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SNAMES

Society of Naval Architects and Marine Engineers Singapore

Propulsion efficiency and beyond

36th Annual Journal 2016/2017

Technology Centre for Offshore and Marine, Singapore (TCOMS)

Society of Naval Architects and Marine Engineers Singapore

36th Annual Journal 2016/2017

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President's Message

Warmest greetings on behalf of the council members and welcome to our 36th Annual Journal.

The publishing of this journal will not be possible without the support from the advertorial and contribution from the members. We would like to express our thanks to the publication committee headed by Dr Chen Hao Liang.

We wish to inform our members that this will be the last issue where our journal is in printed format. The subsequent journals will be published in digital form to defray the printing cost.

SNAMES as a society for the professionals in Naval Architecture and Marine Engineering will stay relevant and will look for ways to help our members overcome the challenges and changes in the horizon. We hope SNAMES can provide a useful platform for our members to share their knowledge to help one another in advancing our technical capabilities. This technical journal is one of such initiatives that have for our members.

We hope you will enjoy reading the articles presented here and looking forward to your support in the publication of our next journal.

For feedback and contribution, please contact us with an email: admin@snames.org.sg

> "We hope SNAMES can provide a useful platform for our members to share their knowledge to help one another in advancing our technical capabilities. This technical journal is one of such initiatives that have for our members."

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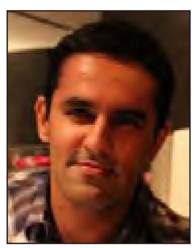

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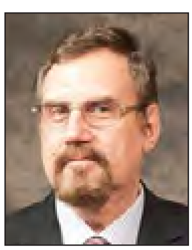

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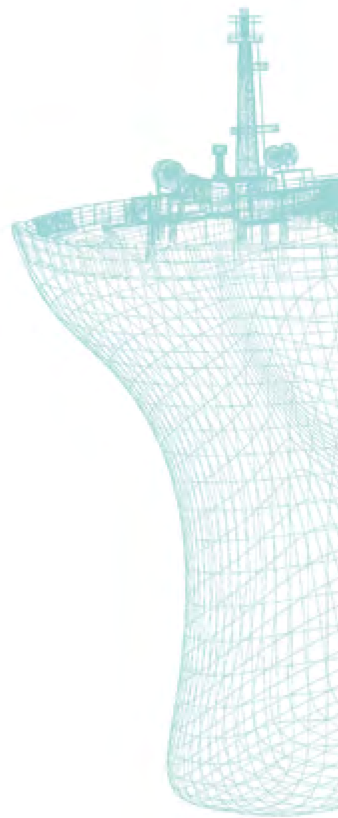

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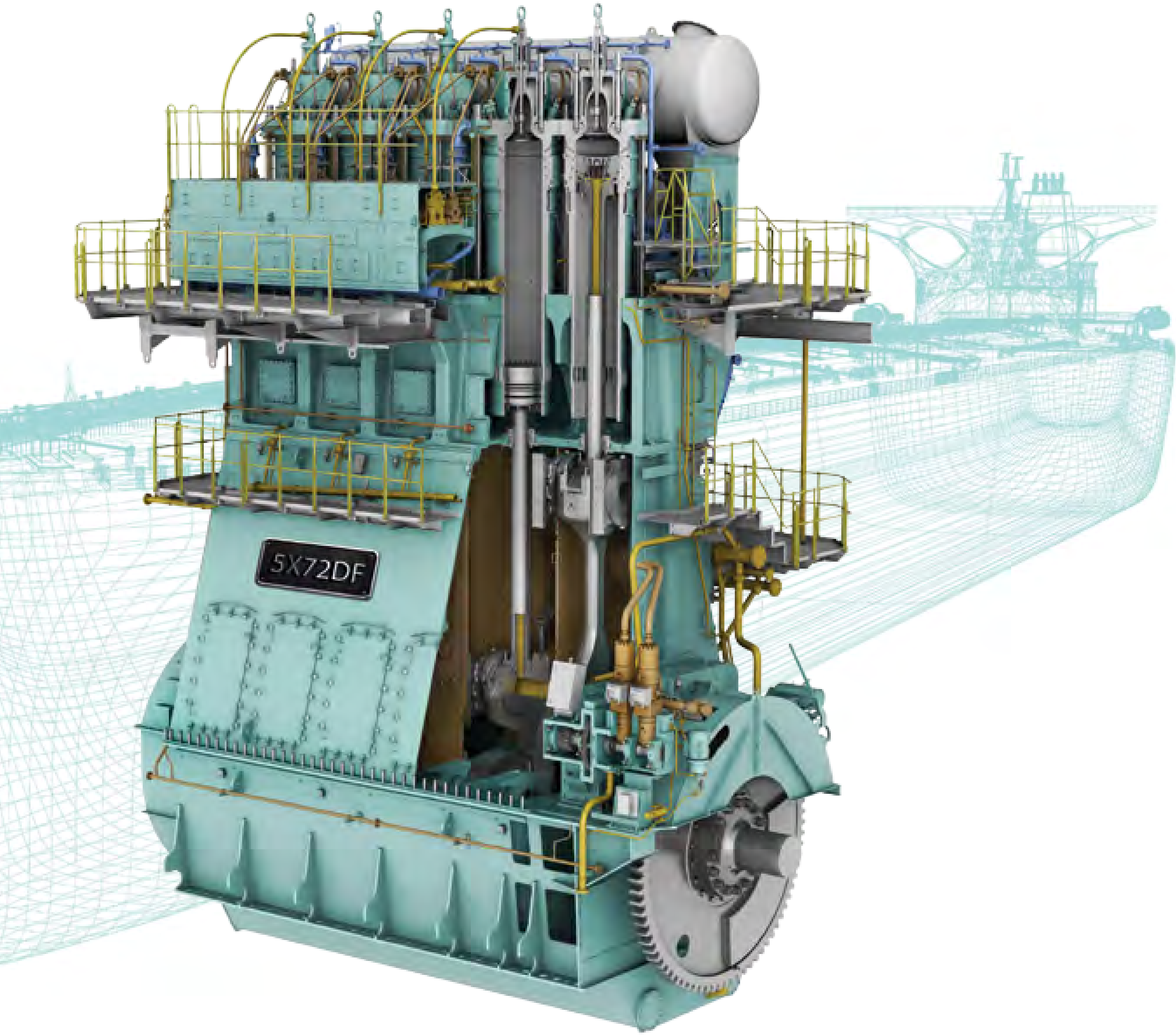

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The 4G Shipyard

"Fooling around with alternating current is just a waste of time. Nobody will use it, ever." —Thomas Edison, 1889.

ABSTRACT

Over the centuries, shipyards have evolved from large basins manually excavated in the ground such as was used to build the Treasure Ships of Admiral Zheng He (in Longjiang, Nanjing, China) to the latest technology employed by DSME in Korea employing floating docks and floating cranes. Taking this a step further, the 4G Shipyard envisions a floating island where floating docks, floating cranes, floating berths and floating workshops. It is self-sufficient in power and potable water. If required, radar and drones provide additional security. This paper discusses the advantage of such a system comparing it to the construction of more traditional shipyards using graving docks, bottom founded piers and quays and gantry cranes. In a country with scarce or heavily populated coastline, or subject to seismic events this solution has its obvious advantage.

The deployability of such a system is an important consideration as the shipbuilding industry wax and wane in a given country. The deployability feature enables shipyards to move from country to country in the course of its economic life, which outlasts the industry life cycle in the country of its origin.

It also makes it easier to employ a more robust business model, which separates the ownership of the shipyard from its management. In the new model, a team of managers can lease a fully functional shipyard for an agreed term. The owner can find a new manager on expiry of that lease at a new more promising location. In this way those with deep pockets but with little inclination or expertise to manage a shipyard can do so, while those with expertise to market and deliver ships can link up with investors to achieve their ambitions. This model is similar to the bareboat charter widely practiced in shipping.

INTRODUCTION

The Titanic in 1912, the Globtik Tokyo in 1973, and the Maersk Mc-Kinney Moeller in 2013, respectively the largest passenger ship, the largest tanker and the largest container ship of their time, each defined a milestone in shipbuilding.

The Titanic and her sister ship, the Olympic, built by Harland and Wolff in Ireland was assembled by riveting each hull plate on a pre-constructed frame on the launch way from which she was finally launched. The mode of construction was plate by plate, the equipment used was overhead cranes and a launch way to skid her into the water.

Six decades on, the 480,000 dwt Globtik Tokyo built by IHI in Japan, was assembled by joining prefabricated blocks made in workshops and assembled in a dry dock. The mode of construction was block by block. The procedure requires a goliath crane and a dry dock flooded to float her (see Fig1). I shall refer to them as 2G shipyards.

Figure 1. A 2G shipyard characterised by its goliath cranes and graving docks. These assets and the piers and wharves are permanent fixed and will have to be demolished when the industry fails. (Image: Courtesy of Waigaoqiao Shipbuilding China)

Nothing much change for four decades, until in 2013 DSME (Daewoo Shipyard and Marine Engineering) built the 18,300-teu Maersk Mc-Kinney Moeller with floating cranes and a floating dock1 using mega blocks, each of which was as wide and as deep as the ship itself. The use of floating cranes enables blocks to be built at a remote site, even outside national borders. Such shipyards are one generation ahead and may be referred to as 3G shipyards.

The three iconic ships, each marks a turning point in the evolution of shipbuilding methodology. The next turning point would bring the industry to a 4th generation shipyard (4G shipyard)

The 4G shipyard will be fully floating. Floating outfitting berths and all workshops will complement floating docks and floating cranes. This paper will examine why such an evolution is inevitable.

FLOATING BERTHS AND FLOATING WORKSHOPS

The two remaining elements for a fully functional shipbuilding production system that have yet to use alternative floating designs are the outfitting berths and workshops.

Would it be possible to berth very large vessels on a floating berth for outfitting? The answer is a resounding yes. Current outfitting berths are designed as bottom-founded structures such as wharves of finger piers. The only reason they are so designed, it would appear, is that such structures are the product of civil and structural engineers who have no training or experience in the application of the principles of buoyancy. That is until recently when floating berths start to appear across the world. They include a 352-metre long berth-cum-breakwater for cruise liner in Monaco, a 200-metre floating berth for submarines in Faslane, Scotland among others.

As for floating workshops, they have been around for many years. For instance, a shipyard in Varna Bulgaria was building a series of floating ferro-concrete workshops since the 1950s. During the Vietnam-US conflict, floating machine shop FMS 811 was based in Cam Ranh Bay to support the US fleet of aircraft carriers.

By making platforms large enough, it would be possible to set up workshops with panel lines to fabricate very large blocks. Platforms of 5 to 9 meters in depth with surface area 30 to 40 hectares are extremely stable, as their meta centres would be several hundred metres above the keel. The structure's hydroelastic response has not yet been analysed but based on the Japan's experience of the 1000-meter long runway, which is only 3 metres in depth, we do not expect any insurmountable problem. Any rolling, pitching or yawing motions will be virtually undetectable. Sizeable jib cranes on the quayside will be able to operate as if on land. Large crawler or rubber-tyred cranes would be able to move about as if on land, as would low-bed transporters.

Large workshops for hull and pipe fabrication including jack-up legs with heavy overhead travelling cranes may be supported by the platform. All shipyard processes such as precision machining, automatic/robotic cutting and welding, dynamic balancing of turbines are unaffected even if the floats are subject to waves and wind.

It would be possible to build as large a block as the capacity of the largest floating crane would be able to lift and transfer to wherever it is need. The block, which may be as large as five hundred to a thousand tons, may be the topside of a FPSO or a semi-submersible or a part of a ship. Alternatively, a new ship may be constructed on the mega-float. If the vessel is less than a thousand ton it may be constructed anywhere on the platform. Larger vessels of say 20,000 tons lightweight may be constructed on an isolated platform nestled within the periphery of the mega float. The isolation allows that portion of the platform to be raised or submerged (by manipulating the ballast water) independently for launching or recovering a vessel. It also prevents bending moments to be induced on the rest of the structure. The alternative method of launching is to skid2 the vessel across to a floating dock. The $7 billion US Navy submarine Zumwalt was launched this way at Bath Iron Works, albeit from land.

Each mega platform is assembled from smaller modules. The size of the modules is determined by constraints of the construction process and launching facility; they may be rectangular, square or hexagonal in shape with a surface area of a quarter to half hectare. The methods of connections between modules may be rigid or flexible depending on what the platform is expected to perform. The sides of the platform are fitted with fenders suitable for the types of vessels to be berthed; alternatively, floating fenders such as those manufactured by Yokohama may be used. It is to be noted that the kinetic energy of the vessel is not only absorbed by the fenders but also by the mass of the mega float (as it yields slightly on impact) and the friction of the water against the bottom of the platform.

THE 4G SHIPYARD

Figure 2 shows an example of a 4G shipyard. The platform is configured in the form of a cross to achieve a good balance between berth space and the deck surface. In this image with a deck surface of 35 hectares, the cross shape has a peripheral berth length of 4000 metres. The berths surround the workshops, project support offices, dormitories and the utilities centre, unlike a conventional shipyard where the berths are on one side and all buildings on the other resulting in workers having to cover very large distances between the ships and shore facilities.

Figure 2. A 4G shipyard operated by a single management (Image:Courtesy of Sixtrees Viz Comms Ltd)

The average distance between any shop and any berth in this arrangement is less than the corresponding average on a 2G shipyard with an equal length of berth. This means time needed to commute between ship and workshop is considerably diminished. With about 3000 people on such a shipyard moving between ship and workshop daily, a minor saving of even ten minutes per day per man translates into 182,500 man-hours per year and present rates represent a revenue generating capacity of about S$10 million.

The deck surface and berth may be expanded simply by adding more modules afloat. Additional modules are constructed off site and completely out fitted with workshops, roads and services before being towed for integration with the existing mega-float. This minimises interference with shipyard activities during the expansion process.

A 35-hectare cross configuration is also interesting as it lends itself to subdivision into four medium-sized yards with its own berth facilities as shown in Figure 3 (or even into eight small standalone yards) sharing a common central facility for administrative buildings, warehouse, canteens, dormitories, training facilities and utilities hub.

For keeping its position, the mega-float is moored with dolphins, which protrudes through recesses in the float or by chains and anchors attached to the bottom of the platform. Both methods allow the float to rise and fall with the tide thereby keeping the deck of the ships berthed alongside at a fixed elevation relative to the deck of the platform. This is a huge advantage as it eliminates the need to adjust mooring lines, or access gangways or utility lines linking ship and shore twice or more times a day as the tide rises and falls.

Roadways on the deck are paved and bounded by drains in the same way as a conventional shipyard. Surface run-offs from precipitation are collected in tanks below deck. With an average of 2.2 metres of rain, the 35-hectare float has a water catchment capacity of more than 700,000 cubic metres per year enough, to meet all the industrial needs of the shipyard(s).

ADAPTING TO FLOATING SHIPYARDS

The scheme is not without critics. The most common objection is "too much trouble taking man and material to and fro from land". The mega-float of this size has enough space to provide housing on board for workers as well as space also for carrying large inventory of pipes, plates, machinery like a conventional 2G shipyard. Steel material and machinery for shipyards are generally imported landed in ports, and then transported many kilometres across the country. The lifting of large bulky cargo on and off trailers and trucks and taking it across the country through congested highways is replaced simply by barging such material from port to the shipyard where large cranes make light work of offloading.

A floating shipyard is no more remote than a shipyard on an isolated island. Examples include 30-hectare Semb Corp Marine's yard set up in 1996 in Karimun Island and the more recent 139-hectare Saipem6 offshore fabrication yard. After a while, a new norm sets in and people adjust. Many shipyards have emerged in Asia from green field sites notably in China (Zhoushan) in the Philippines (Subic Bay) and in Vietnam (Vung Tau) which were one time remote small fishing villages deprived of piped water and electric power. There are some advantages to being remote. Workers are less likely to job hop. Employers find it worth their while to invest in training them. The skills they developed stay with the shipyard.

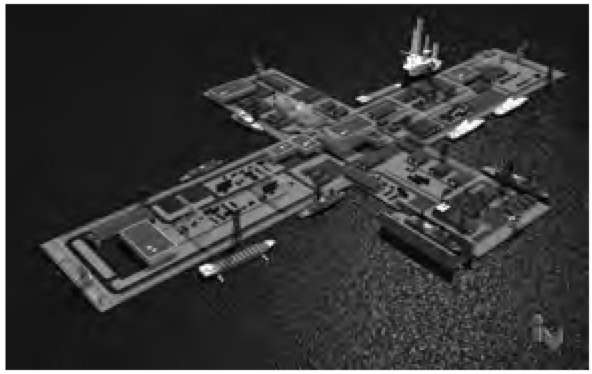

Figure 3. Four medium sized 4G shipyards with shared facilities at its centre. (Image: Courtesy of Sixtrees Viz Comms and IV Consulting)

ECONOMICS

Land for shipyard requires the favourable confluence of a number of environmental factors. It has to be on sound ground as otherwise the construction of the dock and the piers, and wharves are going to be costly. Too soft, and you end up with costly piling, too hard, and excavation may have to be carried out with explosives, too porous or pervious to water results in uplift of the dock floor when dock is dry. The water between the shipyard and the main navigation fairway has to be of sufficient depth for navigation. It should be sheltered from high wind and strong currents. The site should if possible be free from seismic activity and swells from open seas. The plot available to the yard should be well served by good roads and of a shape and size, which offers flexibility in the layout of a shipyard for a specific purpose. The host country has to have a stable political system, a good work culture as well as the infrastructure to facilitate communication, financing and even good hotels for their clients comfort and efficient ports and airports through which much of the shipbuilding material passes. No site is perfect and every site is a compromise.

A 4G shipyard is not a bottom-founded facility and is therefore independent of ground conditions. Being afloat and supported by the buoyancy of water it is not vulnerable to earthquakes or tsunamis. Piled marine structures are especially vulnerable. Rising sea levels pose no threat to a floating shipyard so no provision for this need factored in its design. It need not be sited near land if deep waters are not available. Being in deep water, it often does not require maintenance dredging. Floating docks need not have the massive and energy guzzling pumps as the energy required to dry dock a ship is a fraction of that required to dewater a graving dock with an equal size ship in it.

Examples in China have shown that the shipyard industry can experience cataclysmic shock. The rapid rise and fall of Rongsheng Shipyard3, in the space of three years (constructed in 2007 and in financial distress in 2010) typifies the plight of many other Chinese shipyards. Around 2005, shipyards were booked solid. By late 2008, shipowners were cancelling orders and shipyards were falling like dominoes. One third of the more than a thousand shipyards in China failed financially as bank withdrew support. The main assets of the shipyards are the docks and wharves: not attractive as collateral as they are immovable. No lender wants to provide a lifeline in exchange for assets, which may end up being demolished. In contrast, a lender can have a lien on a floating asset, which can be liquidated transnationally.

The lead-time to deliver a 4G shipyard is less than 2 years, from a commitment made at the Board level. To build a 2G shipyard, site investigation to obtain data necessary for design of the docks and piers and wharves will take at least 6 months. Bidders invited to submit proposals require at least six months to submit their bid if the proposed shipyard is in a third country. They need to find a local partner and understand the contractual environment, tax and local employment laws. To build the dock the contractor has often to first build a whole series of temporary works, including cofferdams, temporary roads from the nearest public road and temporary batching plants and power supply. The actual dock construction is exposed to the greed of man as well as the natural hazards of the ground. It is not uncommon when building a shipyard in a third world country, for the contractor to be subject to extortion by hoodlums and government officials. The contractor must allow for such non-technical cost, which disguised as "preliminaries" ends up being paid by the owner.

The cost of a bare float for supporting the workshops, roads, and other facilities and which also serves as berths, is between $300 and $800 per sq. m, depending on the design loads. The cost to procure mooring rights will differ from country to country. In some country, it may be possible to have it waived if a shipyard provides much needed job opportunities.

The 4G shipyard is dynamic. More modules can be added to increase its footprint as and when the business justifies it. If on the other hand business contracts, you can dismantle part of the shipyard and repurpose it. In developing a 2G shipyard, the owner needs to procure land he does not immediately need just to allow for future expansion, which may or may not happen. If business goes down the slope, you may have to dispose the unused land at a loss.

ASSET MOBILITY

The world's earliest 5-star purposed-built floating hotel4 was built in Singapore. It saw service in Australia, Vietnam and is now the "most opulent hotel" (according to Australian Harley Davidson aficionados Peter and Kay Forwood) in North Korea. Only a floating asset can provide the opportunity for the owner to sell and the buyer to relocate.

Asset mobility is just as important for shipyards. Shipyards are international in character. Singapore shipyards for example are present in all continents except the Antarctica. Asset mobility reduces the cost of exit when things do not go the way shareholders were told they would. Rongsheng in China is stuck with millions of dollars of asset it cannot move to another country. Mitsubishi Heavy Industry was told by Singapore authorities to demolish the huge dry dock when it could no longer sustain the downturn in business in Singapore. (The dock was spared the ignoble faith. Keppel bought it a few years later and turned it around.) Many things can go awry causing business to flounder. The 4G shipyard can be relocated to better pastures with little difficulty. Mooring rights can be abrogated much more easily than land tenure. The options open to the shipyard owner are useful tools for negotiation with worker's union or newly formed government of the host country.

If the owner wish to exit the business completely he can sell it to others to repurpose it as floating port, hotel, apartment, shopping mall or even an a entertainment park. The floating cranes and floating docks will find a ready market since they are used in many areas outside the shipyard industry. Cranes are used for bridge construction, offloading heavy cargos and decommissioning of abandoned rigs. Floating docks may be used for construction of concrete caissons for sub-sea tunnels, ports, and other maritime structures. The terrestrial equivalents goliath cranes and graving docks are not as saleable.

CONCLUSION

Maritime trade routes changes with the landscape of the global economy. Trade across the Pacific Ocean will surpass that across the Atlantic. Asia's dominance in shipbuilding and ship maintenance will remain. However, shipyards in Japan and South Korea will lose their vitality owing to domestic problems: rising labour cost, more profitable business opportunities, and the growing disdain among the young for the jobs in the heavy industries take their toll on both labour and capital necessary to sustain them.

What this means is that shipbuilding and ship repairing hubs will shift to newly emerging economies. India, Vietnam and the Philippines are pushing hard to fill the gaps that Japanese, Korean and eventually Chinese shipyards leave behind.

3G shipyards will be able to cope with the dynamics of such change better than 2G shipyards but they are not as responsive as 4G shipyards. Floating cranes and floating docks by themselves do not make a shipyard.

4G shipyards' deployability from site to site across national borders if necessary, makes it a less risky asset. Investors who would like to consider owning such an asset but not necessarily to operate it will view it favourably. The separation of ownership from management is already widely practised in shipping where the manager, owner and the financial backers could be multiple entities of different nationalities. Such business models open the doors of opportunities for very competent shipyard managers who may not be skilful fundraisers as well.

Thomas Edison and Nikola Tesla were engaged in a "war of currents5". Edison (then his employer) stubbornly rejected Tesla's advice to give up on direct current and put his money in alternating current. Tesla left his boss, strike out on his own with his idea, and soon linked up with George Westinghouse who backed him. But for his ego and attachment to his invention of direct current, Edison could have been credited with everything from the humble fan to high-speed maglev trains.

Could obsession to the familiar, in this case the 2G shipyard and aversion to the potentially disruptive 4G shipyard leave future shipyard investors the way Edison was left behind by Westinghouse with Tesla's vision of alternating current?

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Ivan CK Tam*

Formulating Stability Requirements for Anchor Handling Tug and Supply (AHTS) Vessels

ABSTRACT

The purpose of this study was to develop a decision support guideline for Masters operating AHV. The guidebook was developed with reference to the vessel 'Lewek Swan', whose loading condition was inputted into the shipboard software "CyberMaster" to measure the stability of the vessel.

The methodology of the proposal was to develop departure and arrival loading conditions of the vessel as well as develop a safe stability parameters based on various loading conditions. As a recommendation, the safety parameter graph and guideline will complement the use of shipboard software to enhance the stability of vessels.

The key result was the GZ and heeling lever arm obtained from the forces that resulted from AHO (Anchor Handling Operation), which served as a risk indicator. In conclusion, with this GZ and heeling lever Graph, Masters will have a clearer view and safer guideline to prevent any unsafe operation during AHO.

