MINIMUM GZ CURVE FOR THE VESSEL'S DAMAGED CONDITION



$$* \; GZ_{(MAX)} \geq \frac{\text{HEELING MOMENT (T-M)}}{\text{DISPLACEMENT (T)}} + 0.04 \text{ METRES, BUT MUST} \geq 0.10 \text{ METRES}$$

1) AFTER REACHING EQUILIBRIUM THE DAMAGED LIST 'θ_L' SHOULD NOT EXCEED 7° (FOR ONE COMPARTMENT FLOODING) OR 12° (FOR TWO COMPARTMENTS FLOODING)
2) IF THE VESSEL REMAINS UPRIGHT, IT SHOULD HAVE A MINIMUM GM_T OF 0.05 METRES.
3) THE MARGIN LINE MUST NOT BE SUBMERGED IN THE EQUILIBRIUM CONDITION, NOR AT ANY INTERMEDIATE STAGE OF FLOODING IF THE VESSEL IS CARRYING VEHICLES ON THE BULKHEAD DECK.
   HOWEVER, THE MARGIN LINE OF OTHER VESSELS MAY BE IMMERSED DURING INTERMEDIATE STAGES OF FLOODING IF THERE IS WATERTIGHT DIVISION ABOVE THE FREEBOARD DECK THAT WILL RESTRICT THE FLOW OF FLOOD WATER AND THE VESSEL DOES NOT HEEL OVER MORE THAN 15°
4) THE RESIDUAL DYNAMIC STABILITY MUST EXCEED 0.015 METRE- RADIANS. THIS IS MEASURED FROM θ_E TO EITHER θ_F, <u>OR</u> 22° (FOR ONE COMPARTMENT FLOODING) <u>OR</u> 27° (FOR TWO COMPARTMENTS FLOODING), WHICHEVER OF THESE IS THE LESSER
5) THE RANGE OF DYNAMIC STABILITY MUST NOT BE LESS THAN 7° AT INTERMEDIATE STAGES OF FLOODING AND 15° AT EQUILIBRIUM.
6) THE GZ VALUE MUST EXCEED 0.05 METRES AT INTERMEDIATE STAGES OF FLOODING
7) THE MAXIMUM GZ VALUE MUST NOT BE LESS THAN 0.1 METRES OR THAT GIVEN BY THE ABOVE EQUATION, WHICHEVER IS GREATER. AFTER ALLOWING FOR PASSENGER MOVEMENT TO THE LOW SIDE OF THE SHIP AND <u>EITHER</u> WIND SIDE PRESSURE OF 120 N/M^2 <u>OR</u> FULL DEPLOYMENT OF LIFEBOATS AND DAVIT LIFERAFTS ON ONE SIDE

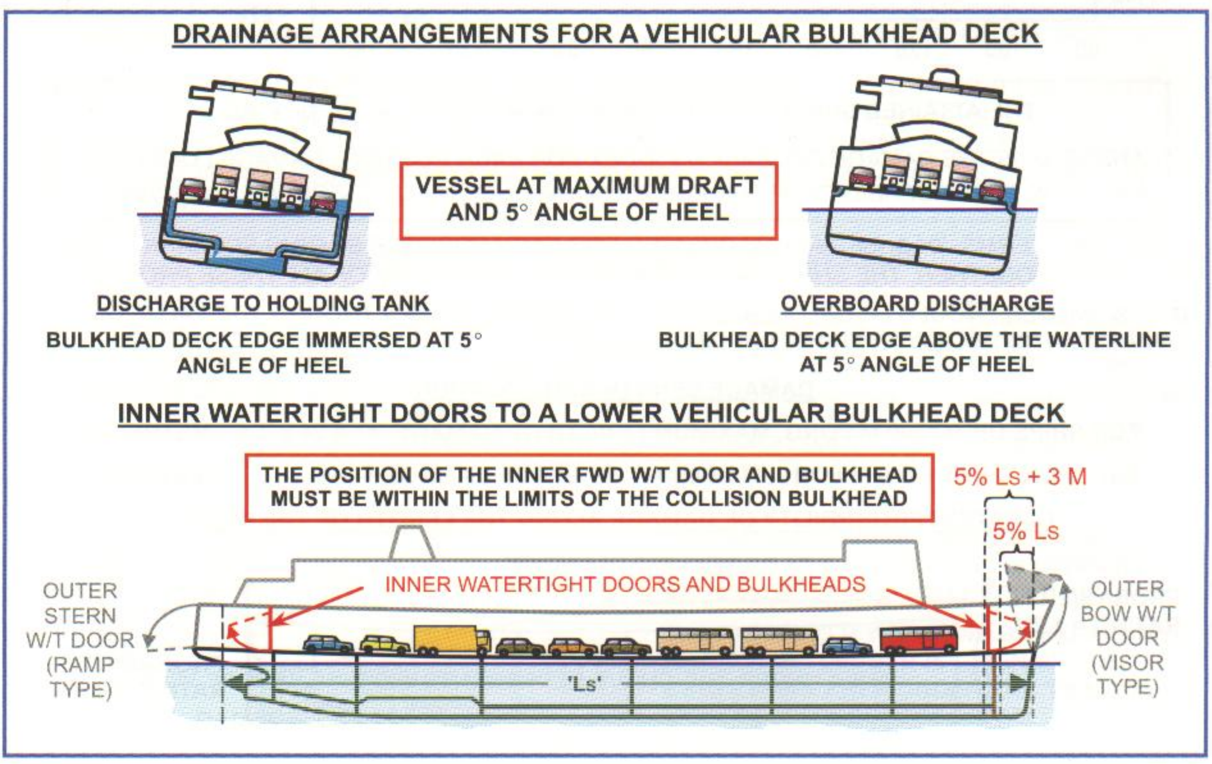
MINIMUM DAMAGE STABILITY CRITERIA FOR PASSENGER SHIPS (Cont.)

Some passenger vessels may require cross flooding systems to meet the Regulations' damage stability requirements and any such arrangements should preferably be self acting but, if not, they must be accessible and operable from above the bulkhead deck. ***Cross flooding must be effective at reducing the list to an acceptable degree within 15 minutes.*** **('SOLAS' Regulation 8 paragraph 5)**

SPECIAL REQUIREMENTS RELATING TO RO-RO VESSELS

Ro-ro ferries are now a common type of passenger vessel and they usually are built to carry vehicles on their bulkhead deck. This is often fully enclosed by the ship's sides being extended upwards for the full length of the hull with watertight doors fitted at the bow and/or the stern to allow the vehicles to drive on and off. The assigned freeboard measured from the bulkhead deck is generally quite small, provided that hatchways and doors leading downwards from such a space are adequately protected by watertight closures, with suitable coamings or sills. However, these vehicle decks are also usually undivided along their entire length as any restriction would reduce the speed at which cars, trucks etc can be parked and secured for what is often a very short sea voyage. This results in a very large free surface area, similar in size to the ship's entire waterplane so the consequences of any significant amount of water being trapped on the car deck are disastrous to the ship's stability. This has been demonstrated dramatically by several passenger ro-ro ferries flooding on the car deck, then rolling over with a large loss of life. In the cases of the 'Herald of Free Enterprise' and the 'Estonia', flooding occurred through the bow doors, but the vehicle deck could also be flooded by a fire activating the ship's sprinkler system.

'SOLAS' Regulation 21, paragraph 1.6 requires that bulkhead decks, used for carrying vehicles, must be equipped with sufficient large drainage scuppers to prevent the accumulation of water in the enclosed space. These scuppers must ***not drain directly overboard if a 5° angle of heel immerses the edge of the bulkhead deck when the vessel is floating at its maximum subdivision waterline.*** Drainage must be directed downwards instead into a holding tank of adequate capacity, built into the ship's bottom. This tank should be fitted with adequate means of pumping the water overboard **Regulation 10, paragraphs 3 & 7** requires that the collision and aft peak tank watertight divisions are extended up into the lower vehicle deck space, where they may be fitted with watertight cargo doors. These extended bulkheads may be stepped, provided they are located within prescribed limits.



SUBDIVISION REQUIREMENTS FOR CARGO VESSELS

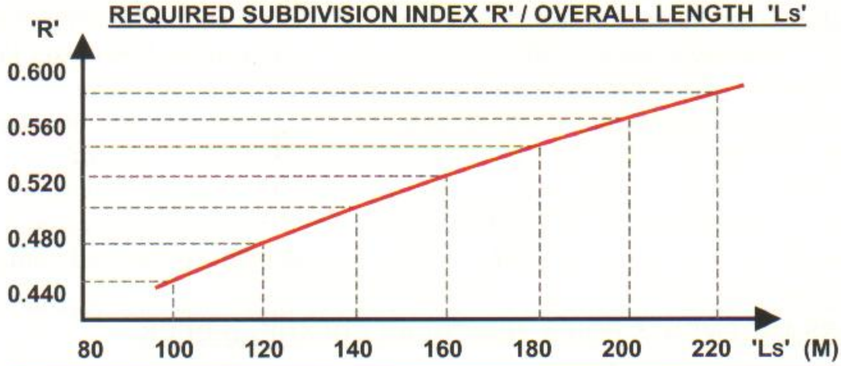
Statutory requirements for bulkhead division in cargo ships, was limited to the provision of the collision bulkhead in the bow region of the hull. The classification societies set criteria with regard to the disposition of other transverse bulkheads, and the Loadline Rules gave consideration to the ship's survivability when assigning the minimum freeboard. However, since 1992, cargo vessels of more than 100 metres in length must comply with ***Regulation 25-1, Chapter II-1, part B-1 of 'SOLAS 1974'***, which gives the subdivision and damage stability requirements for cargo ships.
Unlike the passenger ship subdivision rules outlined in the previous pages, the relatively new cargo ship rules are ***'probabilistic'*** in that they are based upon the probability of each compartment being bilged and the likelyhood of the ship surviving the consequent flooding. The total measure of these probabilities for any given vessel is expressed by the ship's ***'Attained Subdivision Index'*** or ***'A'*** and this must exceed a specified ***'Required Subdivision Index'*** or ***'R'***, based upon the ship's overall length, in order to satisfy the rules.
These rules with explanation and examples, are given in the IMO publication number 871E, entitled **'Explanatory notes to the SOLAS regulations on Subdivision and Damage Stability of Cargo Ships over 100 metres in length'**. This is, however, quite a complex booklet, so the following text only outlines the basic principles of these regulations.

'SOLAS' RULES OF SUBDIVISION FOR CARGO SHIPS OVER 100 M IN LENGTH

THE MINIMUM REQUIRED SUBDIVISION INDEX 'R' FOR A VESSEL = $\sqrt[3]{0.002 + 0.0009\ Ls}$

WHERE 'Ls' IS THE OVERALL LENGTH, IN METRES, OF THE HULL BENEATH THE BULKHEAD DECK

REQUIRED SUBDIVISION INDEX 'R' / OVERALL LENGTH 'Ls'



'Ls' (M)	'R'
100	0.45144
120	0.47914
140	0.50397
160	0.52656
180	0.54737
200	0.56671
220	0.58480

THE ATTAINED SUBDIVISION INDEX 'A' FOR A VESSEL = $\sum$ **'p_i' x 's_i'**

WHERE 'p_i' IS THE DAMAGE PROBABILITY INDEX FOR EACH POSSIBLE DAMAGE LOCATION AND 's_i' IS THE PROBABILITY OF THE VESSEL SURVIVING THE CONSEQUENT FLOODING

THESE TWO TERMS 'p_i' & 's_i' ARE CALCULATED FOR EACH POSSIBLE BILGING SCENARIO, THEN MULTIPLIED TOGETHER. THE SHIP'S VALUE FOR INDEX 'A' IS THE SUM OF ALL THESE PRODUCTS

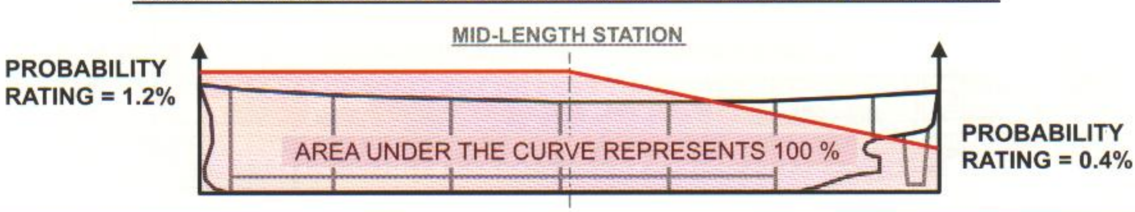
'p_i' & 's_i' VALUES SHOULD BE CALCULATED TO FIVE DECIMAL PLACES THOUGH THE VALUE OF 'A' CAN BE APPROXIMATED TO THREE DECIMAL PLACES

DAMAGE LENGTH AND LOCATION

FOR SHIPS UP TO 200 M LONG, MAXIMUM LENGTH OF DAMAGE = 0.24 x Ls METRES

But FOR SHIPS > 200 M LONG, MAXIMUM LENGTH OF DAMAGE = 48 METRES

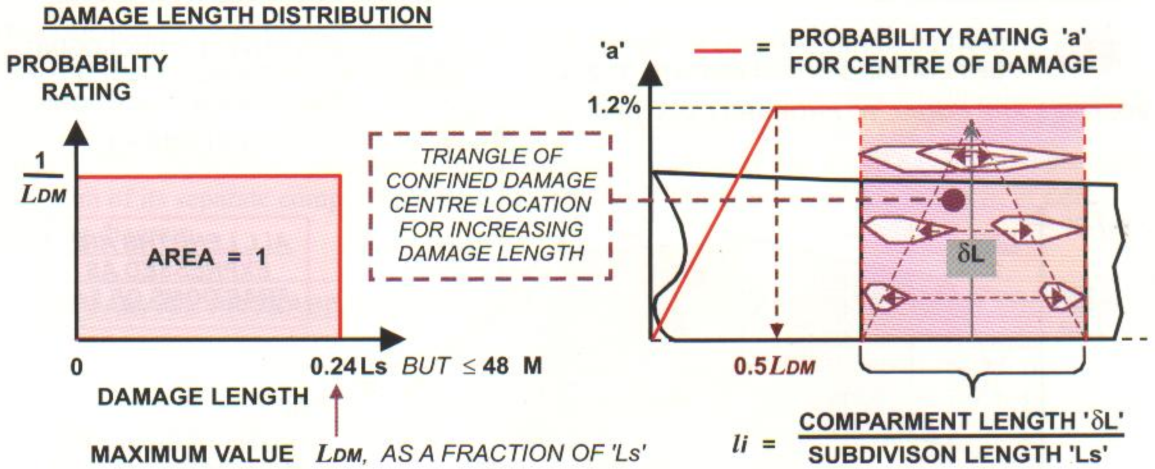
LOCATION PROBABILITY OF DAMAGE ALONG THE LENGTH OF THE HULL



THE PROBABILITY OF A SINGLE COMPARTMENT BEING 'BILGED'

The probability of a space between two transverse bulkheads being flooded would be equal to the area of the location probability curve contained between the two bulkheads. However, the probability of the space being flooded alone, without damage spreading into adjacent compartments, will be less to a degree that depends upon the length of damage. The rules regard this to be random but evenly distributed between zero and 0.24 Ls, where ***'Ls' is the ship's subdivision length.*** (See Page 231)

PROBABILITY OF DAMAGE BEING CONFINED TO A SINGLE SECTION OF HULL LENGTH

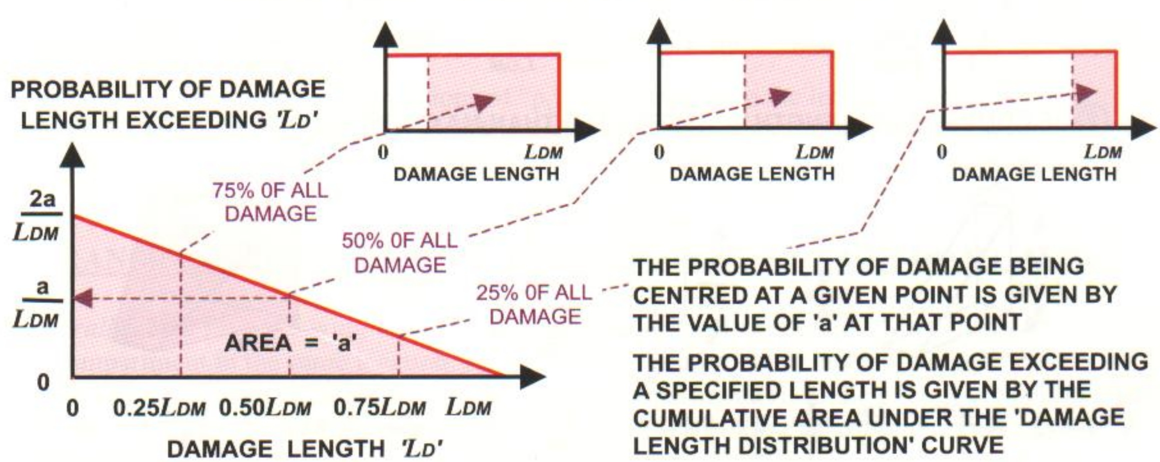


IF DAMAGE HAS OCCURRED AND THERE IS A CERTAINTY OF ITS LENGTH BEING BETWEEN '0' & 0.24Ls, THEN THE AREA UNDER THE DAMAGE LENGTH DISTRIBUTION 'CURVE' IS '1' (I.E. A 100% PROBABILITY THAT DAMAGE LENGTH LIES WITHIN THE RANGE OF THE CURVE). CONSEQUENTLY, THE PROBABILITY RATING FOR ANY PARTICULAR LENGTH OF DAMAGE IN THIS RANGE EQUALS 1/L_{DM}, AS L_{DM} x 1/L_{DM} = 1

THE PROBABILITY OF THE DAMAGE OCCURING BETWEEN TWO BULKHEADS IN THE FWD HALF OF THE HULL LENGTH IS 1.2 x li , WHERE 'li ' IS THE LENGTH BETWEEN THE BULKHEADS, EXPRESSED AS A PROPORTION OF THE HULL'S SUBDIVISION LENGTH. HOWEVER, THE CHANCES OF THE DAMAGE BEING CONFINED TO THIS SPACE DECREASE WITH INCREASING LENGTH OF DAMAGE, AS THE CENTRE OF DAMAGE IS RESTRICTED TO A DIMINISHING REGION OF THE COMPARTMENT LENGTH, (SHOWN BY THE DOTTED TRIANGLE IN THE ABOVE SKETCH)

THE PROBABILITY OF A COMPARTMENT BEING FLOODED ALONE, DEPENDS UPON THE SUM OF THE PROBABILITIES OF DAMAGE BEING CENTRED ON ALL THE LOCATIONS (GIVEN BY THE FACTOR 'a') WITHIN THAT COMPARTMENT AND ALSO THE PROBABILITY OF THE HALF LENGTH OF DAMAGE BEING LESS THAN A LOCATION'S DISTANCE FROM THE NEAREST BULKHEAD THAT DEFINES ONE END OF THE COMPARTMENT.

INVERSE CUMALITIVE DAMAGE LENGTH DISTRIBUTION



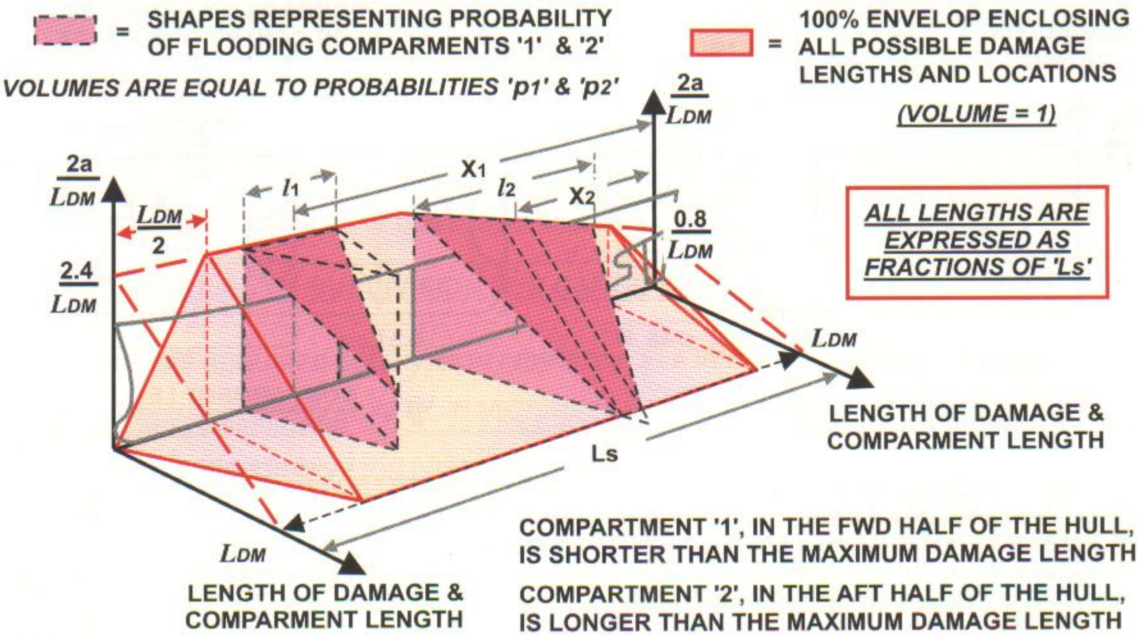
THE PROBABILITY OF DAMAGE BEING CENTRED AT A GIVEN POINT IS GIVEN BY THE VALUE OF 'a' AT THAT POINT

THE PROBABILITY OF DAMAGE EXCEEDING A SPECIFIED LENGTH IS GIVEN BY THE CUMULATIVE AREA UNDER THE 'DAMAGE LENGTH DISTRIBUTION' CURVE

THE PROBABILITY OF A SINGLE COMPARTMENT BEING 'BILGED' (Cont.)

If we take the 'inverse cumulative damage curves, (shown on the bottom of the previous page), for all the points along the ship's length, we build up a three-dimensional graphical shape encompassing all the possible damage lengths and locations. Its volume has a value of '1', representing a 100% probability that damage has occurred. We then plot a 'triangle' of diminishing centre of damage location /damage length for any compartment. This triangle forms the base for the proportion of the total graphical volume that represents the probability of that compartment being flooded alone.

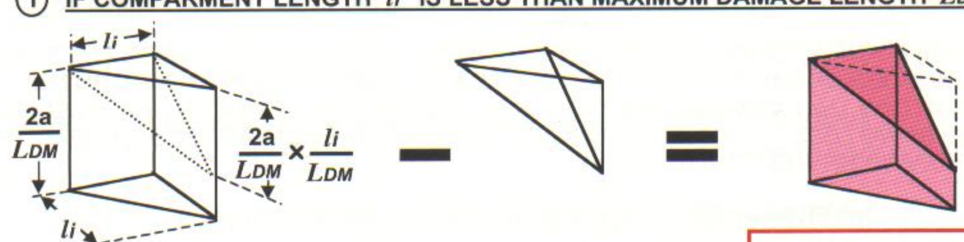
PROBABILITY 'pi' OF FLOODING A SINGLE SECTION OF THE SHIP'S HULL LENGTH



THE PROBABILITY 'pi' OF BILGING A COMPARTMENT, WITHOUT FLOODING ADJACENT SPACES, IS GIVEN BY THE SHADED VOLUMES, MADE UP FROM COMBINATIONS OF PRISMS AND PYRAMIDS

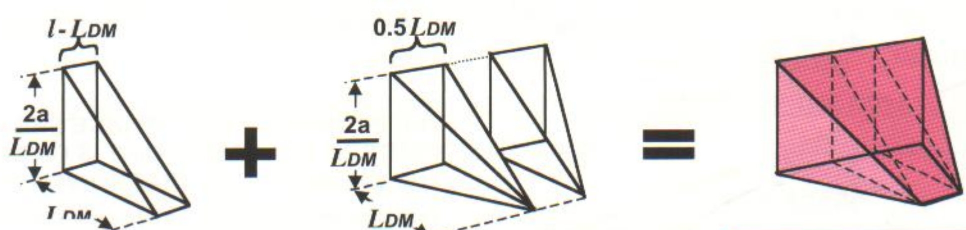
CONSIDER COMPARTMENTS THAT ARE NOT BOUNDED BY THE FWD OR AFT ENDS OF THE HULL

1 IF COMPARMENT LENGTH 'l_i' IS LESS THAN MAXIMUM DAMAGE LENGTH 'L_{DM}'



$$\frac{l_i^2}{2} \times \frac{2a}{L_{DM}} - \frac{l_i^2}{2} \times \frac{1}{3} \times \frac{2a}{L_{DM}} \times \frac{l_i}{L_{DM}} = \boxed{a\left(\frac{l_i^2}{L_{DM}} - \frac{l_i^3}{3L_{DM}}\right) = p_i}$$

2 IF COMPARMENT LENGTH 'l_i' IS GREATER THAN MAXIMUM DAMAGE LENGTH 'L_{DM}'



$$(l_i - L_{DM}) \times \frac{L_{DM}}{2} \times \frac{2a}{L_{DM}} + 2 \times 0.5L_{DM} \times \frac{2a}{L_{DM}} \times \frac{L_{DM}}{3} = \boxed{a\left(l_i - \frac{L_{DM}}{3}\right) = p_i}$$

THE PROBABILITY OF A SINGLE COMPARTMENT BEING 'BILGED' (Cont.)

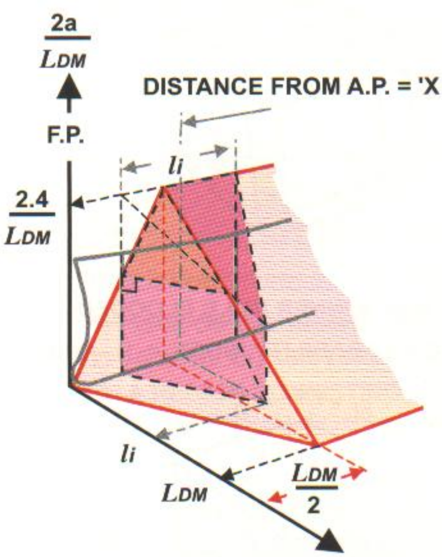
The value of 'a' at any given station along the hull's length, is the probability of the damage centre occurring at that position. For positions *not* within half the maximum damage length 'L_{DM}' of the fore and aft ends of the hull, 'a' is given as follows

$$\text{AT 'Xi' FROM THE A.P. 'a'} = 0.4 + \left(0.8 \times \frac{Xi}{0.5Ls}\right) \quad \underline{BUT} \text{ 'a'} \leq 1.2 \ \underline{\&} \ Ls - 0.5L_{DM} \geq Xi \geq 0.5L_{DM}$$

WHERE 'Xi' & 'L_{DM}' ARE NON-DIMENSIONAL FRACTIONS OF THE SUBDIVISION LENGTH 'Ls'

When calculating the ***damage probability index 'p_i'*** for the compartments shown on the previous page, we can take the value of 'a' for the compartment's midpoint. However, if a compartment either extends across the midship's point or is closer than half the maximum damage length from one end of the hull, then it will include a discontinuity in shape of the 100% 'envelope'. The volume of the resulting three-dimensional probability 'envelope' will still equal 'p_i' but it will have to be calculated as the sum of its separate component prisms and pyramids. An example of this is shown in the following diagram

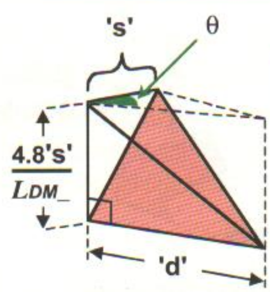
THE DAMAGE PROBABILITY INDEX FOR A TYPICAL COMPARTMENT NEAR THE BOW



LENGTH OF DAMAGE & COMPARMENT LENGTH

THIS IS A FWD COMPARTMENT THAT IS SHORTER THAN THE MAXIMUM LENGTH OF DAMAGE 'L_{DM}' AND CLOSER THAN 0.5L_{DM} FROM THE BOW.

