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Some recent experience with double hull tankers

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John Carlton trained initially as a mechanical engineer and upon completion of these studies he subsequently read for a degree in mathematics. After a period of time in the Royal Naval Scientific Service, where he was concerned with hydrodynamic research, he joined Stone Manganese Marine in 1969. During this time he was involved in the design of marine propellers and bow thrusters as well as undertaking research into propeller off-design performance.
In 1975 he joined Lloyd's Register, first in the Technical Investigation Department where he served for nine years in general engineering analysis and troubleshooting roles and, subsequently, in the Advanced Engineering and Performance Technology Departments. In 1984 he became Deputy Head of the Advanced Engineering Department and in 1987 was appointed in the same role in the Performance Technology Department, In 1992 he became Head of the Technical Investigation Department. Then in 2003 he became the Senior Principal Surveyor for Marine Technology and Investigation and now has global responsibility for these matters within Lloyd's Register. In his present role he also sits on several international technical committees and is author of the book Marine Propellers and Propulsion.

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Synopsis

Lloyd's Register has accumulated a significant amount of trials and service experience with double hull tankers. This experience relates to the design and operation of the hull structure, the machinery systems and the interaction between the hull and its propulsion machinery.

The paper addresses these primary aspects of the design and operation of tankers across the range of ship sizes. In particular, aspects of mechanical damage, structural fatigue and corrosion of the hull structure; the influence of the after body structural design on the propulsion machinery systems; shaft alignment; propulsion; vibration and manoeuvring are discussed.

Introduction

Apart from the normal design evolution processes, the Tory Canyon pollution incident started a process of thought which eventually had a profound influence on modern tanker design. However, following the subsequent Amoco Cadiz incident, the 1978 Protocol was developed which brought into force both the 1973 MARPOL Convention and relevant amendments to SOLAS 74. In the following years several discussions on the potential of segregated ballast tanks took place within the marine industry, however, interest in the double hull tanker concept became central following the Exxon Valdez incident off Alaska. This incident triggered the OPA 90 legislation in the United States of America which introduced the requirement for double hull tankers for port entry and limited the use of single hull tankers to 2015. Following this, the MARPOL 73/78 Convention was amended to require double hull construction in tankers: becoming mandatory in 1994 and in the two or three years prior to this date few single hull tankers were built. Recently, subsequent to the Erica and the Prestige incidents, the European Union has proposed that the phasing-out dates in the Convention be accelerated. This has led to Resolution MEPC.111(50) being adopted on the 4th December 2003 which specifies, subject to some Administration discretion on Category 2 and 3 oil tankers, phase-out dates for single hull tankers.

Set against this background of natural and legislative development, this paper considers recent experience that has been gained with the engineering aspects of modern tanker design; in particular, where problems have been experienced and the nature of these problems. As such, aspects of the hull structure, machinery systems and propulsion of tankers are discussed in the following sections.

Hull Structure

Lloyd's Register's data base on damages to tanker structures has been analysed. This analysis has been made over a ten year period and the data partitioned within the normal ship chartering classification as follows:

Product tankers	5000	to	49999 dwt
Panamax tankers	50000	to	79999 dwt
Aframax tankers	80000	to	124999 dwt
Suezmax tankers	125000	to	199999 dwt
ULCCs/VLCCs		Greater than 200000 dwt	

A sample of 373 ships having an average service life of 7.3 years has been examined and Table 1 gives the sample statistics for each category of ship.

Table 1 Ship sample statistics

Ship Size	Sample	No. in Years	Average Damaged Service	Ships Incidents	Defects
Product	123	8.26	85	401	1185
Panamax	67	9.08	48	276	962
Aframax	84	7.19	40	291	1202
Suezmax	25	6.36	12	34	125
ULCC/VLCC	74	4.69	32	76	211

Examining Table 1, it is seen that the three largest category of ships have the least incidence of damage, averaging at around 47% of the total ships at risk. In the case of the product and Panamax tankers, this incidence statistic rises to about 71%. Furthermore, when an incident occurs it is seen that between 2 and 4 defects are likely to occur in the hull structure.