INTRODUCTION

The lack of stability is due to smaller reserve buoyancy as well as relatively smaller length of compartment as compared to large vessels. In addition, such vessels are more sensitive to damage due to its small frame. As such, the vessel can easily capsize upon impact of dynamic factors during operation and this can pose as a danger for the crews on-board. Hence, a revision in AHV stability is especially essential.

The offshore and AHV industry faces many challenges in overcome stability issues. Despite the IMO's implementation of strict regulations and constant improvement of vessel design by naval architectures, accidents like the Bourbon dolphin (News, 2012) and Stevns Power (Authority, 2004) are happening every other day. The AHO stability issue is considered hazardous and essential to look into. In addition, high demand of AHV in the market increases the urgency of looking into the improvement of the AHO. Evaluating from the existing shortcoming of IMO regulations and the different standard of the classification requirement on stability, the AHV operator do not have a 'standard' guideline for reference.

Although the stability booklet has been made mandatory, the thickness of the contents does not allow quick stability analysis for the Masters. As such, this is where stability software comes into the picture. Due to its capability in allowing faster and more accurate analysis for the Masters, it greatly outshines the analysis of AHTS stability using the stability booklet.

The stability booklet aids the Master by providing information on how the vessel will respond under

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various loading conditions. It is also a booklet that includes guidance on loading and stability calculations. However, these stability booklets are often detailed and lengthy in content. Therefore, a lot of effort and time is required to decipher the information. In addition, the lack of guidelines for reference would result in failure of the Masters in reacting as well as tackling emergency situation within short timeframe. Thus, the main aim of this project is to develop a decision support guideline that prevents an upset in the stability of the vessel, thus ensuring safety of both the crew and the vessel. By complimenting the use of shipboard software with the guidelines, the capability for Masters to enhance the stability of the vessel can be better achieved. In this support guideline, a range of parameters was developed for the Master to analyse as well as make quick decision in operating the vessel.

The aim of this study is to develop a holistic decision making guideline to help the AHV's Master recommend safe stability parameters during operation.

The stability calculation, which is only made before the operation starts do not include the influence of anchor chain/wire and the pulling on anchor acting on the stern of vessel by normal practice. As such, the stability calculation made will not be able to portray a realistic indication of the actual loading condition. Thus, the safety parameter of the ship cannot be determined accurately.

Hence, by calculating the maximum permissible tension on anchor chain/wire and the pulling forces acting on the anchor chain/wire, the maximum forces that will upset the ship's stability can be obtained. This will serve as a guideline for Master to keep the forces within the safety range and not to exceed the maximum tension that might endanger the particular AH work. A numerical example is given to illustrate the method, which can be extracted and made useful for Master onboard during AHO. The heeling lever arm, which was incurred from the maximum permissible tension in anchor chain/wire should be included in the stability calculation as well as serve as an extra safety guideline to aide Master in determining whether it is a "Go" or "No Go" operation. A checklist of stability acts as a summary for Masters for easy references as well as giving a clear picture of what will be used to allow quick analysis.

FORMULATIONS

Basic Assumptions made when calculating maximum permissible tension force during AHO

The calculation will then be done to illustrate the maximum force that the wire/chain is able to sustain without taking the vessel beyond the angles stated above. In this case, the force that we are referring to is the force that acts on the stern roller and transversely to the outer pins.

The heeling moment based on transverse bollard pull must also be shown and allowed for calculation. NMD Anchor Handling guidelines suggest that the vertical component is to be taken as the distance (vertically) from the deck at the tow pins to the centre of the stern thrusters or propeller shaft, whichever is lower.

Alarm or notice should be implemented to indicate the maximum tension force in the wire/chain as well as the point where the lateral force (towing pin/stern roller) is assumed to be applied in the software. The list moment caused by wire tension should be taken into account when checked against the intact stability during AHO.

With NMD Requirements as follows:-
The maximum permissible tension in wire/chain is calculated so that the vessel's maximum angle of heel is limited to the following angles:

1) Angle corresponding to a GZ value equal to 50% of GZ-max
2) Angle giving water on deck when the deck is assumed to be flat
3) 15 degree

Heeling Moment and Vertical Load due to Tension in Wire/Chain

The maximum heeling moment due to wire/chain at the stern of vessel is as shown below in Figure1.

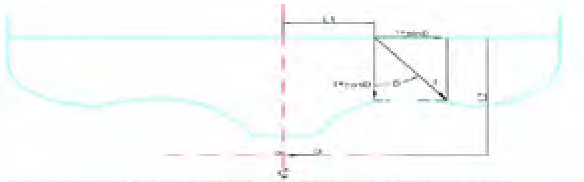


Figure 1 Maximum heeling moment

$$M_w = L_1 \times T \times \cos(d + \theta_{\min}) + L_2 \times T \times \sin(d + \theta_{\min}) \qquad (2)$$

where L_1 represent the torque arm of the horizontal components, L_2 the torque arm of vertical component, T represent the horizontal side-force D and represent the angle of attack if deviation of chain/wire at stern.

Vertical Load due to Tension in Wire/Chain

In order to calculate vertical load, the full load capacity of running in anchor winch is taking into account. Hence vertical load can be calculated as per formula below.

$$M_{vertical} = Max.load \times L_1 \qquad (2)$$

Run-out of Anchor Chain/Wire

During AHO and the anchor chain/wire run-out, the mooring line is running at the maximum force, which is the maximum Dynamic Braking Performance occurred on the winch.

The mooring line has an angle of attack of minimum 25 in relation to the vessel's longitudinal axis in the horizontal plane. (Security, 2008)

Hence, the horizontal plane will take the angle: Transverse components of force or Tension in the wire, T = $F_{braking} \times \sin 2\,3\ ^{\circ}$ (3)

between the horizontal plane and mooring line through the point of attack

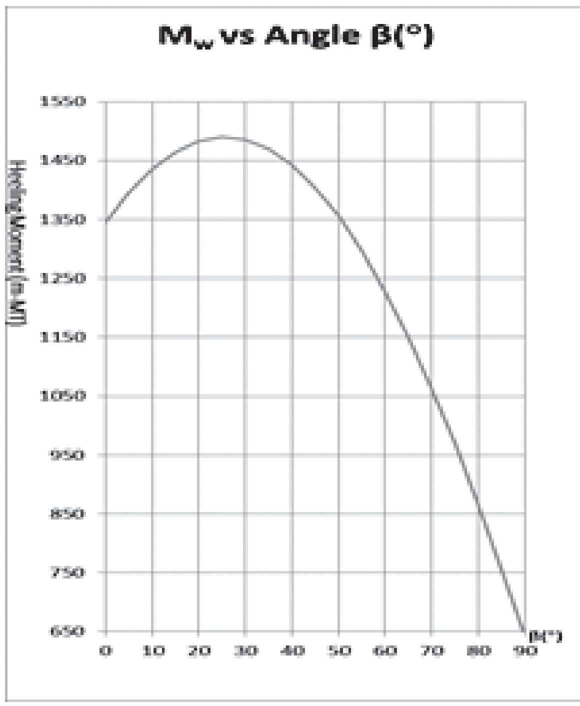
The heeling moment is calculated as the total effect of the horizontal and vertical transverse components of force/tension in the wire or the chain.

Let $\beta (0 \leq \beta \leq 90°)$ be the angle:

By using Equation (1), assuming d to be zero and find the maximum heeling moment which occurred at certain angle β which shown in Figure 2.

Thus, the most unfavourable condition occurs at μ angle is extracted from maximum heeling moment. Therefore, the maximum heeling moment due to wire/chain at the stern of vessel is calculated as below.

$$M_w = L_1 \times T \times \cos(0 + \mu) + L_2 \times T \times \sin(0 + \mu) \qquad (4)$$



Righting Arm Curve

Hydrostatic properties data of ship should be inputed into the stability software. The stablility software should be kept online and running during AHO so that the stability software can be updated at all times. The Master will then be able to refer and generate the righting arm curve as a guideline for stability analysis as shown in Figure 3.

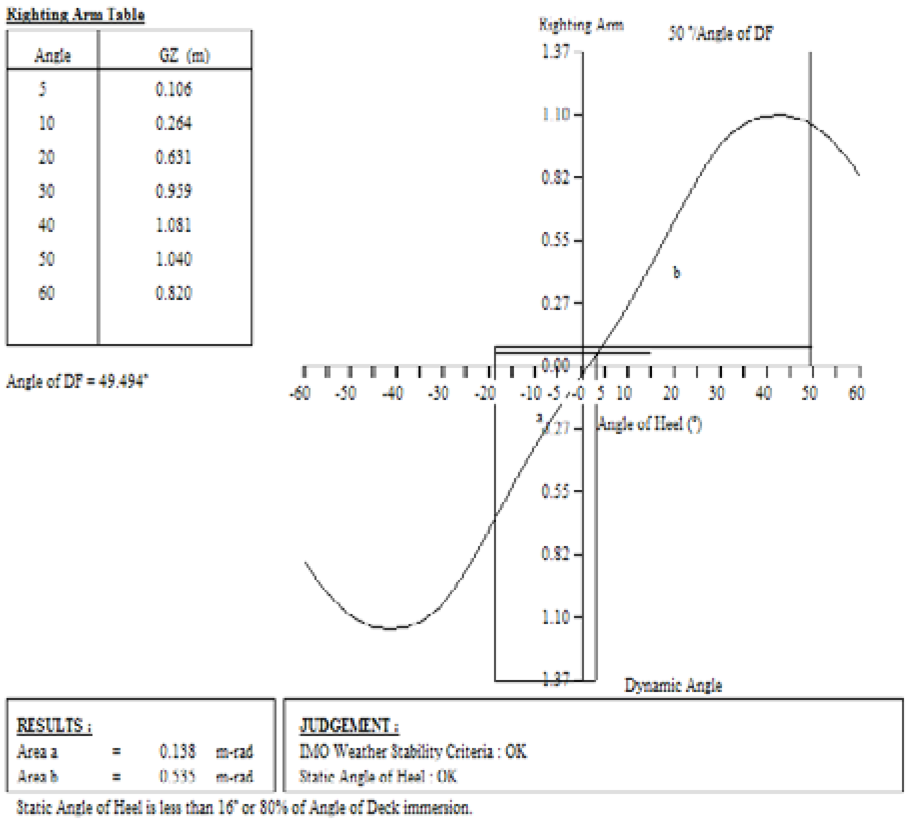


Figure 3 GZ Curve

Angle of DownFlooding and Angle of Heel

The angle of downflooding and angle of heel are intact stability requirement from all regulatory bodies. DownFlood angle is the minimum angle (the lesser to port or to starboard) at which a DownFlood point meets the waterline for a given condition of load, which can be generated from the stability software and angle of rolling is given as below Formula.

$$\textit{Angle of Rolling, } \theta_R R = \theta_0 - \theta_1 \quad (5)$$

where represent angle of heel, in degrees, under action of steady wind and represent angle of roll, in degrees, to windward due to wave action

Heeling Lever Arm

According to IACS, The heeling lever curve should be derived using the following formula:(IACS, 2004) which is the tension on anchor chain/wire incurred at the stern of ship during AHO.

$$b = \frac{0.7 \times T \quad H \quad \cos\theta}{.81 \times \Delta} \quad (6)$$

where b_h presents heeling arm in metre, T is maximum bollard pull in kN, H is the vertical distance in m, between the towing hook and the centre of the propeller and Δ is the loading condition displacement in T.

GZ and Heeling Lever Arm Diagram

GZ curve combined with the enhancement of heeling lever arm, which has incurred from the maximum permissible tension in anchor chain/wire to generate an easy reference diagram as below Figure 4.

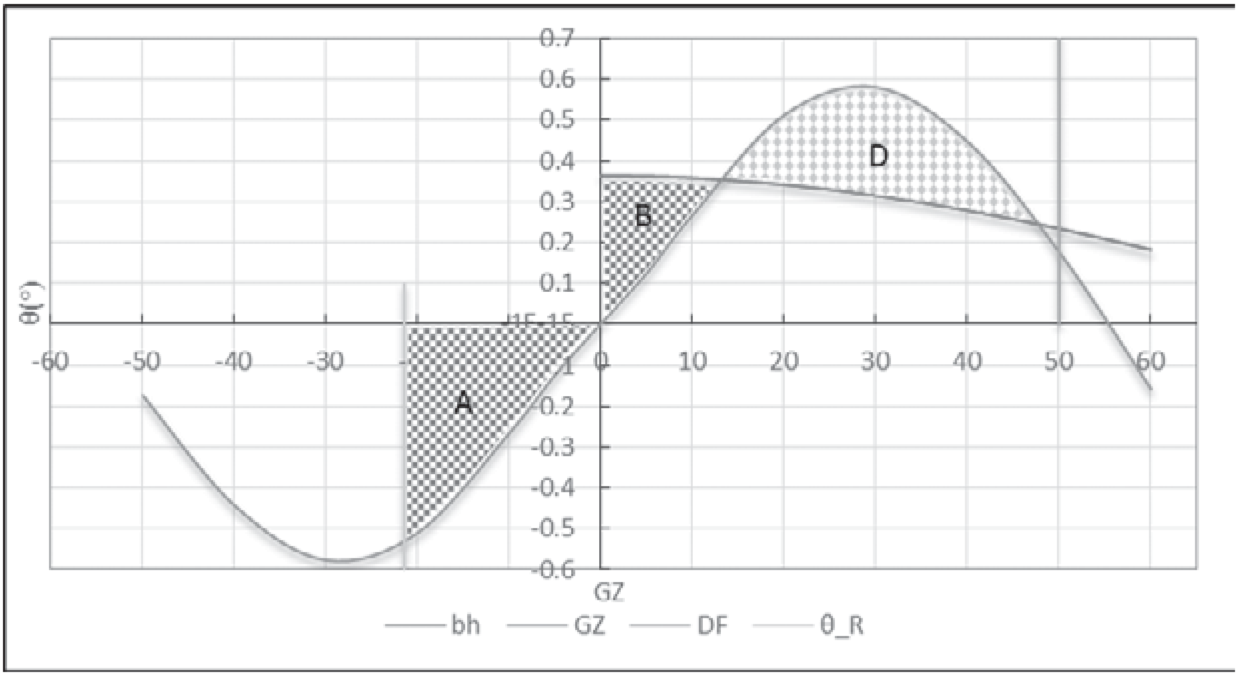


Figure 4 GZ and heeling lever arm diagram

RESULTS AND DISCUSSIONS

Generally, stability software onboard vessels are only able to meet the intact stability and weather criteria. With the heeling lever arm implemented, it assists the Master in determining whether the anchor handling operations are a "Go" or "No Go". From Fig 4., we can determine the area under the curve. In order to ensure a safe range of operation, Area D has to be greater than the sum of Area A and Area B in metre-radians where the boundary had taken into consideration of angle of downflooding and angle of heel.

As the priority for the basis of this paper, the anchor and pulling force exerted on the anchor wire has been calculated and take into account for stability calculation. With the implementation of heeling lever as an enhancement for AH work calculation, it will further provide a safety parameter for the Master for referencing. In addition, the maximum tension load should also be calculated during the operation to provide the worst-case scenario that is likely to occur during AHO. This calculation will serve as a reminder for the Master to keep the stability range in a safe zone before any danger arises during operation. Additionally, it is essential for the calculation to be updated quickly during AHO to ensure that the stability of the ship is kept in a safe parameter at all times.

The heeling moment analysis is extremely important and the calculation of maximum permissible tension of static stability is to determine the influential load incurred during AH work at the stern of ship using the Equation (1).

At all times, the Master should take into account the maximum heeling moment during AH works. This is to determine the force acting at the stern of ship during AHO, which will affect the stability of ship greatly.

From the heeling moment result, we can determine the most unfavorable conditions with a certain angle from Figure 2. From the unfavorable angle, we can determine that the vertical load is the maximum tension load incurred on winch. The maximum tension load also allows us to determine the heeling lever arm, which will affect the stability of ship. This is clearly presented in a simple analysis for Master to refer in area of unit metre-radians.

In conclusion, simply meeting the intact and weather criteria are insufficient to ensure a safe AH work. The requirement of Area D(Area A+Area B), whose calculation includes the influence on anchor wire/chain and pulling on anchor is also an influential factor in affecting ship stability. As such, all criteria must be met in order to proceed further for safe operation.

CONCLUDING REMARKS

The stability calculation, which is only made before the operation starts do not include the influence of anchor chain/wire and the pulling on anchor acting on the stern of vessel by normal practice. As such, the stability calculation made will not be able to portray a realistic indication of the actual loading condition. Thus, the safety parameter of the ship cannot be determined accurately.

In terms of stability software, it is recommended that the software be uploaded online during AHO and updated at all times to diagnose the vessel's condition. This will allow the Master easier reference as well as quick analysis for decision-making.

Hence, the solution of calculating the maximum permissible tension on anchor chain/wire and the pulling forces acting on the anchor chain/wire to indicate the maximum forces that will upset the ship's stability has been proposed. This will serve as a guideline for Master to keep the forces within the safety range and not to exceed the maximum tension that might endanger the particular AH work.

The heeling lever arm, which was incurred from the maximum permissible tension in anchor chain/wire should be included in the stability calculation as well to serve as an extra safety guideline to aide Master in determining whether it is a "Go" or "No Go" operation. A checklist of stability acts as a summary for Masters for easy references as well as giving a clear picture of what will be used to allow quick analysis.

As a recommendation, the stability software can be improved by including influential forces as well as the maximum permissible tension in anchor chain/wire in software analysis to generate an easy reference diagram (GZ and Heeling Lever Diagram in Figure 4) In addition, the software should include the capability to generate Area a, b and d directly. Furthermore, an alarm which will activate when an unsafe stability occur should be included as well. The above-mentioned elements will no doubt allow Masters to avoid stability issue and increase the AHO's safety level in the process. In conclusion, prevention is better than cure and safety measures should be prioritized at all cost.

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Problem Fuel Identification Number (PFIN) - An algorithm to identify fuels with the potential to cause piston ring breakage in marine engines

ABSTRACT

The quality of global bunker fuels has come in for much discussion, controversies and disputes. Adulteration and contamination of bunker fuels supplied from various global bunker ports continues. One of the most insidious and damaging, poor quality bunker fuels are the ones that cause rapid wear and breakage of piston rings in the ship's main engines.

Pistons are key components of the engine that convert the thermal energy from the combustion of the bunker fuel into kinetic energy that moves the crankshaft and propeller. Piston rings fit within the grooves on the piston and are responsible for sealing the gases formed during combustion in the combustion chamber and also for distributing cylinder lubricating oil along the cylinder liner. The presence of excess carbon residue, high asphaltene, poor ignition quality and borderline stability in residual fuels can lead to the deposition of a large quantity of carbon particles in the gap between the piston rings and grooves. This leads to jamming of the piston rings, preventing them from movement in the grooves. This in turn causes high wear of the piston rings and inability of the rings to seal the gases in the combustion chamber. Once this wear reaches a certain critical value, the piston rings break. A main engine with broken piston rings will cause serious blow past and very high wear of the cylinder liner necessitating stoppage of the marine engines at sea. Stopping the ship's engines at sea and repairing or replacing the rings is a complex, hazardous and expensive process.

The aim of this paper was to try to identify the parameters in the residual fuel which were accounted for piston ring breakage and to identify an algorithm to accurately predict which fuels would cause piston ring breakage. We looked at over 61 cases of broken piston rings at Viswa Lab and analyzed the fuel. Data analysis of the parameters was also performed to identify a pattern. We were able to create a proprietary number called Problem Fuel Identification Number (PFIN). We also established, through subsequent case histories, that when the PFIN exceeded 130, there was an 85% chance of piston ring breakage and when the PFIN exceeded 190, there was a 95% chance of piston ring breakage.

This paper describes the background of piston ring breakage and the methodology and validation we used to arrive at the PFIN.

INTRODUCTION

Quality of bunker fuels has always been a widely discussed topic. Nearly 80% of all machinery problems related to the main engine are from fuel and lubricating oil problems. As the world's demand for oil goes up, newer petroleum refining technologies to improve the yield of the most valuable fractions (gasoline, diesel, naphtha etc.) have always made news. Research & Development activities by several major oil companies concentrates on these projects to meet the world's demand of the most valuable oil fractions. Bunker fuels are formulated by blending residual fractions, obtained after primary (atmospheric & vacuum distillation) or secondary refining processes such thermal or catalytic cracking, with various cutter stocks. Common cutter stocks used are cycle oil, diesel, shale oil & recently various other petrochemical byproducts.

While the quality of the lighter fractions has not been altered through these times, trends indicate that the residual fuel quality has definitely been adversely impacted due to the introduction of newer refining and blending technologies. These changes to residual fuel quality affect the ignition & combustion performance[1], stability, increased aromaticity and asphaltene content of the fuels. A major consequence to all these changes is increase in engine damage and operational issues as a result. One such consequence, piston ring breakage, is the subject of this paper.

Piston rings fit within the grooves located on the pistons and the movement of the pistons within the grooves is responsible for sealing the gases in the combustion chamber, spreading of cylinder lubricating oil and for transfer of heat from the piston to the liner. Increased carbon residue, high asphaltene and poor ignition and combustion can lead to carbon particle deposition in the grooves, which prevents the movement of the piston rings. This can lead to poor sealing, increased blow past, lower power, increased fuel consumption and poor spread of lubricating oil with liner wear. In the worst case, this may result in stopping the main engine, replacing the piston rings at sea and can significantly disrupt the ship's operations.

An important objective of this paper was to identify the parameters in the fuel that can cause piston ring breakage and to develop an algorithm to accurately identify when piston ring breakage would occur. The current paper discusses about an algorithm that Viswa Lab has developed to identify a fuel that has a potential to cause piston ring breakage.

METHODS

At Viswa Lab, we reviewed the case histories of 61 ships with 61 fuel oil samples. Asphaltene has traditionally been thought to be the sole parameter causing piston ring breakage. All the test parameters were obtained and examined on these 61 samples to determine a single contributing parameter for piston ring breakage. After this, the data was reanalyzed multiple times to find a group of parameters that worked in concurrence to account for the piston ring breakage. Once the group of parameters and the algorithm was identified, the algorithm was tested on different types of fuels to assess the potential of these fuels to cause piston ring damage/breakage. The consistent validation of the algorithm has led to our patented number called the PFIN, which can reliably predict which fuels can cause piston ring breakage.

RESULTS

A detailed analysis of the fuel oil reports and case histories for 61 ships and fuel oil samples was performed. Asphaltene was not found to be the sole cause of piston ring breakage. In addition, no single parameter, could be identified as the cause of piston ring breakage.

During the reanalysis, the combination of asphaltene, CCAI (Calculated Carbon Aromaticity Index) and MCR (Micro Carbon Residue) was also not found to be the cause of piston ring breakage.

There were instances when one or more of these parameters were abnormal without piston ring breakage.

A systematic reanalysis and search of all the data was performed for a group of parameters which in combination were causing piston ring breakage. This group of parameters were identified as MCR, CCAI, Asphaltene, Xylene Equivalence and Reserve Stability Number. These parameters work together to cause piston ring breakage.

MICRO CARBON RESIDUE (MCR)

This test is used to determine the amount of carbonaceous material present after the evaporation and pyrolysis of the fuel. This test is a part of the standard analysis of fuels and gives an indication of the carbonaceous deposits that can be formed after fuel combustion. Fuels with a high ratio of Carbon to Hydrogen tend to be more difficult to completely burn and hence can result in increased deposits in the engine. The MCR is one of the indicators of the deposit forming tendencies of fuel. Different engines are able to handle fuels with different MCR values.

Method: Micro carbon residue for the fuels was determined by ISO 10370[2]. A 0.1 to 0.2 ml of residual fuel is placed in a glass vial and heated up to 500 C under an inert (nitrogen) atmosphere. The temperature is maintained at 500 C for 15 minutes and then cooled to room temperature. Any carbonaceous material left after this is weighed and reported as a percentage of the weight of the original sample.

Abnormal values are MCR levels greater than 11.5%.

ASPHALTENE

Asphaltenes are defined as the n-heptane insoluble but toluene soluble component of crude oil and residual fuel. They are a complex mixture of different chemical species with a range of chemical compositions and molecular weights. Asphaltenes contain organically bound Vanadium, Nickel, Sulfur and Nitrogen centrally. They are aromatic, large, polar, aggregate together in units called micelles (because of the polarity) and can stay in colloidal suspension (peptized) based on the fuel composition. They typically exist in suspension in highly aromatic fuels and can precipitate out in paraffinic fuels. This accounts for their solubility in toluene but being insoluble in n-heptane. Asphaltenes can create problems when they precipitate out of solution and cause sludge formation. Some asphaltenes may also combust incompletely and can lead to carbon deposition. The after burning of the carbon deposits can cause abrasive wear of the piston and piston rings. After-burning of the fuel is particularly detrimental to the engine as the liners are exposed to higher temperatures than normal. This in turn leads to stresses on the cylinder lubrication causing hot spots. The hot spots lead to deposit formation and ultimately damage to piston, piston rings and the cylinder liner.