THE THREE DIMENSIONAL PROBABILITY PLOT <u>PRODUCES A SHAPE SUCH AS SHOWN BY EXAMPLE 1 ON THE PREVIOUS PAGE, EXCEPT THAT A 'PYRAMID SHAPED' CORNER IS REMOVED</u>



'CROPPED' PYRAMID

$\theta = 63.43^\circ$ *(FOR 2:1 GRADIENT)*

$$\boxed{\text{'s'} = 0.5L_{DM}\,[\,Ls - (Xi + 0.5li)\,]}$$

$$\text{'d'} = \frac{\text{'s'}}{\cos 63.43^\circ} \quad \textit{OR} \quad 2.236\text{'s'}$$

'δV' IS THE VOLUME OF PYRAMID

$$\text{'}\delta V\text{'} = \frac{4.8\text{'s'}}{L_{DM}^2} \times \frac{\text{'s'}}{2} \times \frac{\text{'d'}}{3} \quad \textit{SO}$$

$$\boxed{\text{'}\delta V\text{'} = \frac{0.8(\text{'s'})^3}{L_{DM}^2} \quad \underline{\textit{IF}} \ \text{'s'} \leq 0.5\,li}$$

$$\boxed{\textit{SO,} \ \text{THE COMPARTMENT FLOODING PROBABILITY 'pi'} = 1.2\left(\frac{li^2}{L_{DM}} - \frac{li^3}{3L_{DM}^2}\right) - \frac{0.8(\text{'s'})^3}{L_{DM}^2}}$$

There are many other possible calculations of 'p_i', depending upon the length and location of a compartment, but the examples shown on this and the previous page should be sufficient to illustrate the principles involved. **'SOLAS' Regulation 25-5 Paragraph 1** gives equations for calculating the damage probability index 'p_i' in a form that should cover most compartments of a vessel.

TRANSVERSE PENETRATION OF DAMAGE IN SIDE COMPARTMENTS

The *overall* probability of a wing tank, or side compartment, being bilged, is the same as for a full width compartment in the same location. However, flooding can either be restricted to the wing space(s) alone, or it can extend into adjacent inboard compartment(s), so the overall probability must be split up into separate probabilities, which account for each of these possibilities. Paragraph '2' of Regulation 25 considers the probability of inboard longitudinal bulkheads being damaged.

TRANSVERSE PENETRATION OF DAMAGE IN SIDE COMPARTMENTS (Cont.)

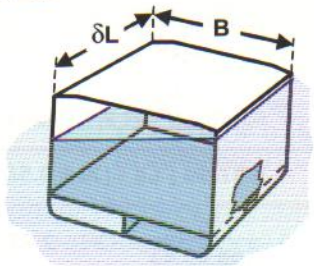
The 'p_i' for a wing tank is calculated in the same way as for a full width space and then multiplied by the ***Reduction Factor 'r'*** to determine its probability of being bilged *without* damage extending into the adjacent inboard spaces. The same basic factor 'p_i' is then multiplied by ***'1- r'*** to determine the probability of flooding the inboard compartments as well. The two probabilities, which relate to the two different damage scenarios, add up to the overall probability factor 'p_i' for that particular section of hull length. The value of 'r' cannot be greater than '1' and depends upon the compartment length and width, relative to the ship's length and beam. (The rules consider short wing compartments to be better stiffened and so more resistant to damage penetration than longer compartments)
The height and vertical extent of damage *below the waterline* is not defined by the regulations. The assumed vertical damage should be such as to cause the least favourable survivability conditions.

THE EFFECT OF LONGITUDINAL BULKHEADS ON THE DAMAGE PROBABILITY INDEX

BILGING SCENARIO FOR COMPARTMENT UNPROTECTED BY WING TANKS

FLOODING PROBABILITY INDEX = p_i

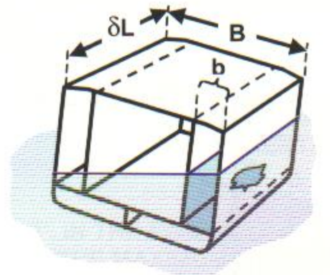
PROBABILITY INDEX 'p_i' OF BEING BILGED INCREASES WITH LENGTH OF THE COMPARTMENT 'δL', RELATIVE TO THE HULL'S OVERALL LENGTH 'L_S' AND THE DISTANCE OF THE HULL'S AFT END TO THE COMPARTMENT'S FWD BULKHEAD



BILGING SCENARIOS FOR COMPARTMENT WITH WING TANK PROTECTION

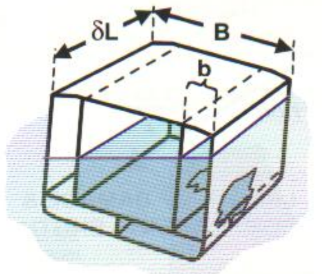
WING TANK ONLY FLOODED

FLOODING PROBABILITY INDEX = r p_i



WING TANK & INNER HOLD FLOODED

FLOODING PROBABILITY INDEX = (r - 1) p_i



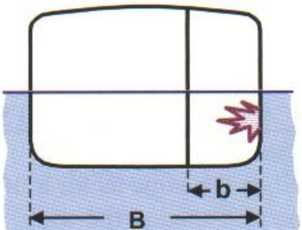
THE OVERALL PROBABILITY OF FLOODING THE WING TANK IS GIVEN BY ITS DAMAGE PROBABILITY INDEX 'p_i'

I.E $\quad r\,p_i + (r - 1)\,p_i = p_i$

IN EACH SCENARIO, WHETHER OR NOT THE DOUBLE BOTTOM TANK IS CONSIDERED ALSO TO FLOOD, DEPENDS UPON WHICH OPTION PRODUCES THE WORST SURVIVABILITY INDEX 'S_i'

CALCULATING VALUES FOR 'r' (PARAGRAPH '2' OF SOLAS REGULATION 25-5)

'r' IS THE PROBABILITY INDEX FOR THE DAMAGE BEING CONTAINED WITHIN THE WING TANK



IF COMPARTMENT LENGTH 'l_i' $\geq 0.2\,\dfrac{b}{B}$

$$'r' = \frac{b}{B}\left(2.3 + \frac{0.080}{l_i + 0.02}\right) + 0.10 \qquad \textit{IF} \quad \frac{b}{B} \leq 0.2 \;.....\; 1$$

$$'r' = \frac{0.016}{l_i + 0.02} + \frac{b}{B} + 0.36 \qquad \textit{IF} \quad \frac{b}{B} > 0.2 \;.....\; 2$$

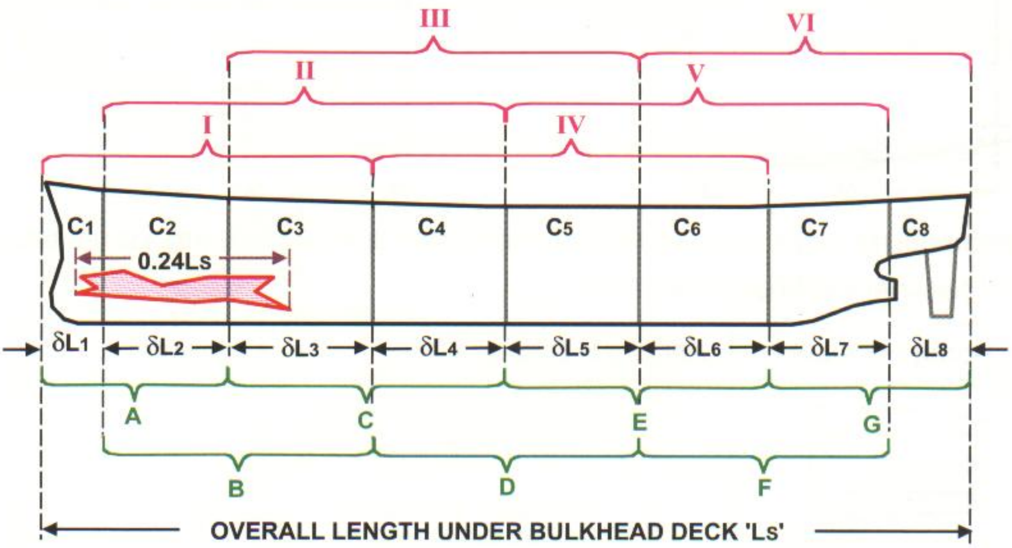
IF COMPARTMENT LENGTH 'l_i' $< 0.2\,\dfrac{b}{B}$

'r' VARIES LINEARLY BETWEEN '1' *IF* l_i = ZERO, & THE VALUE GIVEN BY EQ'N 1 *IF* $l_i = 0.2\,\dfrac{b}{B}$

THE PROBABILITY OF FLOODING TWO OR MORE COMPARTMENTS

The term 'compartment', used in the previous pages, can refer to any section of the hull length that is bounded by watertight bulkheads at either end. As such, it can include further watertight divisions within it and so the same basic set of equations can be applied to groups of adjacent compartments as well as each of the individual watertight compartments that make up the group. A full 'probability of flooding' analysis must account for all the bilging situations (*or damage scenarios*) that can arise from the different possible damage lengths and locations. The following diagram shows how the probabilities of these different scenarios are calculated as the damage is moved along the hull.

LONGITUDINAL DAMAGE PROBABILITY INDEX



A SINGLE PUNCTURE IN THE ABOVE HULL CAN FLOOD ONE OF THE FOLLOWING GROUPS

1) **ONE OF THE INDIVIDUAL COMPARTMENTS I.E. C_1, C_2, C_3 ETC**
2) **A ZONE CONSISTING OF PAIR OF ADJACENT COMPARTMENTS I.E. A, B, C ETC**
3) **A REGION CONSISTING OF THREE ADJACENT COMPARTMENTS I.E. I, II, III ETC**

(EACH INDIVIDUAL COMPARTMENT LENGTH IS LESS THAN THE MAXIMUM LENGTH OF DAMAGE GIVEN AS $0.24L_S$)

THE PROBABILITY INDEX 'p_i' FOR EACH OF THE INDIVIDUAL COMPARTMENTS IS CALCULATED BY APPLYING THE EQUATIONS GIVEN IN SOLAS REGULATION 15-5. §1 THESE RELATE 'p_i' TO THE LENGTH BETWEEN THE BULKHEADS, RELATIVE TO THE HULL'S OVERALL LENGTH, AND THE COMPARTMENT'S POSITION ALONG THE HULL.

THE PROBABILITY OF FLOODING A TWO COMPARTMENT ZONE IS GIVEN AS FOLLOWS

For ZONE 'A' $p_A = P_{12} - (p_1 + p_2)$

I.E. THE PROBABILITY OF BILGING C_1 AND C_2 SIMULTANEOUSLY EQUALS THE PROBABILITY OF THE DAMAGE BEING IN THE COMBINED LENGTH OF $C_1 + C_2$, MINUS THE SUM OF THE PROBABILITIES OF IT BEING CONFINED TO EITHER C_1 OR C_2.

Similarly for ZONE 'B' $p_B = P_{23} - (p_2 + p_3)$ **ETC.**

THE PROBABILITY OF FLOODING A THREE COMPARTMENT REGION IS GIVEN AS FOLLOWS

For REGION 'I' $p_I = P_{123} - (p_{12} + p_{23}) + p_2$

I.E. THE PROBABILITY OF BILGING C_1, C_2 & C_3 SIMULTANEOUSLY EQUALS THE PROBABILITY OF THE DAMAGE BEING IN THE COMBINED LENGTH OF $C_1 + C_2 + C_3$, MINUS THE SUM OF THE PROBABILITIES OF IT BEING CONFINED TO EITHER THE LENGTH $(C_1 + C_2)$ OR $(C_2 + C_3)$ PLUS THE PROBABILITY OF IT OCCURRING IN THE MIDDLE COMPARTMENT C_4.

Similarly for REGION 'II' $p_{II} = P_{234} - (p_{23} + p_{34}) + p_3$ **ETC.**

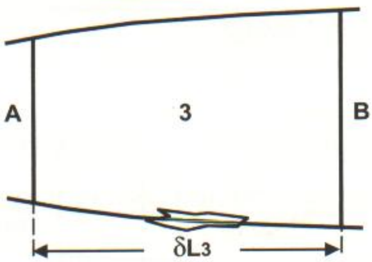
THE SUM OF ALL THE SEPARATE SCENARIO PROBABILITIES IS EQUAL TO '1' AS THERE IS 100% CERTAINTY OF THE DAMAGE BEING SOMEWHERE ALONG THE HULL'S LENGTH

THE PROBABILITY OF FLOODING TWO OR MORE COMPARTMENTS (Cont.)

Wing tank subdivisions often do not coincide with inner compartment bulkheads. Consider, for example, the case of an inner hold space protected by pairs of wing tanks on either side. There are six possible flooding scenarios for the zone contained between the forward and aft hold bulkheads

FLOODING SCENARIO PROBABILITIES FOR A COMPLEX ZONE OF THE HULL

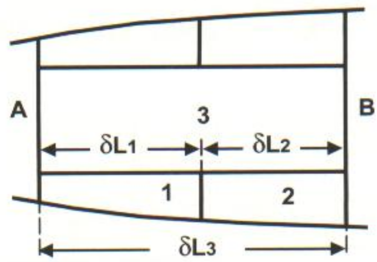
COMPARTMENT UNPROTECTED BY WING TANKS



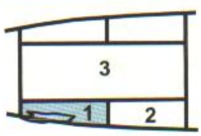
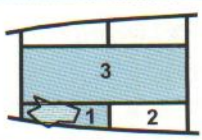
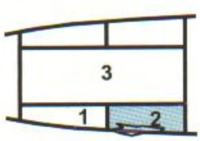
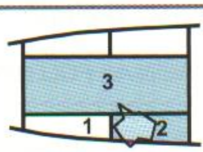
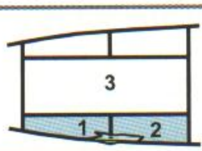
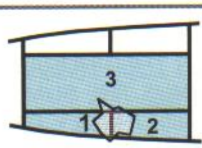
PROBABILITY INDEX = p_3

BASED UPON COMPARTMENT LENGTH 'δL_3'

COMPARTMENT WITH WING TANK PROTECTION



SIDE DAMAGE SCENARIOS ARE LISTED BELOW

FLOODING SCENARIO	SPACES FLOODED	PROBABILITY
	1	$r_1\, p_1$
	1 & 3	$(1 - r_1)\, p_1$
	2	$r_2\, p_2$
	2 & 3	$(1 - r_2)\, p_2$
	1 & 2	$r_{12}\,[p_{12} - (p_2 + p_1)]$
	1, 2 & 3	$(1 - r_{12})\,[p_{12} - (p_2 + p_1)]$

THE SUM OF THE SCENARIO PROBABILITIES = p_{12}

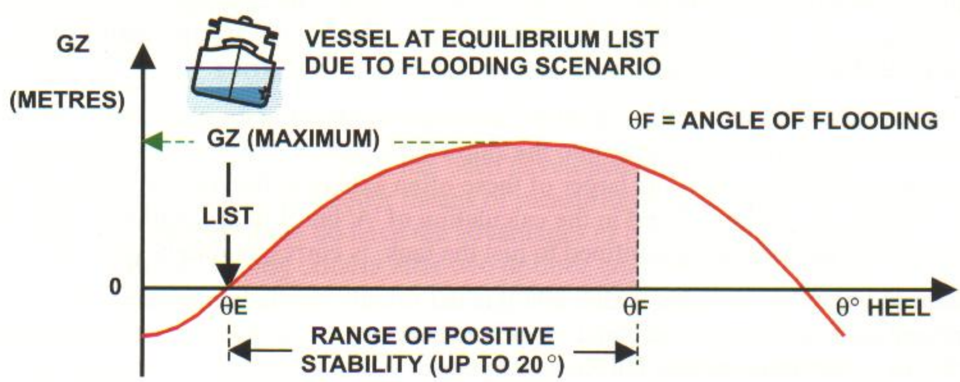
But 'p_{12}' IS BASED ON THE LENGTH 'AB' OR 'δL_3' So 'p_{12}' = p_3

So, SUM OF THE 'p_i' VALUES WITHIN A ZONE = 'p_i' FOR THE ZONE AS A SINGLE COMPARTMENT

THE SURVIVABILITY INDEX FOR CARGO SHIP DAMAGE SCENARIOS

The survivability index 'si' for each flooding scenario is based upon the resulting GZ curve. The means of calculating this are given by **SOLAS Regulation 25, Paragraph 6**.
Paragraph 3 of this Regulation accounts for the probability that flooding will extend into spaces above watertight divisions that are higher than the undamaged waterline, as a consequence of bodily sinkage and heel caused by a flooding scenario. This involves ***'vi'***, the probability index that such a deck will not be breached. 'vi', is based upon the height of a watertight deck above the waterline relative to a considered maximum height of damage. The survivability factor is multiplied by the reducing factor '(1 - vi)' to produce a corrected value 'S (c)'
Survivability factors are determined for the flooding scenarios when the vessel is loaded to its Summer draft and 60% of its Summer draft. ***The average of these two values is the Survivability Index 'si' and is used with the scenario probability factors 'pi' to calculate the ship's Attained Index of Subdivision.***

THE SURVIVABILITY FACTOR 'S i' FOR A GIVEN FLOODING SCENARIO



SCENARIO SURVIVABILITY FACTOR 'S' = $C\sqrt{0.5\,(GZ_{MAX})\,(\text{POSITIVE STABILITY RANGE})}$

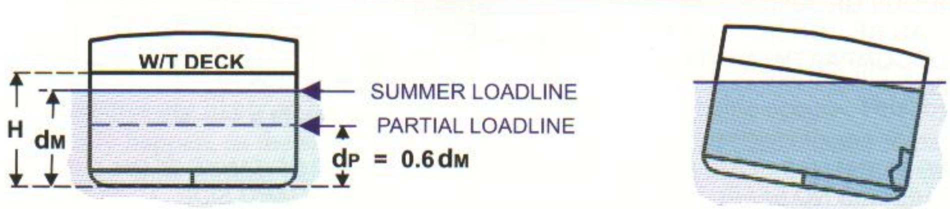
Where THE COEFFICIENT 'C' = $\sqrt{\dfrac{30 - \theta_E}{5}}$ BUT IS NOT GREATER THAN '1'

And THE CONSIDERED MAXIMUM GZ VALUE WILL NOT BE GREATER THAN 0.1 METRES
THE CONSIDERED RANGE OF POSITIVE STABILITY WILL NOT BE GREATER THAN 20°

NOTE THAT 'S' WILL BE ZERO IF 'θ_E' IS GREATER THAN 30° OR THE ANGLE OF FLOODING 'θ_F'

A ZERO VALUE OF 'S' INDICATES THAT THE FLOODING SCENARIO IS LIKELY TO SINK THE SHIP

THE PROBABILITY OF FLOODING EXTENDING UP BEYOND THE UNDAMAGED WATERLINE



PROBABILITY FACTOR OF DECK REMAINING WATERTIGHT 'v_i' = $\dfrac{H - d}{H_{MAX} - d}$

WHERE H_{MAX} IS DEFINED BY THE EQUATIONS IN SOLAS REGULATION 25-6, § 3.3
AND 'd' IS EITHER THE SUMMER LOADLINE DRAFT OR THE PARTIAL LOADLINE DRAFT

THE SURVIVABILITY FACTOR 'S' IS CALCULATED FOR THE VESSEL LOADED TO THE SUMMER AND PARTIAL LOADLINES. IN EACH CASE, 'S' IS MULTIPLIED BY (1 - 'v_i'), TO ACCOUNT FOR THE PROBABILITY OF FLOODING EXTENDING ABOVE THE WATERTIGHT DECK

CORRECTED SCENARIO SURVIVABILITY FACTOR '$S_{(c)}$' = S (1 - 'v_i') FOR EACH DRAFT

Regulation 25-7 states the values of Permeability to be applied to the different types of spaces.

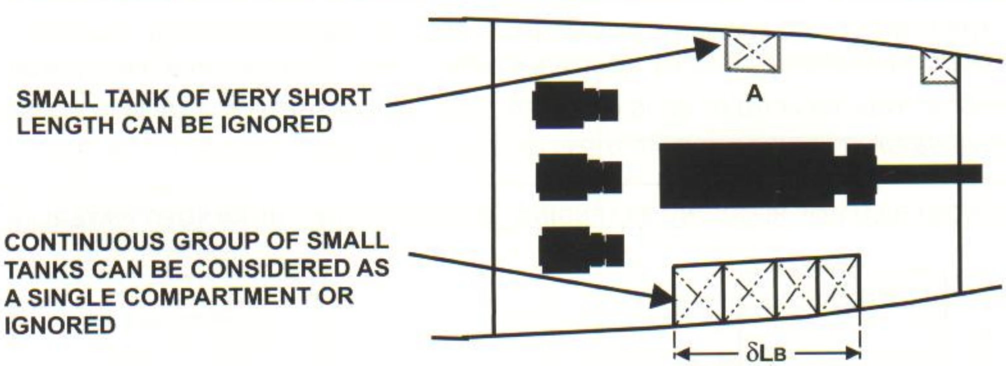
TYPE OF SPACE	PERMEABILITY
STORE ROOM	60%
DRY CARGO SPACES	70%
MACHINERY SPACES	85%
ACCOMMODATION SPACES	95%
VOID SPACES AND EMPTY SPACES	95%

The Attained Index of Subdivision 'A' is the sum of all the '$p_i s_i$' values for the separate flooding scenarios. The situations with a Probability Index, 'p_i', have the greatest influence on the final result and so should have a relatively large Survivability Index 's_i'. It should be appreciated that no scenario makes a *negative* contribution to the overall Attained Index, even if its consequences are disastrous to the ship. Consequently, scenarios with low, or even zero, values of 's_i' are acceptable to the regulations, provided that they also have very low probabilities of occurring.

It is not always necessary to consider every minor watertight division in the ship. In the engine room, there are many relatively small tanks built into the ship's side, such as the settling tanks for the fuel oil, water feed tanks for boilers etc. The larger of these often extend to the full compartment height that they are in and so can be considered in the calculation of 'A' but if their lengths are very short, the probability of side damage being confined to just the tank, is correspondingly very low.

The distribution of watertight division within a hull is not usually totally symmetrical between the port and starboard side, so the Attained Subdivision Index must be calculated for each side and both port and starboard values must exceed the Required Index.
The sum of the probability index for each side of the hull should always be '1', regardless of how the compartments have been subdivided within the zones.

A TYPICAL ENGINE ROOM LAYOUT WITH SMALL AND ASYMMETRICAL TANKS



THE ENGINE ROOM HAS SOME PROTECTION FROM THE GROUP OF TANKS 'B' ON THE PORT SIDE BUT THERE IS NO EFFECTIVE PROTECTION ON THE STARBOARD SIDE., IT WOULD BE ACCEPTABLE TO DISREGARD THE TANKS COMPLETELY AND JUST CONSIDER THE SINGLE SCENARIO OF THE ENGINE ROOM FLOODING IF SIDE DAMAGE OCCURS WITHIN ITS LENGTH

The IMO explanatory notes to the regulations do provide considerable detail as to how compartments should be measured and the way in which flooding scenarios within a zone can be determined when carrying out subdivision calculations.

The ship is to be provided with damage stability information that must include the maximum KG value allowable for any given draft, which ensures that the ship always maintains sufficient survivability, as determined by applying these rules to vessel's design

'SOLAS' SUBDIVISION RULES IN GENERAL

Both the new probabilistic based cargo ship regulations and the older passenger ship subdivision rules concentrate on the possible consequences of damage through collision. The ship's damaged transverse stability is particularly targeted because the greatest risk of life through flooding tends to occur when a ship rolls over very soon after suffering side damage through collision.

Sorting ships into different broad categories (passenger ship, cargo vessel, oil tanker etc.) presents certain problems due to great variety of ship design. For example, the term 'cargo ship' usually includes any vessel that is not a passenger ship or oil tanker. The hull space of a typical cargo ship is largely taken up by the cargo holds, which are totally watertight and segregated from each other. Furthermore, when the ship is fully loaded with minimum freeboard, the holds are closed off and full with cargo, so are relatively low permeability spaces. However, specialised ships such as survey ships, cable layers and dive support vessels are also usually considered by the rules as 'cargo ships', but are, in many aspects, more similar to passenger ships. The hull spaces consist mainly of accommodation, workshops and specialised equipment rooms. It is usually possible to walk from the bow region to the stern at almost any level in the ship and most of the watertight bulkheads have access doorways through them. Consequently, like passenger ships, the enclosed hull spaces generally have a high permeability and the doors in the watertight bulkheads (even though these are built to watertight standards and are normally kept shut at sea) present greater risk of uncontrolled flooding than is the case for true cargo carriers.

The probabilistic approach to subdivision has been developed to account for these wide variations of ship types and is considered to be a better means of measuring a ship's survivability than the older passenger ship rules, though these do have the advantages of being simpler and easier to envisage. A set of minimum damage stability criteria after two or three compartments flooding, is easier to conceive than a minimum subdivision index of, say 0.45. However, the probability approach is likely to be adopted in future regulations and new rules that will combine the probabilistic method with the requirement of minimum damage stability criteria for specified damage conditions, are already being considered in detail for passenger ships.
These proposed passenger ship rules also include, for the first time, comprehensive requirements for cross flooding arrangements.

Probability estimates must be based upon real data, if they are to have any validity. The equations for the probability indices, used in the cargo ship rules, have been determined by analysing 296 collision cases and the data is illustrated in the IMO explanatory booklet. This may appear to be a lot of data but a collision between two ships involves a complex interaction of many factors, such as vessels' sizes, speeds, angle of impact, prevailing weather conditions, their types of construction and their general state of repair. Predicting the probable outcomes of such complicated scenarios requires much more data than would be needed for simpler situations. The wide scatter of the real data points on the various graphs contained in the IMO explanatory notes, tends to suggest that there is not sufficient detailed information for the subdivision index to be any more than a relatively crude estimate of a ship's chances of surviving accidental damage.
Any set of rules involves judging levels of probable risk for different scenarios and the resulting protective measures are likely to be inadequate if the highly improbable occurs. This, by definition, will occasionally happen, as the sinking of the 'Titanic' demonstrated (See chapter 9, pages 227 and 228). In this case, longitudinal damage extended over about the forward 35% of the ship's length but both the existing and proposed regulations do not even consider the possibility of longitudinal damage exceeding 24% of the ship's length. (In fact, for a ship the size of the 'Titanic', the maximum proportion of hull length that the rules allow for damage, is only 18%). The particular way in which the ship struck the iceberg at such a shallow angle caused damage so highly improbable that it is still treated as impossible by the rules. (To be fair, though, a modern all welded hull would not suffer the problem of sheared rivets that seems to have allowed such extensive flooding in the 'Titanic').

There will always be risk in any human endeavour and regulations cannot remove this but the public does not always accept such arguments after a serious accident. The outcry resulting from the loss of the 'Titanic' was so great that it still reverberates around the world, 90 years after it happened.

'MARPOL' SUBDIVISION REQUIREMENTS FOR OIL TANKERS

Oil tankers have always been built with a high degree of subdivision within the hull so that the free surface effect of the oil cargo can be broken up into a considerable number of separate tanks. This produces a strong cellular like structure with openings limited to very small steel watertight access hatches for tank inspection. The risk of a tanker sinking when fully loaded with oil (which is less dense than sea water) was considered to be low and this was reflected in the relatively small minimum freeboard that tankers were allowed to load to. (See Chapter 11 - Freeboard and Loadline assignment). However, in March 1967, the tanker 'Torrey Canyon', loaded with about 120,000T of crude oil, went aground on a reef and broke up near the Scilly Isles at the western approaches to the English Channel. At the time, the resulting oil spillage was the largest ever to have happened (though this dubious record of 'achievement' has been since broken several times) and prompted a series of Inter-government conferences that resulted in the ***'International Convention for the Prevention of Pollution from Ships'***, known as ***MARPOL 1973***, and the ***'International Convention on the Safety of Life at Sea'***, or ***SOLAS 1974***. These two sets of international regulations are updated and amended periodically in response to changes in the shipping industry.

The 'MARPOL' rules have had wide ranging effects upon the construction and operation of tankers with the aim of reducing the risk of oil pollution at sea from these vessels. **Regulation 1 of Annex I of the 1997 edition of MARPOL 73/78** defines the terms used in the rules whilst the hull constructional requirements concerning subdivision and damage control are contained in **Regulations 13, 14, 22, 23, 24 and 25** with supplementary amendments made in 1997 and 1999. These can be briefly summarised as follows:-

All crude oil tankers of 20,000T deadweight or more and products carriers of 30,000T deadweight or more, delivered after 1st June 1982 shall be fitted with dedicated ballast tanks that have sufficient capacity to ensure that the ballast passage drafts meet the following;

1) The amidships ballast draft should not **be less than** 2 metres + 2% of the ship's length.
2) The trim in ballast condition should not **be greater than** 1.5% of the ship's length.
3) The aft draft in ballast condition should be sufficient to **fully immerse** the propeller.