For each ship category the defects to the hull structure recorded at the time of the survey were grouped according to the longitudinal position where they were found in the hull. These groupings were defined as Forward to Midships, Midships and Midships to Aft. Within each of these sub-divisions the location, position within the location and the type of damage has been analysed.

As might be expected, within the data many single instances of defects occur which do not replicate themselves to any appreciable extent within the data sets. In this way there is a degree of randomness about the data set when the defect incidence is low. Accordingly, in Table 2 the defects which have a repeatability of greater than 1% of the number of defects within a ship size category are recorded, rounded to the nearest integer. The defect entries in terms of their longitudinal grouping are listed in order of their magnitude in the Forward to Midships region of the ship, since this region tends to be the most prone to the incidence of defects for all but the Aframax and ULCC and VLCC tankers.

Examination of the data shows that damage to the side tanks features highly for all but the largest of the ships, namely the Suezmax and ULCC and VLCC categories.
To examine the underlying modes of failure for the ship types, the data base has been analysed by defect type in Table 3. Three modes of failure have been defined. The first being from mechanical loading origins such as buckling or setting in of the plating, frames or bulkheads. The second relates to fracture or cracking which may have a dynamic or periodic loading element or, alternatively, a brittle or ductile tearing aspect to its incidence in the data set, while the third category embraces corrosion, erosion or excessive wastage to the structural elements.

Table 3 shows that mechanical damage, such as buckling or set-in types of defect, ranks highly with all ship types. Indeed, for all categories except for the Aframax vessels, it forms the greatest incidence of damage. When it occurs, it most commonly manifests itself as side, or bottom shell, longitudinal or transverse bulkhead distortion.

In the case of fracture or cracking, this originates mostly from either low or high cycle fatigue. However, within the data set there will be instances of tearing occurring, either of a brittle or ductile nature. The three largest sizes of tanker appear to suffer most from this type of defect with the Aframax tankers having the greatest incidence of failure.

Excessive wastage and corrosion is seen to affect the Panamax tankers at a 57% incidence, significantly greater than for the other ship categories which have incidences lying between 5% and 29%. Such defect modes are clearly influenced by coating and corrosion protection integrity and methods, and the accessibility of certain areas of the ship for repair and maintenance. Furthermore, cargo heating and trade routes will also have an influence on these modes of failure.

Table 3 Reported Defect Mode

Ship Size	Mechanical Buckling/ Set-in	Fracture/Cracking	Excessive Wastage/ Corrosion
Product	61%	18%	21%
Panamax	36%	7%	57%
Aframax	27%	60%	13%
Suezmax	36%	35%	29%
ULCC/VLCC	54%	41%	5%

Table 2 Analysis of Principal Areas Damage

Location	Forward to Midships	Midships	Midships to Aft
PRODUCT TANKERS			
All Locations (No.)	471	348	366
All Locations (%)	40%	29%	31%
Side Tanks	21%	14%	14%
Double Bottom Tanks	4%	3%	2%
Wing Cargo Tanks	3%	3%	4%
Centre Cargo Tanks	3%	3%	3%
Wing Ballast Tanks	2%	2%	1%
Hopper Side Tanks	2%	2%	1%
Slop Tanks	-	<1%	4%
PANAMAX TANKERS			
All Locations (No.)	356	299	307
All Locations (%)	37%	31%	32%
Side Tanks	20%	16%	17%
Centre Cargo Tanks	5%	4%	3%
Wing Cargo Tanks	4%	2%	3%
Upper Decks	3%	5%	4%
Wing Ballast Tanks	3%	3%	2%
Deep Tanks	2%	-	-
Slop Tanks	-	-	1%
AFRAMAX TANKERS			
All Locations (No.)	384	383	435
All Locations (%)	32%	32%	36%
Side Tanks	15%	14%	17%
Centre Cargo Tanks	8%	8%	7%
Upper Decks	5%	4%	4%
Wing Cargo Tanks	2%	3%	3%
Slop Tanks	-	-	2%
SUEXMAX TANKERS			
All Locations (No.)	60	34	31
All Locations (%)	48%	27%	25%
Side Tanks	19%	4%	5%
Centre Cargo Tanks	12%	15%	12%
Wing Ballast Tanks	5%	<1%	6%
Topside Tanks	4%	2%	-
Upper Deck	3%	5%	<1%
Double Bottom Tanks	2%	-	2%
Hold	2%	-	-
ULCCs/VLCCs			
All Locations (No.)	58	81	72
All Locations (%)	27%	38%	34%
Wing Cargo Tanks	12%	17%	6%
Side Tanks	5%	9%	9%
Double Bottom Tanks	5%	4%	5%
Wing Ballast Tanks	2%	-	5%
Centre Cargo Tanks	1%	3%	5%
Upper Decks	-	3%	1%
Hopper Side Tanks	-	-	2%