Method: This test was performed based on ASTM D6560 procedure[3]. The test involves taking a specified amount of homogenized heavy fuel oil sample & dissolving with 5 parts of heptane. The test sample is mixed with heptane (30ml/g of sample) and the mixture is boiled for one hour under reflux and stored for 90 to 150 minutes in a dark site. This is then filtered through a 42 Whatman paper and washed with hot heptane to remove any residue. The filter paper is set in a reflux extractor and refluxed with hot heptane for one hour. It is then refluxed with 30 to 60 ml of toluene until the asphaltenes have been dissolved. The dissolved contents are washed with toluene. The toluene is evaporated off and the remains are weighed to determine the asphaltene content of the sample.

Abnormal values are asphaltene levels greater than 10.5%.

CCAI (CALCULATED CARBON AROMATICITY INDEX)

There is no reliable test to determine the ignition properties and CCAI was developed by Shell as a method to characterize ignition properties of a residual fuel. Carbon content and aromaticity of the fuel are thought to be two factors that influence the ignition properties of the fuel. CCAI gives an indication of the ignition delay of the fuel. CCAI is dependent on both the physical and chemical properties of the fuel. Density and viscosity of the fuel are important physical properties of the fuel. Chemical properties that mainly affects CCAI is the aromaticity of the fuel. The higher the CCAI value, the poorer the ignition quality of the fuel. Engine makers prescribe CCAI values for the specific type of engine. CCAI can be between 800 and 880. Values over 860 are considered to have abnormal ignition properties.

CCAI = D – 81 – 141*log [log (V + 0.85)]– 483 log (t+273/323)
D is the density of the fuel at 15 C
V is the viscosity at t degrees C
t is the temperature
Abnormal values are CCAI values greater than 848.

XYLENE EQUIVALENCE (XE)

This test is an indicator of the stability of the fuel. Stability is the property of fuels to remain stable when stored. As described previously, asphaltene exists in fuels in a colloidal suspension. When they precipitate out of the fuel, the fuel is no longer considered stable.

Method: Heavy fuel oil is mixed with xylene. Paraffinic solvent (heptane) is added slowly to this mixture. At each stage of heptane addition, a drop of the mixture is taken out and placed on filter paper. The appearance of two dark rings within each other is consistent with precipitation of the asphaltene. This process is repeated to find the lowest concentration of xylene at which asphaltene precipitation does not occur.

Xylene Equivalent = Xylene (in ml)/ [Xylene (in ml) + Heptane (in ml)] x 100%

Abnormal values are Xylene equivalence levels greater than 30.

RESERVE STABILITY NUMBER (RSN)

This test is an indicator of the stability reserve of the fuel. The stability reserve is estimated by a separability number based on the transmittance of the fuel using the Turbiscan procedure.

Method: The test method is applicable to all heavy fuels containing asphaltene. The fuel is dissolved in toluene. Heptane is added drop by drop and the mixture is evaluated by an optical scanning device. A separability number is assigned based on the transmittance of the fuel sample.

When the number is low (0 to 5), the stability reserve is high and asphaltenes are likely not to flocculate. When the number is between 5 and 10, then the stability reserve is lower and asphaltenes can flocculate if the fuel is exposed to worse conditions. When the number is greater than 10, then the stability reserve is very low and asphaltenes will easily flocculate or have begun to flocculate.

Abnormal values are RSN levels greater than 10.

These 5 parameters work together to cause piston ring breakage. The Asphaltene, Xylene equivalent and Reserve Stability Number predict the asphaltene content of the fuel that has the potential of asphaltene flocculation in the fuel when stored. Flocculated asphaltene can enter the grooves and fix the piston rings in the grooves preventing normal mobility of piston rings. This leads to excess wear and eventual breakage of the rings. The MCR and CCAI are parameters which are related to the carbonaceous deposits and ignition properties of the fuel respectively. These parameters predict excess deposits following the ignition and combustion of fuels and also poor ignition of the fuels. Poor ignition and combustion of the fuel can destroy the cylinder lubrication thereby causing excess wear of the piston rings and the cylinder liners resulting in piston ring breakage.

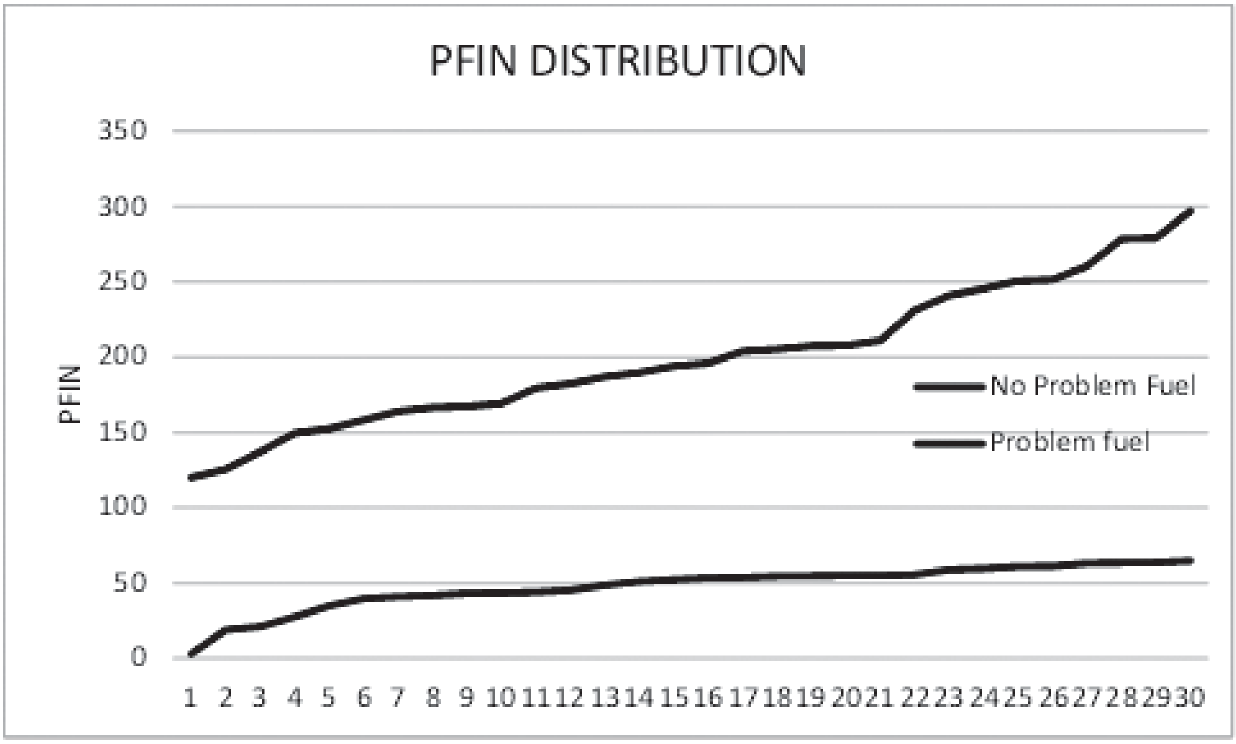

Figure. Distribution of PFIN results for fuels that have caused piston ring breakage (red curve) vs. fuels that have not caused problems (blue curve).

A combined algorithm for all these factors was proposed. This proprietary algorithm was designed to predict piston ring breakage even when one or two out of the five parameters was normal. This algorithm was then tested on 148 samples. Consistent validation of the algorithm was obtained in fuels with piston ring breakage, "normal fuels" without piston ring breakage and fuels with abnormal values without piston ring breakage. We have named this patented algorithm as the Problem Fuel Identification Number (PFIN). We were able to prove that when the PFIN is greater than 130, there is an 85% chance of piston ring breakage. When the PFIN is greater than 190, there is 95% chance of piston ring breakage. We believe that PFIN can be used effectively to differentiate between fuels that can cause piston ring breakage and those that do not cause piston ring breakage.

DISCUSSION

Since the 1980s, ISO 8217 guidelines have been accepted as the standard for bunker fuel quality. Refineries and suppliers focus on increasing the yield of lighter and more valuable fractions from crude oil. While the ISO guidelines provide a framework for fuel quality, there are clearly gaps present. These are areas in quality for which a decisive recommendation is not made by the ISO guidelines. It is these very areas that are exploited when providing a fuel user with an inferior quality fuel.

The resulting bunker fuel contains various contaminants (from the residual fuel) and in addition is mixed with lower viscosity oils including used waste oils and other petrochemical by products to lower the final viscosity of the bunker fuel. This is usually done to satisfy the ISO guidelines for bunker fuel viscosity. Due to the contaminants present both in the residual fuel and in the low viscosity oils/fuel for blending purposes, the quality of bunker fuel has to be vigilantly watched.

The quality of bunker fuels can change constantly based on the refinery feed stock and the blending components used. It is incumbent upon the fuel user to test these fuels to ensure that there is no significant damage to the main engines of the ships. Routine testing of bunker fuel quality at certified laboratories (which do not have a conflict of interest) is the crucial initial step.

Piston rings are a small but essential component of the piston cylinder. Piston ring breakage can result in high blow past, poor cylinder lubrication, cylinder liner wear and can ultimately lead to cylinder and piston ring replacement. This sort of repair necessitates main engine stoppage with demanding and expensive procedures to replace these components.

One of the challenges in shipping is how to identify problem fuels. The identification is problematic due to the ever changing nature of the contaminants present or mixed in with bunker fuels. The objective of this paper was to identify the key parameters that resulted in piston ring breakage from a review of Viswa Lab's data base. We found five factors that worked in conjunction to cause piston ring breakage. These factors were incorporated into our algorithm. The algorithm (PFIN) was tested and validated by Viswa Lab. The patented PFIN is 85% predictive of piston ring breakage when the PFIN is greater than 130 and is 95% predictive of piston ring breakage when the PFIN is greater than 190. This has been subsequently tested and proven on multiple fuel samples. This algorithm is able to predict the vast majority of fuels that can cause piston ring breakage.

Despite such a high level of predictability using PFIN, there are a few instances when piston ring breakage can occur even when the PFIN value is not elevated. These include situations when the scavenge and under piston drains are blocked resulting in water entering the cylinder, styrene/DCPD (dicyclopentadiene) contamination, non OEM piston rings and very high catfine levels. Of these, high catfine levels should be detected by regular bunker fuel analysis and the piston ring type should be known prior to any voyage. Styrene/DCPD contamination can be detected only by Gas Chromatography Mass Spectrometry (GCMS).

In conclusion, bunker fuel quality remains a constantly evolving challenge for fuel users. The majority of main engine equipment problems are caused by poor quality fuels and lubes. Viswa Lab evaluated several cases of piston ring breakage and developed our patented PFIN, which is an algorithm designed to reliably predict the fuel that can cause piston ring breakage.

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3 ASTM D6560 - Standard Test Method for Determination of Asphaltenes (Heptane Insolubles) in Crude Petroleum and Petroleum Products

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Container vessel propulsion selected for the future

ABSTRACT

The operating boundary conditions for container vessels are changing continuously. New regulations for exhaust gas emissions or allowed fuels demand new technical solutions and impact the operation of the vessel.

In 2015 the fuel sulphur level was limited to 0.1% in SECA areas, in 2016 the IMO NOx Tier 3 limit have been introduced and in 2020 a next step for fuel sulphur limitation is expected.

The balance between market demand in transportation of goods and the available vessel capacity is changing, forcing operators to either move for slow steaming or to increase again vessel speed.

Last but not least fuel prices and availability are changing impacting the optimum operation profile and choice of fuel. So LNG has become a new alternative for future shipping.

2-stroke engines designed by Winterthur Gas & Diesel (WinGD) follow the market trends. Today the built-in flexibility of the engine specification allows shipyards and ship owners to adapt the performance specifically for the prospected operation requirements.

The paper describes the optimisation path in the definition of the propulsion system including the engine. It includes as well ideas on how to prepare the vessel propulsion system in new building stage for easy adaptation to new requirements after first years of operation.

1. BOUNDARY CONDITIONS FOR THE ENGINE SELECTION

1.1 CO2 emissions

In 2012 the IMO (International Maritime Organisation), regulatory body for international seagoing vessels, introduced the Energy Efficiency Design Index (EEDI). The vessel specific EEDI is calculated as CO2 emitted by a vessel for every transported cargo in dwt per nautical mile.

The EEDI value to be met depend on the date of contract and is defined in regulation 21 of the Marpol Annex VI (also described in the IMO resolutions MEPC 203(62), see www.imo.org). To ensure a continuous improvement and innovation in the shipping and shipbuilding industry the target levels are reduces step by step till the year 2015 (Fig. 1).

To achieve a low EEDI the following main measures can be considered for the vessel:

1. Reduce the installed engine power
2. Apply engines with lower fuel consumption
3 Install additional energy saving technologies like waste heat recovery
4. Burn gas instead of HFO having less CO2 emission compared for the energy equivalent volume burnt.

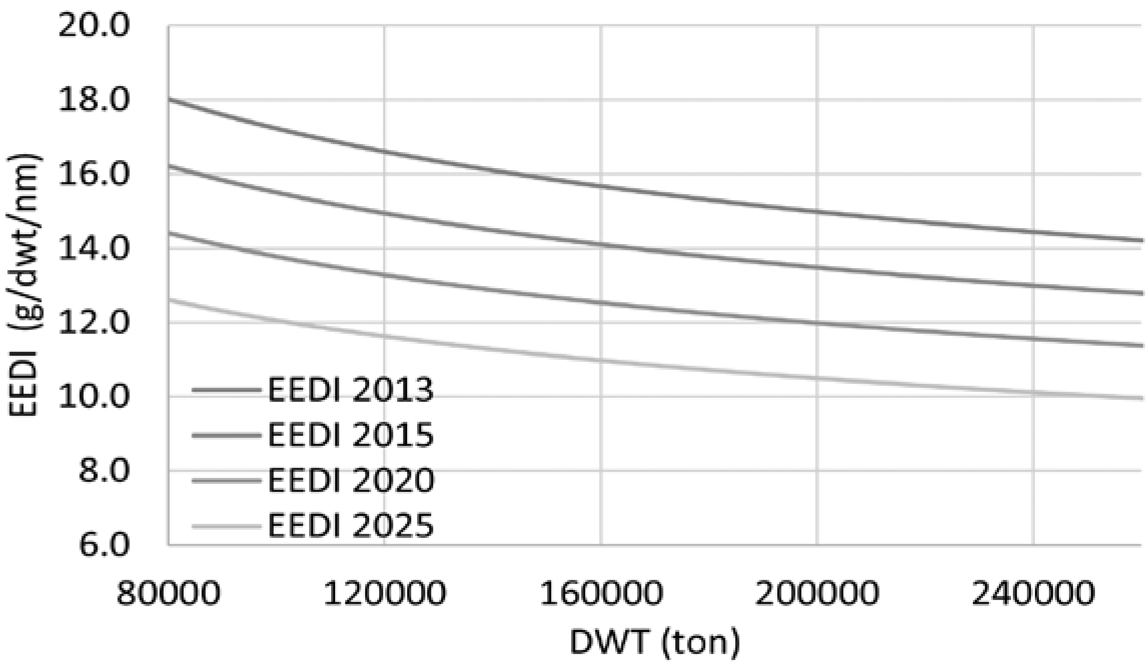


Fig. 1: EEDI target levels introduced by the IMO for very large container vessels

1.2 Container transport capacity and fuel prices

In case of a low ratio of market transportation demand against available container vessel transportation capacity, a reduced vessel speed can be considered to reduce the operating costs and ensure well loaded vessels. Vice versa, high market demand may advise higher vessel speed.

At very low fuel prices (per ton) the fuel operating cost importance is reduced and an increase in vessel speed may be considered to maximize the vessel operating result.

When deciding for a new vessel, defining a scenario for the coming years in view of expected market development, overall available shipping capacity as well as fuel market trends will help to make the right choice of the propulsion system.

1.3 Fuels and related regulations

The worldwide limit of 3.5% sulphur content in fuel introduced in 2012 is intended to be reduced down to a 0.5% sulphur cap. The new limit will be enforced in EU waters from 2020. The IMO will decide on the worldwide sulphur cap in 2018 for start in the year 2020 or 2025.

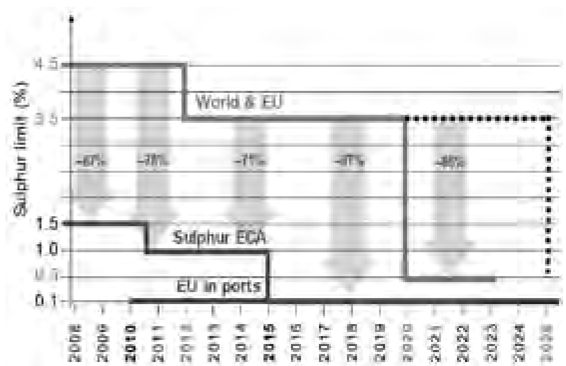


Fig. 2: Present and future IMO sulphur emission regulations.

The sulphur limitations can be respected by either running with low sulphur fuels or by applying exhaust gas scrubbers removing the sulphur oxide (SOx emissions) from the exhaust gas flow.

Another option is the application of LNG as main fuel, cutting down the sulphur emissions to lowest level and reducing additionally CO2 emissions by 25% to 30%.

For the WinGD X-DF engines a break-through in engine technology has been realized. Applying the low pressure gas concept the IMO NOx Tier

III limits are reached automatically, which avoids the investment into an expensive high pressure gas system and a NOx reduction system for the operation in NOx emission control areas – see chapters 3.1.3 and 4.2 for more details on the application of X-DF engines.

1.4 NOx Emission limits

The NOx emission limits implemented by the IMO in the year 2011 (Tier II) were succeded by the IMO Tier III regulations, valid for ships built after 1. January 2016, but only in designated NECA areas.

To meet the low Tier III NOx emission limits diesel engines demand the investment into NOx reducing technologies. For WinGD designed 2-stroke engines the SCR technology is offered. Two installation options are available (see also chapter 3.4):

a) The high pressure SCR (HP SCR) installed between exhaust receiver and turbocharger allows lowest dimensions, but requires a placement in the engine room
b) The low pressure SCR (LP SCR) solution allows the installation of the SCR reactor after the turbocharger outside the engine room.

1.5 Flexibile propulsion setup

Compared to the situation before 2008, vessels and their operators have to face more frequently changing boundary conditions during the vessel life time. The changing demand and supply situation, varying fuel price, different possible fuels and new emission regulations will give advantage to flexible vessel and propulsion designs able to be reconfigured for optimum performance for any actual condition.

This flexibility can be addressed in three different ways, corresponding to different compromise concepts between easiness of flexibility and related costs to achieve it:

– Apply from the beginning an engine design, which can be easily changed to operate efficiently at any new boundary condition.
– Make small investments at initial design to prepare the vessel for later conversion and upgrades
– Make no preparations in newbuilding stage and decide on installation modifications when a major change in boundary conditions will happen
– The following table show a possible scenarios scheme.

Year	2017	2020	2025
Item \ Description	Vessel travel only between Europe and Asia with slow steaming	0.5% fuel sulphur cap intro-duced worldwide. Continue operation with high sulphur HFO and install sulphur scrubber. Good market condition allows higher vessel speed	LNG becomes much cheaper than HFO. Convert propulsion system fully for continuous LNG operation. Extension of operation to US coast.
Engine	X92 derated	X92 with higher rating	Conversion to X92DF
Fuel	HFO & engine and vessel installation ready for LNG	-	LNG
NOx emission level	Tier II	-	Tier III – engine already compliant with X-DF tech-nology
Sulphur strategy	-	Installation of scrubber	LNG operation. Removal of scrubber
Propeller	Propeller optimized for low speed	Change propeller for high speed operation	-

Table 1: Example for a possible ship owner's operation scenario to decide on propulsion and engine technology, reacting on possible coming new regulations and commercial boundary conditions.

2. ENGINE POWER AND SPEED SELECTION

2.1 Vessel speed and operational economy – slow steaming

Slow changes in vessel speed (v) are correlated with high changes in engine power (PE). The correlation shows as:

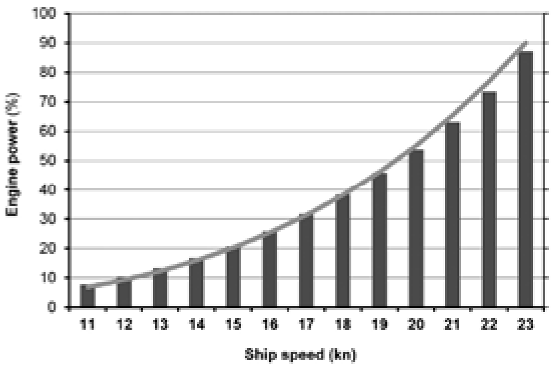


Fig. 3: Power and vessel speed correlation for very large container vessel

$P_E = const * v\beta$

The factor β depends on the actual propeller and hull design. For very large container vessels designed for 21 knots speed β is about 3.4 leading to the following correlation:

The above curve is similar to the theoretical ideal propeller law correlating engine speed (now vessel speed) and power, applying a factor 3.

2.2 Define engine rating

Starting point for any engine selection is the power and propeller speed required to achieve the target vessel speed. Following margins have to be added:

LRM – Light running margin to reflect the changed propeller curve (power versus vessel and propeller speed) due to fouling of hull and propeller and heavy weather conditions

SM – Sea margin to reflect the increased power demand with fouling and heavy weather conditions

EM – Engine margin as mechanical and thermodynamic power reserve for operation at best fuel consumption

For the light running margin WinGD recommends a value between 4% and 7%. Typical values for sea and engine margin are 15% and 10% respectively.

2.3 Propeller selection

As basic law, the lower the propeller rotation speed, the larger can be the diameter of the selected propeller. And the bigger the propeller, the bigger is the achievable maximum propeller efficiency.

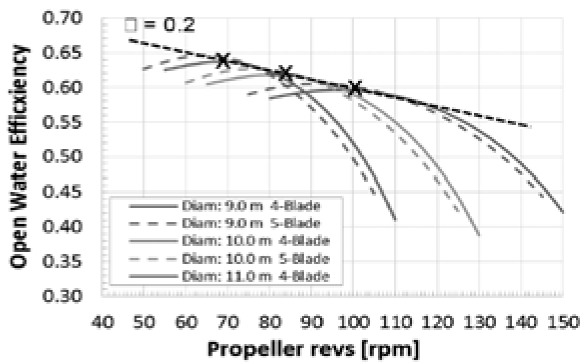


Fig. 4: Achievable propeller efficiencies at given vessel design speed and propeller speed, varying propeller diameter and blade number (example)

Additional to the propeller diameter the number of blades can be increased or reduced to move the efficiency peak to lower or higher propeller speed ranges keeping the same diameter and so avoiding to change hull design.

3. ENGINE SELECTION

3.1 Engine types and their characteristics

3.1.1 Available engines for operation with diesel and LNG

The WinGD X-engines have been designed to meet newest emission regulations at lowest fuel consumption and increased maintenance friendliness.

The target power range between 35000 kW and 70'000 kW can be covered by two engine types:

The X82 engine with 6 to 9 cylinder or the X92 available from with 6 to 12 cylinders.

Both engines, X82 and X92, are available for HFO and MDO/MGO operation, as well as in X82DF and X92DF version for operation with LNG or diesel fuel.

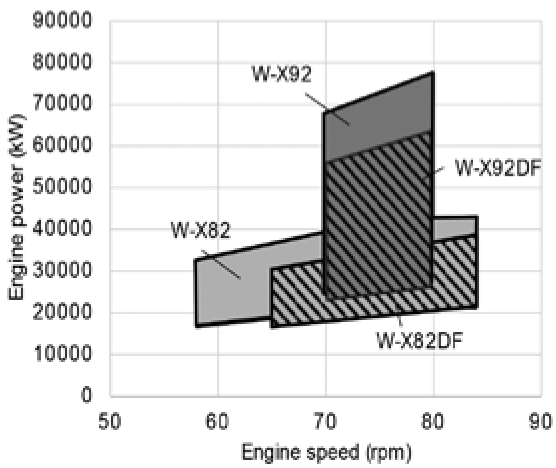


Fig. 5: Power and speed range covered by X82 (6- to 9-cylinder), X92 (6- to 12-cylinder) and related gas engines X82DF and X92DF.