The ballast tanks and their respective piping and pumps must be totally separated from the oil cargo system and should never be used to carry oil. The cargo tanks should not be used for ballast, except when the safety of the vessel is at risk due to severe weather and extra ballast is required. In such circumstances, cargo tanks may be ballasted but the event must be recorded. **(Regulation 13)**

An oil tanker will normally carry about 35% to 40% of its deadweight in ballast when making a ballast passage. Prior to these regulations, it was normal practice to use the cargo tanks for ballast and so, at the end of the passage, there was a considerable amount of oily water to be discharged before loading could start. This problem no longer exists but new vessels must be considerably larger than before for the same cargo carrying capacity as the ballast tanks remain empty on a loaded voyage. Owners of existing tankers were required to select a proportion of the cargo tanks for ballast use only and so reduce the vessel's cargo carrying capacity.

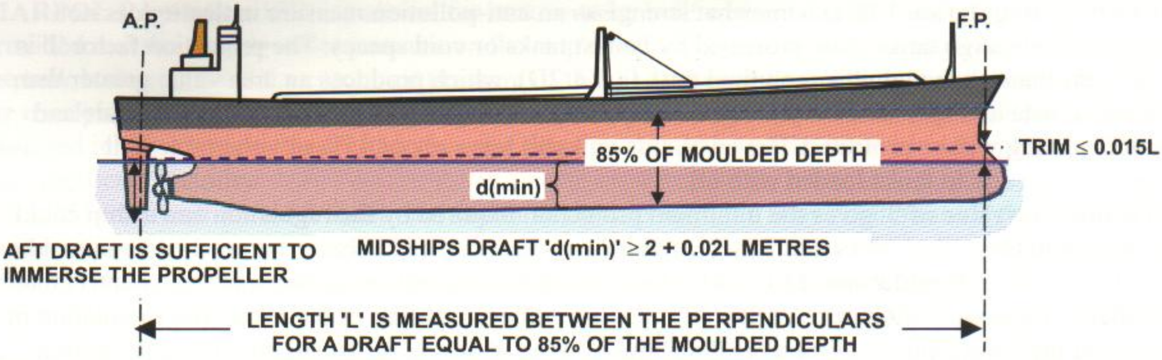
The ballast tanks are to be distributed about the hull to protect the cargo tanks from damage in the event of stranding or collision.

*Initially, this rule **(Regulation 13E)** required that only a proportion of side and bottom plating, based upon the ship's deadweight, was to be protected by the ballast tanks. However, **Regulation 13F** requires that all tankers of 5,000T deadweight or more, delivered after 6th July 1996, should have double bottom and wing ballast tanks protecting the entire length of cargo tank space in the hull. Smaller tankers delivered after this date are to be provided at least with protective double bottom ballast tanks.*

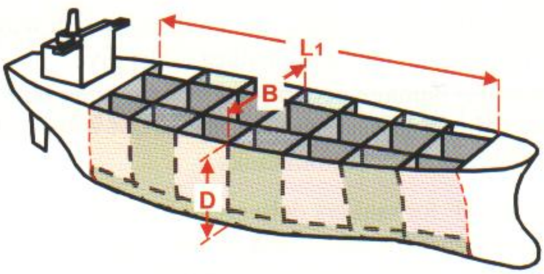
Cargo tanks are to be limited in size to ensure that the oil outflow after hull damage does not exceed certain specified limits.

***Regulations 22 to 24** specify damage lengths, widths and depths to the ship's bottom and side plates and the equations for calculating oil outflows as a result of such damage. The oil outflow is related to the size of oil tanks in the region of the damage and the probable damage penetration through the surrounding protective ballast tanks.*

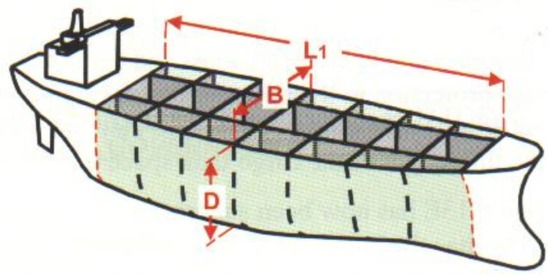
MINIMUM TANKER BALLAST DRAFT CONDITIONS



EXAMPLES OF BALLAST TANK LOCATION FOR VLCC TANKERS DELIVERED BETWEEN 1982 & 1996 (REGULATION 13E OF MARPOL 73/78-ANNEX I)



DOUBLE BOTTOM BALLAST TANKS AND ALTERNATING WING TANKS



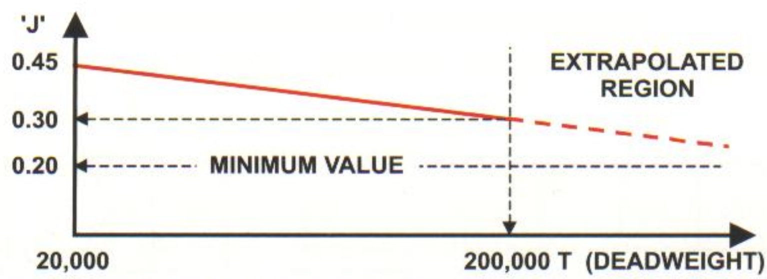
WING BALLAST TANKS EXTENDING OVER THE FULL CARGO LENGTH BUT NO D.B. TANKS

= AREA OF SIDE AND BOTTOM PLATING OVER THE LENGTH OF THE CARGO TANKS

= BALLAST TANKS IN DOUBLE BOTTOM AND ALTERNATING WING TANKS

TOTAL AREA OF SIDE & BOTTOM PLATING OF BALLAST OR VOID TANKS $\geq J\,[L_1\,(B + 2D)]$

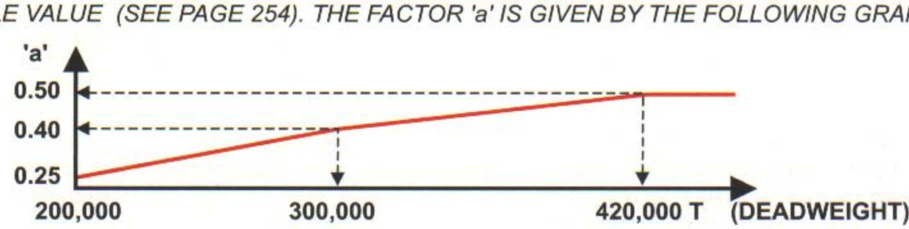
WHERE 'L_1', 'B', AND 'D' ARE SHOWN IN THE DIAGRAM ABOVE. THE FACTOR 'J' IS GIVEN AS



'J' IS GIVEN BY THE ABOVE GRAPH FOR DEADWEIGHTS BETWEEN 20,000 AND 200,000 T. AT DEADWEIGHTS GREATER THAN 200,000T, 'J' IS MODIFIED BY THE FOLLOWING EQUATION

$$'J'_{(DWT \geq 200{,}000T)} = \left['J'_{\text{EXTRAPOLATED}} - \left(a - \frac{O_C + O_S}{4O_A} \right) \right] \text{ BUT MUST NOT BE LESS THAN 0.2}$$

'O_C' AND 'O_S' ARE THE WORST CASE 'HYPOTHETICAL OIL OUTFLOWS' FOR SIDE AND BOTTOM DAMAGE RESPECTIVELY, AS DEFINED BY REGULATION.23 AND 'O_A' IS THEIR MAXIMUMUM ALLOWABLE VALUE (SEE PAGE 254). THE FACTOR 'a' IS GIVEN BY THE FOLLOWING GRAPH



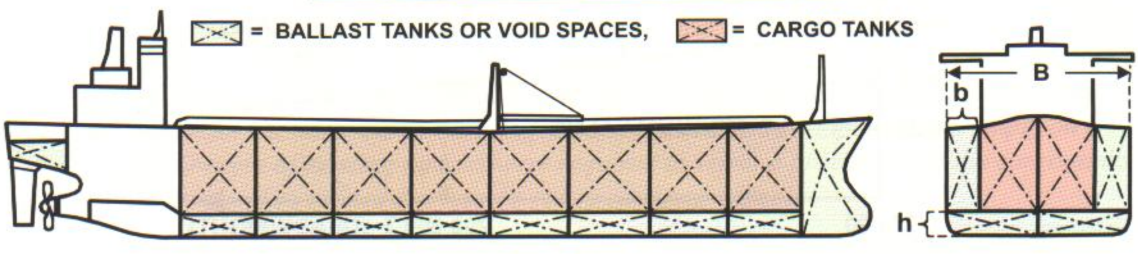
PROTECTIVE BALLAST TANK LOCATION IN TANKERS

MARPOL Regulation 13E is somewhat strange as an anti-pollution measure in that it does not require all the cargo tanks to be protected by ballast tanks or void spaces. The protection factor 'J' is much less than '1', though this is applied to (L_1 (B + 2D), which produces an area value greater than the actual side and bottom shell plating, consequently, if 'J' equals 0.45, about 55% of the side and bottom plating will be protected. However, this will still leave about 45% of vulnerable hull adjoining directly to tanks loaded with oil.
Of course, the value of 'J' gives the minimum protection required by the regulation and a ship could be built with the ballast tanks completely enveloping the cargo space. However, such a ship would also have to meet **Regulations 22 to 24** (to be explained in the following pages of this chapter) that stipulate a maximum allowable outflow of oil as a result of stranding or collision. The calculation of likely oil outflows from a vessel takes the width of wing ballast tanks and depth of double bottom tanks into account. Narrow wing tanks and shallow double bottom tanks in ULCC's would be insufficient protection to reduce the oil outflow to acceptable levels. The smallest possible ship for a given cargo capacity is the most economic to build and operate but ULCC's with the minimum ballast requirements spread over a complete double skin would not comply with Regulation 24. Instead, such tankers could be built with ballast in the double bottom tanks and generous width wing tanks that alternated between ballast and cargo tanks. If damaged, the wing ballast tanks give complete protection to the adjacent centre cargo tanks and if damage occurred in a cargo wing tank, then outflow of oil would be limited to that tank alone. Other designs extended the ballast wing tanks over the full length of the cargo tank space but did not have double bottom tanks

Regulation 13E has now been superceded by Regulation 13F for tankers delivered after July 6th 1996. This has the much more straightforward requirement that the cargo tank space of the hull shall be completely enveloped by protective ballast tanks or void spaces. This has tended to increase the ship's overall ballast capacity, as the minimum allowable outflow requirements must still be met and so it will be easier for ships' officers to ballast the ship satisfactorily in heavy weather.
Regulation 13G requires that ships built to Regulation 13E must comply with the tighter standards set by Regulation 13F within 25 or 30 years of being delivered, depending upon their tank layout.

BALLAST TANK LOCATION FOR TANKERS BUILT AFTER 1996 (REGULATION 13F OF MARPOL 73/78-ANNEX I)

TANKERS OF 5,000 T DEADWEIGHT OR MORE



MINIMUM WING TANK WIDTH 'b' $\geq 0.5 + \frac{D'W'T}{20{,}000}$ M *OR* **2.0 M,** *WHICHEVER IS THE LESSER*

BUT, IN ANY CASE, 'w' MUST NOT BE LESS THAN 1.0 METRES

MINIMUM D.B. TANK HEIGHT 'h' $\geq \frac{B}{15}$ M *OR* **2.0 M,** *WHICHEVER IS THE LESSER*

BUT, IN ANY CASE, 'h' MUST NOT BE LESS THAN 1.0 METRES

NOTE THAT THESE VALUES ARE THE MINIMUM WIDTHS AND HEIGHTS THAT CAN BE MEASURED AT RIGHT ANGLES TO THE SHELL PLATING, AT ANY POINT OF A GIVEN WING OR D.B. TANK.

A TANKER DOES NOT REQUIRE DOUBLE BOTTOM TANKS IF THE PRESSURE OF THE OIL CARGO AND ITS VAPOUR ON THE SHIP'S BOTTOM NEVER EXCEEDS THE EXTERNAL WATER PRESSURE

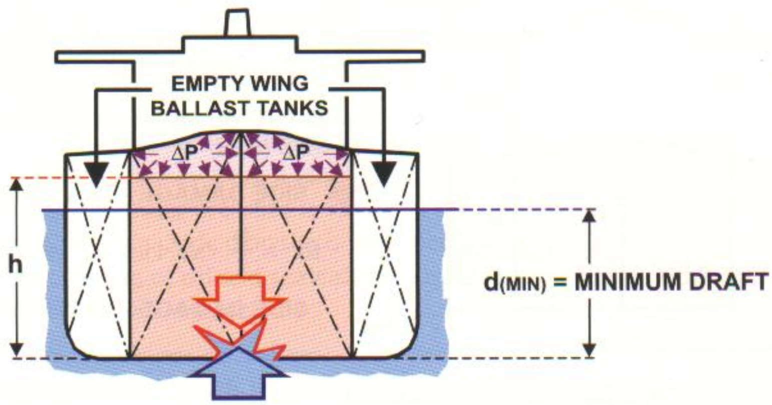
NOT WITHSTANDING THE ABOVE, SMALL TANKERS OF LESS THAN 5,000 T D'W'T ARE REQUIRED TO HAVE AT LEAST DOUBLE BOTTOM TANKS THAT MEET THE ABOVE CRITERIA, EXCEPTING THAT THEY SHALL NOT HAVE MINIMUM HEIGHTS OF LESS THAN 0.76 M

THE REQUIREMENTS FOR DISPENSING WITH DOUBLE BOTTOM TANKS

MARPOL Regulation 13F(4a) allows for tankers to be built without double bottom ballast tanks, provided that the outside water pressure on the ship's bottom is greater than the internal pressure at the tank bottom. This is the combined pressure of the oil cargo and the maximum vapour pressure of the overlying inert gas that is injected into the tank to prevent explosion. If the ship's bottom is damaged in these circumstances, the seawater pressure should be sufficient to prevent oil leakage in any significant quantity.
It would be difficult for most traditional tanker designs to fulfil the above conditions if their full hull depth cargo tanks were fully loaded. However, **Regulation 13F(4b)** allows for horizontal divisions to be built into the centre tanks in order to reduce the internal tank pressure and so meet the above requirements. It also defines the allowable height limits for the positioning of such divisions.

This design, however, is unacceptable to the U.S. authorities, which requires tankers trading in U.S. waters *must be double skinned around the cargo tanks.* (U.S. Oil Pollution Act 90)
It is unlikely that such vessels will be built but if they ever were, dividing centre tanks horizontally would produce a tanker in which it would be possible for lower tanks to be empty whilst upper ones still contain cargo. Such situations could lead to the ship becoming unstable and the stability book would have to specify which loaded conditions would only be acceptable in port

CARGO TANK REQUIREMENTS FOR A TANKER WITHOUT A DOUBLE BOTTOM
(NOT ACCEPTED BY THE U.S. GOVERNMENT)



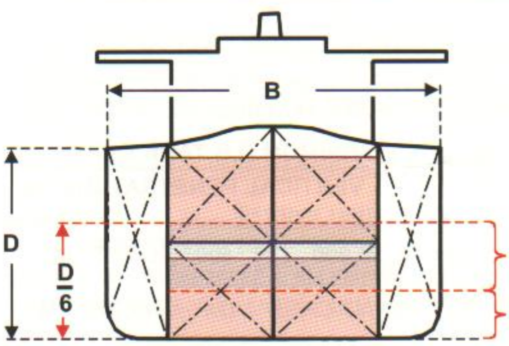
= OIL CARGO OF MAXIMUM DENSITY 'ρ' T / M³ = EXTERNAL WATER PRESSURE

= INERT GAS UNDER PRESSURE 'ΔP' BAR = INTERNAL TANK PRESSURE

INTERNAL PRESSURE AT THE BOTTOM OF THE TANK $'\Delta P + \rho g h' < 1.025\, g\, d_{(MIN)}$

WHERE 'g' = 9.81 M/S² (THE ACCELERATION DUE TO GRAVITY), 'ΔP' (BAR) = GAS PRESSURE 'd(MIN)' (M) = SHIP'S MINIMUM LOADED OPERATING DRAFT, 'h'(M) = DEPTH OF CARGO AND 1.025 T/M³ = THE DENSITY OF STANDARD SEAWATER (THIS SHOULD BE REDUCED IF THE TANKER IS TO OPERATE REGULARLY IN LOWER DENSITY DOCK OR RIVER WATER)

LIMITS OF LOCATING ANY HORIZONTAL CENTRE TANK DIVISION



THE INTERNAL PRESSURE ON THE HULL BOTTOM CAN BE REDUCED BY FITTING AN HORIZONTAL PARTITION. ANY SUCH BOUNDARY MUST BE LOCATED WITHIN THE LIMITS SHOWN IN THE DIAGRAM

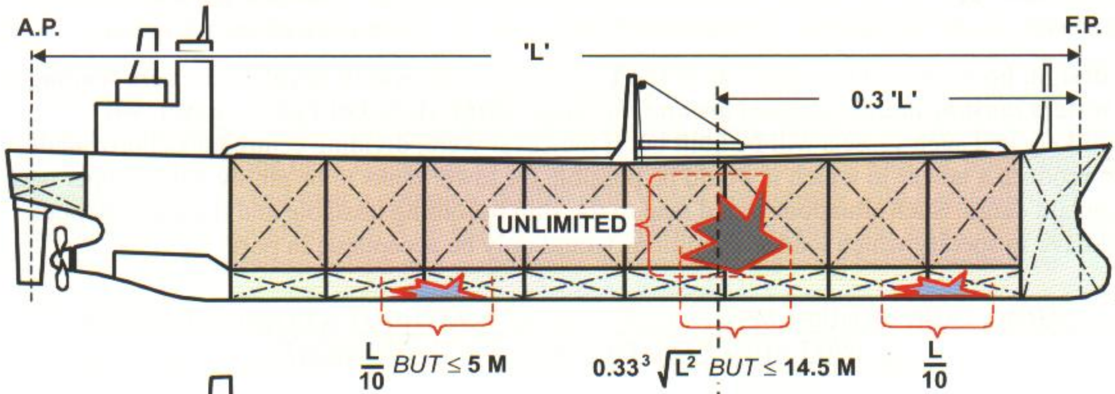
= **UPPER AND LOWER LIMITS OF ANY HORIZONTAL TANK**

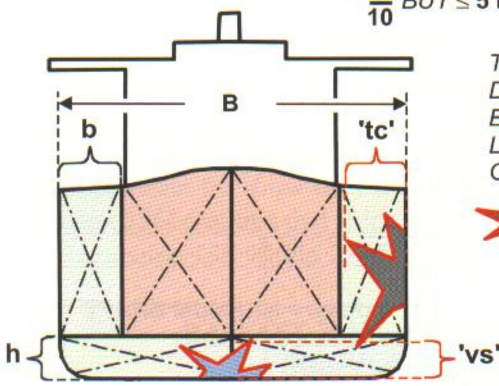
$\frac{B}{6}$ *OR* **6 M** *WHICHEVER IS THE LESSER*

TANKER DAMAGE SCENARIOS AND RESULTING OIL OUTFLOWS

Regulation 22 specifies the extent of damage to the side and bottom plates that must be considered when determining the resulting outflow of oil from the cargo tanks. A set of damage scenarios is identified that includes all the combinations of oil tank leakage for every possible damage location. The oil outflow is then calculated for each damage scenario in accordance to **Regulation 23**. The 'Hypothetical Oil Outflows', 'Oc' and 'Os', are the ***'worst cases'*** of these calculated scenario outflows for side and bottom damage respectively and must not exceed the limits given by Regulation 24.

EXTENT OF DAMAGE CONSIDERED IN DETERMINING HYPOTHETICAL OIL OUTFLOW





THE REGULATIONS CONSIDER SIDE AND BOTTOM DAMAGE SEPARATELY. THE RULES REFLECT THAT BOTTOM DAMAGE TO THE FWD 30% OF THE HULL IS LIKELY TO BE MORE EXTENSIVE THAN FOR THE REST OF THE HULL

SIDE DAMAGE - TRANSVERSE PENETRATION 'tc'

tc $= \frac{B}{5}$ *OR* **11.5 M** *WHICHEVER IS THE LESSER*

SIDE DAMAGE - VERTICAL EXTENT IS UNLIMITED

BOTTOM DAMAGE - TRANSVERSE EXTENT 'ts'

WITHIN 0.3L OF THE F.P. 't's $= \frac{B}{6}$ *OR* **10 M** *WHICHEVER IS THE LESSER, BUT* **'ts' ≥ 5 M,**

ELSEWHERE IN THE HULL **'ts' = 5 M**

BOTTOM DAMAGE - VERTICAL PENETRATION 'vs' $= \frac{B}{16}$ *OR* **6 M** *WHICHEVER IS THE LESSER*

HYPOTHETICAL OIL OUTFLOW CALCULATIONS

OIL OUTFLOW RESULTING FROM SIDE DAMAGE 'Oc' $= \Sigma W_i + \Sigma K_i C_i \quad M^3$

OIL OUTFLOW RESULTING FROM BOTTOM DAMAGE 'Os' $= \frac{1}{3}\left[\Sigma Z_i W_i + \Sigma Z_i C_i\right] \quad M^3$

ΣW_i = VOLUME OF BREACHED WING OIL Tks & ΣC_i = VOLUME OF BREACHED CENTRE OIL Tks

'Ki' AND 'Zi' ARE FACTORS OF VULNERABILITY OF THE CENTRE TANKS PROTECTED BY WING AND DOUBLE.BOTTOM TANKS RESPECTIVELY. THEY CAN HAVE A MINIMUM VALUE OF 'ZERO'

$$K_i = 1 - \frac{\text{MINIMUM WING TANK WIDTH 'b'}}{\text{TRANSVERSE PENETRATION 'tc'}} \quad \& \quad Z_i = 1 - \frac{\text{MINIMUM D.B.TANK DEPTH 'h'}}{\text{VERTICAL PENETRATION 'vs'}}$$

MAXIMUM PERMISSIBLE VALUES OF 'Oc' & 'Os', 'OA' = 30,000 *OR* $400\sqrt[3]{D'W'T}$ **M³**

WHICHEVER OF THESE VALUES IS THE GREATER, BUT, IN ANY CASE **'OA' ≤ 40,000 M³**

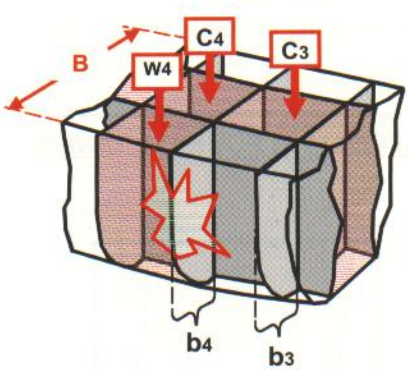
The maximum permissible 'Hypothetical Oil Outflow', 'O_A', varies between 30,000 M^3 for tankers up to 421,875 T deadweight to 40,000 M^3 for tankers over 1,000,000 T deadweight (as yet, no tanker of this size has been built, though some have been proposed). These are very large volumes of oil and actually exceed the total carrying capacity of tankers at the smaller end of the deadweight range that the rules apply to. Regulation 24 is, in practice, meaningless with regard to all but the largest tankers but the MARPOL regulations were originally drawn up at a time when crude oil tanker size was increasing at an extraordinary rate and there seemed no limit to the maximum deadweight of vessels until the oil price crises of 1974 radically changed the tanker market. In particular, the International Committee was anxious to limit the size of oil tanks in the designs of the mega large ULCC's that were being proposed and so was prepared initially to compromise on regulating the smaller tankers.

In 1974, the OPEC nations dramatically increased the price of crude oil and there was a sudden drop of about 10% in the demand for oil on the world market, which resulted in a slump in the tanker business that lasted for about ten years. Many large crude oil carriers were laid up and plans for producing million ton vessels were shelved.
The rules would still apply if these ultra large vessels were to be built but, in the mean while, Regulation 13F has superceded 13E. Small to medium sized vessels built to this more demanding degree of ballast tank protection will easily have reasonable maximum values of the 'Hypothetical Oil Outflows' 'Oc' and 'Os', that are far lower than the limits given by Regulation 24.

It has not been my intention to include market politics in writing this text book but it is impossible to explain the irrationalities of these rules without giving some background to the prevailing economic conditions at the time of the initial negotiations and drafting of the regulations. The oil transportation industry is large and central to the world economy so, consequently, it takes time to change.

The oil outflow calculation for individual damage scenarios is shown below. Notice that the equations allow for a greatly reduced leakage from a tank breached by bottom damage than is the case for side damage. The external seawater pressure is assumed to limit the outflow from a fully submerged hole in the bottom of the ship. Side damage is considered to extend above the ship's waterline and so allow sufficient mixing of oil and seawater in any damaged tank to effectively spread all the effected oil out onto the sea surface.

OIL OUTFLOW CALCULATIONS FOR INDIVIDUAL SIDE DAMAGE LOCATIONS

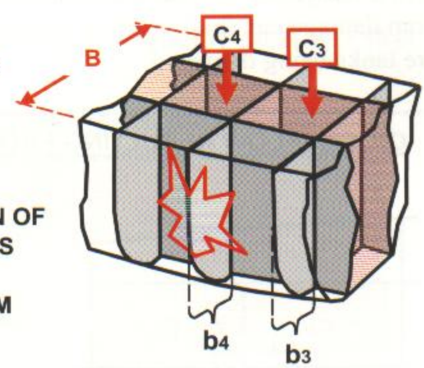


← ① ② →

= OIL TANKS

INBOARD PENETRATION OF DAMAGE 'tc' IS GIVEN AS

$$'tc' = \frac{B}{5} \quad BUT \leq 11.5\ M$$



① **WING TANK 'W4' IS A CARGO TANK BUT WING TANK 'W3' IS A BALLAST TANK**

$$\text{OIL OUTFLOW} = \text{VOLUME Tk 'W4'} + \left(1 - \frac{b_4}{tc}\right) \text{VOLUME Tk 'C4'} + \left(1 - \frac{b_3}{tc}\right) \text{VOLUME Tk 'C3'} \quad M^3$$

② **WING TANK 'W4' AND WING TANK 'W3' ARE BALLAST TANKS**

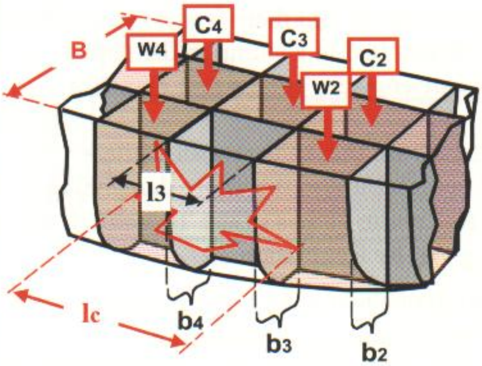
$$\text{OIL OUTFLOW} = \left(1 - \frac{b_4}{tc}\right) \text{VOLUME Tk 'C4'} + \left(1 - \frac{b_3}{tc}\right) \text{VOLUME Tk 'C3'} \quad M^3$$

WHERE b_4/tc AND b_3/tc HAVE A MAXIMUM ALLOWABLE VALUE OF '1' SO THERE IS NO OUTFLOW FROM THE CENTRE TANKS IF b_4 AND b_3 ARE EQUAL TO OR GREATER THAN THE DAMAGE PENETRATION 'tc'

TANKER DAMAGE SCENARIOS AND RESULTING OIL OUTFLOWS (Cont.)

The longitudinal extent of side damage can be sufficient to span over two transverse bulkheads and so effect three wing tanks. If, in such cases, the vessel has alternating ballast and cargo wing tanks (i.e. it is a form of the '13E type' of tanker) _and_ the centre effected wing tank is a ballast tank with wing cargo tanks forward and aft of it, _then_ the volume of the smaller of the two adjoining wing cargo tanks is reduced by the factor 'Si', as defined by **Regulation 23(2)**, in the damage scenario outflow calculation.