Machinery Systems

Today the influences of slow speed engine out of balance forces and moments and their attenuation are relatively well understood in that they do not cause the number of major problems that were common twenty or thirty years ago. This is also the case with the suppression of many auxiliary machinery problems.

In recent years, however, the major area of concern for a great many tankers has been that of developing a satisfactory shaft alignment. Current design practice has been characterised by a combination of relatively short intermediate and tail shafts, directly coupled to the engine flywheel at the forward end, supported by an intermediate plummer bearing and then passing through and supported by the forward and after stern tube journal bearings and the aft end. Within the constraints of contemporary tanker design, such an arrangement has made for a very short-stiff shaft line arrangement. The problems have been two-fold: first, the derivation of a satisfactory alignment through the stern tube bearings and secondly, the attainment of acceptable shear forces and moments at the diesel engine coupling in order to achieve a proper crankshaft alignment.

At the forward end, engine builders have prescribed envelopes of shear forces and bending moments, a typical example of which is shown in Figure 1. Limitations of the type shown are based on the notion of rigid supports for the crankshaft bearings and, in practice, some variability is normally permitted to allow for the flexibility of the crankshaft bearing system. In addition to these bending moment and shear force requirements, the thermal distortion of the diesel engine has to be taken into account as does the internal reactions of the loads within the engine: these reactions include the loadings on the crankshaft introduced by engine components, for example those with cam shaft chain drives whose loadings tend to be distributed between the No. 1 and No. 2 bearings. The internal misalignment which occurs over the aftermost three bearings also tends to reduce the static load on the No. 2 bearing. Furthermore, shaft alignment design calculations frequently do not take account of hull deflection; this often being done by implicit factors rather than explicit calculations.

Commonly, with large tankers it has been found that when the ship's draught increases one or both of the following effects tend to occur: the No 2 crankshaft bearing (from aft) has a tendency to become unloaded or in some cases even top loaded, while excessive hogging of the crank web deflections of the No. 1 crank throw can take place. To overcome these unwelcome engine characteristics a number of actions can be undertaken. First, it has been found that it is beneficial to support the main engine bedplate at its four corners prior to chocking since this will permit the bedplate to sag. After chocking and when in service, this deliberately induced sag, which is typically of the order of 0.2 to 0.4 mm, will tend to reduce due to the thermal hogging characteristic of the engine; caused by the temperature difference between the cylinder blocks and the crankcase as it heats up to service conditions. Secondly, re-alignment of the engine with the ship in an operational ballast condition; this might involve lowering the aft end of the engine by 3 to 5 mm and raising by a similar amount the forward end of the engine. Thirdly, the fitting of eccentrically bored bearing shells to the No. 1, 2 and 3 bearings can sometimes be helpful; the eccentricity being invariably less than 0.3 mm.

In the alternative case of the alignment through the stern tube bearing, it has been found that design loads on the forward bearing have, in some cases, been specified as low as 2 tonnes. Such values are surprising, since by filling the ship's aft peak tank this can reduce loading on this bearing by the order of 5 tonnes. As a consequence, the forward bearing design loading for use in the setting of the alignment should be in the region of at least 10 tonnes to avoid unforeseen circumstances which might unload this bearing. With regard to the after bearing of the stern tube pair there has been a tendency in some instances to ignore the semi-empirical limitation on the relative slope between the bearing and the journal that was developed some thirty to forty years ago for these types of ship. This criterion suggested that this relative slope should not exceed 3 x 10-4 radians and while this is not an exact limit beyond which failure will inevitably occur, experience has shown that it has stood the test of time as a guidance criterion.