3.1.2 Common characteristics of X82 and X92 diesel and gas engines

Basic layout

The improved design of the X-engine generation enhances reliability, engine performance and easy maintainability.

Common proven technology platforms form the basis for the full X-engine portfolio with engine individual adaptations to meet different engine size and power requirements.

RT-flex common rail system

Introduced in the year 1999, the reference RT-flex common rail system has been continuously improved to increase component reliability and lifetime and reduce the maintenance costs.

RT-flex common rail technology provides great flexibility in the engine setting for lower fuel consumption, lower minimum running speeds, smokeless operation at all running speeds, and better control of exhaust emissions.

The integrated redundancy maintains the high reliability of the engines.

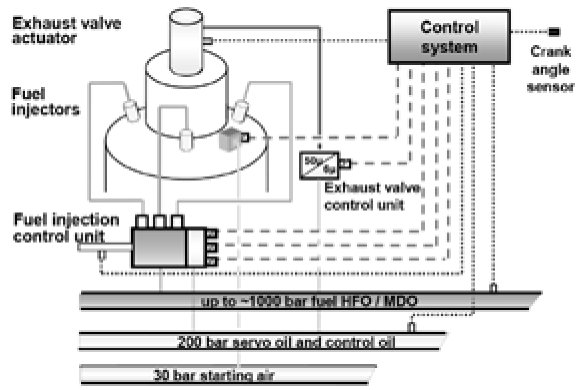


Fig. 6: The unique RT-flex common rail system allowing flexible adjustment of fuel injection characteristics and exhaust valve timing

The ability to regulate the engine's operational performance benefits the manoeuvring capabilities and operation at lowest possible operating speed, as for example during canal transit and port entrance.

Advanced cylinder lubrication concept

The trends to slow steaming in vessel operation and towards high stroke to bore ratios require a dedicated cylinder lubrication system to cope with the more challenging conditions in the combustion chamber.

Fig. 7: Pulse lubricating system with pulse jet technology to distribute the oil evenly on the cylinder liner surface

The X82 and X92 engines apply the latest version of the electronically controlled Pulse Lubricating Systems (PLS) introduced in 2006, replacing the mechanical controlled CLU3 lubrication. The PLS system has been continuously improved and adapted to latest operating boundary conditions and includes the Pulse Jet technology providing best possible cylinder oil distribution on the cylinder liner and piston rings.

Great care has been taken to increase the liner wall temperatures against cold corrosion at slow steaming conditions by additional isolation and bypass cooling, so allowing continuous operation between 10% and 100% engine load with low liner and ring wear.

Automation concept

The engines are controlled by the well proven and validated WECS embedded control system that allows the engine to be operated even if the bridge management system is out of order.

Communication with external systems (Propulsion Control, Alarm Monitoring, and Tier III solutions) is facelifted by CAN or Mod-Bus and is ready for interfacing with new technologies, such as BIG DATA.

Low maintenance costs

WinGD's X82 and X92 engines are designed to achieve as much as five years time between overhauls (TBO).

The TBO of low-speed marine diesel engines are largely determined by the wear of piston rings and cylinder liners. The X-engines' piston-running package, including the Pulse jet lubricating system explained before, ensures longest life time of these components with low lubricating oil consumption.

3.1.3 Low pressure X-DF engines

For the operation with LNG an additional injection system is required to introduce LNG into the combustion chamber for optimum combustion, performance and emission results.

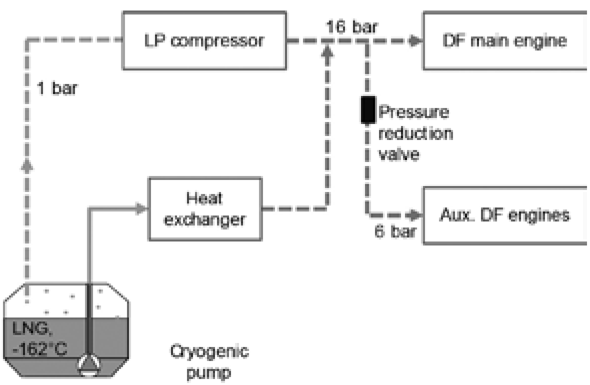
The X-DF engines apply the Otto combustion principle, the only technology able to combine the use of the reliable and cost efficient low-pressure gas injection with high performance and lowest exhaust gas emission levels.

Including the fuel gas supply system (FGSS) the propulsion system looks as follows.

Fig. 8: Low pressure fuel gas supply system (FGSS) for the low pressure X-DF engine with tank, low pressure compressor and pumps, piping. Only low gas pressure components required.

The low-pressure technology reduces NOx to comply with IMO Tier 3 limits without any additional cost intensive reduction technology.

The advantage of the low pressure system from financial investment and operation perspective are described in chapter .



3.2 Engine tuning

The X-engines can be optimised for low, partial or high load operation. The following tuning options are available:

1. Standard Tuning: high load tuning, Delta Tuning: part load tuning,
2. Delta Bypass Tuning: part load tuning, optimised for increasing steam production above 50% engine load, and reduced fuel consumption below 50% engine load.
3. Low-Load Tuning: optimised for engine loads below 75 %.

4. TC cut off: one of three turbochargers can be cut off to improve low load performance.

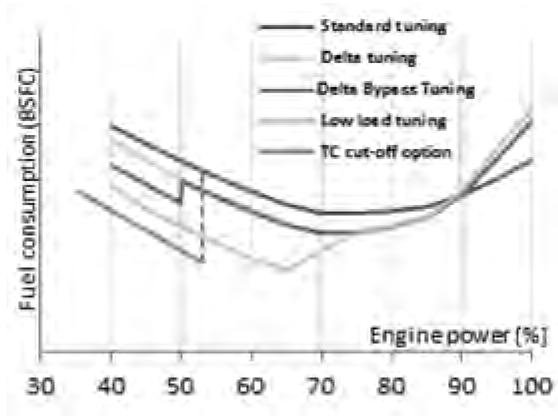


Fig. 9: Engine tuning options to optimize fuel consumption for individual operational profile

Additional option is the "steam production control" (SPC). The SPC allows to vary the bypass rate continuously to set the exhaust gas temperature exactly according to the need of the economizer for steam production. This ensures to get the best overall fuel consumption.

3.3 Waste heat recovery (WHR)

Waste heat recovery is an effective technology for simultaneously cutting exhaust gas emissions and reducing fuel consumption. These WHR plants thus allow to cut exhaust gas emissions, deliver fuel savings of up to 10%, and improve the ship's EEDI.

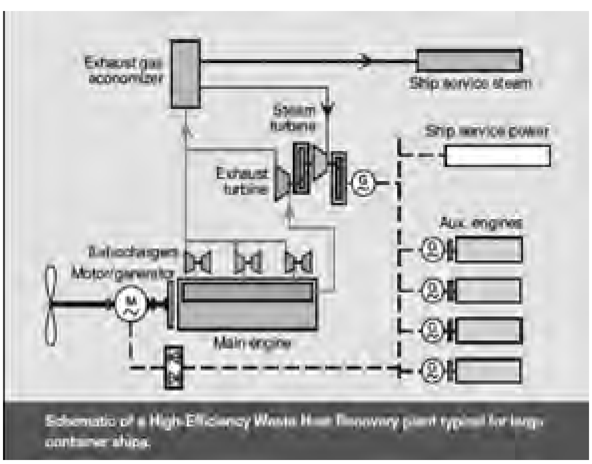


Fig. 10: Layout diagram with waste heat recovery system

In the WHR plant, a turbo-generator combines the input from a steam turbine with an exhaust gas power turbine to generate electrical power, while steam from the economiser is available for the ship's service heating.

3.4 NOx Emission reduction technology

X-engines comply to IMO Tier 3 NOx limits thanks to the designed interface with SCR systems and, as alternative, with the X-DF engine concept.

3.4.1 SCR solutions

The SCR technology reduces emissions of nitrogen oxides (NOx) by means of a reductant (typically ammonia, generated from urea) at the surface of the installed catalyst.

Compared to the EGR technology, SCR is a proven technology used for decades in land based power plants as well as on marine engines. Due to the lower investment and maintenance costs of SCR as well as the imminent risk of EGR for engine reliability issues by recirculating polluted exhaust gas into the cylinder, SCR is the preferred solution to comply with the IMO Tier 3 NOx limits.

The SCR technology is available as high pressure SCR (HP SCR) and low pressure SCR (LP SCR). HP SCR can be operated with MDO/MGO or HFO, whereas presently offered solutions with LP SCR is limited to MDO/MGO operation.

A specific engine tuning in Tier III allow to minimize fuel consumption whilst providing the required exhaust gas temperature for the SCR operation.

3.4.2 Operation with LNG

As discussed before the low pressure X-DF engine operated with LNG comply with IMO Tier III NOx and SOx limits.

Comparing with the SCR system, the installation of an LNG fuelled propulsion system including the fuel gas supply system (FGSS) requires significant higher investments into a new vessel.

Therefore from total cost of ownership point of view and investment in X-DF operation makes only sense, if LNG can be bunkered to much lower prices than with HFO or MDO.

If the operational savings with LNG are not big enough the operation with diesel and the installation of an SCR system for IMO Tier 3 compliance will be the preferred solution.

3.5 Flexible operation adaptation

3.5.1 Dual tuning

Dual Tuning can be selected when a special operating profile is required. All X-engines can be built and certified with two different tuning combinations.

Dual tuning provide customers with benefits in terms of specific fuel consumption, and improved exhaust gas flow and temperatures.

The engine's NOx certification will be provided by individual Technical Files and EIAPP certificates for each tuning.

3.5.2 Dual rating

X-engines can be designed and shop tested for two different engine ratings. This means that the engine can have two optimised CMCRs points following the same propeller curve.

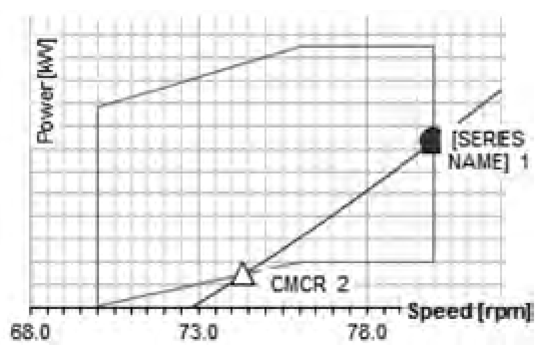


Fig. 11: Selection of 2 engine rating points on same propeller curve

As a consequence, the ship operator can select two possible optimal service speeds, depending on the market conditions.

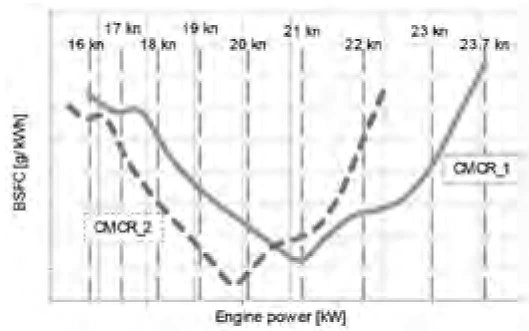


Fig. 12: Dual rating - selection of two CMCRs on same propeller curve allowing savings in low load operation with CMCR_2.

3.5.3 Change from diesel to LNG fuel

Both engines, the X82-B as well as the X92 can be converted for operation as dual fuel engine with focus on LNG operation. X-engines have been designed to be DF ready, meaning that the standard diesel engine can be converted to a low-pressure X-DF engine. The major vessel retrofit part concerns the installation of the LNG fuel tanks and the fuel gas supply system (FGSS):

- LNG tank
- Cryogenic pumps and evaporators for cold LNG to produce LNG at max 16 bar
- Optional compressor pumps for boil of gas (BOG)
- Gas valve unit to control gas pressure and supply before engine

The original diesel engines are modified to X82DF or X92DF respectively by:

- Installation of the low pressure gas fuel injection system
- Adding the pilot fuel injection system for ignition of the LNG engine
- Include the control system for LNG operation
- Adapt the turbocharging system for the LNG operation

4 NEWBUILDING CASES

4.1 X82 and X92 for 14000 TEU vessel

The target vessel shall have the following specification:

– Transportation capacity: 14000 TEU
– Vessel design speed: 22 Kn
– CMCR 1 power with 76 rpm: 42500 kW

As option a dual rating shall be investigated, to improve the fuel consumption for low load operation, applying the same propeller:

Vessel 2nd design speed: 19.3 Kn
- CMCR 2 power with 66.5 rpm: 28500 kW (along propeller line from CMCR 1)

To improve the propulsion efficiency the impact of a larger propeller shall be investigated for engine layout at 72 rpm:

CMCR 3 required power with 72 rpm: 42000 kW (constant speed coefficient =0.2)

And vessel 4th design speed: 20.3 Kn
CMCR 3 required power with 70 rpm: 33200

To cover the above demand the following engines are investigated:

- 9X82
- 7X92
- 8X92

In the layout field for power and engine speed the selected points look as follows:

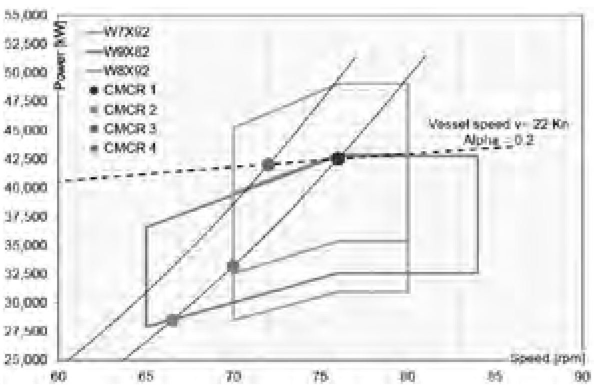


Fig. 13: CMCR selection, propeller curves in the X82 and X92 engine layout field

All 3 engines can be applied for CMCR 1. The lower propeller speed option is available only with 8X92, whereas 9X82 engine allows the widest dual rating step.

When operating the vessel at 20 Kn the best fuel consumption is obtained with the 8X92, saving at least 2.3% fuel costs compared to 7X92 and 9X82. By taking the larger propeller for CMCR 3, the fuel costs can be reduced by additional 0.5%, reaching 2.8%.

The derated engines (CMCR 2 and 4) give the best results when running at 18Kn vessel speed. Compared to 20 Kn speed, 22% to 23% fuel savings can be achieved for the transport of one TEU from A to B. Lowest fuel cost per TEU in this case can be achieved with the derated 8X92, having about 1% advantage compared to the derated 7X92 and 9X82 engines.

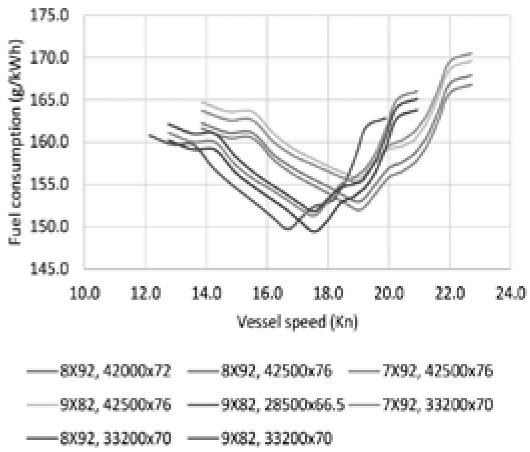


Fig. 14: Power specific fuel consumption drawn over ship speed.

4.2 12X92 and 12X92DF for DF ready 20000 TEU vessel

The target vessel shall have the following specification:

– Transportation capacity: 20000 TEU
– Vessel design speed: 22 Kn
– Power at CMCR with 78 rpm: 60000 kW

It is assumed the ship owner expects to change operation to LNG, when the HFO sulphur cap is reduced worldwide to maximum 0.5%. Same vessel speed shall be run with diesel and LNG. The engine therefore shall be easily convertible to X-DF for dual fuel operability.

At the time of conversion also IMO Tier III NOx compliance shall be achieved.

Best selection for this requirement is the 12X92/12X92DF engine.

A change towards LNG operation start to makes sense in case of a much lower price of LNG compared to low sulphur HFO (LSHFO). In the case study it shall be assumed, that LSHFO cost 500 USD / ton and LNG 400 USD / ton.

With about 24'000 USD savings per day running continuously 80% load, a yearly saving of about 6 Mio USD can be considered. In view of investments for LNG operation into the vessel including LNG fuel handling infrastructure and engine, which has to be expected in the range of 30 to 50 Mio USD, the payback time would be 5 to 9 years.

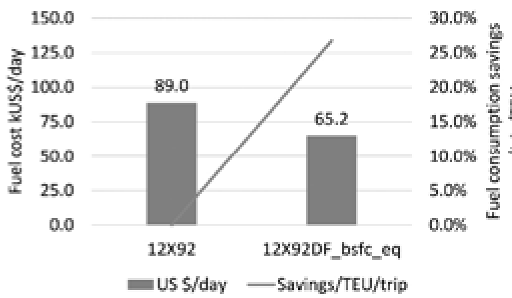


Fig. 15: Fuel costs per day and savings with LNG compared to diesel at continuous 80% load assuming fuel price of 400USD/ton for LNG and 500 USD/ton for low sulphur HFO.

Important is to notice the Tier III compliance of the 12X92DF compared to the 12X92 engine, avoiding installation of NOx reducing technology on or after the engine. As well CO2 and particulate matters can be reduced considerably. All these low values are achieved not only in ECA areas, but all time the engine is operated with LNG.

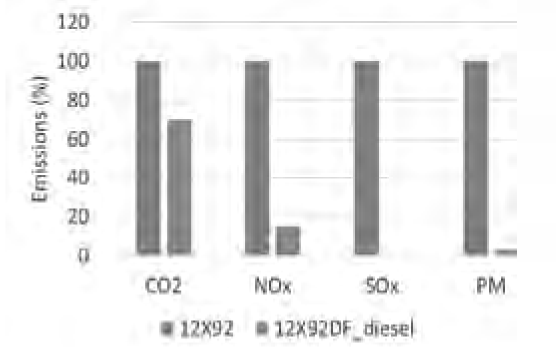


Fig. 16: Emissions reduction from 12X92 in diesel operation towards 12X92DF in LNG mode.

5. CONCLUSIONS

Varying market conditions, different operation profiles and new upcoming regulations require individual optimisation of the engine and propulsion system for large container vessels.

Due to the wide range of optimisation possibilities, the X82 and X92 marine 2-stroke diesel engine and their relative versions X82DF and X92DF for LNG operation represent optimum solutions as propulsion engines for very large container vessels with freight capacities between 8000 TEU to 22000 TEU.

Several engine tunings and ratings to optimize the operation for different operating profiles and varying fuel cost scenarios, as well as the possibility to use any diesel fuel or LNG, provide means to find the optimum installation for any need. As such these engines can meet customers demand at the initial stage as well as after some years of operation with lowest modification costs to adapt for any future new boundary conditions regarding operational profile, applied fuels and environmental regulations.

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Low-pressure X-DF engines: ideally suited to power LNG carriers and gas-fuelled merchant vessels

ABSTRACT

In 2013 Winterthur Gas & Diesel Ltd. (WinGD), formerly Wärtsilä's 2-stroke division, introduced low-pressure low-speed engines to the emergent gas-fuelled vessel market. In the meantime, the first engine of the X-DF family successfully passed the type approval test. Many X-DF engines are factory acceptance tested and will start commercial operation in the next months.

This paper explains the low-pressure dual-fuel key technology and demonstrates the paramount advantages under different aspects: environmental benefit, installation cost optimisation (CAPEX) as well as operational benefits (OPEX).

BACKGROUND

In 2012 the IMO (International Maritime Organisation), regulatory body for international seagoing vessels, introduced the Energy Efficiency Design Index (EEDI). The vessel specific EEDI is calculated as CO2 emitted by a vessel for every transported cargo in dwt per nautical mile.

In a thorough analysis of the benefits of high-pressure gas injection and low-pressure gas admission on two-stroke dual-fuel (DF) engines, assumptions were based extensively on data acquired in successful tests conducted by the predecessors of Winterthur Gas and Diesel (WinGD) on both two- and four-stroke DF engines with high-pressure gas injection technology. It is noteworthy that the results of that research and development from the mid-1980s aligns well with the technology of low speed engines with high-pressure gas injection now reaching the market, especially in terms of gas compression requirements and the parameters for the proportions of liquid and gaseous fuel. As with modern DF engines with high-pressure gas injection, a gas injection pressure of around 300 bar and pilot oil injection of 3 % to 5 % of the total heat release were employed.

Targeting maximum added value to ship owners and operators, among the key economic drivers in the process were excellent environmental-friendliness, simple and maintenance-friendly installation combined with lowest possible investment costs. In addition, port-to-port operation on gas was taken into account and the paramount consideration of the safety of the total gas installation, i.e. engine and ancillary system, including redundancy in case of gas system failures, was studied exhaustively. Based on WinGD's findings, low speed two-stroke engines with low-pressure gas admission and employing the Otto Cycle were developed and officially introduced to the market in 2013. The basic operating principle is adopted from the four-stroke DF engines with "micro-pilot" injection of liquid fuel i.e. representing only 1 % or less of total heat release, with which Wärtsilä established the reputation of four-stroke DF engines from the late

1980s. As shown in , in the two-stroke version of Wärtsilä's DF technology, low-pressure gas admission takes place via admission bores located at the mid stroke of the piston, where compression is still relatively low, instead of at top dead centre against maximum cylinder compression in the case of high-pressure gas injection. Ignition is assisted by pre-chambers housing the pilot injectors. The location of the pilot injector and of the main injector for diesel-operating or fuel-sharing operating mode is shown in .

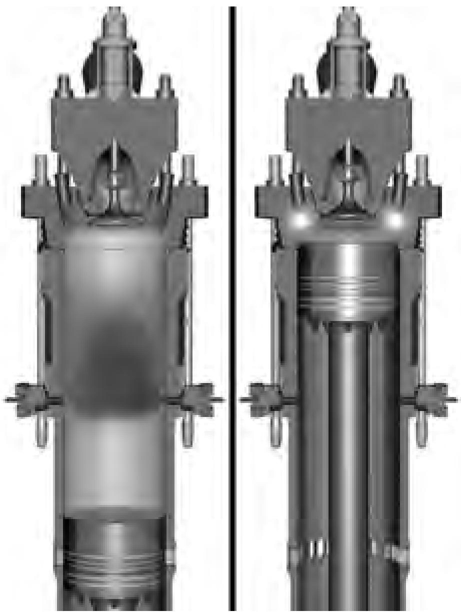

Figure 1: The 2-stroke DF principle with gas admission (left) and ignition (right).

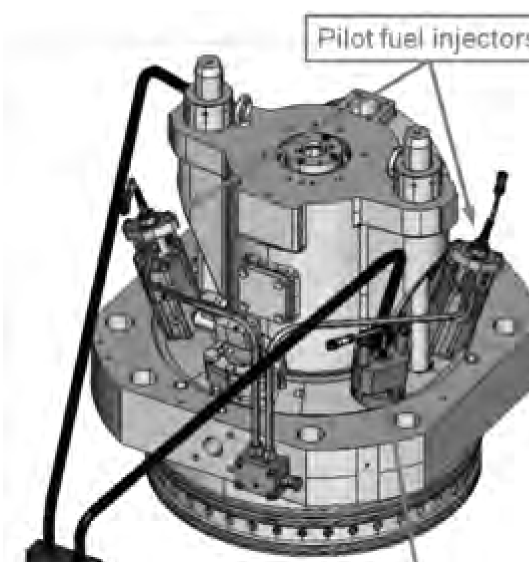

Figure 2: Main- and pilot-fuel injector arrangement.

EMISSIONS

A comparison of emissions from the 72 cm bore WinGD W-X72DF engine, a high-pressure dual-fuel engine, both in gas operating mode, with a conventional pure diesel engine operating on HFO is shown in . The major emissions characteristics are detailed in the following section.

The higher hydrogen-to-carbon ratio of natural gas in comparison to liquid fuels leads to a direct reduction in carbon dioxide (CO2) emissions. However, Total Unburned Hydrocarbon (THC) emissions also need to be considered. In the Diesel Cycle combustion process on a high-pressure DF engine, gas is burned in the gas jet flame, i.e. combustion is locally rich and consequently most gas-fuel is burned. Hence, emissions of unburned hydrocarbons, mainly methane, are kept very low. In the Otto Cycle combustion process on a low-pressure DF engine, a lean fuel-gas pre-mixture is burned. Thus, it is possible to have localised areas of fuel-gas/air mixtures which are too lean and/or too cold, with some fuel-gas possibly not completely burned. Consequently, emissions of unburned hydrocarbons and methane (also known as methane slip) can be higher. Wärtsilä's medium speed DF engine development showed that HC emissions and methane slip can be significantly reduced by optimising the air/gas mixing process and avoiding dead volumes and crevices in the combustion chamber which can create areas of excessively lean mixture. Also, a suitable ignition method is required for the whole fuel-gas/air mixture to maximise complete combustion, and is provided by pre-chamber technology.