OIL OUTFLOW RESULTING FROM SIDE DAMAGE TO THREE WING TANKS, TWO OF WHICH ARE CARGO TANKS



= OIL TANKS

DAMAGE

INBOARD PENETRATION $'tc' = \frac{B}{5}$ *BUT* ≤ 11.5 M

LONGITUDINAL LENGTH $'lc' = \frac{1}{3}\sqrt[3]{L^2}$ *BUT* ≤ 14.5 M

WHERE 'L' IS THE SHIP'S LENGTH

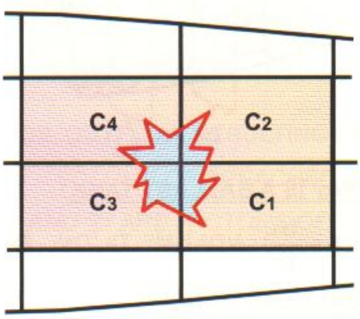
WING TANK ' W_2' IS SMALLER THAN WING TANK ' W_4'

REDUCTION FACTOR $'Si' = 1 - \frac{l_3}{l_c}$

$$\text{OIL OUTFLOW} = \begin{cases} \text{VOLUME Tk } 'W_4' + \left(1 - \frac{b_4}{tc}\right) \text{VOLUME Tk } 'C_4' + \left(1 - \frac{b_3}{tc}\right) \text{VOLUME Tk } 'C_3' \\ + \left(1 - \frac{l_3}{l_c}\right) \text{VOLUME Tk } 'W_2' + \left(1 - \frac{b_2}{tc}\right)\left(1 - \frac{l_3}{l_c}\right) \text{VOLUME Tk } 'C_2' \quad M^3 \end{cases}$$

Bottom damage scenarios will also include multiple tank damage and **Regulation 24(4)** states that, if four centre tanks are simultaneously effected by a particular bottom damage location, then the scenario oil outflow is reduced by using a factor of 1/4. If any less than four tanks are involved in bottom damage, the normal reduction factor of 1/3 is used, as given by the equation on page 255. Bottom damage can involve wing and centre oil tanks, though the following example shows only centre tanks being damaged.

OIL OUTFLOW RESULTING FROM BOTTOM DAMAGE AFFECTING FOUR OIL TANKS

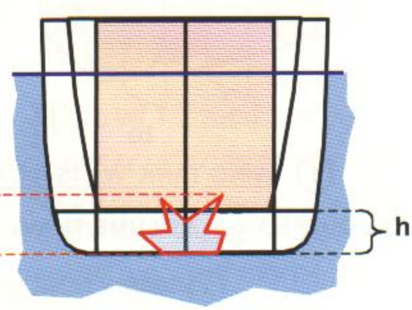


= OIL TANKS

DAMAGE

VERTICAL PENETRATION

$'vs' = \frac{B}{16}$ *BUT* ≤ 6 M



$$\text{OIL OUTFLOW} = \frac{1}{4}\left(1 - \frac{'h'}{'vs'}\right)(C_1 + C_2 + C_3 + C_4) \ M^3$$

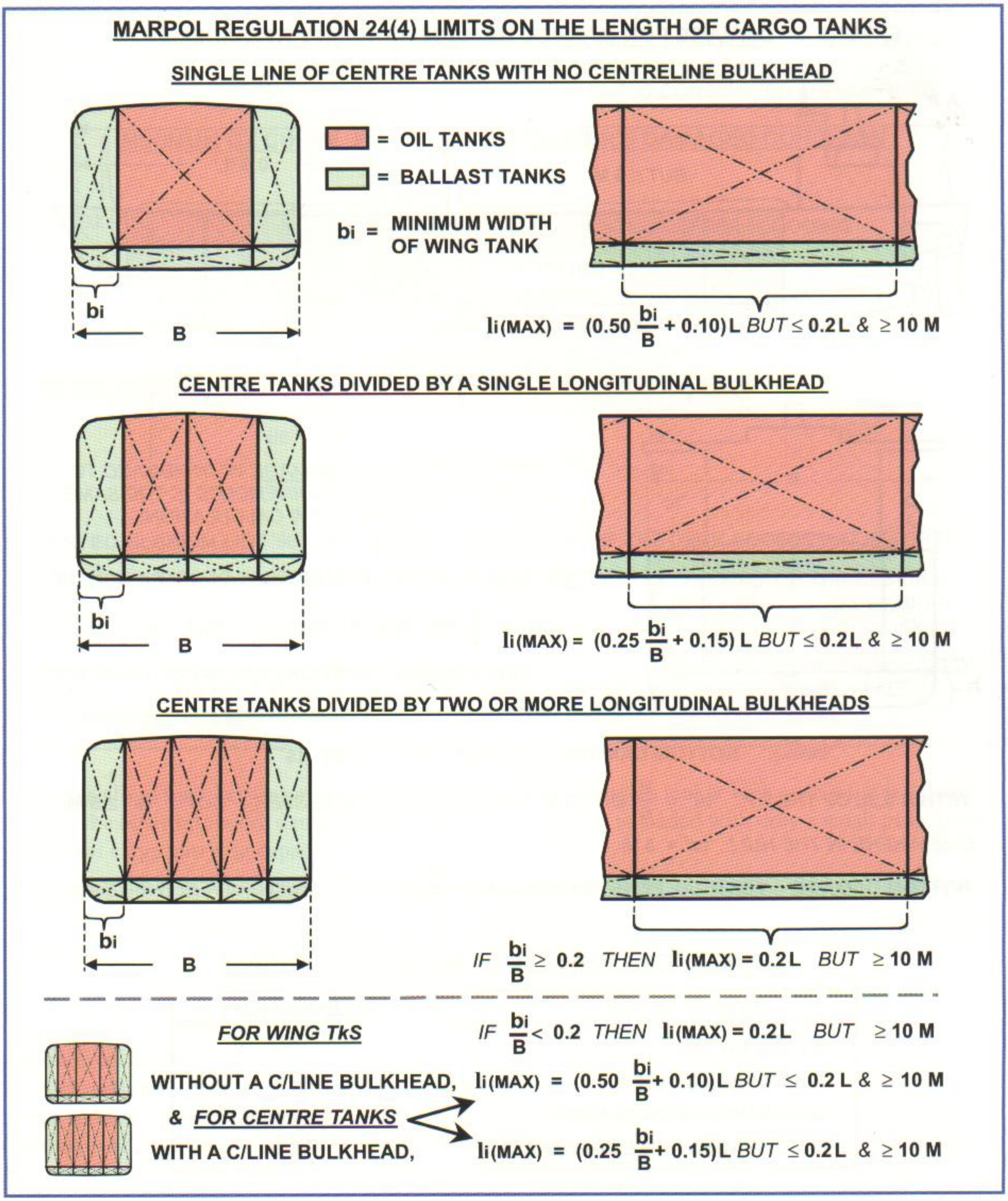
THE MAXIMUM VALUE FOR h/vs IS '1'. THE ABOVE TANKS HAVE FULL WING AND DOUBLE BOTTOM BALLAST TANK PROTECTION. h/vs WOULD EQUAL 'ZERO' IF THERE IS NO D.B. PROTECTION

MARPOL LIMITS ON CARGO TANK SIZE

Regulation 24(3) gives limits to the size of oil cargo tanks. Wing cargo tanks must not exceed 75% of the 'Hypothetical Oil Outflow', appropriate to the tanker being considered. If however, a wing cargo tank is longer than the longitudinal extent of damage 'lc' *and* is wider than the transverse penetration of damage 'tc', then the tank volume may be increased to the limit of the 'Hypothetical Oil Outflow'. Centre tanks must not exceed 50,000 m^3.

These very large volumes for individual tanks are, in reality, far bigger than would be practical for most existing tankers. However, as stated before, the rules were worked out at a time when tanker size was increasing dramatically and the rules were aimed at regulating the very largest of ULCC's.

Regulation 24(4) specifies limits on the length, 'li' of cargo tanks for a vessel 'L' metres long , as shown below. Note that the minimum limit for the maximum cargo tank length is 10 metres.

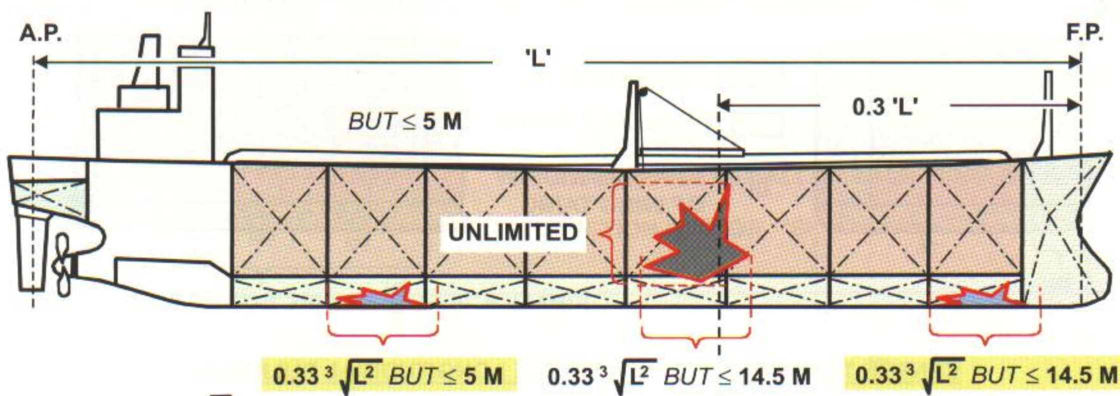


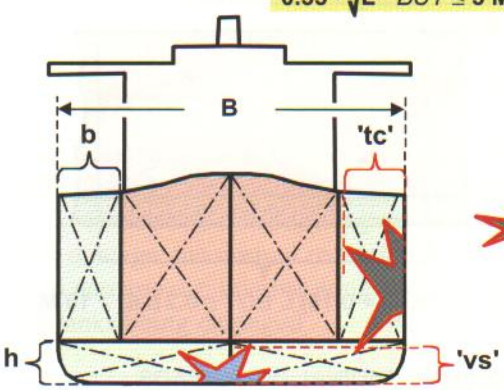
MARPOL STABILITY REQUIREMENTS FOR TANKERS

Regulation 25A (*Amendment MEPC.75(40)*) states that tankers built or converted after Feb 1st 2002 must comply with the I.M.O. merchant ship minimum intact stability requirements as outlined in Chapter 3, page 63. These intact criteria apply equally to ballast and loaded voyages.
A tanker in port must maintain an upright fluid GM value of 0.15 metres or more. All stability calculations must allow correctly for the free surface effect of slack fluid in the tanks.

Regulation 25(3) sets out minimum acceptable damage stability criteria for tankers after suffering damage to an extent specified in Regulation 25(1) & (2). Unfortunately, the limits of damage are not quite the same as those regarding the oil outflow calculations. The differences are in the longitudinal extent of bottom damage, both in the forward 30% of hull length and the remainder of the vessel. These differences are highlighted in yellow in the following diagram.

EXTENT OF DAMAGE FOR MARPOL DAMAGE STABILITY REQUIREMENTS





THE EXTENT OF SIDE DAMAGE WITH REGARD TO DAMAGE STABILITY REQUIREMENTS IS THE SAME AS FOR THE OIL OUTFLOW CALCULATIONS

SIDE DAMAGE - TRANSVERSE PENETRATION 'tc'

tc $= \frac{B}{5}$ *OR* **11.5 M** *WHICHEVER IS THE LESSER*

SIDE DAMAGE - VERTICAL EXTENT IS UNLIMITED

BOTTOM DAMAGE - TRANSVERSE EXTENT 'ts'

WITHIN 0.3L OF THE F.P. 'ts' $= \frac{B}{6}$ *OR* **10 M** *WHICHEVER IS THE LESSER, BUT* **'ts' ≥ 5 M,**

ELSEWHERE IN THE HULL **'ts' = 5 M**

BOTTOM DAMAGE - VERTICAL PENETRATION 'vs' $= \frac{B}{16}$ *OR* **6 M** *WHICHEVER IS THE LESSER*

PERMEABILITIES TO BE USED IN BILGING CALCULATIONS

TYPE OF SPACE	PERMEABILITY
STORE ROOMS	60%
MACHINERY SPACES	85%
ACCOMMODATION SPACES	95%
VOID SPACES AND EMPTY SPACES	95%

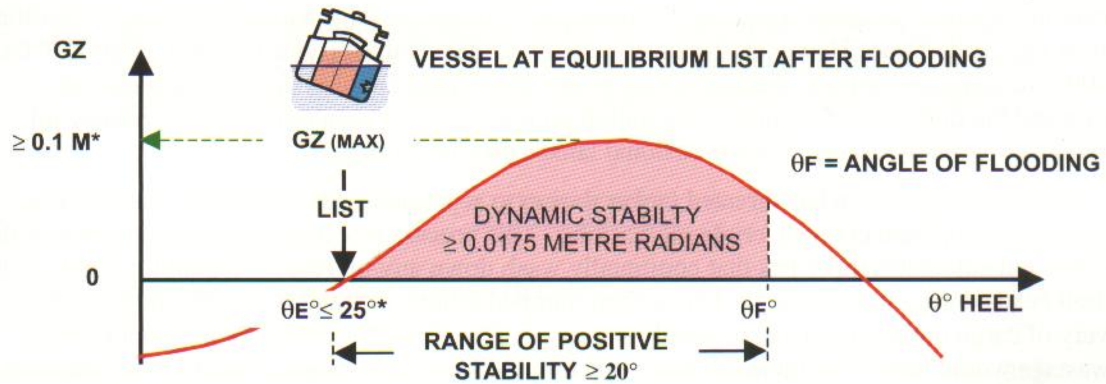
MARPOL STABILITY REQUIREMENTS FOR TANKERS (Cont.)

The MARPOL minimum damage stability requirements, given by **Regulation 25(3)**, *apply only to loaded tankers*. **Ballast conditions need not be considered**. The extent of damage, defined by the previous page, will result in at least two adjacent compartments flooding, though the engine room can be considered separately as a single flooded space, for tankers of less than 225 metres in length.

MARPOL MINIMUM DAMAGE STABILITY CRITERIA FOR LOADED TANKERS

IF LBP ≥ 225 M	*THEN THE LONGITUDINAL EXTENT OF DAMAGE CAN EXTEND ACROSS ANY BULKHEAD TO FLOOD ANY PAIR OF ADJACENT COMPARTMENTS*
IF 225 > LBP > 150 M	*THEN THE LONGITUDINAL EXTENT OF DAMAGE CAN EXTEND ACROSS ANY BULKHEAD EXCEPT THE BOUNDARIES OF THE MACHINERY SPACE WHICH IS CONSIDERED TO FLOOD SEPARATELY AS A SINGLE SPACE*
IF 150 M ≥ LBP	*AUTHORITIES GOVERNING THESE REGULATIONS WILL CONSIDER THE DAMAGED STABILITY OF AN INDIVIDUAL VESSEL, AND RELAX SOME OF THE MINIMUM REQUIREMENTS IF THIS IS CONSIDERED APPROPRIATE*

MINIMUM GZ CURVE FOR A LOADED TANKER'S DAMAGED CONDITION



*** THE EQUILBRIUM LIST 'θE°' MAY BE INCREASED TO 30°, IF THE DECK EDGE IS NOT IMMERSED**

EQUALIZATION SYSTEMS WILL NOT BE CONSIDERED IN DETERMINING 'θE' OR THE RESIDUAL DYNAMIC RANGE OF STABILITY BUT SUFFICIENT RESIDUAL STABILITY MUST BE MAINTAINED THROUGHOUT ANY TIME PERIOD THAT WHEN THEY ARE IN USE, IF A VESSEL IS SO EQUIPPED

FREE SURFACE EFFECTS OF EACH 'FULL' BUT SLACK CARGO TANK SHOULD BE CALCULATED FOR A 5° LIST BUT THE REGULATING AUTHORITIES CAN REQUIRE FREE SURFACE EFFECTS OF PARTIALLY FILLED TANKS TO BE CALCULATED AT GREATER ANGLES OF HEEL

The rules allow for a very large angle of equilibrium heel, though a tanker with such a list is unlikely to meet the other minimum damaged stability criteria unless it is loaded with only a small proportion of its total cargo carrying capacity.

All tankers should be provided with an approved stability book that explains the normal sequence of loading and discharge that maintains adequate stability and trim whilst not exceeding the permissible bending moments. The book should also give all the loaded conditions that meet the MARPOL regulations and data regarding the stability of the possible damaged conditions.

Although the ballast tanks are normally segregated from the cargo system, there are blanked connections that allow cargo tanks to be ballasted in the event of very severe weather. It may be equally possible to use these connections to transfer oil out of damaged tanks into empty ballast tanks, in the event of stranding or collision. However, stability must be maintained and excessive bending moments avoided in either a 'free floating ' or 'grounded' hull. The Master must be provided with readily accessible and relevant information before deciding upon such a course of action.

THE OVERALL EFFECT OF THE MARPOL CONVENTION

The 'MARPOL' requirements for segregated ballast tanks in tankers can only really be effective at preventing oil pollution from a tanker accident in the early stages of the incident. If a tanker goes aground, the regulations provide time for off-loading cargo or salvage operations. However, neither of these operations is necessarily easy or even possible if the vessel grounds on a coast exposed to severe weather. The large tankers 'Torrey Canyon', 'Amoco Cadiz' and 'Braer' all broke up shortly after going aground on exposed reefs with the result that their entire cargoes ended up in the ocean. The only loss of life in all three cases was that of one of the team attempting to salvage the 'Torrey Canyon', which perhaps makes the point of how difficult this kind of operation can be in such circumstances. Many maritime nations have responded to this risk by restricting the routes that tankers can follow through their waters.

MARPOL's most significant contribution in reducing oil pollution from tankers is probably the way in which the regulations have greatly reduced the routine disposal of oil contaminated water.
Prior to the 1980's tankers were ballasted by filling a proportion (usually at least one third) of the cargo tanks with seawater that was then used to wash down the tanks in readiness for the next load. The tank washings were allowed a certain time to separate out most of the oil residue from the water, which then could be pumped overboard from underneath the highly contaminated oily sludge that would float on the top of the tank. In theory, only water with very low traces of oil was allowed to be pumped directly into the open ocean and the remaining 'sludge' was discharged to reception facilities ashore at the loading port. However, the sheer quantity of dirty ballast (typically about 70,000 T for a 200,000 T deadweight tanker), combined with erratic provision of shore facilities, commercial pressure and the difficulty of enforcing regulation on the high seas, resulted in a lot of routine oil pollution that probably exceeded the occasional tanker disaster spillage.

The 'MARPOL' Convention has directed tanker design to segregate cargo tanks from ballast tanks and equip 'black oil' tankers with 'crude oil washing'. This process re-cycles a small proportion of the cargo through automatic spray jets that continually wash down any oil residue remaining stuck to the tank bulkheads during the discharge. This is then pumped ashore with the rest of the cargo so the recovery of cargo during discharge is greatly improved. (Previously, a considerable proportion of this wastage would have been pumped overboard into the sea). Furthermore, much of the structural stiffening can be placed on the ballast side of tank dividing bulkheads. The new designs of tankers tend to have much smoother tank walls and this further increases the recovery rate of oil on discharge whilst greatly reducing the need for any tank cleaning with water on a ballast voyage.
The initial cost of building tankers has increased as a result of the 'MARPOL' regulations but there are real cost benefits in reducing wastage in oil cargoes and these are likely to become more significant as oil will increase in its value with the shrinking of easily accessible oil reserves.

The 'MARPOL' regulations, taken as a whole, are far more wide reaching than the limited part dealt with by this chapter and affect routine operation onboard all ships. Procedures for the disposal of toxic substances, sewage and garbage are covered as well as oil waste.
Coastal water ballast is, however, one related growing concern of several maritime nations that is not yet included in 'MARPOL' but may well be incorporated into future revised regulations. Biological organisms are transported in ballast water taken from coastal waters of one region of the world and then pumped out in another. Problems arise when alien species take root in a new habitat where there is no native biological control and the local marine ecology is drastically altered. Severe damage can be caused to the local economy by adverse effects upon commercial fishing and tourism. Consequently, the governments of the U.S.A., Canada, Australia and New Zealand have passed legislation to outlaw the discharge of ballast from foreign coastal regions in their own local waters. Ships trading with such countries must ensure that any ballast loaded in shallow water elsewhere is not discharged into the country's own coastal waters. Deep-water ballast is considered to be 'clean' and so any ballast that is to be discharged prior to loading must be replaced with deep-water ballast before arrival in the loading port. The authorities can demand to see records of ballast operations that includes the loading and discharge locations of the all the ballast recently carried.
Ballast operations on passage must be carefully planned to ensure that the ship always retains adequate stability with a suitable trim and draft for the sea conditions it is sailing in.

CHAPTER 11

THE LOADLINE REGULATIONS FOR MERCHANT SHIPS ENGAGED IN INTERNATIONAL TRADE

SUMMARY

THE LOAD LINE REGULATIONS ARE EXPLAINED IN THE FOLLOWING SECTIONS

1) A BRIEF HISTORY AND OUTLINE OF THE CURRENT REGULATIONS
2) DEFINITIONS AND SPECIFICATIONS FOR THE MARKINGS ON THE HULL.
3) THE CONDITIONS OF FREEBOARD ASSIGNMENT.
4) THE DETERMINATION OF THE MINIMUM FREEBOARDS.
5) THE ASSIGNMENT OF THE TIMBER FREEBOARDS.
6) THE LOAD LINE ZONES AND SEASONAL AREAS

The contents of this chapter only cover the main points of the regulations. The text is not intended to explain every variation possible within the rules, which can only be fully understood by studying the regulations themselves

CONTENTS

THE BACKGROUND AND AIMS OF THE LOADLINE REGULATIONS

The potential dangers of overloading a ship have long since been recognised by the maritime world and international law. Taking an overloaded ship to sea has been a criminal offence in Britain for over one hundred years. Of course, limiting the carrying capacity of a ship can restrict its earning capability so there was considerable initial resistance to government regulations. However, in the second half of the nineteenth century, after a period of increasing losses of ships, many of which were considered to be grossly over insured, the insurance companies were prepared to support the British politician, Samuel Plimsoll, in his campaign to make overloading ships illegal. In 1875, the British Government passed the 'Unseaworthy Ships Act' that required British merchant ships to be marked with a what became known as the 'Plimsoll Line' indicating its maximum allowed draft. In 1930, an International Load Line Convention produced an internationally agreed set of rules for determining the maximum loaded draft for a merchant ship of any nationality.

Some marine insurers, notably Lloyd's of London, had established their own means of assessing a ship's seaworthiness prior to the passing of government legislation. The insurance business created the 'classification societies' (such as Lloyd's of Britain, Bureau Veritas of France and the American Bureau of Shipping etc.) to assess under-writing risks. These institutions set the technical standards of a ship's design and construction, then oversee its building and operational maintenance. Classification societies give a vessel a class rating (such as 100A1), depending upon its standard of construction, and these are used in the commercial world as a measure of a ship's suitability for chartering, though, in reality, only the highest class rating is considered acceptable by the industry. Lloyd's, for example, produce 'The Lloyd's Rules for the Construction of Steel Ships', which contains detailed guidance to the 'scantlings' (i.e. thicknesses of plating, grades of steel used etc.), frame spacing and methods of construction that must be employed for a ship to meet their class standards. It is commercially essential for a merchant ship to be listed and regularly inspected by one of the classification societies but this is not specified as an actual direct legal requirement.
The societies employ surveyors who make periodic inspections of a ship to ensure that the class standards are maintained. These surveyors can also act as agents for the government to ensure that a vessel is meeting minimum legal requirements of safety equipment etc. The same surveyor often carries out two consecutive surveys, one to ensure that the ship's class is being kept up properly whilst the other can be acting as a government agent inspecting safety standards. (The first is a contractual requirement to make the ship commercially employable, whilst serious failure in the second can result in the ship being arrested.)
Governments accept the classification societies as being responsible for ensuring that the loadline rules are properly applied to a merchant vessel during its construction and each society has two identifying letters that are included in the loadline marks on the sides of ships that they oversee.

The main aim of the loadline regulations is to ensure that a ship always has sufficient reserve buoyancy to remain seaworthy in any sea conditions that the vessel is likely to encounter. The basic maximum permitted draft is known as the ***'Summer Loadline'*** and is indicated by a line painted amidships on each side of the ship. This is determined by what the loadline regulations consider to be a minimum safe freeboard for the vessel on the basis of its size and design features.
The rules, however, require additional seasonal / regional loadlines to account for the ship operating in Winter, Winter North Atlantic or Tropical ocean areas of the world.
The rules also require that the ship is built to sufficient strength for the maximum allowable operating displacement and set down certain structural criteria to ensure the safety of the crew and minimise the risk of flooding in heavy seas.

It should be appreciated that these regulations apply only to merchant ships engaged in International trade. Warships are exempt and fishing vessels must comply with different regulations that have the same broad aims but are less specific in detail. Both naval and fishing vessels are sometimes brought into merchant service. Many commercial survey ships are either converted from ex deep-sea trawlers or naval hydrographic survey ships. I also know of at least two Russian Navy ex-submarine carriers that have been converted to commercial cable ships. A company, intending to buy such a ship, should research thoroughly into the cost of meeting merchant ship regulations.

A BRIEF OUTLINE OF THE LOADLINE REGULATIONS

The current regulations are given by the ***'1966 International Convention on Load Lines'*** with subsequent ***amendments*** and a ***'Protocol of 1988'*** all of which are published by the I.M.O. The International Convention states its general aims and intentions in thirty-four Articles that appear at the beginning of the publication. The actual detailed regulations are specified in three Annexes and various periodic Amendments.

ANNEX I consists of Regulations 1 to 45, which are divided into four main chapters as follows:-

Chapter 1 - 'General', Regulations 1 to 9.

Regulation 1 requires that a ship must be built with adequate strength for the maximum loaded displacement that it will operate with and delegates the responsibility for determining the details of construction to the classification societies
Regulation 2 lists types of vessels that require special consideration under the rules.
Regulation 3 defines terminology used in the rules.
Regulations 4 to **9** set out the detailed requirements of the loadline marks on the hull.

Chapter 2 - 'Conditions of Assignment of Freeboard', Regulations 10 to 26.

Regulation 10 specifies the stability information that must be onboard a vessel.
Regulations 11 to **26** give details of constructional requirements for maintaining the weathertightness of the hull, the safety of the crew and ensuring that drainage is adequate to prevent seawater accumulating on the exposed decks in heavy weather.

Chapter 3 - 'Freeboards', Regulations 27 to 40.

Regulation 27 defines the characteristic features that categorise all merchant ships into one of two different types, with regard to freeboard assignment. Type 'A' vessels are ships built to carry liquid bulk cargoes, such as oil tankers, whilst type 'B' vessels are all other merchant ships.
Regulations 28 and **29** lists Summer freeboards, against ship's length, for both type 'A' and 'B' vessels with a standard hull form.
Regulations 30 to **39** detail the corrections to be applied to the tabulated Summer freeboard in order to account for a ship's design features (e.g. block coefficient, sheer, extent of superstructures etc.) that are different from the standard hull form
Regulation 40 defines how the other loadline freeboards (Fresh, Tropical, Winter and Winter North Atlantic) are related to a ship's Summer freeboard.

Chapter 4 - 'Special requirements for ships assigned Timber Freeboards', Regulations 41 to 45

Regulations 41 to **44** specify additional constructional requirements that a ship must meet if it is to be granted a reduced minimum freeboard whilst carrying a timber deck cargo. **Regulation 45** tabulates the freeboard reductions that can be allowed on the basis that the timber deck cargo counts as additional reserve buoyancy.

ANNEX II defines the seasonal and geographic limits of the Tropical, Summer, Winter and Winter North Atlantic Loadline Zones, both in words and with an accompanying map

ANNEX III illustrates the form of certification and records that should be carried onboard a vessel

The ***'Supplement to the 1966 International Convention on Load Lines'*** contains various amendments, the most significant being;-

Amendment to Regulation 27, concerning the definition of 'Type A' and 'Type B' ships.

This amendment allows for a type 'B' ship to be considered for a reduced freeboard assignment, similar or equal to that of a type 'A' ship of the same length, provided that the vessel meets certain requirements regarding hatchway provision, crew safety and damage stability in the event of a prescribed extent of damage.