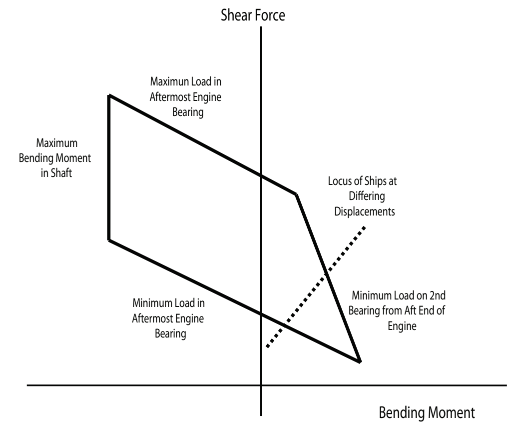

Figure 1: Typical Slow Speed Diesel Engine Coupling Loading Diagram

Furthermore, there have been instances where incorrect detail design has precipitated early failure. For example, the design of keeper rings which have not fully supported the aft end of resin bearings.

With regard to shaft alignment procedures these have been found lacking on a number of occasions. As a basic premise the engine should be aligned with the ship afloat and in a normal service ballast condition unless a significant body of experience has been accumulated with a particular class of nominally identical ships. Even if this is the case, then caution should be excised about extrapolating this experience too far. As discussed previously the main engine, prior to chocking, should be supported at its four corners to induce a bed-plate sag. While in this condition the loads on No.1 and 2 bearings should be more heavily biased towards the No.2 bearing with the No.1 bearing only just loaded. Following this initial procedure the alignment should be checked at full load draught with the engine hot. In some cases it has also been found beneficial to recheck the static shaft alignment some months after the ship has entered service because, due to the ship relaxing as it weathers its first few storms in the sense of structural shake down, the alignment has been found to change. Recognising that in many cases the locus of operation on the engine coupling shear force ñ bending moment diagram is frequently across the cusp of the diagram, Figure 1, the shake-down process of the ship may load the shafting system such that it works outside of its design constraints.

Propulsion

Much work in the 1960's and early 1970's concentrated on the propulsion environment in which the propellers were required to work. At that time the wake field was a primary concern in balancing the after body lines so that unpleasant cavitation or radiated hull surface pressure effects of either an extreme U or V form hull shape should not be experienced.

There have been some instances in recent years where the inflow into the propeller disc has not been fully optimised. In particular, a cause of concern has been the after body transition from the full tanker body to the relatively slender form in way of and immediately ahead of the propeller station. If this transition is not correctly made then flow separation in the vicinity of the top of the propeller aperture can take place. When this occurs very rapid changes of flow velocity, frequently unstable in character, can take place in the upper regions of the propeller disc. This may then manifest itself in the unstable cavitation growth and collapse which not only has implications for premature propeller blade erosion but also for high levels of radiated hull surface pressures. These high pressure amplitudes very likely embrace several blade rate harmonic components as well as introducing broadband spectral effects: both aspects often having adverse implications for ship local and global vibration modes. As a consequence, the flow characteristics and their stability should always be thoroughly investigated at model scale.

With regard to manoeuvring, few problems have been encountered with tankers with the exception of some ships at the lower end of the size range. In these cases there has been a tendency towards course instability which has been attenuated by further attention to the rudder design.

Conclusions

This paper has endeavoured to draw together a number of hull structural, machinery, propulsion and manoeuvring experiences that have occurred over the last ten years with double hull tankers. In doing so it has drawn particular attention to the defects experienced by the different sizes of tanker, both in terms of their modes of failure and location. Clearly, there is a much greater detail of information than can be contained in a single paper such as this and this may be addressed in future and more specific papers. Additionally, contemporary problems of achieving a satisfactory shaft alignment and propulsion configurations have also been discussed based on a series of complex issues which Lloyd's Register has been called upon to resolve in recent years.

Acknowledgements

The Author wishes to thank the Committee of Lloyd's Register for permission to publish this paper and also to thank his many colleagues for their help and assistance in its preparation. In particular, thanks are due to Richard Neasham, Gavin Lee and Mike Magill and to the Author's many colleagues in the Lloyd's Register's Technical Investigation Department.