SULPHUR OXIDE EMISSIONS

Sulphur oxide (SOX) emissions are significantly reduced by both DF technologies, since natural gas is practically sulphur-free and any remaining sulphur oxide emissions are directly linked to the sulphur content of the pilot-fuel and the required pilot-fuel injection quantity. Less specific pilot-fuel is needed for low-pressure DF engines and consequently low-pressure technology provides a marginal

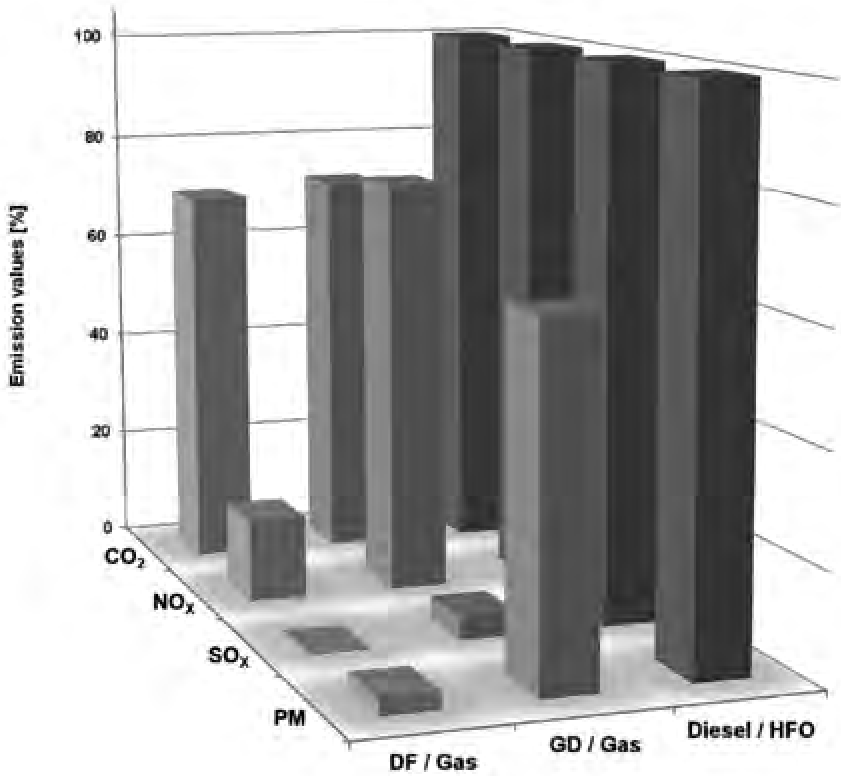

Figure 3:Typical diesel and DF engine emissions.

environmental advantage in terms of SOX.

NITROGEN OXIDES (NOX)

In the rich diesel combustion process, local high temperature peaks are created in the combustion chamber and lead to NOX formation. By contrast, the combustion process of the lean burn Otto Cycle ensures very equal combustion temperature distribution which ultimately keeps NOX emissions significantly lower than the IMO Tier III limit for Emission Control Areas (ECAs) at all times.

PARTICULATE MATTER (PM)

The difference between lean and rich combustion processes has an influence on PM emissions, which are significantly reduced by the Otto Cycle combustion process. The remaining PM emissions of a low-pressure DF engine are mainly related to Diesel Cycle combustion of the pilot-fuel and combustion of some cylinder lubricating oil. A high-pressure Diesel Cycle dual-fuel engine also helps to reduce PM emissions in comparison to a conventional diesel engine, since natural gas is, compared to liquid fuel, a significantly lighter hydrocarbon fuel and also has a higher hydrogen-to-carbon ratio. However, PM emission levels still reach almost two thirds those of a conventional diesel engine due to rich combustion. Besides the clear environmental impact of PM emissions, legislative limitations are expected in the near future.

DEVELOPMENT CHALLENGES

The challenge for a high-pressure DF engine solution is principally the handling of the fuel-gas when pressurised to at least 300 bar. The two-stroke engine test carried out in 1986 showed that handling fuel gas at 300 bar is, in principle, possible and field experience with medium speed high-pressure DF engines confirmed this conclusion, while on the other hand the demanding high-pressure gas system hampered the market breakthrough. The development of low-pressure DF engines is more demanding. The engine requires the correct homogeneous gas/air mixture which should be evenly distributed within the whole combustion chamber. Hence, the air and gas needs to be evenly mixed in the combustion chamber to achieve a homogeneous mixture. Local areas of over-rich fuel-gas concentrations could lead to self-ignition, while areas of over-lean fuel-gas concentrations might not ignite and, consequently, unburned gas would be lost in the exhaust. The compression ratio must also be reduced for Otto Cycle gas operation, but kept as high as possible for the diesel operation mode. WinGD's modern, fully electronically controlled engines provide the basis for ingenious solutions to these challenges.

SAFETY

Irrespective of which technology is applied, safety aboard the ship has to be at the same level as with conventional diesel engines. Double-walled pipes are required for both solutions, so that leaking gas cannot enter the gas-safe engine room, and the leakage can be detected immediately. The amount of fuel-gas in the engine room piping system is generally similar for both solutions, as the fuel-gas is either transported in larger diameter pipes at lower pressure/density or in smaller pipes at higher pressure/density. While a fracture in a pipe or a pipe connection carrying high-pressure gas has a higher expansion energy potential, both solutions can be made safe.

ENGINE PERFORMANCE

A high-pressure Diesel Cycle DF engine applies the same combustion process as a conventional diesel engine. Thus, in general no basic performance differences are expected. With an Otto Cycle low-pressure DF engine, performance differences need to be considered due to the different combustion process. The maximum power output of a DF engine employing the Otto Cycle is lower than that of an engine employing the Diesel Cycle. This can be seen as a drawback and a deeper evaluation was needed to establish if a power density deficit is acceptable or not. Most conventional diesel engines are selected in the lower range of their rating fields due to the fuel-saving potential of derating. Low-pressure DF engines have quite similar performance within their whole rating field, and are even slightly better in the upper range. Thus, the rating point can be freely selected within their entire rating field. The theoretical drawback of the reduced maximum engine output is, in most cases, practically no issue, as can be seen in , which shows the selected rating points of current WinGD X62 diesel engine projects in comparison to the W-X62DF's rating field.

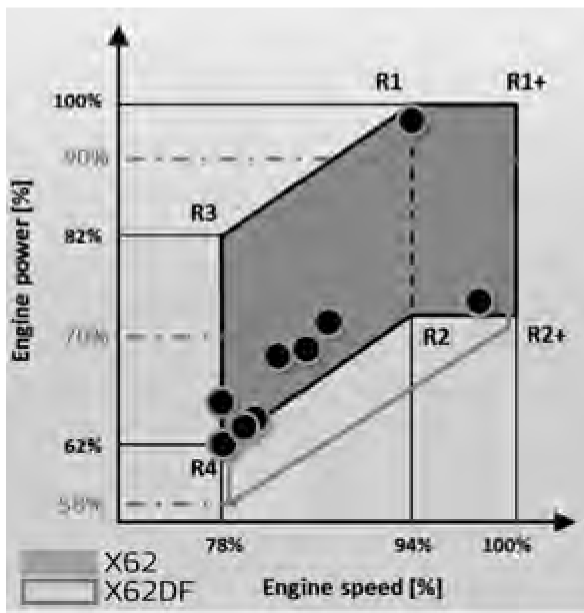

Figure 4: Rating field comparison, conventional diesel engine versus dual-fuel engine with low-pressure gas admission.

METHANE NUMBER DEPENDENCY

As the fuel-gas/air mixture is being compressed, uncontrolled self-ignition needs to be avoided. Since the fuel-gas is a mixture of different gases, its resistance to self-ignition, indicated by the methane number (MN), can vary. For fuel-gas with a lower stability against self-ignition, maximum engine output might be limited. The engine

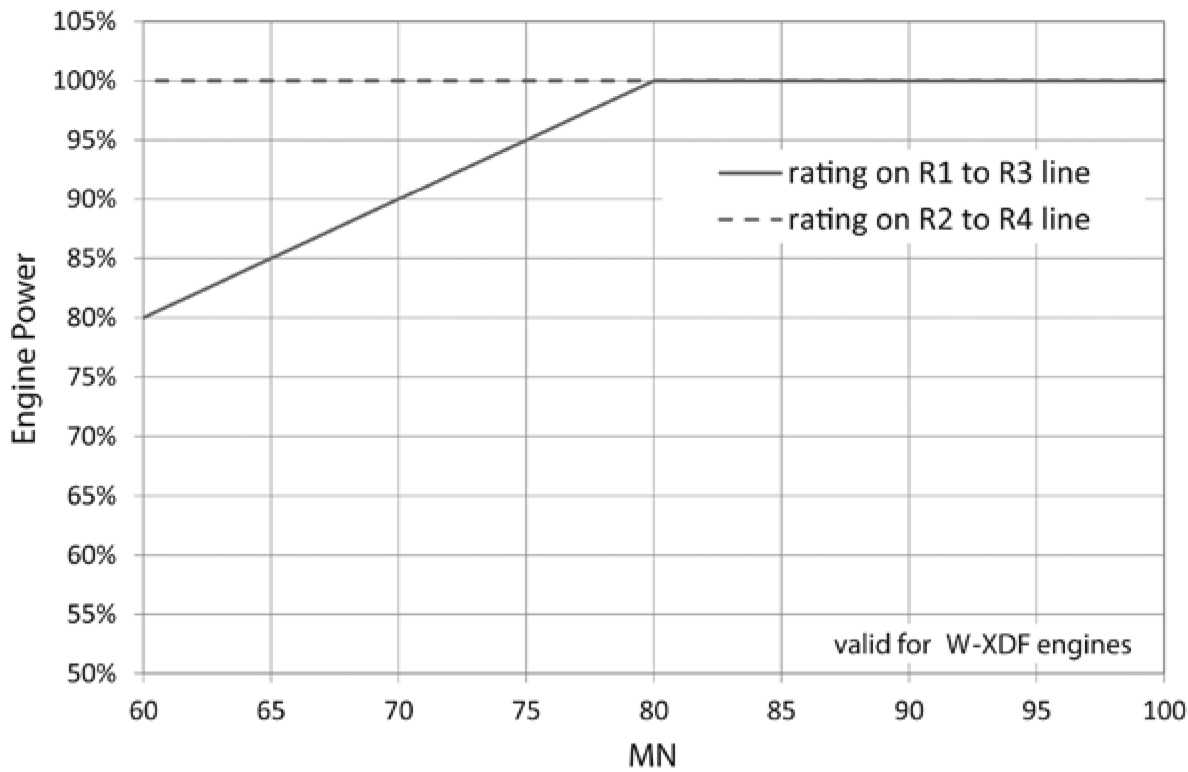

Figure 5: WinGD dual-fuel engine, power output in gas operation.

output which can be reached under all conditions will be at least as indicated in . For conditions colder than the tropical design condition, more engine output can be attained. Considering that the majority of available fuel-gas has a methane number in the range of 70 to 90, an LNG-fuelled vessel can always be operated up to its service speed, which is typically reached at 85 to 90 % of the contracted maximum power output. For LNG carriers, using LNG as both cargo and fuel, the situation is somewhat different. Natural boil-off gas (NBOG) has a high methane number at about MN100, since the lightest fraction, i.e. methane, evaporates first. For forced boil-off gas (FBOG) a controlled evaporation is usually carried out, which allows the heavier fractions to be returned to the cargo tank, while delivering the light methane to the engine. Thus, the methane number of the gas supplied to the engine is usually clearly above MN 80, even though the methane number of the remaining LNG in the tank during ballast voyage can be much lower. Thus, while it can be concluded that the dependency on methane number needs to be considered as an engine design criteria, it is only of minor relevance to vessel operation.

EFFICIENCY

As is well known, a Diesel Cycle engine generally has the best efficiency of all internal combustion engine types taking into account the entire load range. However, at high engine loads the efficiency of an Otto Cycle engine is similar or even better. shows a schematic comparison between the specific energy consumption of an Otto Cycle engine to that of a Diesel Cycle engine as shown in . Over the entire load range, low-pressure technology can be expected to be at a disadvantage. However, the high-pressure gas supply, inherent to the high-pressure solution, consumes more electrical power than a low-pressure supply. In addition, high-pressure technology requires exhaust gas treatment to achieve low NOX emissions. Depending on the selected treatment system, additional chemicals will be consumed, additional electric power is required and engine performance is reduced. Thus, engine efficiency should be considered in terms of overall system performance, and this is the subject of the case studies as followed.

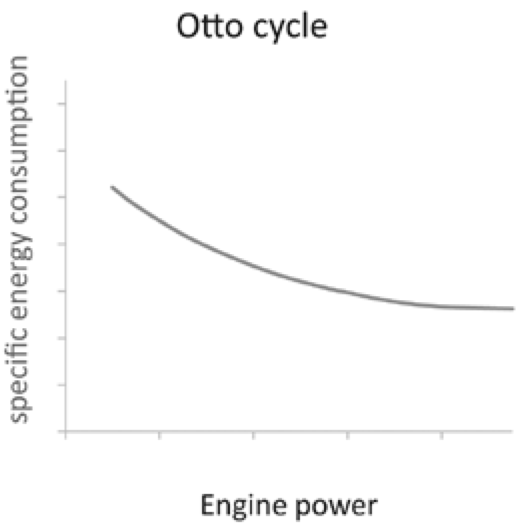

Figure 6: Specific energy consumption for the Otto Cycle (schematic).

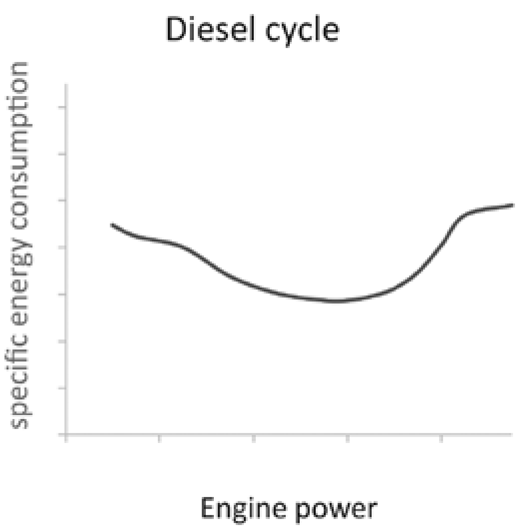

Figure 7: Specific energy consumption for the Diesel Cycle (schematic).

GAS HANDLING

The main difference between the two DF solutions is the fuel-gas pressure, and designing for higher pressures can be considered more challenging. System component costs are at a clearly higher level, more energy is needed to reach the required pressures and maintenance costs will also be higher. For example, cryogenic centrifugal pumps are sufficient for the pressurisation of LNG in the low-pressure concept, but for the high-pressure concept reciprocating (piston) pumps with higher maintenance needs in addition to centrifugal supply pumps are assumed. Looking at the handling of boil-off gas on LNG carriers, several types of gas compressor are available for the low-pressure concept such as centrifugal, screw or piston types. For the high-pressure concept only piston compressors are currently available from a limited number of makers. Moreover, the higher electrical power demand for compressing the fuel-gas to higher pressures is taken into account. Thus, in terms of fuel-gas handling, the low-pressure concept exhibits clear advantages.

CASE STUDY LNG CARRIER

Case description

Given that the major market for two stroke DF engines is the new generation of higher capacity LNG carriers, WinGD executed a case study based on a carrier in the capacity range 170,000 m3 to 180,000 m3, having a twin engine propulsion plant consisting of either two of WinGD's five cylinder, 72 cm bore W5X72DF or two 5 cylinder engines of similar bore diameter with high-pressure gas injection. Each main engine is rated 12,500 kW @ 69.0 rpm. Wärtsilä auxiliary engines and installed gas compression equipment are as follows:

Low-pressure:

– 2 x W-8L34DF + 2x W6L34DF
– 2 x centrifugal compressor
– LNG supply pump.

High-pressure:

– 2 x W-9L34DF + 2x W6L34DF
– 2 x piston compressor
– 1 x HP LNG piston pump
– LNG supply pump.

Vessel keel-laying is considered to be after 2016, but before the introduction of any of the new IMO Tier III ECAs, so that emission control is only required in US waters. LNG is loaded at a terminal in the Gulf of Mexico and needs to be transported via the Panama Canal to Asia. Transit through the Panama Canal is at an average speed of 10.0 kn and normal service speed 19.5 kn. Power demand for part-load operation is estimated by the mathematical power-speed prediction formula with an assumed beta factor (exponent) of 3.5. The same power demand is assumed for both laden and ballast voyages. The contracted maximum continuous rating (CMCR) is assumed to be 2 x 12,500 kW at an engine speed of 69 rpm. The service speed can be attained at a continuous service rating (CSR) power of 90 % of the CMCR. Whenever possible, the engines operate in gas mode. In the United States ECA, the engine with high-pressure gas injection needs to operate an exhaust gas treatment system. Exhaust gas recirculation (EGR) is assumed to be the selected solution. Outside the ECA, HFO is assumed as the pilot-fuel for the engine with high-pressure gas injection engine, while MDO is assumed for the W5X72DF. No switching to HFO is taken into account, as the pilot-fuel quantity is so minor. The fuel prices considered are detailed in TABLE 1. Different ship owners/charterers assume gas prices in their own calculations in different ways. This study uses an average gas price of 450 USD/ton, equivalent to 9.5 USD/mmBTU. This does not consider the recent fuel price crash, but an estimated long-term price level. However, the influence of the LNG price on the final result is minor, as shown in . In laden voyages only NBOG consumption is assumed, but depending on the tank insulation quality, which affects the boil-off rate (BOR), some forced boil-off might be needed in order to reach service speed. The W5X72DF engines are fed by a centrifugal compressor, which is currently the generally accepted and proven solution, although more efficient screw or piston compressors could be applied. The engine with high-pressure gas injection is fed via a high-pressure piston compressor. In ballast voyage additional forced boil-off is considered. For the W5X72DF, LNG is supplied by pumps to the evaporator and then pressurised by the compressor. For engines with high-pressure gas injection, the gas is supplied in two parallel streams with NBOG supplied via the piston compressor, while the LNG for the FBOG supply is pressurised by high-pressure piston pumps before the gas is evaporated in a high-pressure evaporator. This allows electrical power savings for the high-pressure gas injection solution, since the part-load power savings of the compressor are more significant than the additional power demand from the high-pressure LNG pump. The same principle could be applied for the W5X72DF solution. Development projects for such further electrical power saving solutions are currently ongoing. As an electrical base-load for ship and engine room operation 2,000 kW is assumed. Depending on the technology selected, additional electrical power is required to operate the gas compressor and/or the LNG pump, as well as the EGR system for the engine with high-pressure gas injection. The EGR power consumption only takes into account the blowers and no water treatment system, as its power demand is not as significant since clean fuel-gas is burned. During the laden voyage through the Panama Canal, power demand for the gas combustion unit (GCU) is assumed to be due to its own air blower as well as the pressurised gas supply. While gas burned by the GCU is not considered here, it is lower for the low-pressure solution, since at least some gas is utilised by the engines during slow steaming and consequently does not need to be burned by the GCU, while the engine with high-pressure gas injection cannot utilise any gas since a pure diesel operating mode is required.

Table 1: Specific energy consumption for the Diesel Cycle (schematic).

Fuel type	Fuel price		LHV	Max sulphur content
	[USD/ton]	[$/mmBTU]	kJ/kg	[%]
LNG	450	9.5	50,000	0.0%
MGO	850	21.0	42,800	0.1%
MDO	650	16.1	42,707	0.5%
HFO	450	11.7	40,500	3.5%

FUEL CONSUMPTION

In calculating the total fuel consumption of the main engine and gensets per round trip, as shown in , it can be seen that the W5X72DF solution has marginally higher gas consumption compared to an engine with high-pressure gas injection, but lower liquid fuel consumption in about the same marginal range. Comparing the distribution of fuel costs between main engines and gensets ( ) it can be seen that the W5X72DF causes marginally higher fuel costs than an engine with high-pressure gas injection but these are overcompensated by the lower fuel costs for the auxiliary engines, resulting in some, if only minor, savings. The influence of the assumed LNG price in relation to fixed liquid fuel prices can be seen in . As long as the LNG price per energy content is not higher than HFO, the W5X72DF clearly saves fuel costs.

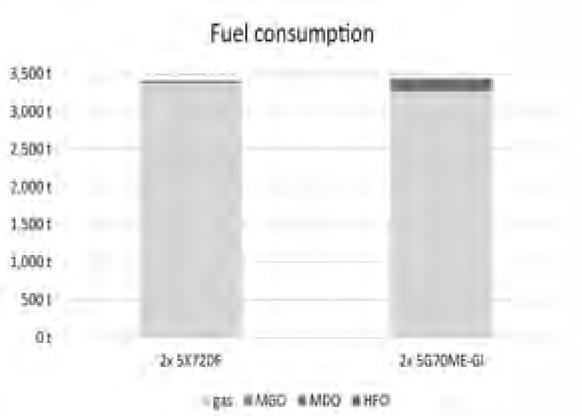

Figure 8: Fuel consumption per round trip, maine engine and generator engines.

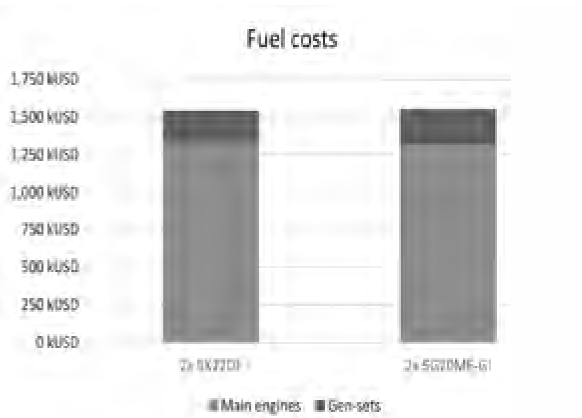

Figure 9: Fuel costs per round trip, maine engine and generator engines.

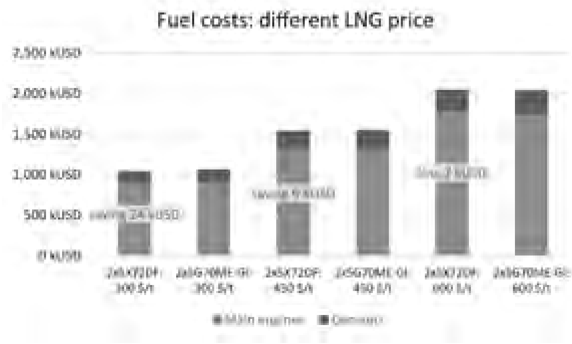

Figure 10: Fuel costs per round trip, maine engine and generator engines, considering fixed liquid fuel prices and variable LNG prices.

INSTALLATION COSTS

The main differences derive from the fuel-gas handling system. It needs to be designed for 16 bar(g) or at least 300 bar(g). In addition, an exhaust gas treatment system is needed for the high-pressure injection solution to comply with IMO Tier III NOX emissions, while the low-pressure solution complies as long as it is operated on gas. Taking the above into account, the same indicative costs are assumed for both main engine solutions. The gensets for the high-pressure gas injection solution are slightly more expensive, since in total two cylinders more need to be installed to cover the higher electrical power demand. The main difference in investment costs is due to the gas system. With the highest cost impact, low or high-pressure compressors are needed. In this study two sets of compressors, one in operation and one on standby are assumed. Mean values from different makers are used for indicative prices. gives an indication regarding the average investment cost saving of applying low-pressure technology.

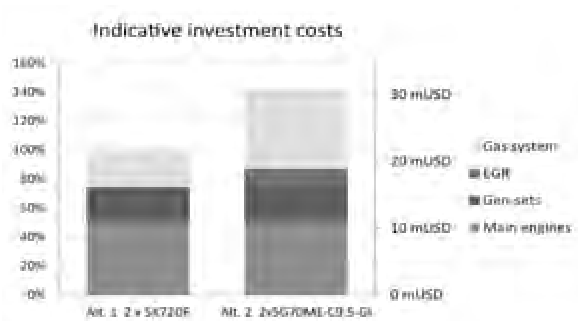

Figure 11: Indicative machinery solution investment costs – LNG carrier.