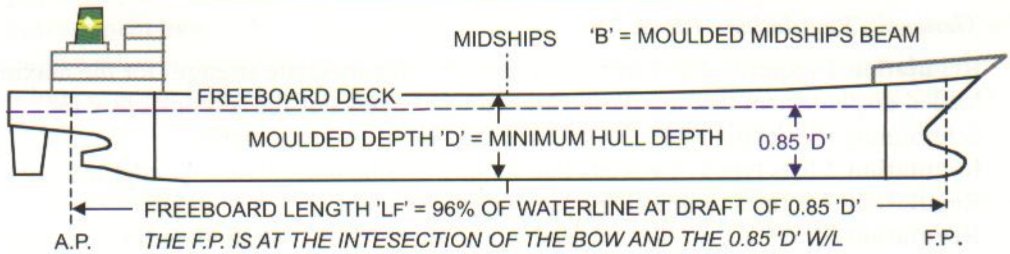
The supplement also contains 'Unified Interpretations of the Regulations', which contains more detail with regard to the details of structural requirements of the 'Conditions of Assignment of Freeboard' that minimise the risk of flooding and provide adequate crew safety.

DEFINITIONS OF SUPERSTRUCTURE, THE FREEBOARD DECK AND LENGTH

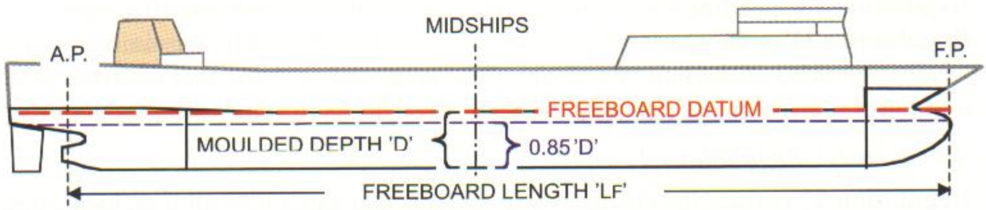
Regulation 3 of Chapter 1 contains definitions of a ship's ***Freeboard Length 'LF'***, ***Beam 'B'***, ***Moulded Depth 'D'***, ***Block Coefficient 'Cb'***, ***'Freeboard Deck'*** and ***'Superstructures'***,
These particular definitions, for the purpose of the rules, are explained in the following diagrams

THE FREEBOARD DECK, MOULDED DEPTH, FREEBOARD LENGTH AND BLOCK COEFF[N]

THE UPPERMOST CONTINUOUS DECK AS THE FREEBOARD DECK



A LOWER ENCLOSED STEPPED DECK AS THE FREEBOARD DECK IN A RO-RO SHIP



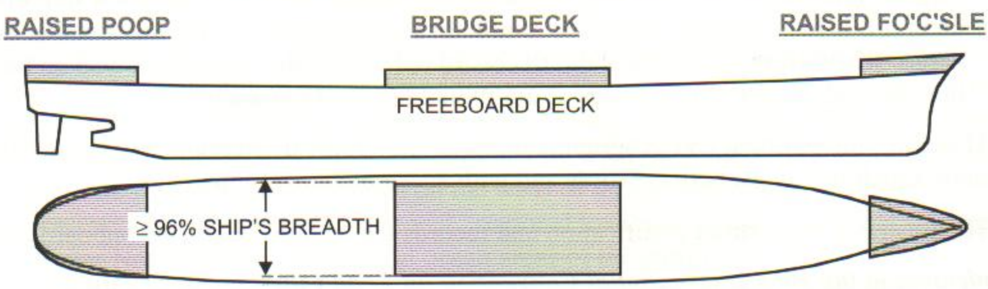
THE UPPERMOST CONTINUOUS DECK IS USUALLY THE FREEBOARD DECK, BUT IN SOME SHIP DESIGNS, IT IS MORE SUITABLE TO MEASURE FREEBOARD FROM A LOWER DECK.
THE UPPER DECK IS THEN CONSIDERED TO BE FULL HULL LENGTH SUPERSTRUCTURE

$$\textbf{BLOCK COEFICIENT 'C}_b\textbf{'} = \frac{\textbf{DISPLACED VOLUME AT DRAFT 0.85 'D'}}{\textbf{'L}_F\textbf{' x 'B' x 0.85 'D'}}$$

(THE FORE AND AFT PERPENDICULARS ARE OFTEN CALLED THE 'FORE AND AFT TERMINALS')

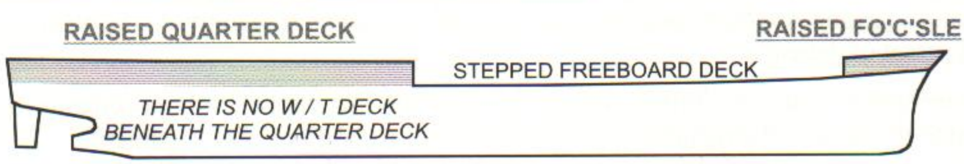
ENCLOSED SUPERSTRUCURES

ENCLOSED SUPERSTRUCTURES ARE WEATHERTIGHT DECKED STRUCTURES THAT SPAN ACROSS THE FREEBOARD DECK TO WITHIN 4% OF ITS WIDTH FROM EITHER SIDE OF THE SHIP. THEY PROVIDE RESERVE BUOYANCY AND ARE CLASSIFIED AS FOLLOWS



THE BRIDGE DECK AND POOP GENERALLY CONTAIN ACCOMODATION OR WORKSPACES.
DECKHOUSES ARE STRUCTURES THAT DO NOT EXTEND WITHIN 4% OF THE SHIP'S SIDES.

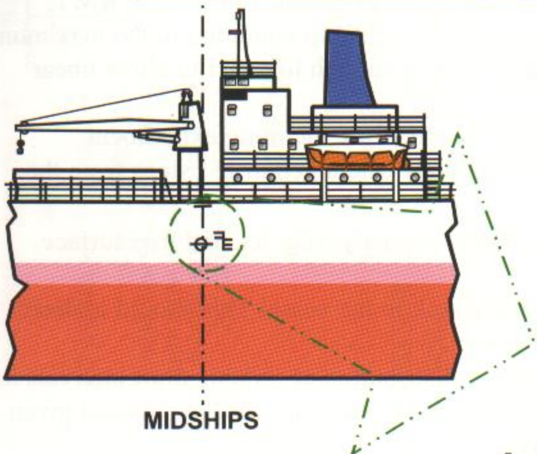
RAISED QUARTERDECKS ARE PRODUCED BY A STEP IN THE FREEBOARD DECK AND ARE NOT BUILT OVER AN UNDERLYING WEATHERTIGHT DECK, UNLIKE OTHER SUPERSTRUCTURE

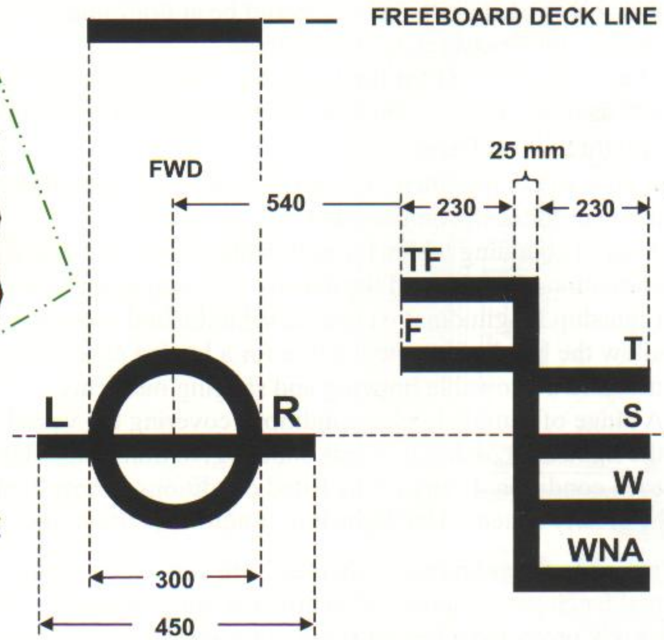


THE LOADLINE MARKINGS ON THE HULL

The freeboards of a vessel are measured from the upper edge of the freeboard deck at its lowest point on the midships station, though an extra allowance can be made if an exposed freeboard deck is partly or totally covered by wood sheathing. If the vessel has a rounded gunwale then the freeboards are measured from the upper limit of the ship's vertical side at the midships station. (**Regulation 3-6**) This datum for measuring freeboards must be marked by a line on the hull, known as the ***'deck line'*** **Regulations 4** to **8** detail precisely how this and the loadlines are to be marked on the hull

THE LOADLINE MARKS ON A BRITISH REGISTERED CARGO VESSEL





ALL DIMENSIONS ARE IN 'mm'

THE APPRORIATE FREEBOARDS ARE MEASURED FROM THE UPPER EDGES OF THE LOADLINE MARKS

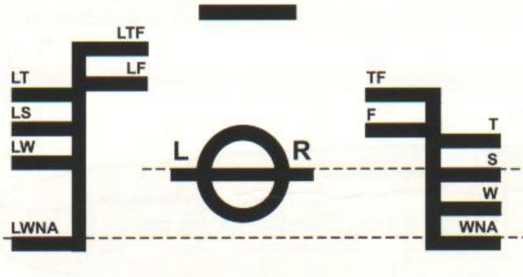
THE POSITIONS OF THE MARKS ARE TO BE PERMENANTLY INDICATED

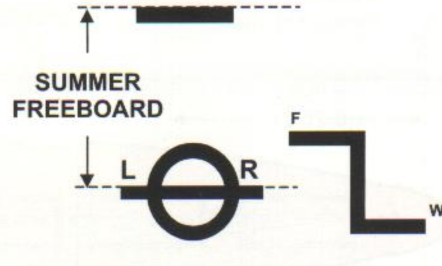
'LR' = LLOYD'S REGISTER
(I.E. THE CLASIFICATION SOCIETY RESPONSIBLE FOR ASSIGNING THE SUMMER FREEBOARD)

THE 'WNA' LOADLINE ONLY APPLIES TO VESSELS OF LESS THAN 100 METRES IN LENGTH

TF = TROPICAL FRESH WATER LOADLINE
F = FRESH WATER LOADLINE
T = TROPICAL ZONE LOADLINE
S = SUMMER ZONE LOADLINE
W = WINTER ZONE LOADLINE
WNA = WINTER NORTH ATLANTIC ZONE LOADLINE

SPECIAL MARKS FOR TIMBER SHIPS AND SAILING VESSELS





ADDITIONAL LUMBER MARKS ARE PLACED FORWARD OF THE LOADLINE DISC AND INDICATED BY THE PREFIX 'L'

THERE IS NO REDUCED FREEBOARD ALLOWED ON THE WNA LOADLINE

SAILING VESSELS ARE ONLY REQUIRED TO MARK THE FRESH AND WNA LOADLINES IN ADITION TO THE SUMMER LOADLINE THROUGH THE DISC

THE CONDITIONS OF FREEBOARD ASSIGNMENT

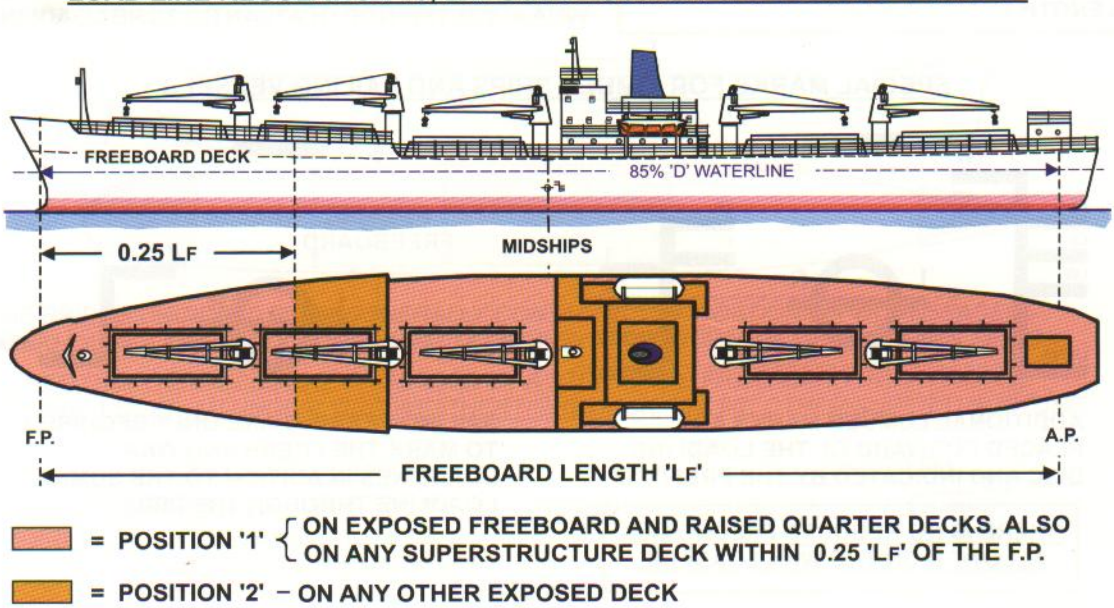
Chapter 2 specifies the conditions that a ship must meet in order to qualify for freeboard assignment.

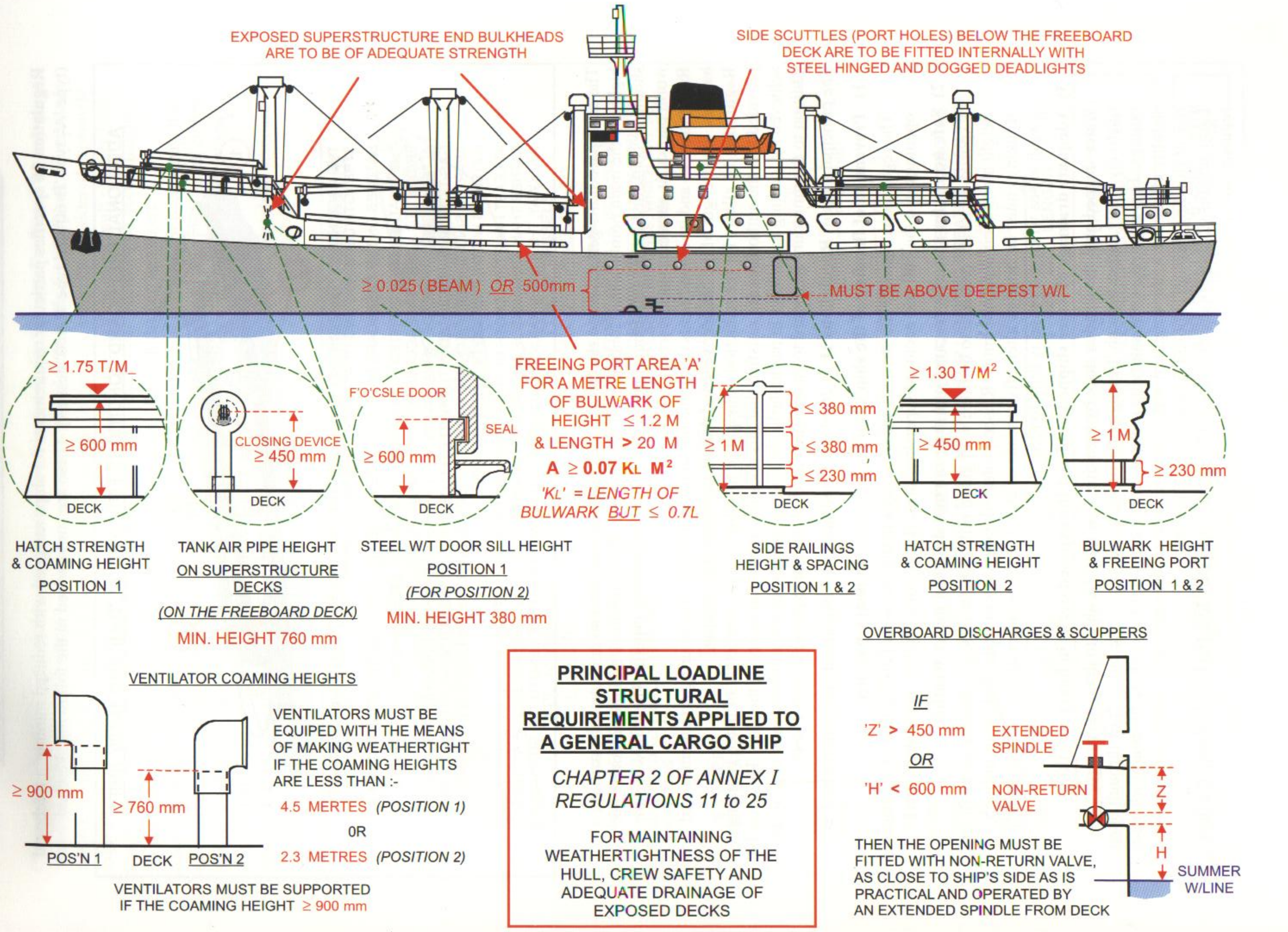
Regulation 10 requires that a ship be provided with sufficient information to allow the master to ensure that that the vessel is loaded safely with adequate stability, suitable trim and within acceptable stress limits of the hull. This is contained in the 'Approved Stability Book', which is provided by the shipbuilders and a copy is sent to the marine authority of the ship's flag state (The MCA in the case of British registered ships) who then check and endorse it with their official stamp. The stability book, combined with ship's data onboard, should provide the following;-

1) A table of hydrostatic data giving values of;- Displacement, Deadweight, TPC, FWA, KM_T, LCB, LCF and MCTC, for salt water drafts ranging from the lightship condition to the maximum possible loaded draft. Values should be at draft intervals close enough together to allow linear interpolation between each increment
2) The VCG and LCG for the Lightship conditions and a record of the inclining experiment
3) KN data allowing for the KN value at any draft and trim to be calculated in 15° steps from the upright to 90° of heel.
4) A table of compartment data giving values of:- volumetric capacity, vcg, lcg and free surface moment for each compartment.
5) A set of sounding tables for each ballast, fuel, and water tank in the vessel This should include corrections for trim and the density of liquid in the tank.
6) Lightship longitudinal weight distribution and buoyancy distribution at suitable draft intervals to allow the bending moment curve for a loaded condition to be plotted and checked against given maximum allowable hogging and sagging moments
7) A range of sample loaded conditions covering the vessel's normal operating range and including the lightship condition. These should give transverse stability, trim and bending moment data for each condition. If any of the listed conditions is unsuitable for proceeding to sea, then this should be clearly stated. (The 'lightship' condition is often unseaworthy)

The remaining regulations in chapter 2 are concerned with ensuring that the risk of flooding through external hatchways, doorways, ventilators, air pipes, ports etc. is minimised whilst the crew are adequately protected when carrying out essential duties on the exposed deck in heavy weather and the weather deck has sufficient drainage.
The main points of these regulations are shown by the example of a general cargo ship, illustrated on the next page. The criteria for hatchways, ventilators, air pipes and external doors depends on whether they are in a 'position 1' or a 'position 2' category, as defined in the sketch below.

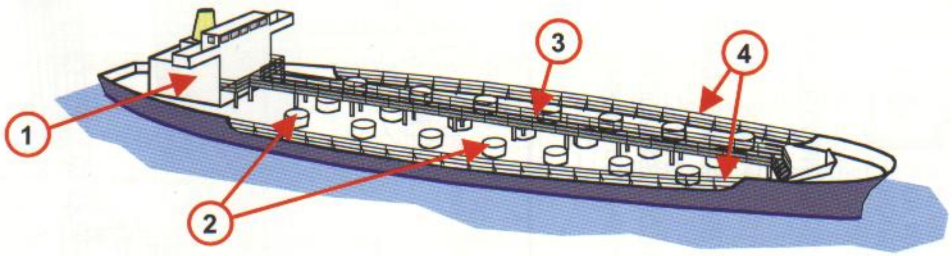
LOAD LINE REGULATION 13, EXPOSED POSITION CATEGORIES '1' AND '2'




EXPOSED SUPERSTRUCTURE END BULKHEADS ARE TO BE OF ADEQUATE STRENGTH
SIDE SCUTTLES (PORT HOLES) BELOW THE FREEBOARD DECK ARE TO BE FITTED INTERNALLY WITH STEEL HINGED AND DOGGED DEADLIGHTS
≥ 0.025 (BEAM) OR 500mm
MUST BE ABOVE DEEPEST W/L
FREEING PORT AREA 'A' FOR A METRE LENGTH OF BULWARK OF HEIGHT ≤ 1.2 M & LENGTH > 20 M
A ≥ 0.07 KL M²
'KL' = LENGTH OF BULWARK BUT ≤ 0.7L
≥ 1.75 T/M_
≥ 600 mm
DECK
HATCH STRENGTH & COAMING HEIGHT
POSITION 1
CLOSING DEVICE ≥ 450 mm
DECK
TANK AIR PIPE HEIGHT
ON SUPERSTRUCTURE DECKS
(ON THE FREEBOARD DECK)
MIN. HEIGHT 760 mm
F'O'CSLE DOOR
SEAL
≥ 600 mm
DECK
STEEL W/T DOOR SILL HEIGHT
POSITION 1
(FOR POSITION 2)
MIN. HEIGHT 380 mm
≥ 1 M
≤ 380 mm
≤ 380 mm
≤ 230 mm
DECK
SIDE RAILINGS HEIGHT & SPACING
POSITION 1 & 2
≥ 1.30 T/M²
≥ 450 mm
DECK
HATCH STRENGTH & COAMING HEIGHT
POSITION 2
≥ 1 M
≥ 230 mm
DECK
BULWARK HEIGHT & FREEING PORT
POSITION 1 & 2
VENTILATOR COAMING HEIGHTS
≥ 900 mm
≥ 760 mm
POS'N 1
DECK
POS'N 2
VENTILATORS MUST BE EQUIPED WITH THE MEANS OF MAKING WEATHERTIGHT IF THE COAMING HEIGHTS ARE LESS THAN :-
4.5 MERTES (POSITION 1)
OR
2.3 METRES (POSITION 2)
VENTILATORS MUST BE SUPPORTED IF THE COAMING HEIGHT ≥ 900 mm
PRINCIPAL LOADLINE STRUCTURAL REQUIREMENTS APPLIED TO A GENERAL CARGO SHIP
CHAPTER 2 OF ANNEX I REGULATIONS 11 to 25
FOR MAINTAINING WEATHERTIGHTNESS OF THE HULL, CREW SAFETY AND ADEQUATE DRAINAGE OF EXPOSED DECKS
OVERBOARD DISCHARGES & SCUPPERS
IF
'Z' > 450 mm
OR
'H' < 600 mm
EXTENDED SPINDLE
NON-RETURN VALVE
Z
H
SUMMER W/LINE
THEN THE OPENING MUST BE FITTED WITH NON-RETURN VALVE, AS CLOSE TO SHIP'S SIDE AS IS PRACTICAL AND OPERATED BY AN EXTENDED SPINDLE FROM DECK

Regulation 26 specifies particular requirements for ships assigned with reduced minimum freeboard (type 'A', type 'B-60' and type 'B-100' vessels). These are summarised in the following sketch.

ADDITIONAL FREEBOARD CONDITIONS FOR TYPE 'A', 'B-60' AND 'B-100' SHIPS



1 THE MACHINERY CASING SHOULD BE ENCLOSED IN A SUPERSTRUCTURE OR AN EQUIVALENT STRENGTH DECK HOUSING OF AT LEAST STANDARD HEIGHT.

2 HATCHES ON THE FREEBOARD DECK ARE TO BE CLOSED BY STEEL WEATHERTIGHT COVERS

3 THE WEATHER DECK IS FITTED WITH A PROTECTED RAISED STEEL WALKWAY (THE 'FLYING BRIDGE') TO ALLOW SAFE ACCESS FOR THE CREW.
ALTERNATIVELY UNDERDECK WALKWAYS ALONG EACH SIDE OF THE HULL THAT ARE WELL LIT, GAS-TIGHT AND VENTILATED WITH PROTECTED DECK ACCESSES NOT MORE THAN 90 METRES APART. A SINGLE UNDERDECK WALKWAY IS ACCEPTABLE, PROVIDED THAT IT IS AT LEAST 0.2 (SHIP'S WIDTH) INBOARD OF THE SHIP'S SIDE.

4 AT LEAST HALF THE LENGTH OF THE FREEBOARD DECK IS TO BE PROTECTED BY OPEN RAILS RATHER THAN BULWARKS. IF UNDERDECK WALKWAYS ARE PROVIDED , INSTEAD OF A RAISED WALKWAY, THEN OPEN RAILS SHOULD BE FITTED ALONG THE ENTIRE LENGTH THE SHIP'S SIDES

The examples of the general cargo ship on the previous page and the tanker shown above only highlight the main points in the Conditions of Assignment. The regulations and the amendments should be consulted for the full details and, in some cases, alternative criteria. The following are worth particular attention;-
Regulation 15 and its amendments detail the minimum criteria for the securing arrangements of hatches, the strength of the hatch covers and that of any supporting beams
Regulation 16 allows for the reduction of height or the elimination completely of hatch coamings, provided that the authorities are satisfied that this does not impair the safety of the ship. This may be achieved by locating hatchways in particularly sheltered positions, or the authority may impose an addition to the minimum freeboard to ensure such hatches are not exposed to heavy seas.

The principal aims of **Regulations 11** to **26** of the 'Conditions of Assignment', are summarised as:-

1) **Prevention of flooding of the ship's reserve buoyancy in heavy weather**

   The regulations consider the strength, effective weathertightness and disposition of all exposed openings into the hull and enclosed superstructures. (E.g. doors, hatches, ventilators, etc.)

2) **Safe Access for the crew to carry out their normal duties in heavy weather**

   Exposed decks are required to be fitted with guard rails or bulwarks, of adequate strength and height along their outboard limits. Raised protected gangways or underdeck walk-through trunkings must be fitted as alternative means of crossing the weather deck if it is frequently awash at sea when the vessel is fully loaded.

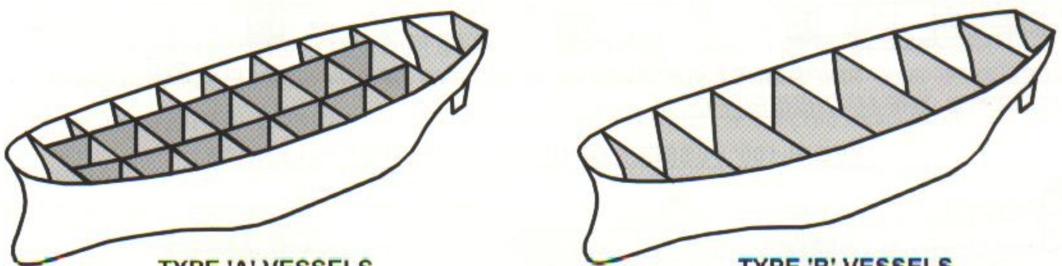
3) **Adequate drainage of water shipped onboard the exposed decks in heavy weather**

   Water shipped onto the weather deck by breaking seas must be able to drain overboard relatively freely, either through 'freeing ports' of sufficient area in the bulwark or through open railings. This improves crew safety by reducing the 'wetness' of the deck, reduces any adverse stability effect due to the weight of water accumulating on the upper deck and minimises the risk of flooding through exposed doorways, air pipes etc.
   Large lengths of bulwark are discouraged on vessels with very low freeboards where the deck is regularly awash.

FREEBOARD TABLES FOR TYPE 'A' AND TYPE 'B' SHIPS

Chapter 3 contains the information and equation for calculating the Summer Freeboard for any given vessel and how this value relates to the other seasonal/regional freeboards (i.e. Tropical, Winter and WNA). **Regulation 27** states that tankers, known as type 'A' vessels, are to be considered separately from all other ships, which are categorised as type 'B' vessels. The hulls of ships built to carry bulk liquids are divided into many weathertight tanks with only small steel weathertight access hatches. As such, they are less likely to founder by the sea flooding through the hatches so are allowed a smaller Summer Freeboard than a dry cargo ship of the same size and hullform. Subsequently, the rules have permitted freeboard reductions to Type 'B' ships if they are fitted with steel hatch covers and have sufficient subdivision to meet certain damage stability criteria, described in the revised **Regulation 27**, given by the '1988 Protocol'. These are 'B-100' and 'B-60' vessels.
Regulation 28 lists tabulated freeboard values/ship length for types 'A' and 'B' vessels of standard hullform.