CONCLUSION

The low-pressure DF solution adopted by Winterthur Gas & Diesel provides benefits in the majority of aspects. It is the most environmentally friendly solution, always complies with IMO Tier III emissions rules when operated on gas, even when outside ECAs. The associated investment costs are lower especially for LNG carriers. The overall fuel costs for operating a vessel are similar for both high- and low-pressure solutions. Using the assumed fuel prices and operating profiles, the low-pressure solution can even result in some savings. An overview of the comparative merits of low- and high-pressure DF technology for two-stroke engines is provided in TABLE 2. It is clear from the weighting of different topics into five different levels, from great benefit to significant drawback must always be somewhat subjective, and different standpoints will result in different results. However, as is the intention of this table, it indicates the trends from which WinGD has made the decision to apply on two-stroke engines DF technology employing low-pressure gas admissions and the Otto Cycle, as used on the Wärtsilä medium-speed engines which established the industry standard technology for four-stroke DF applications.

Table 2: Weighted overview low and high-pressure dual-fuel technology

		Low-pressure technology	High-pressure technology
Environment	GHG	+	++
	SO_x	++	+ (++)[1]
	NO_x	++	✓ (+)[2]
	PM	++	+
Technical aspects	Power output	✓	++
	MN dependency	✓	++
	FGHS selection	++	✓
Financial	OPEX	++[3]	+
	CAPEX	++	-- (-)[4]
Summary	Conclusion	++	✓ (+)[4]

GHG: greenhouse gas, i.e. contribution to global warming
PM: particulate matter
FGHS: Fuel-gas Handling System, i.e. number of available component suppliers
++: large benefit+: benefit
✓: meets requirements
-: drawback
--: significant drawback

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[1] WINTERTHUR GAS & DIESEL: GTD 1, General Technical Data, 2015

[2] MAN DIESEL & TURBO: CEAS Engine Calculations v1.8.54, 2015

1 Depending on sulphur content of pilot-fuel.

2 If NOx treatment system (EGR or SCR) is in operation.

3 If considering: savings for HFO treatment system operation,potential gas system efficiency improvements, e.g. optimised compressor layout, displacement type compressor installation, etc. gas burned by the GCU of a LNGC, lower maintenance costs.

4 Less significant CAPEX disadvantage for LNG fuelled vessels.

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Interaction between the Soil and the Anchor of a Semi-Submersible

ABSTRACT

From an analysis of deepwater platform operating in the Gulf of Mexico areas, the drag anchor and the plate anchor appear to be the preferred solutions to a traditional deep-water mooring in water depth up to 1500m. With advancing technology, the drag and plate anchors can be easily sized and their ultimate holding capacities have increased manifold, up from the old-fashioned methods of using a deadweight anchor. This paper will focus on designing the mooring lines and determining the mooring forces through the use of various software (Excel, Orcaflex, MOSES), followed by soil properties analysis, and lastly the sizing methods of the anchor and its holding capacity.

KEYWORDS

Semi-submersible, drag anchors, plate anchors, ULS, ALS, soil capacity, anchor holding power

1. INTRODUCTION

From the analysis of deepwater platform operating in the Gulf of Mexico areas, the drag anchor and the plate anchor appear to be the preferred solutions to a traditional deep-water mooring in water depth up to 1500m. For the anchors to perform on the seafloor, the seabed soil characteristics need to be studied for a better understanding of the interaction between the two components.

The research conducted in the following sections will cover the analysis of the anchoring system of a semi-submersible and the soil. Researches are presented with a view of introducing the basic knowledge of sizing an anchor for a semi-submersible, using both empirical formulas method and catalogue method.

In the following sections, the environmental conditions in the Gulf of Mexico are first defined, based on DNV guidelines, which include water depth, significant wave height, wave period, wind speed and surface current velocity. Then, a single line mooring analysis for catenary mooring is presented in Microsoft Excel, with a user interface system, to allow the ease of mooring computation.

The ring typed semi-submersible is selected and modelled, using MOSES, by Bentley Systems (2013) and OrcaFlex, by Orcina Ltd. (2015). RAO results from MOSES are computed and imported into OrcaFlex, where it is then used to simulate the motions of the semi-submersible and the mooring lines in the stated environmental conditions. This will give the forces and motions of the semi-submersible and mooring lines, in

which the resulting uplift force from the bottom of the line will be used to calculate the holding capacity power needed for the anchors.

The soil types in the Gulf of Mexico will be discussed and their properties parameters given. Based on the soil types, the anchors chosen to be compared are the deadweight anchor, drag anchor and the plate anchor. A basic method, by Taylor (1982), of sizing a deadweight anchor will be first used to determine base dimensions and weight of the anchor needed. The drag and plate anchors types are then studied and selected to meet the requirements of the semi-submersible and field conditions, to finally determine the holding capacity of each anchor by using a Vryhof anchors (2005) anchor catalogue. The three anchors will be compared to determine the best type of anchor to be recommended for the semi-submersible.

1.1 Aim and Objectives

This report aims to calculate the mooring forces on the semi-submersible and give an analysis of the anchor holding capacity of the semi-submersible on the soil.

The objectives of this report include:

a. Selection of environmental conditions
b. Modelling of the semi-submersible
c. The analytical method of a single mooring line of the semi-submersible for input into OrcaFlex software
d. The analytical method of the anchor holding power capacity

1.2 Review of Literature

The research conducted in the following sections involved the analysis of the anchoring system of a semi-submersible and the soil. Reviews of the literature on selected relevant works intend to introduce the area of interest.

The environmental conditions of the area were taken into account to be used for the seakeeping analysis of the semi-submersible. DetNorskeVertitas AS OS-C 103 (2012) presented a detailed environmental condition of the Gulf of Mexico, with the significant wind, wave, current to be experienced. Wanvik et al. (2001) and Mungall et al. (2004) agreed with the environmental conditions according to reports and findings from the Offshore Technology Conference 2001.

There are different types of semi-submersibles used in the industries. With variations of columns mixed with different shaped pontoons, the designs of semi-submersible are endless. The advantages in the ring type pontoon semi-submersible were recommended by Wanvik et al. (2001), Mungall et al. (2004), Halkyard et al. (2002); with modelling by OrcaFlex, based on demo dimensions provided by Orcina Ltd. (2015).

During operation, spread catenary mooring is normally deployed to keep the semi-submersible in place. The calculations for mooring proposed by Faltinsen (1993), MIT (2011) gave basic calculation methods for the catenary equations. Using DetNorskeVertitas AS OS-E 301 (2013) safety design standards, and materials (chains and wires) compared by Massachusetts Institute of Technology (2011) and Bruijn et al. (2006), the mooring lines specifications can be obtained.

For the design of anchors, standards given by DetNorskeVertitas AS RP-E 301 (2012) and DetNorskeVertitas AS RP-E 302 (2002) provided the recommendation for design of drag and plate anchors holding power. Works by Senol (2012), Sincock and Sondhi (1993) with drag anchors; and James Forrest et al. (1995), Wilde et al. (2001) with plate anchors, displayed the various calculation methods used in the designing and sizing of anchors. Although there are no established standard methods of anchor sizing, a deadweight method could be improvised to design the anchor size.

For the anchor to function effectively in the seabed, the soil types and properties have to be considered. Soil analysis proposed by Taylor (1982) used sediment property estimation method to determine the soil type when a detailed physical survey is not practical. However, Aubeny et al. (2001), DetNorskeVertitas AS RP-E 302 (2002), Al-Khafaji et al. (2003) gave the properties of the soil by conducting geophysical and geotechnical surveys. These provided improved results to the soil conditions data, which can be defined as hard soil, layered soil and soft to medium soil respectively, and their geometrical properties better understood.

From the reviews above, it can be established that a few particular areas of the topic require additional work and research intending to improve the results of the proposed topic.

2. Environmental Conditions

For any design of offshore platforms or floating systems, obtaining the environmental conditions is the foremost step to start the project. Using DetNorskeVertitas AS OS-E301 (2013), a summary of the 100-years return period for the Central Gulf of Mexico are presented in Table 1.

Table 1 – Environmental conditions of GOM

Conditions	Values
Water Depth	500 m
Wave Height	15.8m
Period	13.9 s
Wind Speed (1 hr @ 10m)	48 m/s
Surface Current	2.4 m/s

According to Le Méhauté (1976), the graph for the validity of several theories for periodic waves can be used to obtain the deepwater relationship. From this graph, the linear wave theory (airy) can be adopted for all the relevant force calculations and simulations.

3. EXCEL PROGRAMME OF MOORING LINES

To calculate the mooring forces, basic details of the mooring lines have to be established. Based on Faltinsen (1993), the mooring equations were given, with the main equations being listed below in Equations (1), (2) and (3).

The minimum line length required: ℓ_{min}:

$$\ell_{min} = h(\frac{2T_{max}}{wh} - 1)^{\frac{1}{2}} \quad (1)$$

Horizontal length of the mooring line X:

$$X = \ell - h\left(1 + 2\frac{T_H}{wh}\right)^{\frac{1}{2}} + \frac{T_H}{w} cosh^{-1}\left(1 + \frac{wh}{T_H}\right) \quad (2)$$

Maximum tension in the line T_{max}:

$$T_{max} = T_H + wh \quad (3)$$

h – Depth of the water (m)
ℓ – Length of mooring line (m)
TH – Horizontal tension (kN)
w – Weight of the material per meter (N/m)

An Excel programme was created using the Excel data validation codes to produce a user interface system to allow for the ease of calculation of the mooring forces. Table 2 shows the user interface system. The mooring programme provided swift, accurate solutions to allow several mooring formulas to be computed with the input of the basic values (such as depth of water, mooring line length, etc.).

Table 2 – Excel mooring programme

INPUT BY USER

Mooring Components	Values
Depth of water (m)	500
Mooring Line Length (m)	1500
Types of mooring lines used	Chain
Lines Diameter (mm)	Chain Wire Polyster Dyneema
Weight (kg/m)	
Weight (N/m)	[illegible]
Pre-tension + Environmental Force (kN)	1500
Safety Factor	1.7
Minimum Breaking Load (kN)	7596

OUTPUT CALCULATIONS

Mooring Components	Values
Max Tension	2574.195
Max Tension with Safety Factor	4376.1315
Minimum Line Length (m)	1336.763006
Horizontal Force TH (kN)	1500
Horizontal Distance x of Minimum Line (m)	980.8095287
Vertical Force TZ (kN)	2871.888273
Total Horizontal Footprint X (from topview) (m)	1144.046523
Length of Mooring on Seabed (m)	163.2369945
Will mooring line fail?	NO

The mooring programme required the user to know the environmental conditions and the properties of the mooring line, to be used. The safety factor should be based on references from DNV, API, etc. In this paper, the partial safety factor was established on DetNorskeVertitas AS OS-E 301 (2013), consequence class: class 2, where mooring failure is likely to result in unacceptable consequences.

4. SEMI-SUBMERSIBLE CONFIGURATION

The use of OrcaFlex (Orcina Ltd., 2015) software allowed the analysis of the semi-submersible and it's anchoring system to be done efficiently and accurately. Based on demo dimensions given by OrcaFlex, the model and its environment were designed and scaled to the preferred dimensions. With results obtained from the Excel mooring programme, the required mooring lines lengths were then input into the model.

The mooring line configurations were established based on a spread 12 lines mooring catenary pattern at each of the four corners of the semi-submersible. Two different configurations were proposed – 45° gap between lines, Figure 1, and 30° gap between lines, Figure 2. Basic static analysis demonstrated that the 30° gap resulted in smaller forces on the mooring lines. Thus the 30° gap configuration would be used for latter simulations.

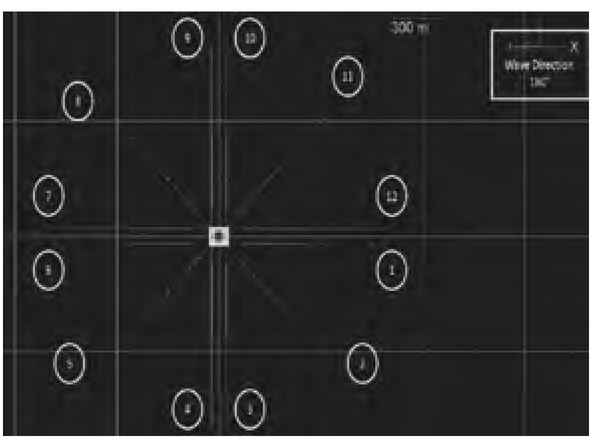

Figure 1 – 45° gap between the mooring lines

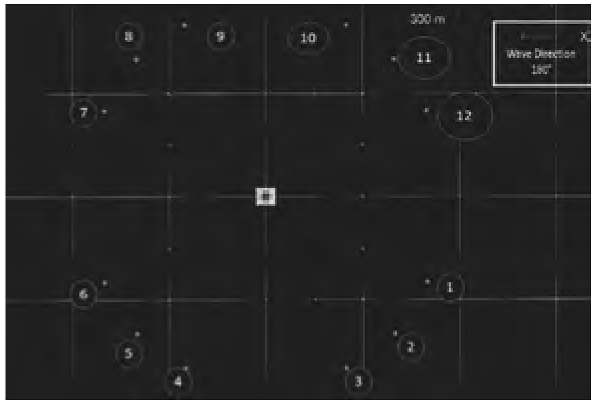

Figure 2 - 30° gap between the mooring lines

With a depth of 500m, 12 single line mooring made entirely of chains were not feasible, due to cost and weight factors. Therefore, a combination of chain-wire-chain was proposed, shown in Table 3, which in turn reduced the cost of the anchoring system, as well as reducing the overall weight of each mooring line. This allowed the semi-submersible to have higher payload i.e.; more economical benefits.

Table 3 – Chain-wire-chain combination and length

Sections: 3 Total length = 1300.000m

No.	Line Type	Section Length (m)
1	Chain	100.000
2	Wire	600.000
3	Chain	600.000

MOSES (Bentley Systems, 2013) software must be used to model the semi-submersible, Figure 3, and run the simulation using "3D DIFF" theory in the frequency domain, to obtain the RAO of the semi-submersible. The displacements RAOs output file could then be formatted and imported into OrcaFlex, with the heave and surge RAO showed in Figure 4 and 5. It was done so because even though OrcaFlex was able to model the semi-submersible, it was unable to produce any initial RAOs readings, thus the need to use MOSES for the generation of the RAO.

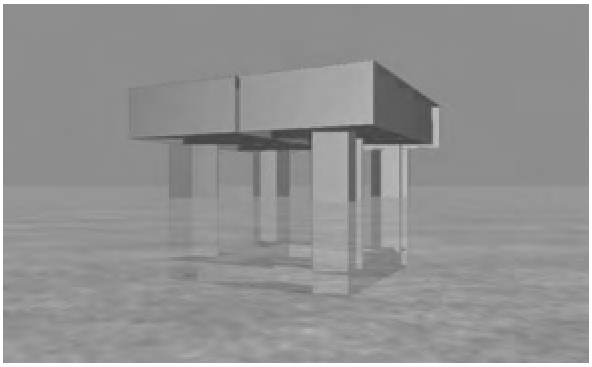

Figure 3 – MOSES model of the semi-submersible

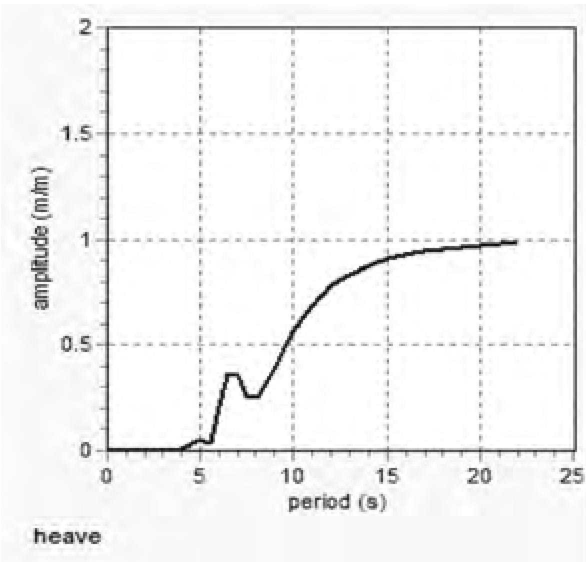

Figure 4 – 180° Heave RAO of the semi-submersible

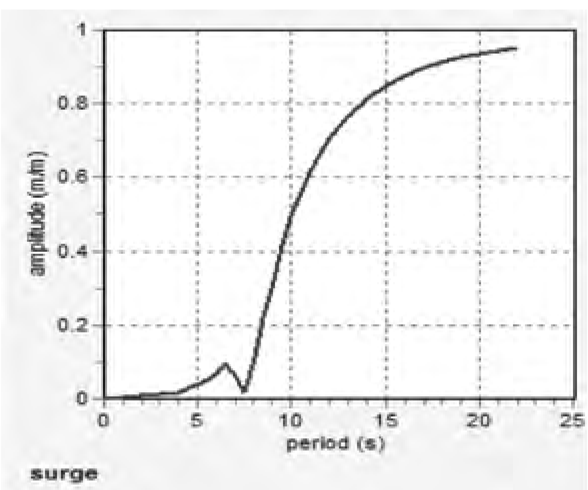

Figure 5 – 180° Surge RAO of the semi-submersible

5. SIMULATION BY ORCAFLEX

In OrcaFlex, the semi-submersible model was run in static analysis and then followed by dynamic simulations. The dynamic approach was taken concerning "Chapter 2 Section 2 – 2.2.11" of DetNorskeVeritas AS OS-E 301 (2013) for the analysis of mooring system behaviour. A time frame simulation (-5 to 0 seconds) was given for the static analysis of the entire semi-submersible and mooring system, and (0 to 50 seconds) for the dynamic analysis to allow the simulation to stabilise into the equilibrium state.

In this report, the Ultimate Limit State (ULS) and Accidental Limit State (ALS) were taken into consideration for the calculation of the forces on the mooring lines. Consequence Class 2 was chosen, where mooring line failure would result in unacceptable failure such as loss of life, capsize or sinking. Consequence Class 2 was selected because it gave the highest partial safety factor, to ensure the safety and integrity of the design semi-submersible and mooring system.

The semi-submersible was subjected to environmental loadings at head seas 180° and quartering sea 135°, with the shaded view given in Figure 12. Forces at beam seas 90° would not be given due to the ring-shaped nature of the semi-submersible, which results in similar forces as head seas 180°.

5.1 Head Seas (180°) ULS

In the Ultimate Limit State, the simulation was done based on a ring-typed semi-submersible, with all 12 mooring lines tensioned and anchored to the seabed.

In head seas (180°) ULS, the mooring lines "Line 1" and "Line 12" as shown in Figure 2, experienced the highest tension as the semi-submersible undergoes in the head seas forces, which resulted in the semi-submersible to drift to the left with the wave direction.

The max tension of the mooring line "Line 1" at the semi-submersible End A was computed to be around 1750kN, shown in Figure 6. It illustrates that the mooring Excel calculation done in section 3 was a good rough estimate of the tension of each line. The under/over-prediction of the tension of each line may be due to the possibility that the Excel calculation took only one type of mooring material while OrcaFlex took the chain-wire-chain combination into account.

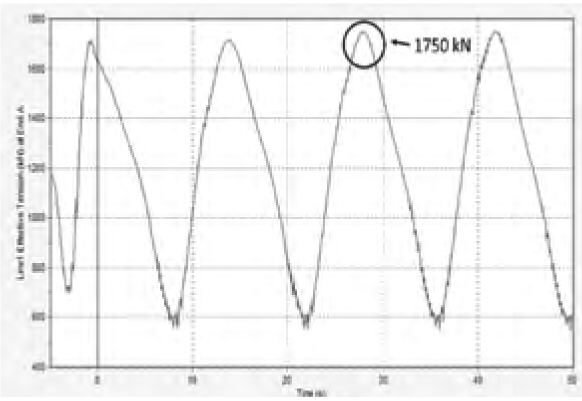

Figure 6 – Line 1 End A tension at ULS in head seas (180°)

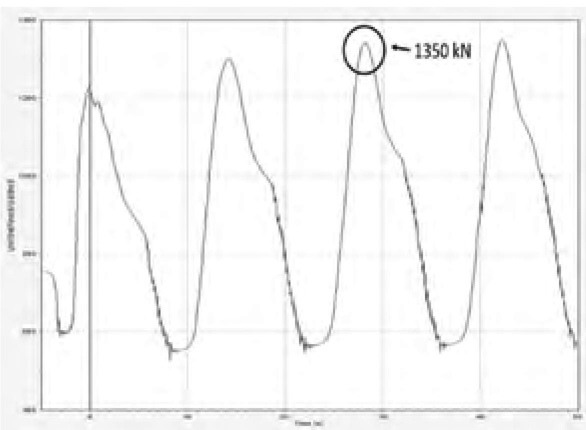

Figure 7 - Line 1 End B force at ULS in head seas (180°)

The End B force at the anchor point, where the line is connected to the anchor, was computed to be around 1350kN, Figure 7. The forces were noticeably smaller due to the following reasons:

1) The wave forces at the bottom are almost non-existent (deep-water relations).
2) Forces exerting the anchor are largely horizontal force; vertical force is at minimal with the weight of the chain at the catenary end keeping the mooring line on the seabed.

These two reasons could also be used to account for the End B forces in the remaining experiments for 135° ULS, and 135° and 180° ALS.

The zig-zag-ing in the line tensions/ forces may be due to the chains moving within their gap and the interaction between the seabed.

5.2 Quartering Seas (135°) ULS

At quartering seas (135°) ULS, the semi-submersible experienced the environmental forces coming at 135°. It caused tension on the lines to be dependent mainly on line 1, 2 and 3, with the highest tension computed on "Line 2" at about 2700kN, shown in Figure 8 and 9.

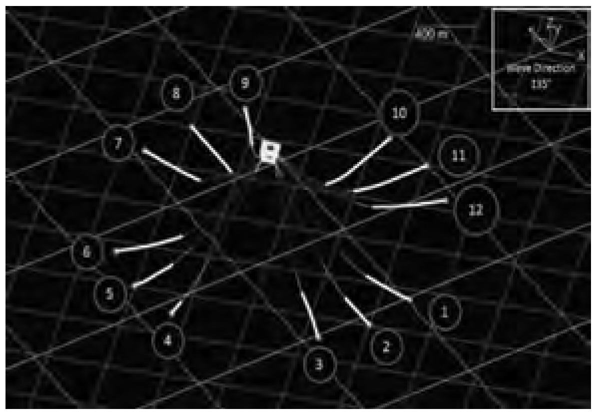

Figure 8 - ULS mooring configuration in quartering seas (135°)

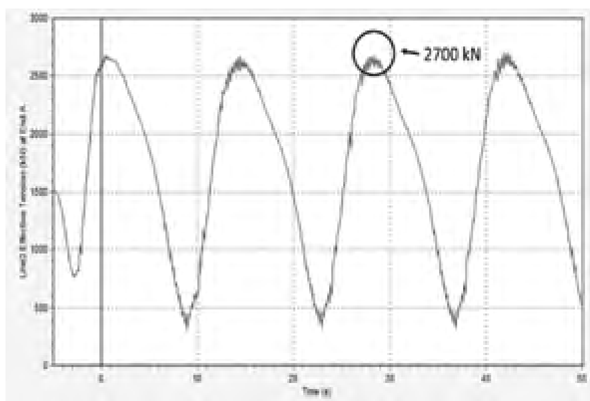

Figure 9 - Line 2 End A tension at ULS quartering seas (135°)

Comparing both the ULS 135° and 180° simulations, the highest tension computed was on "Line 2" at ULS 135°. DetNorskeVertitas AS

OS-E 301 (2013) ULS Partial Safety Factor of 1.7 for quasi-static analysis was used to find the utilisation factor:

$$u = \frac{T_{c-mean}\,\gamma_{mean} + T_{c-dyn}\,\gamma_{dyn}}{S_c} \quad where\ u\ \leq 1 \qquad (4)$$

u – Utilisation factor

$T_{c\text{-mean}}$ – Mean tension (kN)

γ_{mean} – Partial safety factor on the mean tension

$T_{c\text{-dyn}}$ – Dynamic tension (kN)

γ_{dyn} – Partial safety factor on dynamic tension

The utilisation factor "u" was found to be 0.7203, proving that the designed mooring properties were suitable to the environment.