TYPE 'A' AND 'B' VESSELS WITH REGARD TO FREEBOARD ASSIGNMENT



TYPE 'A' VESSELS
(TANKERS)

THE LONGITUDINAL HULL FRAMING RESULTS IN A HIGH DEGREE OF HULL SUBDIVISION

ACCESS TO THE UNDERDECK COMPARTMENTS IS LIMITED TO SMALL STEEL WEATHERTIGHT HATCHES

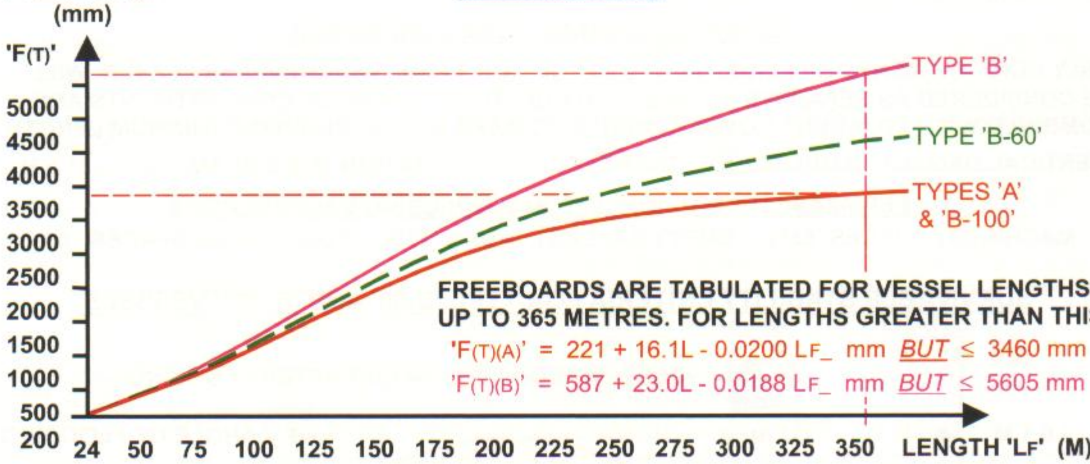
TYPE 'B' VESSELS
(ALL OTHER VESSELS)

THE TRANSVERSE HULL FRAMING RESULTS IN A LIMITED DEGREE OF HULLSUBDIVISION

ACCESS TO THE UNDERDECK COMPARTMENTS IS THROUGH LARGE HATCHES, WHICH MAY BE EQUIPED WITH WOODEN COVERS

TABULATED SUMMER FREEBOARD VALUES

(REGULATION 28)



$'F_{(T)}'$ FOR TYPE 'B-100' = $'F_{(T)(A)}'$ & $'F_{(T)}$ FOR TYPE 'B-60' = $'F_{(T)(B)}' - 0.6\{'F_{(T)(B)}' - 'F_{(T)(A)}'\}$

WHERE 'B-60' AND 'B-100' SHIPS ARE TYPE 'B' VESSELS WITH STEEL HATCHES AND ADEQUATE SUBDIVISION TO SURVIVE ONE OR TWO ADJACENT COMPARTMENT FLOODING RESPECTIVELY

REGULATION 27-10 GIVES TABULATED FREEBOARD CORRECTIONS FOR TYPE 'B' SHIPS WITH VULNERABLE HATCH COVERS OF REDUCED STRENGTH, PLACED IN POSTION '1' CATEGORY

DAMAGE SURVIVABILITY CRITERIA FOR TYPE 'B-60' AND 'B-100' SHIPS

The development of large bulk carriers since the late 1960's has resulted in type 'B' ships that have a significant degree of longitudinal subdivision due to wing ballast tanks being incorporated into the hull. The cargo hatches, though large, are generally of substantial steel construction with relatively high coamings, compared to the older general cargo vessels. Such ships of more than 100 metres in length can qualify for reduced type 'B' freeboard if, when fully laden, they meet specified minimum damage stability requirements after being flooded in a single hull compartment (Type 'B-60' ships with a partial freeboard reduction towards type 'A') or two adjacent fore and aft compartments. (Type 'B-100' ships with a freeboard equivalent to type 'A'). These requirements are shown below.

THE DAMAGE SCENARIOS TO BE CONSIDERED FOR TYPE 'B-60' AND 'B-100' SHIPS

TYPE 'B-60' AND 'B-100' VESSELS MUST EXCEED 100 METRES IN LENGTH

TYPE 'B-60' VESSELS OF UP TO 150 METRES IN LENGTH

THESE TYPE 'B-60' SHIPS MUST SURVIVE THE FLOODING OF ANY SINGLE COMPARTMENT EXCLUDING THE MACHINERY ROOM.

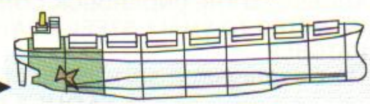
TYPE 'B-100' VESSELS OF UP TO 150 METRES IN LENGTH

THESE TYPE 'B-100' SHIPS MUST SURVIVE FLOODING OF ANY TWO ADJACENT FORE AND AFT COMPARTMENTS EXCLUDING THE MACHINERY ROOM.*

TYPE 'B-60' AND 'B-100' VESSELS OF MORE THAN 150 METRES IN LENGTH

← TYPE 'B-60' VESSEL

TYPE 'B-100' VESSEL →



IF A TYPE 'B-60' OR 'B-100' SHIP EXCEEDS 150 M IN LENGTH, THE MACHINERY ROOM MUST BE CONSIDERED AS A FLOODABLE COMPARTMENT.

EXTENT OF DAMAGE TO BE CONSIDERED

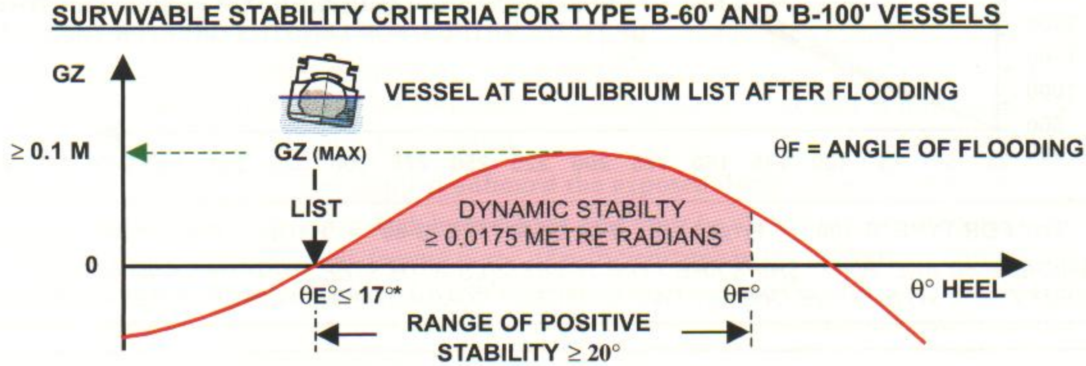
* ONLY COMPARTMENTS OF LENGTHS $\geq 0.33\sqrt[3]{L}$ OR 14.5M, WHICHEVER IS LESSER, WILL BE CONSIDERED AS SEPARATE IN THESE CALCULATIONS. SHORTER COMPARTMENTS ARE COMBINED WITH ADJACENT COMPARTMENTS TO MAKE UP THE REQUIRED MINIMUM LENGTH

VERTICAL DAMAGE IS UNLIMITED, TRANSVERSE PENETRATION = 0.2 BEAM *BUT* $\leq$ 11.5 M

PERMEABILITIES TO BE USED IN THE BILGING CALCULATIONS

MACHINERY SPACES 85%, EMPTY BALLAST TANKS 95%, FULL CARGO SPACES 0%

SURVIVABLE STABILITY CRITERIA FOR TYPE 'B-60' AND 'B-100' VESSELS



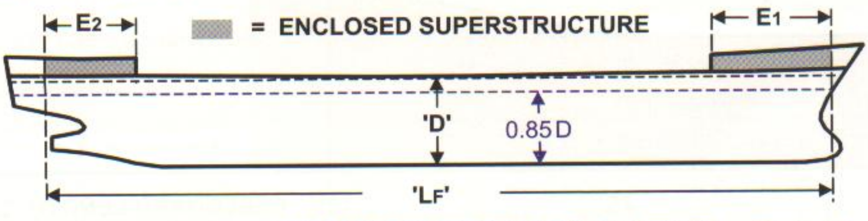
*** IF THE DECK EDGE IS IMMERSED, THEN θE°≤ 15°**

FREEBOARD CORRECTIONS FOR HULLFORM FEATURES

The following corrections are applied to a vessel's tabulated freeboard to account for design features that enhance or reduce its reserve buoyancy, when compared to the standard hullform.

FREEBOARD CORRECTION FOR SMALLER TYPE 'B' SHIPS WITH SHORT SUPERSTRUCTURES

THE TABULATED FREEBOARDS CONSIDER THAT TYPE 'B' SHIPS OF LESS THAN 100 METRES IN FREEBOARD LENGTH 'L_F' HAVE STANDARD ENCLOSED SUPERSTRUCTURES EXTENDING OVER AT LEAST 35% OF THE SHIP'S LENGTH 'L_F'. IF THE ACTUAL SUPERSTRUCTURE LENGTH IS LESS, THE FREEBOARD MUST BE INCREASED AS FOLLOWS

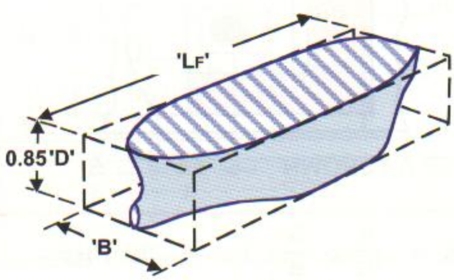


WITHIN 'L_F', **THE EFFECTIVE SUPERSTRUCTURE LENGTH 'E' = E_1 + E_2**

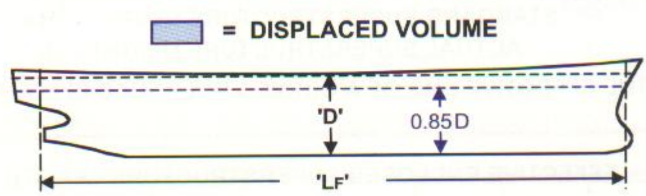
IF 'E' < 0.35 'L_F, *THEN* **CORRECTED FREEB'D = FREEB'D(TAB) + 7.5 (100 - L_F) $\left(0.35 - \frac{E}{L_F}\right)$ mm**

THE REGULATION (29) MAKES NO ALLOWANCE FOR A FREEBOARD REDUCTION IF 'E' > 0.35 'L_F'

FREEBOARD CORRECTIONS FOR HULL PROPORTIONS



BLOCK COEFFICIENT 'C_b' = $\dfrac{\text{DISPLACED VOLUME}}{L_F \times B \times 0.85D}$



CORRECTION FOR BLOCK COEFFICIENT 'C_b' AT A DRAFT OF 85% MOULDED HULL DEPTH

THE TABULATED FREEBOARD RELATES TO A BLOCK COEFFICIENT OF 0.68, IF THE SHIP'S 'C_B' VALUE EXCEEDS THIS THEN THE RATIO OF RESERVE BUOYANCY TO SHIP'S DISPLACEMENT IS DIMINISHED, SO THE FREEBOARD MUST BE INCREASED TO RESTORE THE BALANCE

FOR A SHIP'S VALUE OF 'CB' > 0.68, CORRECTED FREEB'D = FREEB'D (TAB) x $\dfrac{C_b + 0.68}{1.36}$

THE REGULATION (30) MAKES NO ALOWANCE FOR A FREEBOARD REDUCTION IF 'C_b' < 0.68

CORRECTION FOR FREEBOARD LENGTH TO MOULDED DEPTH RATIO

THE TABULATED FREEBOARD RELATES TO A LENGTH TO MOULDED DEPTH RATIO OF 15:1. IF A SHIP'S RATIO IS LESS THAN THIS THEN THE RATIO OF RESERVE BUOYANCY TO DISPLACEMENT IS DIMINISHED, SO THE FREEBOARD MUST BE INCREASED TO RESTORE THE BALANCE

FOR A SHIP WHERE 'D' > $\frac{L_F}{15}$ & 'L_F' < 120M, FREEB'D CORRECTION = + $\frac{L_F}{0.48}\left(D - \frac{L_F}{15}\right)$ mm

FOR A SHIP WHERE 'D' > $\frac{L_F}{15}$ & 'L_F' > 120M, FREEB'D CORRECTION = + 250 $\left(D - \frac{L_F}{15}\right)$ mm

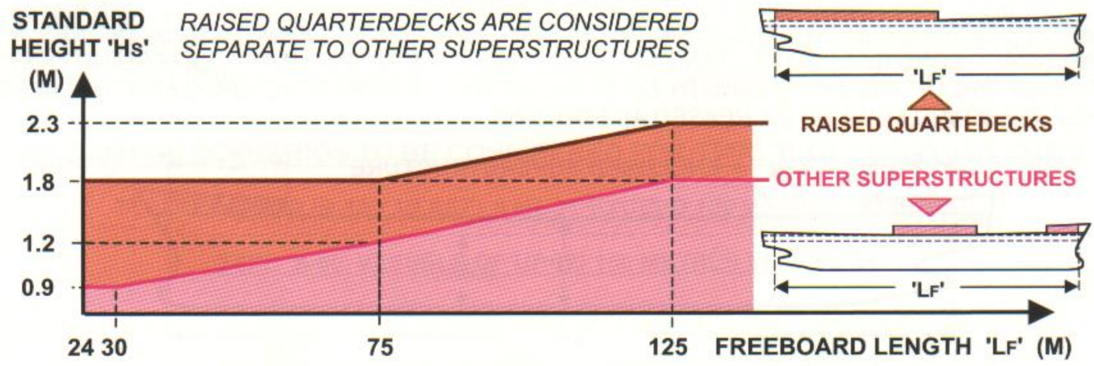
THE REGULATION (31) ALLOWS FOR THE CORRECTION TO BE NEGATIVE IF 'D' < $\frac{L_F}{15}$ AND

THE SHIP HAS AN ENCLOSED MIDSHIPS SUPERSTRUCTURE OF A LENGTH AT LEAST = 0.6 'L_F'

FREEBOARD CORRECTION FOR SUPERSTRUCTURES

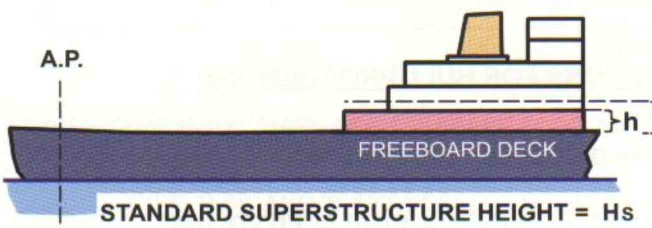
Enclosed superstructures of a significant height are important for providing reserve buoyancy above the freeboard deck. **Regulations 33** to **37** gives the criteria for assessing their effectiveness.

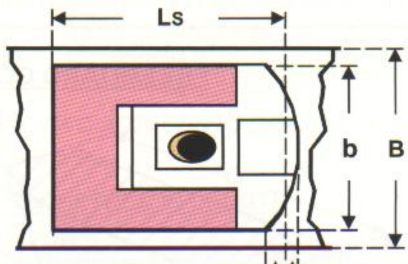
REGULATION 33 - STANDARD SUPERSTRUCTURE HEIGHTS



REGULATION 34 TO 36 - SUPERSTRUCTURE EFFECTIVE LENGTH

THE EFECTIVE LENGTH OF ENCLOSED SUPERSTRUCTURES <u>WITHIN THE FREEBOARD LENGTH</u>, IS DETERMINED AS SHOWN IN THE FOLLOWING DIAGRAMS :-





STANDARD SUPERSTRUCTURE HEIGHT = Hs

ACTUAL SUPERSTRUCTURE HEIGHT = h

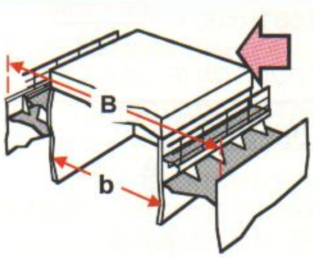
THE EFFECTIVE END OF A CURVED BULKHEAD IS LOCATED AS SHOWN → $\frac{2X}{3}$ $\frac{X}{3}$

EFFECTIVE ENCLOSED SUPERSTRUCTURE LENGTH 'E' = $\frac{h}{Hs} \times \frac{b}{B} \times Ls$ ***IF*** **'h' ≤ 'Hs'**

WHERE 'Hs' = REQUIRED STANDARD HEIGHT, 'B' = SHIP'S BREADTH AT THE SUPERSTRUCTURE
'h' = ACTUAL SUPERSTRUCTURE HEIGHT & 'b' = ACTUAL SUPERSTRUCTURE BREADTH

<u>THERE IS NO INCREASE ALLOWED IN EFECTIVE SUPERSTRUCTURE LENGTH IF 'h' > 'Hs'</u>

<u>RAISED QUARTERDECKS</u> WILL BE ACCOUNTED FOR IN THE SAME MANNER AS SHOWN ABOVE, EXCEPTING THAT THEIR EFFECTIVE LENGTH WILL NOT BE CONSIDERED TO EXCEED 0.6 'LF'



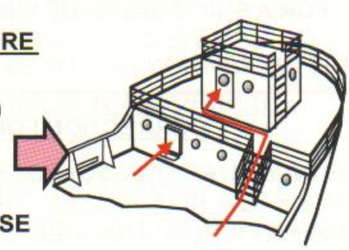
<u>HATCH TRUNK AS A SUPERSTRUCTURE</u>

TRUNK WIDTH 'b' ≥ 0.6 'B'

(SEE AMMENDED REGULATION 36)

<u>POOP AS A SUPERSTRUCTURE</u>

POOP CAN BE ACCESSED BY THE EXTERNAL LADDER AND DECK HOUSE



<u>'TRUNKS'</u> ARE BULKHEAD STRUCTURES THAT OPEN DIRECTLY INTO THE SPACE BELOW THE FREEBOARD DECK. **(E.G. MACHINERY CASINGS AND HATCHWAY COAMINGS)** THEY MAY BE CONSIDERED AS EFFECTIVE ENCLOSED SUPERSTRUCTURES, PROVIDED THAT;-
THEY ARE OF EQUIVALENT SUPERSTRUCTURE HEIGHT AND STRENGTH,
HAVE AN AVERAGE WIDTH AT LEAST 60% OF THE SHIP'S BREADTH AT THAT POINT
PROVIDE ADEQUATE PROTECTION FOR CREW AND FITTINGS AT ITS EXPOSED DECK LEVEL.

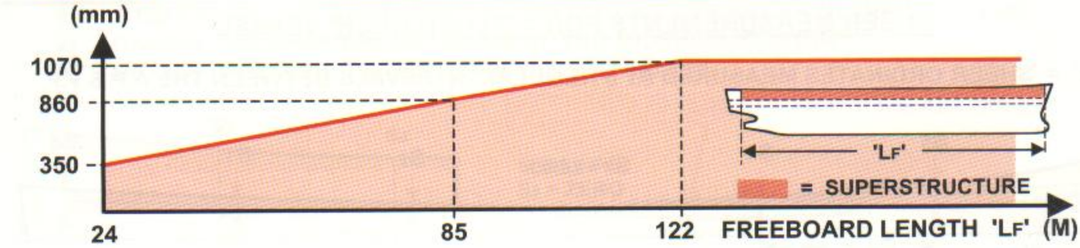
<u>POOPS AND BRIDGE DECKS</u> ENCLOSING WORK SPACES OR ACCOMMODATION MUST HAVE ALTERNATIVE ACCESS IN ADDITION TO WEATHERTIGHT DOORS ON THE FREEBOARD DECK, IN ORDER TO QUALIFY AS ENCLOSED SUPERSTRUTURES

FREEBOARD CORRECTION FOR SUPERSTRUCTURES (Cont.)

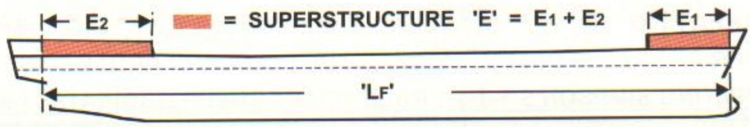
Regulation 37 gives the freeboard deductions that are allowed for effective enclosed superstructure length as a proportion of the ship's freeboard length.

REGULATION 37- FREEBOARD DEDUCTION FOR SUPERSTRUCTURE

REDUCTION FOR ANY SHIP WITH A FULL LENGTH ENCLOSED SUPERSTRUCTURE ABOVE THE FREEBOARD DECK

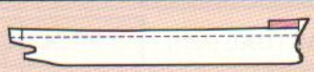


THE FREEBOARD DEDUCTION ALLOWED TO A SHIP DEPENDS ON THE EFFECTIVE LENGTH OF ITS SUPERSTRUCTURES AS A PROPORTION OF ITS HULL LENGTH AND THE TYPE OF VESSEL.



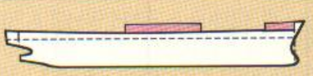
ALL TYPE 'A' VESSELS

E / LF	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
PERCENTAGE OF FULL HULL LENGTH SUPERSTRUCTURE DEDUCTION	0	7%	14%	21%	31%	41%	52%	63%	75.3%	87.7%	100%



(1) TYPE 'B' VESSELS WITH F'O'CSLE[1] BUT NO BRIDGE DECK

E / LF	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
PERCENTAGE OF FULL HULL LENGTH SUPERSTRUCTURE DEDUCTION	0	5%	10%	15%	23.5%	32%	46%	63%	75.3%	87.7%	100%



(2) TYPE 'B' VESSELS WITH F'O'CSLE[1] AND BRIDGE DECK ≥ 0.2 LF[2]

E / LF	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
PERCENTAGE OF FULL HULL LENGTH SUPERSTRUCTURE DEDUCTION	0	6.3%	13.7%	19%	27.5%	36%	46%	63%	75.3%	87.7%	100%

% DEDUCTIONS FOR INTERMEDIATE E / LF VALUES ARE FOUND BY LINEAR INTERPOLATION

NOTES

1. *USE TABLE '1' FOR ANY TYPE 'B' SHIP IF THE FO'C'SLE EFFECTIVE LENGTH ≥ 0.4 'LF'*
   IF THE FO'C'SLE LENGTH 'f' < 0.07 'LF' THEN THE % REDUCTIONS ARE MULTIPLIED BY;

   REDUCTION FACTOR FOR SHORT FO'C'SLES (TYPE'B' SHIPS) $= 5 \times \dfrac{0.07\,L_F - 'f'}{0.07\,L_F}$

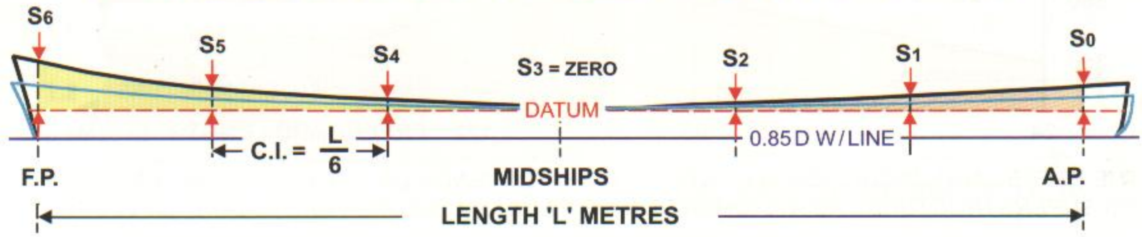
2. *INTERPOLATE BETWEEN TABLES '1' & '2' FOR BRIDGE DECK LENGTHS < 0.2 'LF'*

FREEBOARD CORRECTION FOR SHEER

The tabulated freeboards are based upon a standard sheer profile, measured at seven equally spaced stations along the hull. A process, based upon Simpson's 1-3-3-1 rule of area estimation (See Page 12), is applied separately to the sheer measurements from the aft perpendicular to midships and the forward perpendicular to midships to produce measures of effective sheer aft and forward respectively of the uppermost continuous deck. **Regulation 38** prescribes how a ship's sheer is assessed to determine whether the vessel has extra sheer or a sheer deficiency, relative to the standard profile.

SHEER MEASUREMENTS FOR A 'FLUSH DECK' VESSEL

'Sx' = SHEER ORDINATES MEASURED AT 6 REGULAR INTERVALS BETWEEN THE A.P & F.P



= SHEER AREA FWD — = STANDARD SHEER PROFILE = SHEER AREA AFT

THE ACTUAL SHEER PROFILE IS COMPARED WITH THE STANDARD PROFILE AS SHOWN BELOW

APPLYING SIMPSON'S 1-3-3-1 RULE TO AFT SHEER ORDINATES (mm)

STATION	STANDARD SHEER	MULTIPIER	PRODUCT	ACTUAL SHEER	MULTIPIER	PRODUCT
A.P.	25.0 (L/3 + 10)	1	+ 25.0(L/3 + 10)	S_0	1	+ S_0
L/6	11.1 (L/3 + 10)	3	+ 33.3(L/3 + 10)	S_1	3	+ $3S_1$
2L/6	2.8 (L/3 + 10)	3	+ 8.4(L/3 + 10)	S_2	3	+ $3S_2$
MIDSHIPS	ZERO	1	ZERO	ZERO	1	ZERO
		STANDARD (STD)	Σ PRODUCT		ACTUAL (ACT)	Σ PRODUCT

APPLYING SIMPSON'S 1-3-3-1 RULE TO FWD SHEER ORDINATES (mm)

STATION	STANDARD SHEER	MULTIPIER	PRODUCT	ACTUAL SHEER	MULTIPIER	PRODUCT
F.P.	50.0 (L/3 + 10)	1	+ 50.0(L/3 + 10)	S_7	1	+ S_7
5L/6	22.2(L/3 + 10)	3	+ 66.6(L/3 + 10)	S_6	3	+ $3S_6$
4L/6	5.6(L/3 + 10)	3	+ 16.8(L/3 + 10)	S_5	3	+ $3S_5$
MIDSHIPS	ZERO	1	ZERO	ZERO	1	ZERO
		STANDARD (STD)	Σ PRODUCT		ACTUAL (ACT)	Σ PRODUCT

THE AREA UNDER THE FWD OR AFT SHEER PROFILE = 3 x 0.125 C.I. Σ PRODUCT mm²

& THE EFFECTIVE SHEER, FWD OR AFT $= \dfrac{3 \times 0.125 \text{ C.I. } \Sigma \text{ PRODUCT}}{3 \times \text{C.I.}}$ **mm**

EFFECTIVE SHEER (FWD OR AFT) = 0.125 Σ PRODUCT, *FOR STANDARD AND ACTUAL SHEER*

& SHEER EXCESS ('δS_A' OR 'δS_F') = 0.125 (Σ PRODUCT(ACT) − Σ PRODUCT(STD)) mm

(A SHEER DEFICIENCY AFT OR FWD PRODUCES A NEGATIVE VALUE OF 'δS_A' OR 'δS_F')

FREEBOARD CORRECTION FOR SHEER (Cont.)

Regulation 38 defines how sheer should be measured in vessels with other than flush deck profiles. It allows for the superstructure height, *in excess of standard,* to increase effective sheer of a ship.

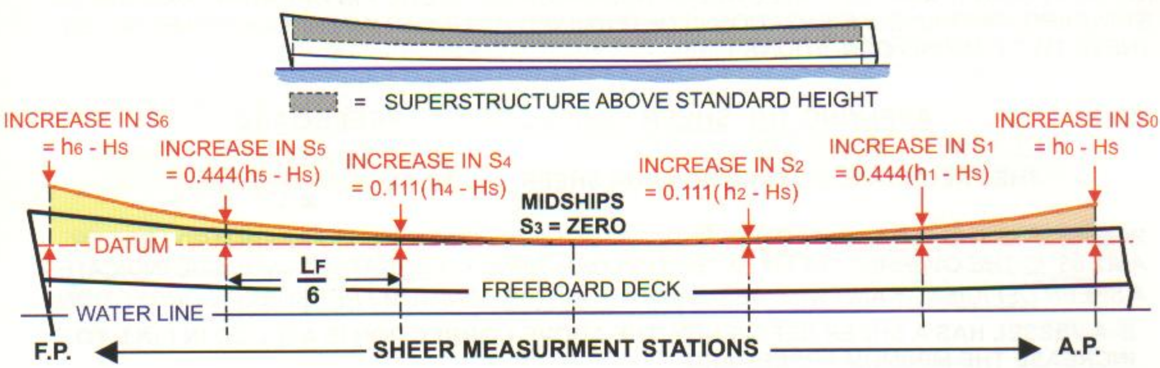
1 - SHEER MEASUREMENTS FOR A STEPPED PROFILE DECK WITH NEGATIVE SHEER

THE PROFILE BELOW IS TYPICAL OF SOME TUGBOATS. NOTE THAT THE DECK IS STEPPED AND THAT ITS LOWEST POINT IS AFT OF THE MIDSHIPS STATION. THE ***FREEBOARD DATUM IS ;-***

1) **STEPPED BY THE SAME VERTICAL EXTENT AS THE BREAK IN THE DECK LINE**
2) **DRAWN THROUGH THE DECKLINE AT THE MIDSHIPS STATION**
3) **DRAWN PARALLEL TO THE DESIGNED WATERLINE**

2 - SHEER MEASUREMENTS OF A FULL LENGTH SUPERSTRUCTURE DECK OF GREATER THAN STANDARD HEIGHT

IF A VESSEL HAS A FULL LENGTH SUPERSTRUCTURE ABOVE THE FREEBOARD DECK, THEN ***SHEER IS MEASURED AT THE UPPER DECK*** *AND THE PROFILE IS ADJUSTED TO GIVE EXTRA SHEER IF THE SUPERSTRUCTURE HEIGHT 'hx' EXCEEDS THE STANDARD HEIGHT 'Hs'. THIS GIVES CREDIT TO OVER-STANDARD HEIGHT OF SUPERSTRUCTURE THAT IS NOT ALLOWED BY THE 'SUPERSTRUCTURE FREEBOARD CORRECTION', OUTLINED ON PAGES 273 & 274*

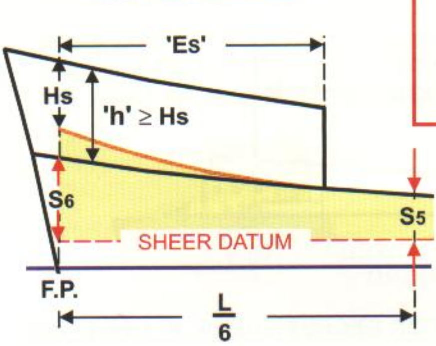


3 - SHEER MEASUREMENT OF A VESSEL WITH A RAISED FO'C'SLE AND POOP

IF A SHIP HAS A RAISED FO'C'SLE AND / OR A RAISED POOP, THEN EXTRA SHEER WILL BE CREDITED, PROVIDED THAT THE SUPERSTRUCTURES ARE EITHER;-

1) *OF AT LEAST STANDARD HEIGHT 'Hs' WITH GREATER SHEER THAN THE FREEBOARD DECK*
2) *OF A HEIGHT 'h' THAT IS GREATER THAN STANDARD HEIGHT 'Hs'*

RAISED FO'C'SLE



SHEER CREDIT TO FWD SHEER 's' IS GIVEN AS ;-

$$'s' = \frac{'h' - Hs}{3} \times \frac{'Es'}{'L'} \quad \text{mm}$$

WHERE SUPERSTRUCTURE EFECTIVE LENGTH 'Es' $\leq$ 0.5 L

's' IS TO BE ADDED TO THE EFFECTIVE FWD SHEER

THIS IS THE EQUIVALENT OF ADJUSTING THE SHEER PROFILE TO FOLLOW THE ORANGE LINE OPPOSITE

EFFECTIVE AFT SHEER WILL BE INCREASED IN THE SAME MANNER FOR VESSELS WITH RAISED POOPS

FREEBOARD CORRECTION FOR SHEER (Cont.)

The forward and aft sheer deficiency or excess, 'δS_F' and δS_A', are combined to give a single sheer difference 'δS', between the standard and actual sheer profiles, which is then used in the correction of freeboard for sheer. This procedure puts more emphasis on a fwd sheer deficiency than an excess aft.

THE SINGLE MEASURE OF EXCESS OR DEFICIENT SHEER

THE OVERALL SHEER EXCESS OR DEFICIENCY 'δS' = $\dfrac{'\delta S_A' + '\delta S_F'}{2}$

PROVIDED THAT BOTH 'δS_A' AND 'δS_F' ARE EITHER BOTH POSITIVE (EXCESS SHEER) OR NEGATIVE (SHEER DEFIENCY)

IF 'δS_F' IS NEGATIVEAND 'δS_A' IS POSTIVE (DEFICIENT SHEER FWD BUT EXCESS SHEER AFT)

THEN **THE OVERALL SHEER DEFICIENCY 'δS' = $\dfrac{'\delta S_F'}{2}$**

THE RULE DOES NOT ALLOW FOR A FWD DEFICIENCY OF SHEER TO BE COMPENSATED BY AN EXCESS SHEER IN THE AFT HALF OF THE HULL

IF 'δS_F' IS POSITIVEAND 'δS_A' IS NEGATIVE (EXCESS SHEER FWD BUT DEFICIENT SHEER AFT)
AND THE AFT ACTUAL EFFECTIVE SHEER $\geq$ 75% 0F THE STANDARD EFFECTIVE SHEER

THEN **THE OVERALL SHEER EXCESS OR DEFICIENCY 'δS' = $\dfrac{'\delta S_A' + '\delta S_F'}{2}$**

BUT IF THE AFT EFFECTIVE SHEER < 50% 0F THE STANDARD EFFECTIVESHEER

THEN **THE OVERALL SHEER DEFICIENCY 'δS' = $\dfrac{'\delta S_A'}{2}$**

THE RULE DOES ALLOW FOR EXCESS SHEER FWD TO FULLY COMPENSATE FOR DEFICIENT AFT SHEER, PROVIDED THAT THE DEFICIENCY IS LESS THAN 25% OF STANDARD SHEER AFT. NO SUCH COMPENSATION IS ALLOWED IF THE SHEER DEFICIENCY IS GREATER THAN 50% OF STANDARD. PARTIAL COMPENSATION IS DETERMINED BY LINEAR INTERPOLATION BETWEEN THESE TWO LIMITING CONDITIONS

APPLYING THE SHEER CORRECTION TO FREEBOARD

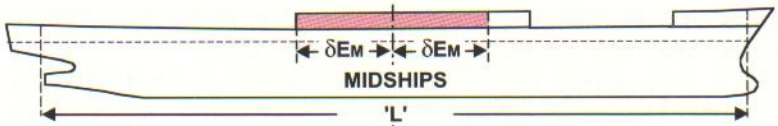
THE FREEBOARD CORRECTION FOR SHEER = - 'δS' $\left(0.75 - \dfrac{'E'}{2\,'L'}\right)$ mm

WHERE 'E' IS THE TOTAL EFFECTIVE SUPERSTUCTURE LENGTH, 'L' IS FREEBOARD LENGTH AND 'δS' IS THE OVERALL SHEER EXCESS OR DEFICIENCY. A NEGATIVE 'δS' VALUE INDICATES A SHEER DEFICIENCY AND WILL RESULT IN A POSITIVE MINIMUM FREEBOARD CORRECTION.

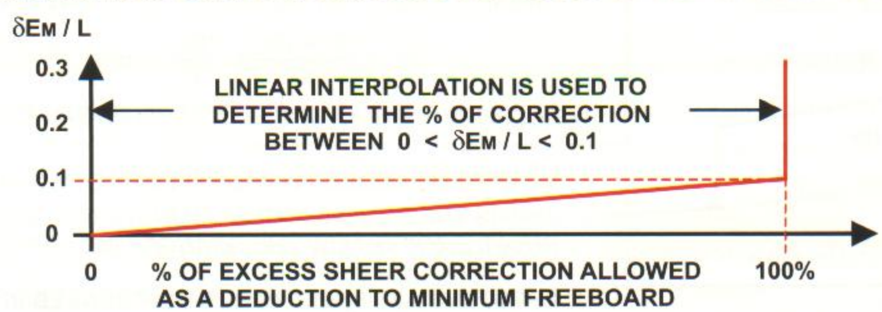
IF A VESSEL HAS A SHEER DEFICIENCY, THE ABOVE CORRECTION IS APPLIED IN FULL TO INCREASE THE MINIMUM FREEBOARD.

IF A VESSEL HAS A SHEER EXCESS, THE CORRECTION MAY BE APPLIED TO REDUCE THE MINIMUM FREEBOARD, PROVIDED THAT THE FOLLOWING RULES ARE OBEYED

①



δE_M = THE MINIMUM LENGTH OF A MIDSHIPS SUPERSTRUCTURE FWD OR AFT OF MIDSHIPS



② **THE MAXIMUM REDUCTION IN FREEBOARD ALLOWED FOR EXCESS SHEER = 1.25 'L' mm**

FREEBOARD CORRECTION FOR SHEER (Cont.)

The previous page shows how overall sheer excess or deficiency, 'δS' is multiplied by a factor that accounts for the length of all the vessel's superstructures as a proportion of freeboard length. The resulting 'sheer correction' will be added to give an increase in minimum freeboard, if the vessel has an overall deficiency in sheer.
If, however, a vessel has excess sheer, the extent to which the minimum freeboard may be reduced depends upon the effective length of any midships superstructure. This is to ensure that the midships deck edge is not submerged at very small angles of heel.

THE MINIMUM ALLOWED BOW HEIGHT AT THE SUMMER DRAFT

Regulation 39 specifies a minimum allowable bow height that must be maintained when the vessel is floating to the summer loadline at its design trim. The assigned Summer Freeboard for a vessel must be increased, if necessary, to ensure that the minimum bow height requirements are met

MINIMUM BOW HEIGHT 'HB'

THE MINIMUM BOW HEIGHT 'HB', MEASURED AT THE F.P. AT THE SUMMER W/L, IS GIVEN BY;-

$$'H_B' = 56\,'L_F'\left(1 - \frac{'L_F'}{500}\right) \times \frac{1.36}{'C_b' + 0.68} \text{ mm,} \quad \underline{IF} \text{ FREEBOARD LENGTH } 'L_F' < 250 \text{ M}$$

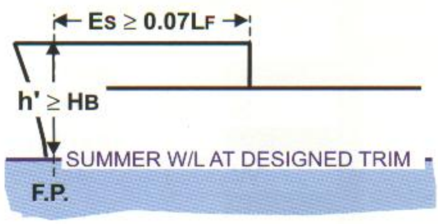
OR

$$'H_B' = 7000 \times \frac{1.36}{'C_b' + 0.68} \text{ mm} \quad \underline{IF} \text{ FREEBOARD LENGTH } 'L_F' \geq 250 \text{ M}$$

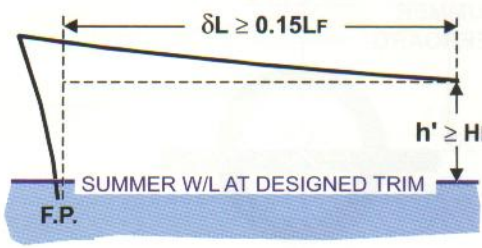
WHERE THE BLOCK COEFF^NT $'C_b' = 'L_F' \times \text{BEAM} \times 0.85(\text{DEPTH})$ *BUT* $'C_b' \geq 0.68$

THE FREEBOARD MUST REMAIN GREATER THAN THE MINIMUM BOW HEIGHT AFT OF THE F.P, FOR THE FOLLOWING SPECIFIED LENGTHS, DEPENDING UPON WHETHER THE BOW HEIGHT IS ACHIEVED BY SHEER OR A RAISED FO'C'SLE

RAISED FO'C'SLE CRITERIA TO ACHIEVE MINIMUM BOW HEIGHT



FWD SHEER CRITERIA TO ACHIEVE MINIMUM BOW HEIGHT



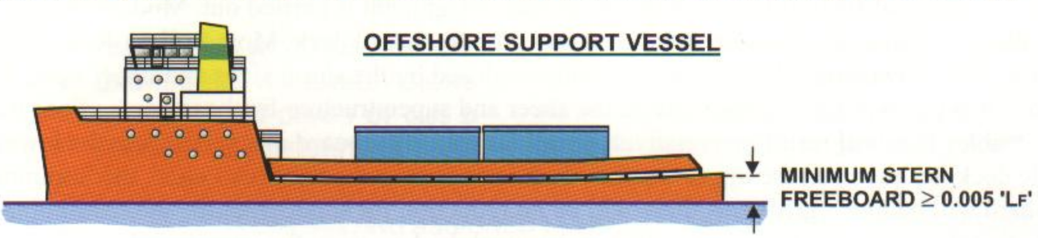
TRIM REQUIREMENTS FOR HIGH FO'C'SLE. LOW AFT DECK HULLS

CHAPTER 3, PAGES 65 & 66, DESCRIBES HOW THE 'FREE TRIM EFFECT' INCREASES THE STERN TRIM OF VESSELS WITH THIS HULLFORM WHEN THEY HEEL OVER DURING ROLLING.

SUCH SHIPS ARE REQUIRED AT ALL TIMES TO MAINTAIN A STERN FREEBOARD $\geq 0.005\ 'L_F'$

THESE SHIPS ARE OFTEN DESIGNED TO SAIL WITH A SLIGHT BOW DOWN TRIM TO MAINTAIN THIS MINIMUM STERN FREEBOARD AND ENSURE SUFFICIENT RESERVE BUOYANCY AFT. THIS MUST BE ALLOWED FOR IN DETERMINING THE MINIMUM FULLY LOADED BOW HEIGHT

OFFSHORE SUPPORT VESSEL



DETERMINING A SHIP'S ASSIGNED MINIMUM FREEBOARDS

The freeboard deck must be decided upon so that the following measurements can be determined;- **Moulded Depth 'D', Moulded Beam 'B', Freeboard Length 'LF' and the Block Coefficient 'Cb'** The calculation is then carried out in the following sequence

1 The tabulated freeboard for the vessel's length is determined from the tables appropriate to vessel type (**Regulation 28**) and increased, if necessary, for type 'B' vessels as follows:-
   (a) Addition for under-strength exposed hatch covers (**Regulation 27-10**)
   (b) Addition for insufficient effective superstructure length if 'LF' < 100 metres (**Regulation 29**)

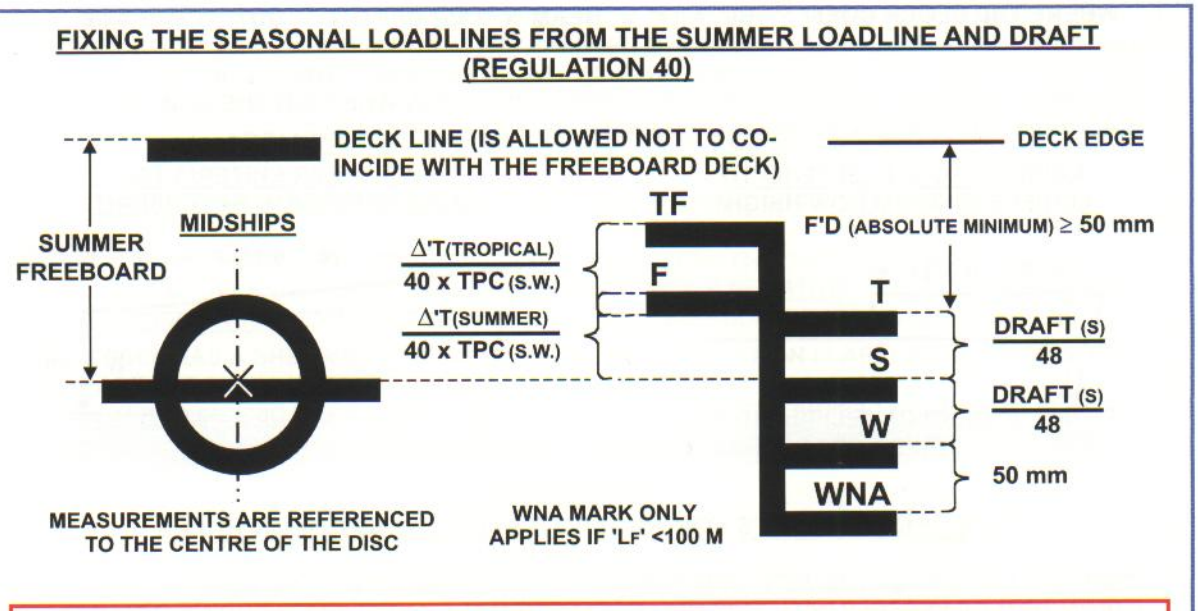
2 If the 'Cb' value > 0.68, the freeboard is then reduced by the factor given in **Regulation 30**.

3 Freeboard can now be corrected with the following additions or subtractions
   (c) Length to Moulded depth ratio (**Regulation 31**)
   (d) Superstructure length as a proportion of freeboard length (**Regulations 33** to **37**)
   (e) Excess or deficient Sheer, relative to the standard profile (**Regulation 38**)

4 The Freeboard is increased, if necessary, to ensure that the minimum bow requirements are met (**Regulation 39**)

5 The 'Deck Line' may be located amidships but somewhere other than the freeboard deck edge, if this is more practical for the particular design of the ship. The assigned freeboard, measured from the deck line, must then be adjusted to ensure that the loadlines are the correct distance below the freeboard deck. (**Regulation 32**)

6 The additional seasonal freeboards can be determined as these are measured from the Summer Loadline and the Summer midships draft, as shown below. (**Regulation 40**)



THE ABSOLUTE MINIMUM S.W. FREEBOARD, (I.E. THE TROPICAL), MUST NOT BE LESS THAN ;- 50 mm *OR,* FOR VESSELS WITH REDUCED STRENGTH HATCH COVERS, 150 mm

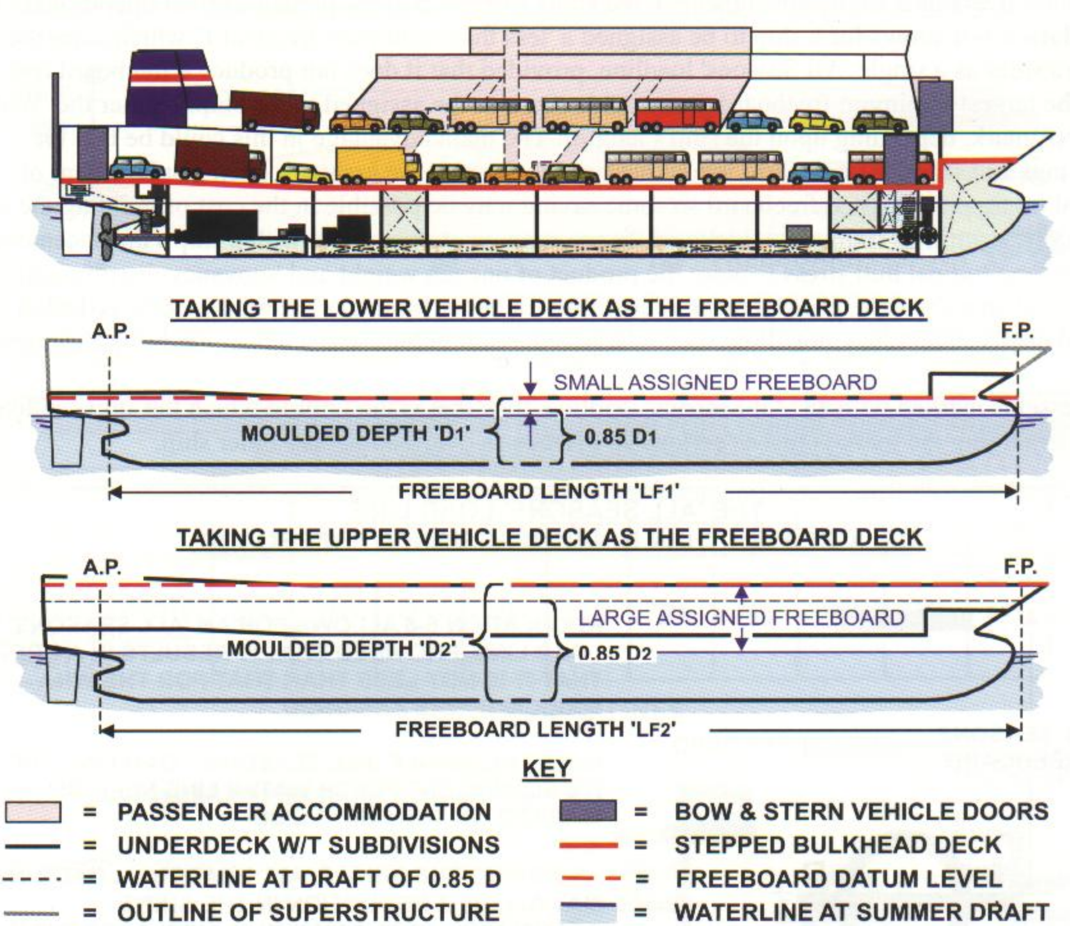
There is a degree of flexibility in how the freeboard assignment is carried out. Modern ship design often allows a choice of which deck to be taken as the freeboard deck. Most Roll on-Roll off ferries are built with a lower vehicle deck that is totally enclosed by the ship's sides and upper open deck and so if it is taken as the freeboard deck, the sheer and superstructure freeboard deductions will be considerable. This will result in a relatively small Summer Freeboard measured from the lower vehicle deck. If, however, the upper weather deck is chosen as the freeboard deck, the Summer Freeboard will be much greater as there will be no reduction for superstructure and restrictions for the loading doors but the datum line will now be much higher up on the vessel's side.

Consequently, measuring the freeboard from either the lower or the upper vehicle deck should result in producing approximately the same actual Summer Draft, though there may be slight advantages in one approach over the other, depending upon the details of a particular vessel.

There are also the 'Conditions of Freeboard Assignment' to consider, such as the requirements of weathertightness of the access arrangements to the lower vehicle deck. If the vessel is a passenger ship, then it will be impractical to move large numbers of passengers quickly from their cars to the above accommodation through regulation weathertight doors with their high sill steps. Consequently, vehicle decks usually have easy access up to the passenger spaces above via large stairwells fitted typically with pneumatic sliding doors that are flush with the deck. These stairwells often connect the upper open vehicle deck to the lower one and as the doors are only fire resistant but not weathertight, there is a possibility of flooding the lower deck through the stairwells. The effectiveness by which the ship's design protects the lower vehicle deck from flooding, will be an important factor in the consideration of freeboard and may result in additional freeboard being assigned to the vessel, depending upon where it is measured from.

Finally, it is quite possible that the ship's assigned freeboard will be restricted by the requirements of the subdivision regulations, as outlined in Chapter 10, whichever deck is taken as the freeboard deck. The lower vehicle deck will generally be the 'bulkhead deck' so the risk of flooding spaces beneath it must be minimised. Access downwards from it to machinery spaces, store rooms or other cargo compartments, must be through 'dogged' weathertight hatches or doors with coaming or sill heights, as given by chapter 2, of the load line regulations.

DIFFERENT OPTIONS FOR FREEBOARD ASSIGNMENT FOR A RO-RO FERRY



IN EACH CASE, VALUES OF FREEBOARD LENGTH, MOULDED DEPTH AND BLOCK COEFFICIENT WILL DIFFER, THOUGH THE RESULTING ASSIGNED SUMMER DRAFTS SHOULD BE VERY SIMILAR

DETERMINING A SHIP'S ASSIGNED MINIMUM FREEBOARDS (Cont.)

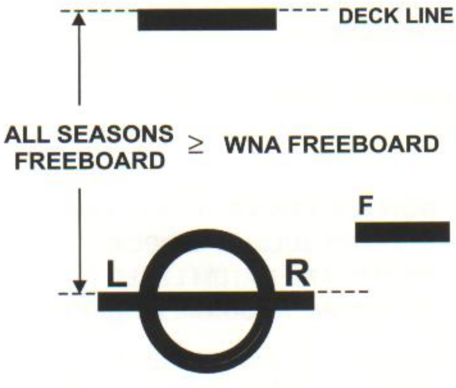
Freeboard requirements need careful consideration right at the start of the design stage of the vessel. In the past, shipowners have tended to want ships that had the maximum possible carrying capacity for their size so they sought the minimum possible assigned freeboard for their vessels. Up to 1970, a high proportion of the world's merchant fleet consisted of the general cargo ships built with generous sheer and prominent superstructures, especially raised fo'c'sles, (as shown on page 268). These ships often had a relatively low minimum freeboard, particularly if they were designed to operate as tramp ships, which were frequently chartered to carry bulk grain or mineral ores. The current 1966 Load Line Regulations were drawn up with these ships very much in mind.

Bulk carriers generally are still built for maximum deadweight capacity but much of the modern shipping consists of specialised vessels, which do not carry high density cargoes, so the emphasis on achieving minimum possible freeboards has generally diminished (See Chapter 3, page 66)
Car carriers and ro-ro vessels have a much smaller proportion of their displacement weight that is actually cargo, compared with the older type of ship, as vehicles take up a lot of space for relatively little weight. Container ships, similarly, tend to carry medium to low density manufactured goods and a considerable proportion of the volume within many containers is either packaging or broken stowage. Even tankers, since the MARPOL regulations (See Chapter 10) will not load to the full capacity of their hulls for they are now required to have separate ballast tanks, which cannot be used for cargo on the laden voyage, as was the case in the past.
There is also a growing number of commercial merchant ships providing off-shore services, such as surveying, dive support etc. but as such, do not carry any cargo at all (though they are classified as 'cargo vessels' in most regulations).

In today's shipping industry, some shipowners may not actually want their vessels to be assigned minimum freeboards for deadweights that the ships will never carry in their normal operational life. **Regulation 6-6** allows for a ship to be assigned a 'less than minimum freeboard', which is marked on a ship's sides as a single 'All Seasons' loadline, provided that it does not produce a freeboard less than the largest minimum freeboard that would otherwise be assigned to the ship. (Either the 'Winter' or WNA mark, depending upon the ship's length). The main advantage in this could be that the scantlings and strength of the hull would not have to support the greater loaded displacement of normal minimum assigned freeboard so some saving may be possible in the cost of building the ship. Generally, scantlings (hull plate thickness, frame spacing etc.) increase with a ship's displacement, though longitudinal hull stress is more the product of uneven weight and buoyancy distributions than simply the ship's weight. (See Chapter 8, covering 'Bending Moments'). Consequently, reducing the loaded displacement may not allow such a large reduction in hull scantlings as might be expected.

If a vessel is marked with an 'All Seasons' loadline, then that is the limit to which the ship can load and submerging the mark is just as serious an offence as overloading any other ship.

THE 'ALL SEASONS' LOAD LINE, OR 'GREATER THAN MINIMUM ASSIGNED FREEBOARD'



REGULATION 6-6 ALLOWS FOR AN 'ALL SEASONS' LOAD LINE, PROVIDED THAT IT RESULTS IN A DRFT THAT IT IS NOT LESS THAN THAT FOR THE WNA MARK IF IT WAS ASSIGNED

ONLY THE SINGLE 'ALL SEASONS' LOAD LINE AND ITS EQUIVALENT FRESH WATER LINE NEED BE MARKED ON THE SHIP'S SIDE.

ASSIGNING THE SINGLE 'ALL SEASONS' LOAD LINE MAY ALLOW A REDUCTION IN THE SHIP'S SCANTLINGS AS THE MAXIMUM DISPLACEMENT IS CORRESPONDINGLY REDUCED

THE ASSIGNMENT OF TIMBER FREEBOARDS

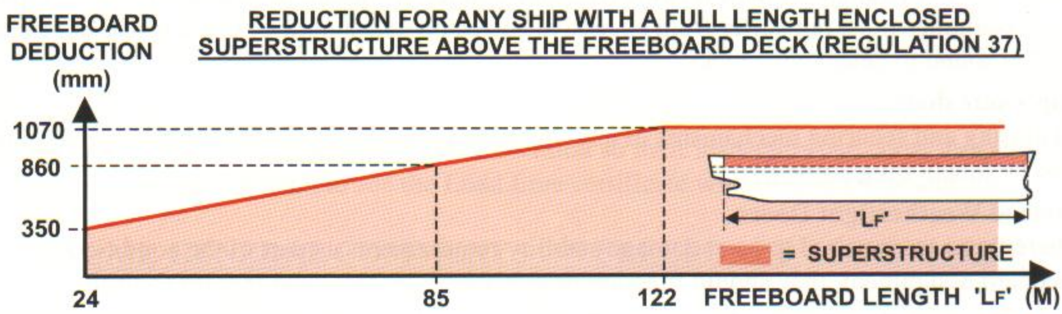
Ships regularly engaged in the timber trade can be assigned reduced ' lumber freeboards' that allow for an increase in maximum draft when the vessel is carrying a deck cargo of timber. The regulations consider a deck cargo of wood to be additional reserve buoyancy above the freeboard deck, provided that it is well secured and covers the entire length of the ship's cargo deck up to at least standard superstructure height. **Regulations 41** to **44** of the Load Line rules detail the 'Special Conditions of Assignment for Timber Freeboards', which have already been outlined in Chapter 5 that deals with stability requirements for ships operating under special circumstances. (See pages 100 to 102)

Regulation 45 gives a special table for the ***alternative percentage*** of 'Superstructure Deduction' that is to be applied in the freeboard assignment to obtain the minimum 'Lumber Summer Freeboard' and specifies the adjustments made to this to locate the other seasonal / regional lumber load lines.