5.3 Head Seas (180°) ALS

In the Accidental Limit State, the simulation was based on a single line failure, to test the integrity of the mooring system. It was decided that the line with the highest tension should fail while the 11 remaining lines withstand the environmental loadings and remain in position.

At head seas (180°) ALS, "Line 1" would be released from the semi-submersible at the start of the simulation to mimic an emergency disconnection system in the case of breakage of a mooring line. The line was designed shorter and remain anchored to the seabed so as to prevent line clashing (the possibility of the snapped line getting in contact with another line or object), which would result in an unstable simulation and the eventual crashing of the semi-submersible.

The results were not surprising, with "Line 2" showing the highest tension at 1900kN, Figure 10, as the environmental induced forces were distributed among the remaining 11 lines. The figure also shows a comparison of the forces on the failed "Line 1" on the seabed, which was almost negligible.

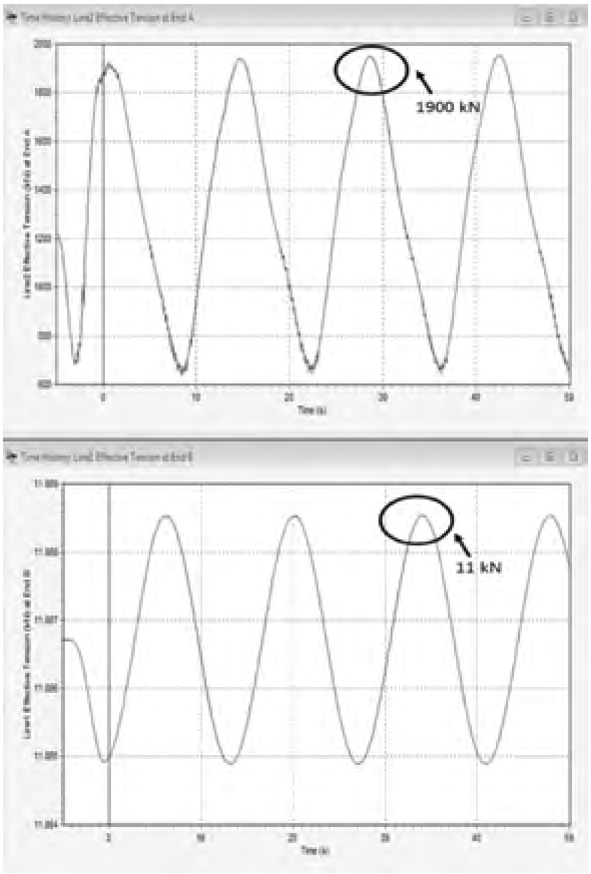

Figure 10 –Comparison between Line 2 End A tension and Line 1 End B force at ALS in head seas (180°)

5.4 Quartering Seas (135°) ALS

At quartering seas (135°) ALS, "Line 2" would be released. As proven in subsection 5.2, the semi-submersible was largely dependent on three (3) main lines for stationkeeping, thus removing one line would result in a significant increase in the tension experienced by the other two lines. As expected, the highest tension was on "Line 3" End A at 3700kN, Figure 11, a substantial jump from previous numbers. The End B tension was given to be 3200kN.

The tension range in "Line 3" was also the highest experienced so far in all simulations, suggesting that fatigue stress might be a long-term issue, but this will not be discussed as it is not within the scope of this article.

Again, using DetNorskeVeritas ALS Partial Safety Factor of 1.1, the utilisation factor "u" was found to be 0.533, attesting to the reliability of the mooring system.

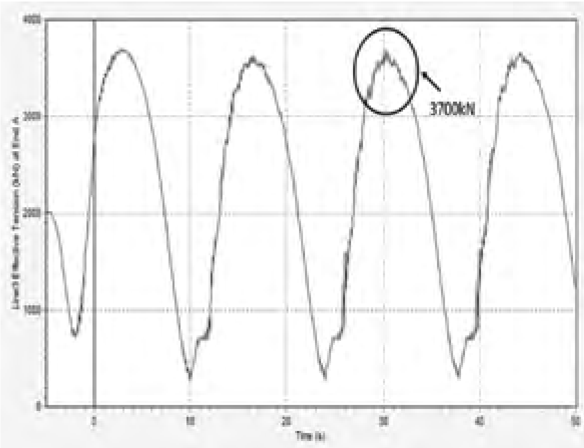

Figure 11 - Line 3 End A tension at ALS quartering seas (135°)

6. SOIL CAPACITY AND ANCHOR HOLDING POWER

Before the development of high holding power anchors, basic soil mechanics empirical formulas were used to calculate the ultimate holding capacity of an anchor. However, it is difficult to predict the exact holding power, as it can only be established after deployment in the field using specialised sophisticated anchor data software. Therefore, to start off the sizing of an anchor, knowing the properties of the soil are the first step to the prediction of the anchor dimensions and weight needed. It allows the selection of the type of anchors that will be used. The subsequent step is to use the empirical formulas, together with the maximum pulling force of the mooring line and the soil properties, to determine the anchor holding power.

6.1 Soil Parameters

Firstly, the main properties of the soil to be determined were the types of soil, undrained soil shear strength (Su), unconfined unconsolidated tests (UU), and clay sensitivity (St). On-site assessments of the Standard Penetration Test (SPT), or Cone Penetrometer Test (CPT), provided estimations of the values that could be compared with laboratory test results.

Studies by Z.A. Al-Khafaji et al. (2003) suggested that the soil in the Gulf of Mexico consisted of mainly very soft to stiff clay. By substituting these clay type properties into empirical formulas, it showed that the geotechnical properties of the clay would affect directly the anchor to be used and its holding power. Using catalogue from Vryhof Anchors (2005), it could be concluded that the properties and parameters of the clay to be based on American Society for Testing and Materials (ASTM) results, which accounted for the undrained shear strength of the clay-typed soil in the Gulf of Mexico to be around 35 kPa, based on Table 4 and 5, which was the mean average shear strength of the very soft to stiff clay type.

Table 4 – American (ASTM) and British (BS) standards for undrained shear strength (Vryhof Anchors, 2005)

Undrained Shear Strength (kPa)		
Consistency of Clay	ASTM D-2488	BS CP-2004
Very soft	0 - 13	0 - 20
Soft	13 - 25	20 - 40
Firm	25 - 50	40 - 75
Stiff	50 - 100	75 - 150
Very stiff	100 - 200	150 - 300
Hard	200 - 400	300 - 600
Very hard	>400	>600

Table 5 – Relation between shear strength and test values (Vryhof Anchors, 2005)

S_u kPa	UU kPa	SPT N	CPT MPa
0 - 13	0 - 25	0 - 2	0.0 - 0.2
13 - 25	25 - 50	2 - 4	0.2 - 0.4
25 - 50	50 - 100	4 - 8	0.4 - 0.7
50 - 100	100 - 200	6 - 15	0.7 - 1.5
100 - 200	200 - 400	15 - 30	1.5 - 3.0
>200	>400	>-30	>3.0

These parameters could then be input into OrcaFlex environmental data for seabed soil properties. It would, in turn, allow the generation of the End B forces on the mooring line and anchor, to determine the anchor holding power needed.

6.2 Anchor Holding Power

After getting the information on the end forces at the seabed, sizing the anchor could be done by empirical formulas. Because there is currently no established approach of sizing anchors, this paper used a simplified method deduced by Taylor (1982) to size a deadweight anchor, followed by using a Vryhof anchors (2005) catalogue to size a drag/ plate anchor (Refer to drawings in Appendix).

DEADWEIGHT ANCHOR

Applying empirical formulas presented by Taylor (1982), design procedures for deadweight anchors on a cohesive (clay) seafloor, the submerged weight of the anchor could be derived. General parameters such as horizontal and vertical loads and average soil undrained shear strength, Equation (5), were pre-requisites information needed to derive the anchor holding power "R", Equation (6). Because the two main equations (5) and (6) were derived in imperial units (as done by Taylor (1982)), further conversions had to be done to convert the resulting answers to metric units.

Average Soil Strength S_{ua}

$$Sua = 0.5 * (Su + Suz) \qquad (5)$$

Anchor lateral load resistance R

$$R = B^2 * (S_{uz} + 0.2\, S_{ua}) \qquad (6)$$

S_u – Undrained Shear Strength at depth zero (psi)
S_{uz} – Undrained Shear Strength at penetrated depth (psi)
S_{ua} – Average Soil Strength (psi)
B – Anchor breadth (inches)

Although it was fairly accurate on obtaining the weight of the anchor needed, other factors such as scouring, seafloor slope, and cyclic loading were not taken into consideration in this analysis, which was the limitations of the formulas. Through a trial and error process, the final required submerged weight of the anchor equated to almost 25 tonne based on the highest End B force (3200kN) at quartering seas (135°) ALS.

DRAG ANCHOR

The drag anchor was initially chosen to be the main anchor because:

1) It is suitable for use in the soft to hard clay
2) Wide range of size and capacity
3) Can provide continuous resistance even if maximum capacity is exceeded
4) Low resistance to vertical loading, but not a problem with catenary mooring

To accurately acquire the weight of the anchors, catalogue from Vryhof Anchors (2005) was used to give a prediction of the anchor holding power. Stevpris MK5 anchor was selected, with the Ultimate Holding Capacity UHC, formula (8), calculated to be around 350t at 7t anchor weight.

$$UHC = A * W^{0.92} \qquad (7)$$

A – Parameter depending on soil, anchor and anchor lines
W – Weight of anchor (tonnes)

It showed that the deadweight formulas gave an over-estimation, nearly 3.5 times the anchor weight needed to achieve the holding capacity, Table 6. The reasons for this might be due to the design of the drag anchor, which provides added efficiency to the addition of the UHC.

PLATE ANCHOR

The plate anchor was chosen as an alternative as an anchor for the MODU for these reasons:

1) Has higher holding capacity-to-weight ratio than other types of anchors
2) Accurate placement of the anchor on the seabed

The catalogue was again used to give the area required to meet the Ultimate Pull-out Capacity UPC of the selected Stevmanta Vertical Loaded Anchor (VLA). The UPC area was found to be about

8 m2, as given by Equation (8), much smaller than the area needed for the empirical formula.

$$UPC = N_c * S_u * A \tag{8}$$

Nc – Bearing Capacity Factor (Around 7 to 10)
Su – Undrained Shear Strength of clay (kPa)
A – Stevmanta fluke area (m2)

Table 6 – Comparison between calculated and catalogue results of anchors

Anchor Types	Calculation Methods	Forces to meet	Required Weight/ Area
Deadweight Anchor	Empirical Formulas	3200 kN	25 tonne
Drag Anchor	Vryhof Catalogue		7 tonne
Plate Anchor	Vryhof Catalogue		8 m2

These formulas showed that sizing an anchor for the use of semi-submersible would be very simple and straightforward. It also showed that for the mooring system of a semi-submersible, the drag anchor was one of the most effective anchors to be used on its simplicity of sizing and weight-to-power ratio.

However, there were more design considerations, like the fluke angle, mooring lines used, and penetration depth in the soil, to be factored in to ensure the precise and accurate holding power of the anchor. Nonetheless, this paper demonstrated a basic foundation of designing the anchors.

7. DISCUSSION AND CONCLUSION

The aim stated at the start of this paper is met by using the various methods to obtain the various objectives, which is to analyse the forces of the mooring lines of a semi-submersible, and to size an anchor to meet the anchor holding capacity on the soil.

Rules provided by DetNorskeVertitas made up the bulk in this paper, as DNV's criteria are simple to understand and apply.

MOSES and OrcaFlex are two of the major software used for the simulations of the semi-submersible. By exploring and learning OrcaFlex though online papers/ instructions, the basic of OrcaFlex can be understood. Through the use of OrcaFlex, the process of model designing, the different components that contributed to the forces, and the calculations of the mooring line length which affected the forces, are fundamentally grasped and the simulations generated to the best possible results.

The simulation results for 0°, 45° and 90° are not given as the semi-submersible is a ring-typed semi, which means the forces will be similar for perpendicular angles. It is only applicable to a ring-typed semi-submersible, thus, if other types of semi-submersibles (twins-hull, round-typed) are used, tests for 0° to 180° have to be performed for more accurate results.

Finally, the quality and results of this paper are produced to the best standards given the time and resources. There are desires to do more advanced simulations to illustrate more analysis results of the mooring system, but given limited time, these shall be discussed in the next section.

8. RECOMMENDATIONS FOR FUTURE WORKS

From what was given in this thesis, steps for the foundation of sizing up an anchor were produced. The ULS and ALS for both 135° and 180° calculated by OrcaFlex were fairly accurate and assisted in the design of the mooring system by finding the forces on the anchor end. But to survive long-term environmental loadings, the Fatigue Limit State (FLS) have to be taken into account to ensure structural reliability.

The calculations of the FLS can be a major topic on its own due to the complexity of the works that needs to be performed. FLS includes, but not limited to, single/ multi-lines fatigue analysis, as well as the effects of Vortex Induced Vibrations (VIV) and Cyclic Stress Loadings on moorings.

Another area of studies that could be done is experimenting with different software to simulate the handling of the anchors. Besides OrcaFlex, specialised software like "GMOOR" from Global Maritime, and "Stevtrack Anchor Data Acquisition System" from Vryhof Anchors are all powerful programs for anchor simulations and calculations. Aspects like the touchdown force, embedment force, and the proof-loading force are a few objectives that can be taken into consideration.

Abbreviations

3D DIFF	3D Diffraction
ALS	Accidental Limit State
End A	The connection point of the mooring line to the semi-submersible
End B	The connection point of the mooring line to the anchor
MODU	Mobile Offshore Drilling Unit
RAO	Response Amplitude Operator
UHC	Ultimate holding capacity of drag anchor
ULS	Ultimate Limit State
UPC	Ultimate pulling capacity of plate anchor

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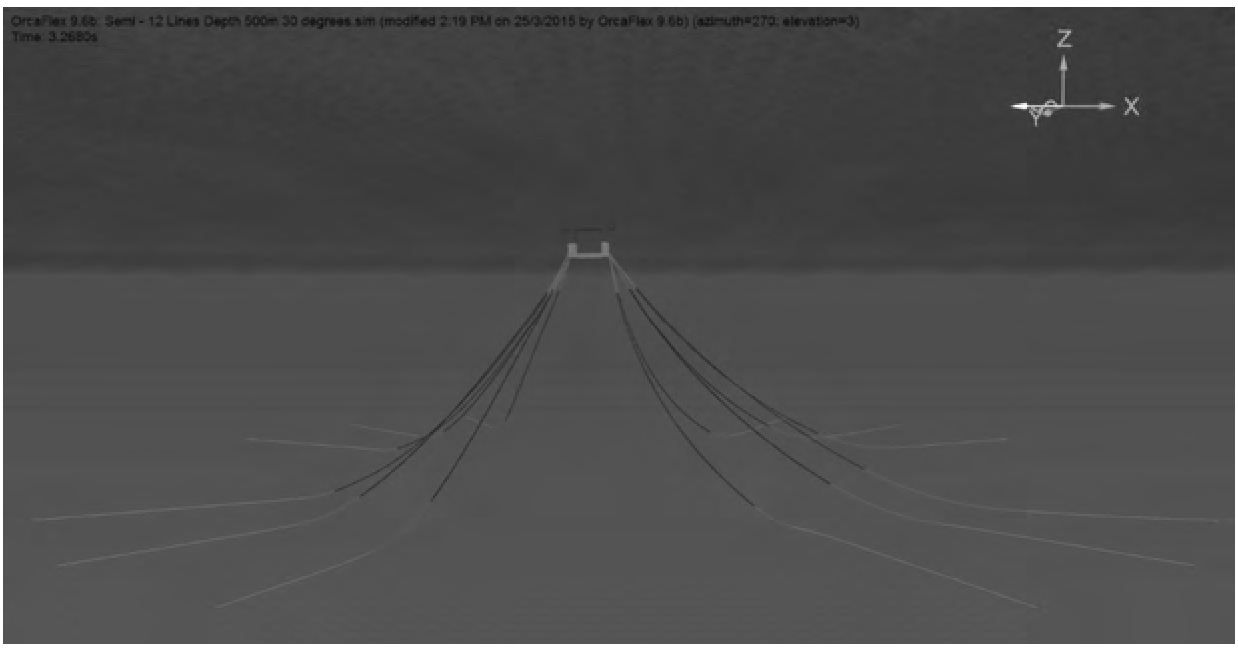

Figure 12 – Full View of Semi-Submersible with mooring lines in Head Seas (180°) ULS

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Improving the Propulsion Efficiency by means of Unconventional Tip Propellers

ABSTRACT

A large number of Propulsion Improvement Devices have been developed in the past and are routinely “reinvented” whenever the price of fuel increases or the freight rates are low, as it is now the case. On the contrary propellers have undergone very few modifications.

Only tip propellers, most noticeably of CLT type, have been installed in relatively high number at full scale, showing their superior characteristics when compared to conventional propellers.

Despite this fact, tip propellers are still unbeknown to most Ship Owners, Technical Managers and Shipyards. In addition their use has been curtailed by the need of different treatment at model scale when compared with conventional propellers.

The goal of the present paper is to offer an overview of 40 years of development of tip plate propellers and to present the latest experiences and R&D projects in respect to CLT propellers.

INTRODUCTION

In the current market conditions fuel efficiency is of paramount importance, despite fuel being extremely cheap in comparison to the past. This is in addition to more general considerations in respect to pollution prevention, Energy Efficiency Design Index and Operational Indicator (EEDI and EEOI) and Ship Energy Efficiency Management Plan (SEEMP).

Historically much has been achieved in reducing the advance resistance of the ship hull, naked and appended. In addition many propulsion improving devices (PIDs) have been invented, later abandoned and then reinvented and reintroduced. Nowadays the PID portfolio spans over pre-swirlers, swirl recoverers, ducts, hull fins, rudder fins, bulbed or twisted rudders, hub caps... either alone or combined one with the other.

On the contrary propellers have gone through very little innovation: little is worth mentioning apart from the introduction of high skew, a continuous improvement of the annular profiles and the use of unconventional tip shape, with the purpose of increasing the propulsive effiency and achieving a good cavitation behaviour.

In principle the careful optimization of the hull (main dimensions, bow and stern shape), followed by the selection of a tip plate propeller and of a bulbed and twisted rudder will result in a high propulsion efficiency (hence low EEDI and EEOI).

If the above is performed effectively, the use of further PIDS (e.g. pre or post stators) is likely to bring only marginal gains. The exception are vessels with unfavourable main dimensions and non-optimized hulls.

It should be noted that propellers in general, and tip plate propellers in particular, are compatible with any kind of PID, while PIDS are not always compatible with one another.

In any case the design of the propeller and of the selected PID must be integrated; off-the-shelf designs will not be compatible with one another and should be best avoided.

For the same reason, in principle, to retrofit a PID on an existing vessel to increase the propulsion efficiency is bound to modify the wake on the propeller disk and the power absorption characteristics of the propeller. The net result is that part of the gains are curtailed by the fact that the original propeller will work in an off-design conditions. Therefore, it is better to retrofit a high efficiency propeller, of tip plate type, and, eventually, retrofit a PID at the same time, rather than to retrofit a PID only and keeping the existing propeller.

GENERAL OVERVIEW OF UNCONVENTIONAL TIP PROPELLERS

The first claims about the potential advantages of tip loaded propellers (Tip Vortex Free propellers, TVF propellers) were published in October 1976 in "Ingeniería Naval" by Prof. G. P. Gomez, later contributions by Klaren, Sparemberg, Anders, De Jong, Kappel... followed.

Since then the tip plate propeller concept has evolved continuously and nowadays, unbeknown to most, they are the dominant choice in respect with ship propulsion.

Currently two different types of tip propellers are the most extended for ship propulsion: the Kappel propeller and the CLT propeller. It should be noted that Tip Raked and Wide Chord Tip (WCT) propellers, despite having a modified tip geometry, can be regarded as conventional propellers.

There are notable difference between the design principles of Kappel and CLT propellers, however one underlying goal is common:

To inhibit the natural unloading of a wing (or propeller blade), which is due to the "short circuit" of the pressure and suction side across the wing tip, by resorting to non-planar lifting surfaces.

In the case of conventional propellers, the high pressure and the low pressure side communicate across the propeller tip, as a consequence the differential pressure at both sides of the tip is null, therefore the outer annular sections do not contribute significantly to the total thrust of the propeller. In order to generate thrust also in way of the propeller tip the Kappel propeller design minimizes the flow over the tip while in the CLT propeller design the tip plates act as barriers avoiding water flow over the propeller tip.

The readers are already familiar with the application of this principle in civil aviation: winglets are fitted at the wing tips to provide for the generation of additional lift.

Figure 1 The principle of generating additional Lift by means of winglets

It should be noted that propeller featuring advanced section profiles or innovative chord radial distribution are currently available for design and installation on commercial ships, however these propellers are by all means conventional propellers and do not allow for the jump in propulsion efficiency allowed for by unconventional tip plate propellers.

KAPPEL PROPELLERS

Very little literature is available on the subject of Kappel propellers.

Kappel propellers were developed by Prof. J. J. Kappel starting from early eighties.

Prof. J. J. Kappel design company was incorporated into MAN Diesel & Turbo in 2012.

According to the Kappel design philosophy the propeller tip is bent forward, and it belongs to the suction side of the propeller. The transition between the blade and the tip is smooth, the tip is hydro-dynamically loaded and it is designed so as to contribute actively to the generation of lift and thrust.

The first Kappel propeller was fitted in 2005 to M/V Nordamerika, Dampskibsselskabet Norden A/S, Copenhagen.

The gain in efficiency of Kappel propellers in respect to an equivalent state-of-the-art conventional propeller is reported as ranging from 3 to 5%.

The actual number of full scale installations of Kappel propellers to date is not known to the authors.

CLT PROPELLERS

Extensive literature has been published on the subject of Contracted and Loaded Tip propellers (CLT Propellers).

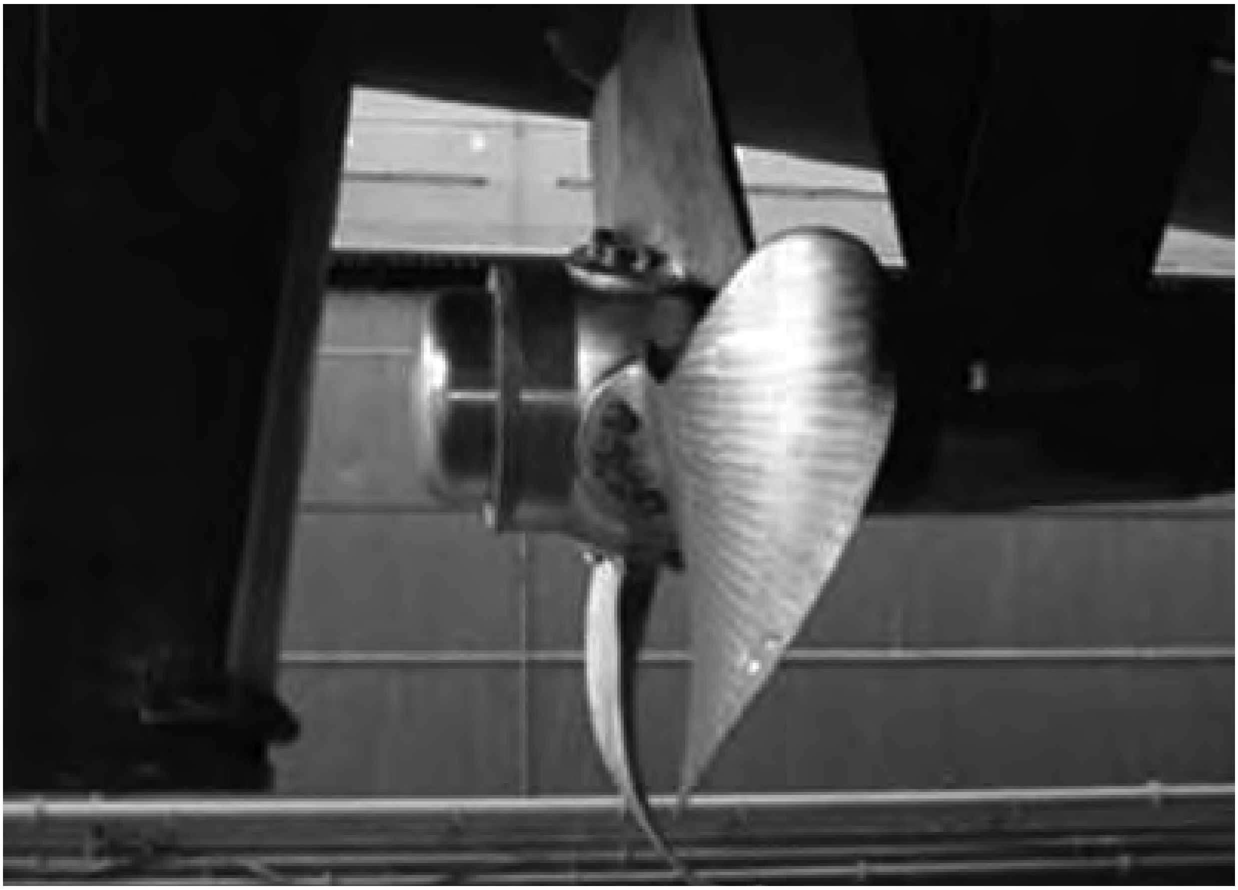

Figure 2 A CP Kappel propeller, note the propeller tip curled toward the suctions side.