The Lumber loadlines and the special timber minimum stability criteria only apply to the vessel when it is loaded with deck timber stows that meet the timber conditions of assignment. The normal marks limit the drafts for any other loaded condition of the ship.

Not all the ships that carry timber on deck have reduced timber freeboards assigned to them. It is for the shipowner to decide whether or not to build a ship that meets all the special timber conditions of assignment and many choose not to, in which case, the ship's draft will be restricted by the normal loadline even when it is loaded with timber on deck. **Regulations 41** to **44** would not apply to such a vessel but they still provide a useful guide to safe methods of securing the timber deck stow

TIMBER FREEBOARD DEDUCTION FOR SUPERSTRUCTURE

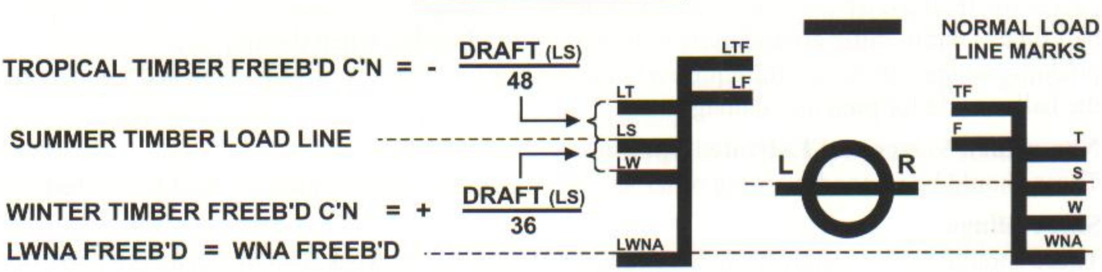


THE EFFECTIVE FREEBOARD LENGTH AND THE DEDUCTION ALLOWED FOR A FULL LENGTH SUPERSTRUCTURE FOR TIMBER SUMMER FREEBOARD IS DETERMINED THE SAME AS FOR ANY VESSEL, BUT A MORE GENEROUS PERCENTAGE OF THIS DEDUCTION IS APPLIED.

SUPERSTRUCTURE DEDUCTION FOR LUMBER SUMMER FREEBOARD
(USE INTERPOLATION FOR INTERMEDIATE VALUES OF 'E / L F')

E / LF	0	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
PERCENTAGE OF FULL HULL LENGTH SUPERSTRUCTURE DEDUCTION	20%	31%	42%	53%	64%	70%	76%	82%	88%	94%	100%

TIMBER LOAD LINES



THE ASSIGNMENT OF TIMBER FREEBOARDS (Cont.)

The table shown on the previous page somewhat surprisingly allows 20% of the full superstructure deduction to be used in calculating the Summer Timber Freeboard of a ship with no effective superstructure at all. The special conditions of assignment for timber load lines include a requirement that the ship shall have a raised fo'c'sle length at least 7% of the freeboard length so, clearly, the table of percentage deduction values cannot be applied for 'E/LF' values less than 0.07. The 20% deduction for a zero length of superstructure is simply included for easy interpolation and is beyond the valid range of 'E/LF' values

ROUTINE INSPECTION OF COMPLIANCE WITH LOADLINE CONDITIONS

Government surveyors annually inspect ships to ensure that the load line regulations are being obeyed and, in particular, the conditions of assignment are being maintained onboard. This 'Load Line' survey can cover any aspect of the rules discussed in the previous pages, though a surveyor is unlikely to re-measure the position of the marks unless he has a suspicion that they have been unlawfully moved or they are not clearly visible at all. The surveyor will usually expect to see evidence that weathertight integrity and crew protection from the sea on the external decks are kept to the required standards. The following list highlights some of the main points of possible concern.

1) **Hatches**
   The weathertightness of the covers must be satisfactory and the surveyor may require this to be tested with water pressure from a fire hose. Rubber seals of steel hatches must be in good condition and closure arrangements (dogs, cleats etc.) should be satisfactory. Wooden hatch boards, their tarpaulins, battens, locking bars and wedges all should be in a good state of repair. Hatch coamings must not suffer from excessive corrosion and should be chalk tested or scanned by ultra sound to detect any cracks.

2) **Ship's side doors**
   All closing and securing mechanisms of any openings in the ship's side below the freeboard deck must be functioning correctly

3) **Internal Weathertight Doors**
   Where these are required to meet damage stability requirements as part of the conditions of freeboard assignment, they should function properly both by local and remote control

4) **Superstructure doors and side scuttles (i.e. 'port holes')**
   These should have rubber seals in good order and close satisfactorily without excessive corrosion on the steel rims around the openings preventing an effective weathertight seal. Scuttles should be fitted with the appropriate toughened glass. All 'dogged' handles should be moveable, well greased and tighten up satisfactorily against their steel wedges on the inside of the door framing. Deadlights, where required, must be capable of satisfactory closure and securing so the hinges and screw down 'port dogs' should be well maintained. Any portable 'storm shutters' should fit and be capable of securing satisfactorily on their mountings

5) **Deck Penetrations such as ventilator trunks tank air pipes etc.**
   There should be no excessive corrosion around any pipe or trunk that penetrates an exposed deck, below the required minimum height clearance of vents and air pipes above the deck. Closure devices must be in good working order. Anti-flooding air pipes vent overside via the tops of circular chambers, which contain hollow ball floats. Any water flooding into the chamber causes the float to rise and seal off the vent until the water has drained back overboard. These ball floats tend to rattle around quite a lot within the chamber when the ship is rolling and pitching, so the surveyor will require a random sample of these vents to be opened up to inspect the balls inside for puncture damage.

6) **Non-return Valves and Extended Spindles**
   These should be in good working order and any extended spindles must be well lubricated

7) **Side railings**
   These should not have significantly damaged sections nor be weakened by excessive corrosion.

COMPLIANCE WITH LOADLINE CONDITIONS (Cont.)

8) Underdeck walkways (Type 'A', type 'B-100' and type 'B-60')
These should be adequately light, well ventilated and gas tight seals should be effective

9) The Approved Stability Book
A surveyor will wish to see this and satisfy himself that it has been updated for any significant modifications that have been made to the vessel. If a ship has a approved stability computer, there should be records of cross checks between the stability book and computer on standard loaded conditions

The above list is by no means exhaustive but I hope it covers most aspects of the vessel's general upkeep that relate to the load line regulations. It is the responsibility of the ship's officers to be familiar with how the conditions of freeboard assignment apply to their particular vessel.

If a ship has suffered any hull damage such as buckling plates and frames on coming alongside heavily against a jetty (the nautical equivalent of a 'car parking shunt'), then this could effect the ship's structural strength. In these circumstances when the damage is usually relatively minor, the classification society often specifies temporary measures to be taken immediately after the accident and then gives the shipowner a limited time period to carry out full repairs. (This is an example of what is known as a 'Condition of Class' in which the vessel will lose its class rating if the repair work is not satisfactorily completed within the time scale allowed)
The maintenance of adequate structural strength of the ship is a condition of load line assignment that is generally left in the hands of the classification societies. However, a Load Line surveyor would expect to see the classification society's written reports on the damage and subsequent repair work to satisfy himself that the correct action is being taken within an acceptable time scale.

If the ship's 'Flag State' surveyor is satisfied that the Load Line requirements are being met, then he will issue a signed, dated and stamped endorsement stating that the ship is in compliance with the Load Line Convention.
Most maritime nations are signatories to the convention so their inspectors can come onboard any vessel in their ports and demand to see the Load Line certificate with in-date endorsements.

SOME GENERAL COMMENTS ON FREEBOARD ASSIGNMENT

The 'Conditions of Freeboard Assignment' have to be complex to allow for most conceivable variations in the continually changing design of merchant ships. Furthermore, Article 9 of the Convention allows for authorities to grant exemption from any part of the regulations that prevent development in ship design, provided that the aims of the conditions of assignment are met by suitable alternative arrangements or the vessel concerned is restricted in its trading areas. (E.g. the rules could be relaxed in the case of a ferry engaged solely on short crossings in sheltered waters but this would be clearly stated as a particular condition of assignment for that vessel.) It is for the shipowner to present a good case when applying for exemption from any particular requirement.

It should also be appreciated that there is a certain degree of 'ad hoc' in the way in which regulations come into being and this causes some overlap between the load line rules and other regulations. Passenger ship requirements concerning the drainage from enclosed vehicle decks, closure arrangements of side shell doors, the need for scuttles below the freeboard deck to be of the non-opening type etc. are part of the 'SOLAS' subdivision rules for passenger ships, (See Chapter 10) rather than the load line rules. Similarly, the provision of adequate operational stability information is one of the conditions of freeboard assignment in the load line regulations but provision of damage control information is a requirement of SOLAS in the case of passenger and cargo ships, whilst MARPOL covers the same requirement with regard to tankers.
In addition to complying with the load line rules, many vessels will also have to meet additional related requirements of these other regulations.

The Load Line Convention covers all merchant ships engaged in international trade but many maritime nations apply its standards either partially or in full, to their coastal merchant fleet.

LOAD LINE ZONES AND SEASONAL AREAS

The Tropical, Summer, Winter and WNA freeboard zones are based upon the following weather criteria

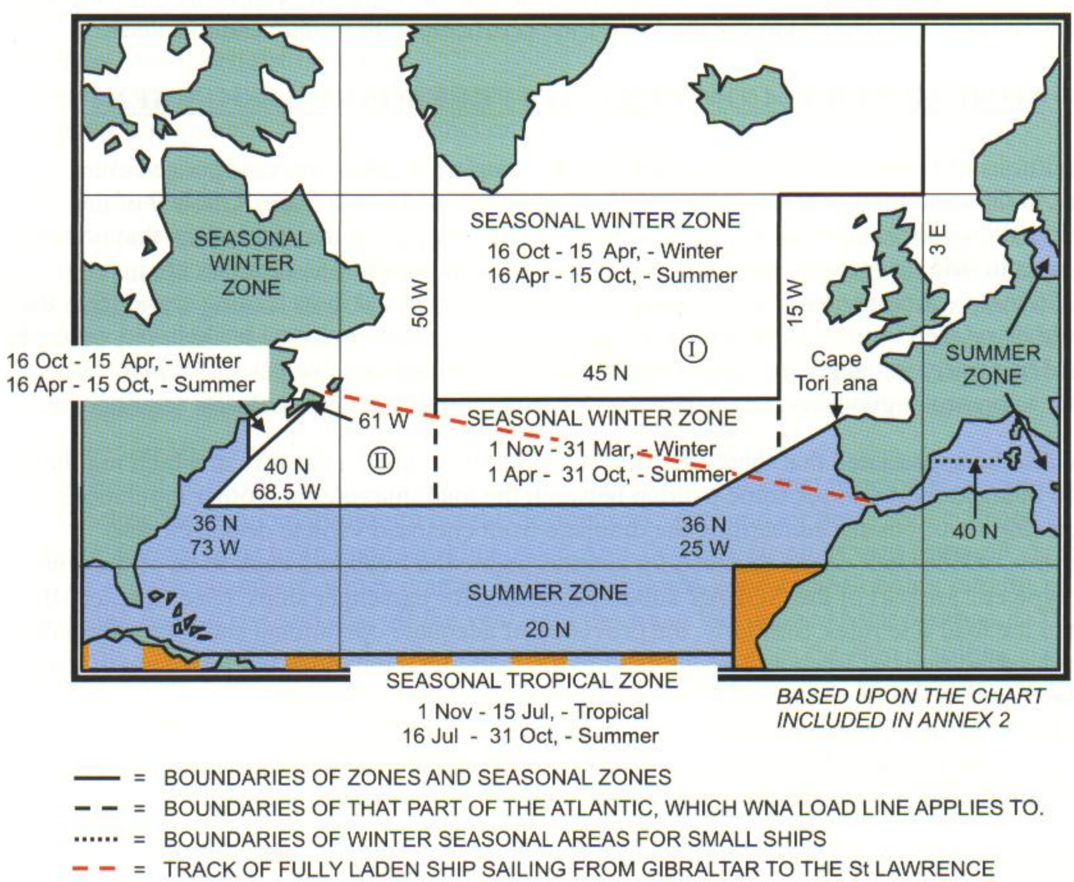
> **Summer Zones** - Regions where not more than 10% of wind speeds exceed 34 knots.
>
> **Tropical Zones** - Regions where not more than 1% of wind speeds exceed 34 knots and no more than 1 tropical cyclone occurs in any 5° latitude /longitude square in the same calendar month within a ten year period.
>
> **Winter Zones** - All other regions.

Annex II of the Load Line Convention contains **Regulations 46** to **52**, which detail the geographical limits of these zones and seasonal areas. (I.e. areas of the ocean that alternate zone category between Tropical and Summer or Summer and Winter, depending upon the season.) A World Chart is also included to illustrate the Zones and Seasonal Areas.

It is a criminal offence for the Master and shipowner to sail a vessel into a zone when, in the upright condition, the relevant midships zone load line is below the still waterline This law makes no allowance for the vessel being sagged. (*It should be appreciated that most cargo vessels will sag when fully laden).*

Many authorities around the world will police the load line law by checking on vessels before they sail from one of their ports. However, the law still applies on the open seas and if a ship departs from a Summer Zone port but passes through a Winter zone, whilst on passage, the master is still obliged to ensure that the ship is not exceeding the Winter draft on entering the Winter zone.
The ship can load beyond the Winter deadweight but sufficient fuel and water must be consumed so that its draft is less than the Winter marks when it changes zones. The section of the Zone chart shown below, illustrates the example of a ship sailing from Gibraltar to the Gulf of St Lawrence

NORTH ATLANTIC LOAD LINE ZONES AND SEASONAL AREAS



THE MAXIMUM DEPARTURE DEADWEIGHT FOR A VOYAGE PASSING THROUGH DIFFERENT ZONES

There is often a commercial requirement to load a ship with cargo, fuel and water to its maximum carrying capacity, particularly if it is on charter to carry a bulk cargo. The Master, however, cannot allow the ship to enter a loadline zone in an overloaded state so the following type of calculation must be carried out to determine the maximum sailing deadweight.

Consider a ship making the voyage from Gibraltar to the Gulf of St Lawrence, as shown on the previous page, in December. The ship is chartered to carry 7500 Tonnes of cargo and must sail with full water and stores. The fuel costs are favourable in Gibraltar so the Master is instructed to take the maximum bunkers allowed by the load line drafts

DEADWEIGHT CALCULATION TO OPTIMISE DEPARTURE DRAFT

SHIP'S PARTICULARS				
DEADWEIGHTS (TONNES)	SUMMER 8655		WINTER 8325	
MAXIMUM CAPACITIES (TONNES)	FUEL 1000	WATER 240	STORES 115	
CONSUMPTION (TONNES / DAY)	FUEL 24		WATER 5	
VOYAGE DETAILS FOR DECEMBER SALING				
PASSAGE DURATION BY ZONE	3 DAYS IN SUMMER		5 DAYS IN WINTER	
VOYAGE REQUIREMENTS				
MAXIMUM LOAD (TONNES)	CARGO 7500	FUEL ?	WATER 240	STORES 115

DEADWEIGHT CALCULATION

FUEL CAPACITY FOR WINTER DEADWEIGHT = 8325 - (7500 + 240 + 115) TONNES
= 470 TONNES

BUT THE VESSEL WILL CONSUME THREE DAYS OF FUEL AND WATER BEFORE ENTERING THE WINTER ZONE SO IT CAN SAIL WITH AN EXTRA WEIGHT OF FUEL EQUAL TO THIS CONSUMPTION

FUEL ON DEPARTURE FROM GIBRALTAR = 470 + 3 (24 +5) TONNES
= 557 TONNES

DEADWEIGHT ON DEPARTURE GIBRALTAR = 7500 + 557 + 240 + 115 TONNES
= 8412 TONNES

THE MEAN DRAFT AND TPC FOR THIS DISPLACEMENT SHOULD THEN BE FOUND FROM THE SHIP'S HYDROSTATIC DATA. THE FOLLOWING CHECK ON THE ABOVE CAN THEN BE APPLIED.

DEPARTURE DRAFT - (TPC x 29 T) SHOULD = WINTER LOAD LINE DRAFT

IF THE CHECK CONFIRMS THE CALCULATION THEN THE FUEL ESTIMATE IS APROXIMATELY CORRECT BUT THE MEAN DRAFT SHOULD BE OBSERVED DURING THE BUNKERING AND THE LOADING OF FUEL MUST BE STOPPED AT THE CALCULATED DEPARTURE DRAFT. THE VESSEL WILL BE ALMOST CERTAINLY SAGGED AND THE TARGET DRAFT WILL BE EXCEEDED IF THE ESTIMATED FULL FUEL FIGURE IS ACTUALLY LOADED

It is a legal obligation of the Master to ensure such calculations are made to avoid overloading the ship when it enters a more restricted load line zone than that of the departure port. It is also essential that the owners appreciate these restrictions when tendering for charter contracts.

Cable laying ships discharge their cargo at sea and the stability and ballasting must take care of this. A ship should also sail in an upright condition with minimal trim and possess seaworthy stability

EPILOGUE

A ship can be regarded as simply a set of figures and calculations and this book covers the major theory, regulations and equations that are concerned with stability and trim of a ship, both in the intact and damaged condition. However, I hope to have also helped to provide an insight into gaining a feel as to how a ship behaves at sea.

Though a ship is not a living thing, it certainly is a dynamic one that is constantly interacting with the turbulence of the sea by rolling, pitching and flexing to the waves. The 'whole' is greater than the 'sum of the individual parts' and so we should appreciate that the basic calculations we routinely make are only a fairly crude indicator of the ship's actual behaviour as all the different aspects of it are occurring simultaneously. The sums are important but we should also always try to be sensitive to the response of the vessel to different sea conditions and loaded states. If the motion of the ship 'feels wrong' in heavy seas, then we should try to understand what is going on then do something about it if possible. We also need to remember that actions that keep a ship out of trouble in bad conditions are generally carried out before it is in that situation. A good standard of maintenance along with well-secured cargo and equipment are essential to avoiding problems in heavy seas.

Finally, for those of us who live and work at sea, the ship is our home as well as workplace and there is only a centimetre or so of steel plate separating us from an alien world in which we really cannot expect to survive for very long without the sanctuary of the vessel. This is not usually uppermost in our day to day thoughts but it is worth keeping in the back of our minds.

Ian Clark October

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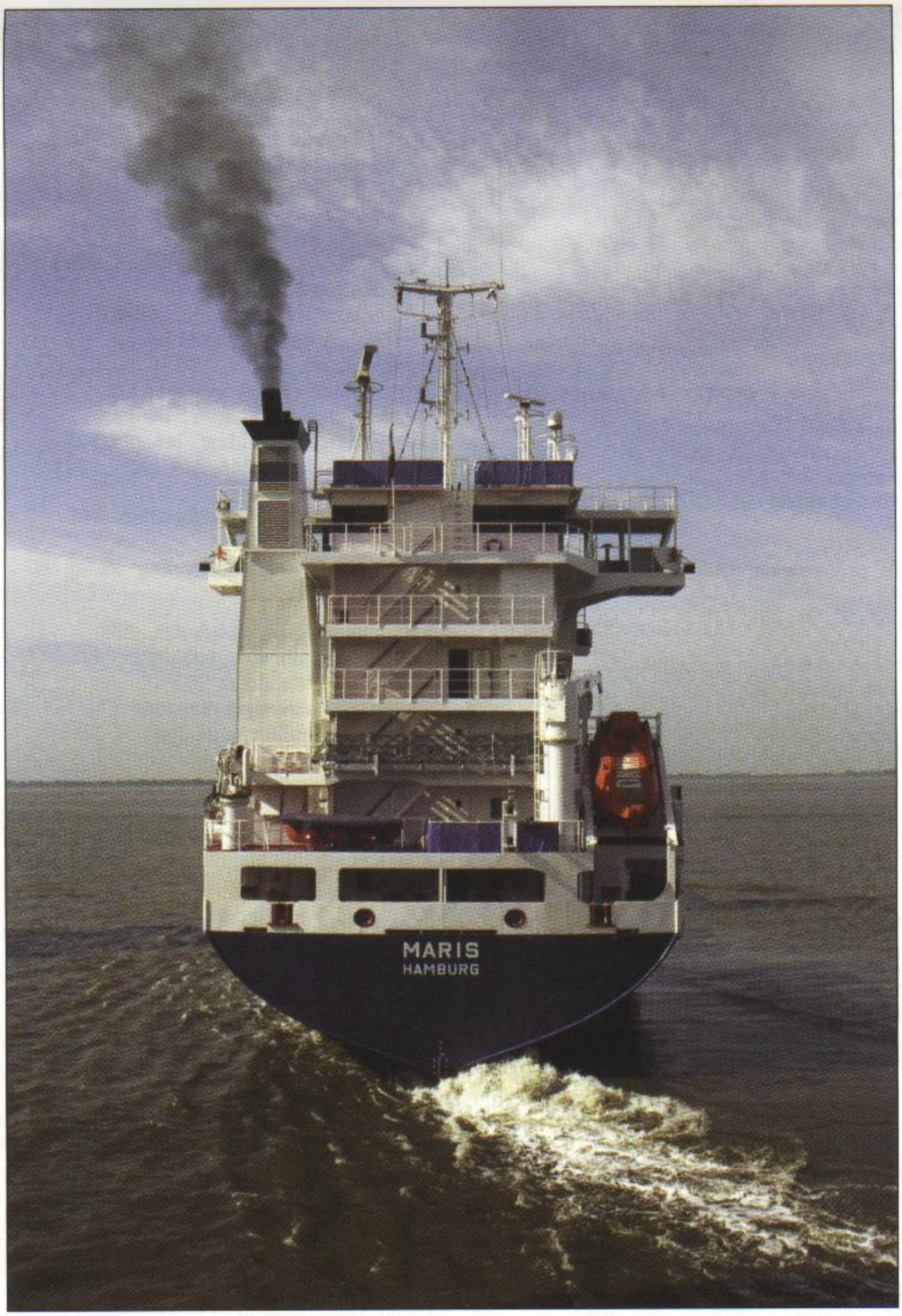

Container ships must always sail with sufficient freeboard to remain seaworthy. MV Maris on the Weser river departing from Big Stromkaje container terminal.

ABOUT THE AUTHOR

Ian Clark started his sea-going career in 1969 as a cadet with the Ocean Steamship group, serving on general cargo ships trading from Europe to the Far East and West Africa. He then worked as third and second mate with P&O General Cargo group, the British Antarctic Survey and Bank Line, during which time he gained his First Certificate and BSc in Nautical Studies from Liverpool Polytechnic. Mr Clark then left the sea in 1978 to become a secondary school physics teacher for twelve years, during which time he also studied for an MSc in Marine Geotechnics at Bangor University and designed and built a 28 foot racing yacht in partnership with a friend. He returned to the Merchant Navy in 1990 as first mate on small survey ships working in the North Sea, gaining his Master's certificate in 1992, and has since worked mainly on off-shore support vessels with some relief work on ro-ro ferries. He is currently Chief Mate on cable ships with Global Marine Systems.

In writing this book, he has explained the principles of applied physics that underlie the subject whilst always trying to keep the various aspects of trim, stability and strength within a practical context. An aspiring marine officer should find the different sections of the book useful throughout their career at sea, from cadet to marine superintendent.

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'THE MANAGEMENT OF MERCHANT SHIP STABILITY, TRIM AND STRENGTH'

Errata

Page 17 ,- In equation for WPA, Add term *'WPA'* after the word *'WHERE'*

Page 44 ,- In equation for KNθ, *'Sin θ'* to replace *Tan θ'*

Page 50 ,- In equation for GZ at θ° of heel, *'minus'* sign to replace *'multiplication'* sign

Page 53 ,- In lower GZ curve., Curves 'A' and 'B' should separate at the point of deck edge immersion of the vessel with the smaller freeboard

Page 64 ,- In the six MCA minimum stability criteria, Original point '4' to be removed (it applies to the IMO rules). Original point '5' to be split into two points, becoming the new correct points '4' and '5'

Page 68 ,- Add additional information, *'The MCA require areas to be calculated by Simpson's Rules'*

Page 73 ,- In first equation for G0Gv , *'G_0G_1'* to replace *'G_0G_V'*

Page 74 ,- In equations for G_0G_V, The symbol *'B'* to replace *'W'*

Page 81 ,- In Moments table, Total athwartships moments and offset of 'G' to port to be *'217.8'* and *'0.044 M'* respectively

Page 83 ,- Ambiguity in equation for maximum draft at θ° of heel for a vessel with rise of floor., *'d0' ……… Tanα'* to be enclosed in brackets.

Page 86 ,- Incomplete information for head of the derrick, Height *'h'* and distance *'X'* to be defined as measured from the keel and centreline respectively. Equation altered to agree. Also *'G_0'* label to add to circular insert.

Page 88 ,- Equations altered to conform with page 86

Page 90 ,- In equation for angle of heel θ°, *'Tan θ'* to replace *'Sin θ'*

Page 91 ,- In equation for Angle of Loll, Negative symbol to be inserted before the number *'2'* within the square root sign. (A negative GM value multiplied by -2 produces a positive value with a real square root)

Page 92 ,- In equation for Heel with negative upright GM, *'B_0M_0'* to replace *'Tanθ'*

Page 106 - In of shift of grain stows in hatchways , *'15°'* to replace *'10°'*

Page 106,- In applying correction to allow for vertical shift in Kg of grain stow, Factors *'1.12'* and *'1.06'* to be applied to volumetric moments of grain stows, <u>not</u> their Kg values.

Page 107,- In of shift of grain stows in hatchways , *'15°'* to replace *'10°'* in diagram

Page 108,- Omission from point '3' of stability criteria, Add *' or 40°'* to explanatory note to point 3

Page 108;- Clarification to label in GZ diagram, Add the word *'NET'* before *'GZ'* in label of max. GZ.

Page 109,- Change of reference publication, Replace *'SOLAS'* reference with *'International Grain Code'*

Page 109- Miss-print in measurement values in label box for *'saucering diagram'*

Page 127,- In equation for Trim, The symbol *'d_F'* to replace *'d_B'*

Page 133,- In MCTC equation, *'0.01'* to replace *'δt'*

Page 140,- In units in equation for WPA, Square symbol *'^{2}'* to replace cube symbol *'^{3}'*

Page 145;- In assumption, Replace red outlined box in diagram

Page 155- In equation used to derive equation for T'w, *'Vw'* to replace *'Lw'* on Left hand side of equation.

Page 167,- In equation for *'total mass'*, delete cube symbol *'^{3}'*, in last but one equation, replace *'2π'* with *'$2\pi/\sqrt{12}$'* and *'0.5 √ LENGTH 'L''* with *'0.14 √ LENGTH 'L''*

Page 168;- Use above equation correction to replace *'6 SECONDS'* with *'1.73 SECONDS'*

Page 169;- Replace *'6 SECONDS'* with *'1.73 SECONDS'*

Page 184,- Extra labelling in bending moments diagram, Additional buoyancy/metre values added

Page 209,- In value of weight increase, *'461.25'* value to replace *'4.61.25'* in equations

Page 214,- In definition of equation term, *'Tan θ'* to replace *'Sin θ'*

Page 286- In equation for Departure draft, Daily draft change *'(TPC x 29 T)'* to be multiplied by *'3'*.

Page 288- Change reference, Replace reference to U.K. *'Merchant Shipping Grain Regulations'* with *'I.M.O. 'International Grain Code'*