CLT Propellers were developed from the early 80's and they are the direct development the Tip Vortex Free Propellers (TVF propellers), developed in late 70's, by adapting the geometry of the tip plate to the contraction of the fluid vein crossing the propeller disk.

Subsequently the design company SISTEMAR was established with the purpose of designing and further developing CLT propellers.

CLT propellers are characterized by the following:

- The tip chord is finite.
- An end plate is fitted at the blade tip, located on the pressure side.
- The blade tip bears a substantial load.
- The end plate is virtually unloaded.
- Low to moderate skew.

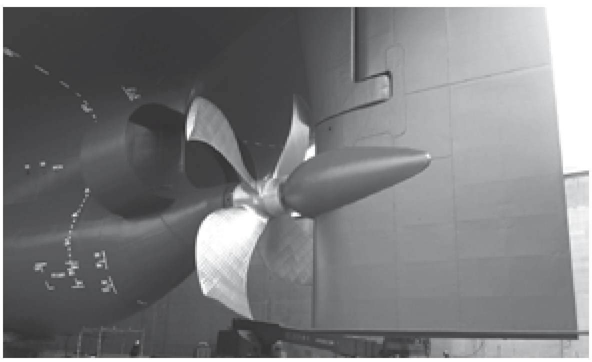

Figure 3 A FP CLT Propeller installed to 120,000 DWT bulker, with WED and bulbed rudder.

CLT propellers are characterized by the following:

- The tip chord is finite.
- An end plate is fitted at the blade tip, located on the pressure side.
- The blade tip bears a substantial load.
- The end plate is virtually unloaded.
- Low to moderate skew.

It should be underlined that in the CLT propeller, contrary to Kappel propellers, the end plates are hydrodynamically unloaded and that they operate as barriers so as to avoid the communication of water between the pressure and the suction side of the blades, allowing to establish finite load at the tip of the blade.

The gain in efficiency of CLT propellers in respect to an equivalent state-of-the-art conventional propeller ranges from 4 to 8%.

Up to date CLT propellers, both of fix and controllable pitch type, have been successfully installed on more than 300 vessels, of very different types:

Tankers, Product carriers, Chemical carriers, Bulk carriers, Cement carriers, General cargoes, Container ships, Reefers, Ro–Ro, Ro–Pax, Fishing vessels, Trawlers, Catamarans, Hydrofoils, Patrol boat, Landing crafts, Oceanographic ships, Yachts.

The application range has been extremely wide:

- Up to 300,000 DWT
- Up to 22 MW per propeller
- Up to 36 knots.

THEORETICAL CONSIDERATIONS

Considering for a moment the propeller design theories we shall recall the following facts:

- the larger the gradients of the spanwise distribution of circulation
- the larger the induced velocities
- the higher the hydrodynamic pitch angle
- the greater the angle formed by lift and thrust vectors
- the lower the "useful" quota of the lift
- the lower propeller open water efficiency

Let us define the parameter ε as the ratio between the suction in front of and the pressure jump across the propeller disk (Δp). In other words ε defines how the propeller thrust is obtained by combining the under-pressure existing at the suction side of the propeller blades:

$(p_o - \varepsilon \Delta p)$

with the over-pressure existing at the pressure side of the propeller blades:

$(p_o + (1-\varepsilon)\, \Delta p)$

As recalled above, according to the lifting line theory, by reducing the gradient of the spanwise distribution of circulation the induced velocities are also reduced and, consequently, according to Bernoulli, the suction at the propeller disk is also decreased, which means to decrease ε.

According to the Classic Momentum Theory the ideal propeller open water efficiency can be expressed as a function of the non dimensional propeller specific load CTH:

η0 = 2 / (1 + (1 + CTH)0.5)

Where:

CTH = T / (0.5 ρ A V2)

Such formulation, however, is not valid for non conventional tip plate propellers since the influence of the parameter ε is not taken into account. According to the New Momentum Theory, presented to SNAME, New York, in 1993, the ideal efficiency of a propeller is as follows:

η0 = 1 / (1 + ε CTH)0.5

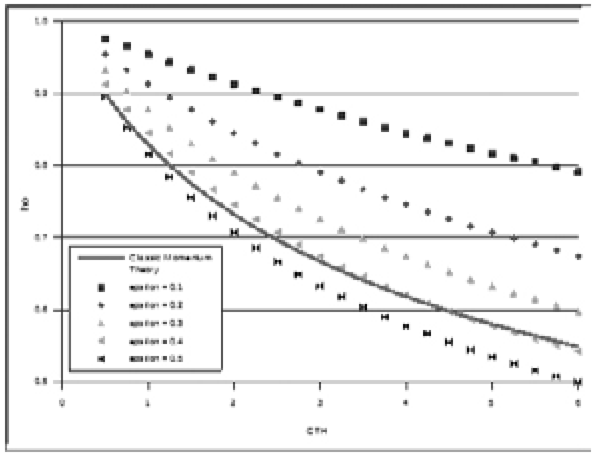

Figure 4. Ideal efficiency according to either the classic and the new momentum theory.

In Figure 4 both formulations of the open water efficiency are plotted; the following comments can be made:

According to the New Momentum Theory η0 increases when ε decreases;

The New Momentum Theory allows for greater ideal efficiency than Classic Momentum Theory in case of ε parameter having a low value.

CLT propellers and New Momentum Theory were presented at the SNAME New York Metropolitan Section in 1993.

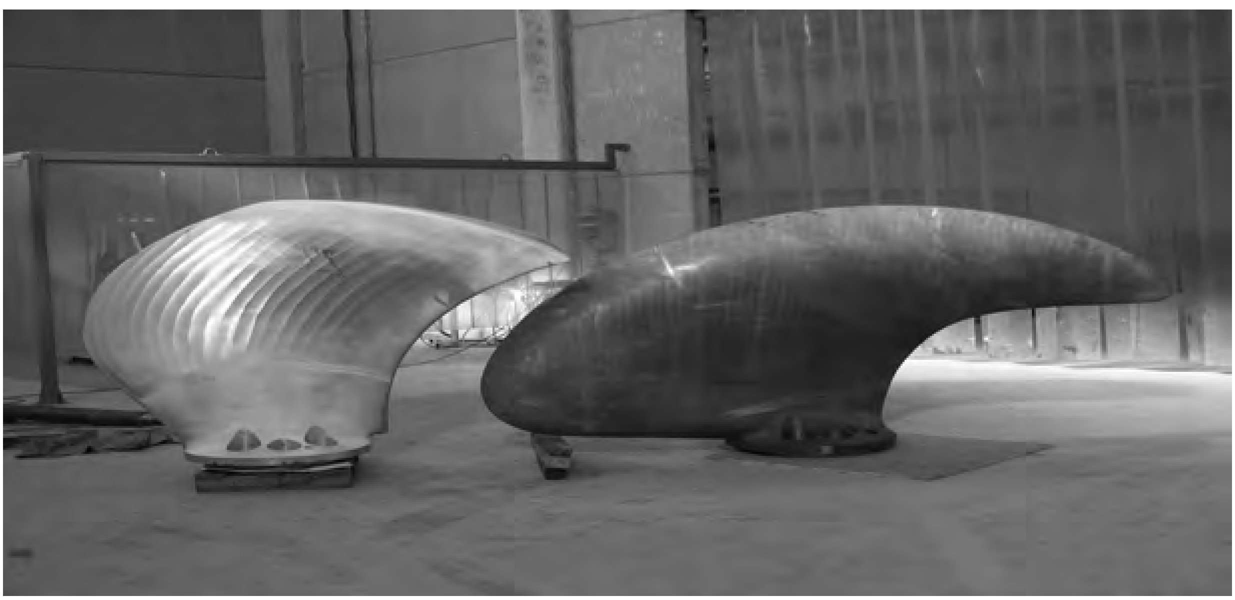

Figure 5. The striking difference between two equivalent propeller blades: CLT, left, and high skew conventional propeller blade, right, for a modern Ro-Pax.

COMPARISON BETWEEN CONVENTIONAL, KAPPEL AND CLT PROPELLERS

It is very important to realize that when comparing propellers one must always refer to "equivalent propellers", otherwise biases can be introduced in the comparison.

Equivalent propellers are propellers which, regardless of their actual design, have been designed for the same design conditions and under the same constraints.

Let us now compare equivalent Conventional, Kappel and CLT propellers on the basis of the theoretical considerations expressed in the previous chapter.

From the theory several observation can be made better to understand the difference between conventional, Kappel and CLT equivalent propellers.

Conventional propellers due to their nature, sport high gradients in respect to the spanwise circulation and generate no lift at the propeller tip.

Kappel propellers sport a somewhat lower gradients in respect to the spanwise circulation and generate some lift at the propeller tip.

CLT propellers sport the lowest gradients in respect to the spanwise circulation and generate the highest lift at the propeller tip.

The different spanwise circulation distributions of the three different type of propeller have very different gradients and, thereby, different induced velocities and different ε coefficients. In particular ε is in the range of 0.4 for conventional propellers and about 0.1 for a CLT propellers, while ε coefficient for Kappel propellers lies somewhere in between.

The above also clarifies and justifies the reason why Kappel propellers are more efficient than conventional propellers and why CLT propellers are more efficient than both conventional and Kappel propellers.

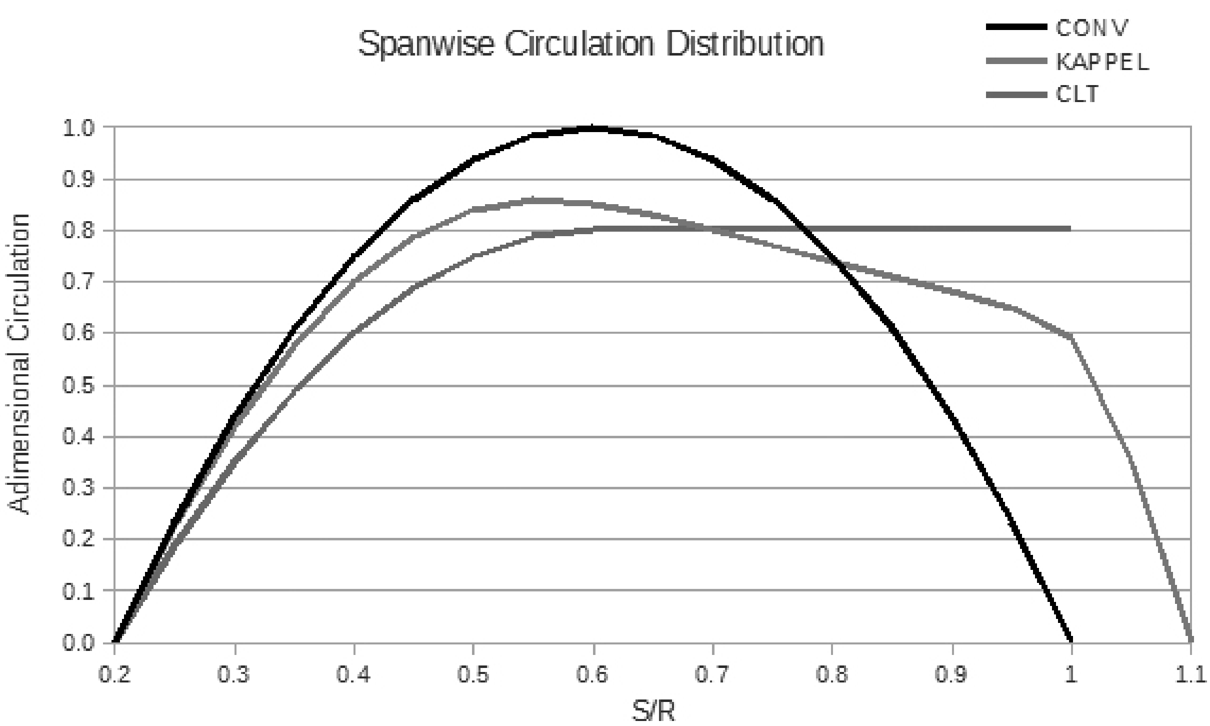

Figure 6. Spanwise Circulation Distribution of Conventional, Kappel and CLT propellers.

As far as the position of the tip plate it should be recalled that a fin towards the suction side, as for Kappel propellers, has detrimental in respect to the the cavitation behavior

while a fin towards the pressures side, as for CLT propellers, has a beneficial effect on cavitation

R&D ON CLT PROPELLERS

SISTEMAR has actively participated in many R&D projects with the aim to further update the design and experimental technology related with CLT propellers. Some of these projects are described herebelow.

1997 – 2000 "Optimization of ship propulsion by means of innovative solutions including tip plate propellers." CEHIPAR, NAVANTIA, SISTEMAR. This project resulted in the development of an ad hoc extrapolation procedure for open water tests of CLT propeller. The extrapolation is based on the ITTC-78 method adapted for CLT propellers by considering the presence of the end plates and the scale effects on lift forces.

During 1999 a new type of mean lines was developed by SISTEMAR with the aim to improve further the efficiency of CLT propellers by reducing the under-pressure on the suction side and increasing the overpressure on the pressure side.

2001-2003 "Research on the cavitation performance of CLT propellers, on the influence of new types of propeller blades annular sections and the potential application to POD's" CEHIPAR, NAVANTIA, SISTEMAR. This R&D project resulted in the development of a new procedure for cavitation tests and pressure fluctuation measurements with CLT propeller at model scale.

2003 – 2005 "Research on the performance of high loaded propellers for high speed conventional ferries" CEHIPAR, NAVANTIA, SISTEMAR, TRASMEDITERRANEA, TSI. The aim of this research was the design, model scale test and full scale installation and tests of CLT propeller CP blades for a large and modern conventional Ro-Pax with the goal of comparing state-of-the-art high skew conventional and CLT CP blades. The project showed that the installation of CLT CP blades allowed for both a large increase in efficiency and a relevant decrease in pressure pulses and vibrations.

2005 – 2008 "SUPERPROP: Superior Life Time Operation of Ship Propeller" an EU sponsored R&D project aimed at studying the influence of different maintenance policies on the hydrodynamical performance of tugs and trawlers. Within this project a CLT propeller was successfully retrofitted on a trawler.

In 2009 SISTEMAR was invited by CEHIPAR and VTT to participate as subcontractor to the SILENV EU FP7 R&D project. CLT propellers were analysed by means of CFD calculations and model tests and were identified as an effective measure to decrease the noise and vibration levels on board.

In 2006 A.P. Moeller Maersk, selected CLT propellers as the single most promising device and a joint R&D campaign was launched. CLT propellers were designed for a 2,500 TEU container vessel, a 35,000 DWT product tanker and a VLCC and were tested at model scale at HSVA, Hamburg. The CLT propeller for the 35,000 DWT product tanker was also tested at CEHIPAR and it was retrofitted on the M/S Roy Maersk at the end of October 2009. Full scale trials resulted in an increase in propulsive efficiency and in a decrease in noise and vibration levels. Full scale cavitation observations were in good agreement with the model scale ones.

TRIPOD, "TRIple Energy Saving by Use of CRP, CLT and PODded Propulsion" was a European FP7 project conducted from 2010 onward by ABB, VTT, AP MOLLER MAERSK, CEHIPAR, CINTRANAVAL-DEFCAR and SISTEMAR. The main goal was the development and validation of a new propulsion concept for improved energy efficiency of ships through the integration of three existing propulsion technologies: podded propulsion (POD), CLT propellers and counter-rotating propellers (CRP). Different propulsion configurations for the 8.500 TEU's container vessel "Gudrun Maersk" were analysed by means of model tests and computations: single screw ship with conventional propeller and CLT propeller, CRP arrangement with conventional and with CLT propellers. It was concluded that the CLT + CRP principle is the most efficient configuration, however it is also the most expensive one, while the best overall solution is the use of a single CLT propellers in a standard Diesel mechanical configuration as it allows for the second best overall efficiency at a fraction of the cost.

Under the National International Cooperative Opportunities in Science and Technology Program (NICOP) the OFFICE OF NAVAL RESEARCH of the U.S. NAVY (ONR) contracted SISTEMAR in 2012 for a two years R&D project called "Energy Efficient Contracted-Loaded Tip (CLT) Propellers for Naval Ships". The objectives of the project was to develop and to demonstrate that CLT-type propellers for naval ships can improve efficiency by an amount comparable to the one demonstrated for commercial ships (i.e. 6 - 8%) over conventional open propellers without sacrificing cavitation performance. In the course of the project Tip Plate Propellers designed by both NSWCCD and SISTEMAR have been compared by means of model tests and computations. This project has provides the US Navy with direct experience on energy efficient tip plate propellers.

An ongoing R&D project co-sponsored between SISTEMAR and Spanish Ministry of Defence is dealing with the optimisation of the new generation of CLT propellers for military vessels. The Spanish BAM Corvette Class, already fitted with CLT propellers, has been selected to develop an optimisation process of CLT propeller blades; different CLT design alternatives are being simulated with CFD codes and tested at model scale.

Recent developments performed within this R&D project have confirmed that a smooth transition from blade to end plate can improve the cavitation behavior in way of the tip plate without any penalty on efficiency. Such change ahs lead to a new generation of CLT propellers. Differences with previous CLT propeller design are clearly shown in Fig. 7.

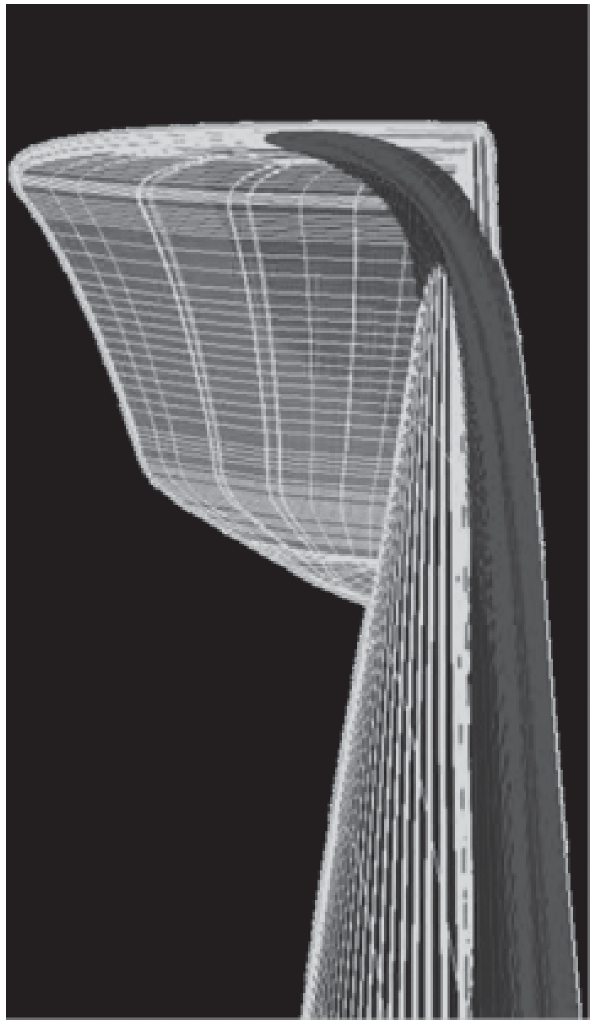

Figure 7. Comparison of tip region of traditional and new generation CLT propellers.

EEDI, EEOI, PROPELLERS AND PIDS

IMO, through MARPOL, has started scrutinizing the propulsion efficiency of the vessels by means of both EEDI and, eventually, EEOI.

Figure 8. Superabundance of PIDs in a recent newbuilding, this is definitely a dont!

In principle the careful optimization of the hull (main dimensions, bow and stern shape), followed by the selection of a tip plate propeller and of a bulbed and twisted rudder will result in a high propulsion efficiency (hence low EEDI and EEOI).

If the above is performed effectively, the use of further PIDS (e.g. pre or post stators) is likely to bring only marginal gains. The exception are vessels with unfavorable main dimensions and non optimized hulls.

It should be noted that propellers in general, and tip plate propellers in particular, are compatible with any kind of PID, while PIDS are not always compatible with one another. Despite this fact PIDs are sometimes installed one on top of the other, with little knowledge and with marginal increase in efficiency, if any.

It should be underlined that the design of the propeller and design of the selected PID must be integrated; off-the-shelf designs will not be compatible with one another and should be best avoided.

For the same reason, in principle, to retrofit a PID on an existing vessels to increase the propulsion efficiency is bound to modify the wake on the propeller disk and the power absorption characteristics of the propeller. The net result is that part of the gains are curtailed by the fact that the original propeller will work in an off-design conditions. Therefore it is better to retrofit a high efficiency propeller, of tip plate type, and, eventually, retrofit a PID at the same time, rather than to retrofit a PID only and keeping the existing propeller.

LATEST PROJECTS AND INSTALLATIONS IN THE ASIA PACIFIC REGION

The first application of a CLT propeller in Korea was in 1993, fitted in the cement carrier "Goliath" (15,700 DWT) built by Hanjin Heavy Industries for the Australian owner Goliath Portland Cement. The 5,25m diameter CLT propeller was manufactured by Navalips Cádiz (Spain). Between 1993 and 1995, eleven CLT propellers were fitted on oil tankers and cargo ships lower than 6,000 DWT, in basis to a collaboration agreement signed with the Company Japan Hamworthy. Said propellers were manufactured in Japan by Nakashima Propellers.

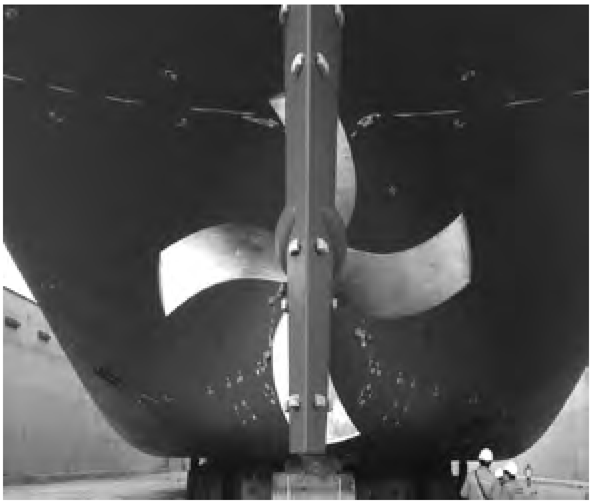

Figure 9. FP CLT propeller fitted on "JS Greenstone", 12,000 m3 LEG carrier.

In 2001 CLT propellers were fitted in Korea on 37,000 DWT oil product tankers "Flores" and "Sicilia" built by Hyundai Mipo Dockyard for the Portuguese Owner Soponata and the Italian Owner Arminter, respectively. The CLT propellers diameter was 5,45 m. and they were manufactured in Spain by Navalips Cádiz. However, after the closing down of Navalips Cádiz, it was of first importance to stablish some licence agreements with manufacturers of Far East with the aim to manufacture CLT propellers at competitive prices for said market.

The first license agreement for the manufacturing of CLT propellers in China was signed in 2010 with Yuanhang Propeller Manufacturing (YHC). In 2013 a licence agreement was signed with one of the main manufacturers in the world, Dalian Marine Propeller Company (DMPC), with the capability of manufacturing propellers up to 11 m diameter and up to 100 t mass. The first order was for a series of four LEG ships (Liquefied Ethylene Gas) with a capacity of 12,000 m3, manufactured by Sinopacific Offshore & Engineering for Greenships Gas (Jaccar Group) of Singapore.

The second CLT propellers order in China was confirmed in May 2014 for "M/V Castillo de Malpica" and "M/V Castillo de Navia", two 120,000 DWT bulkcarrier newbuildings at Shanhaiguan New Shipbuilding Shipyard, SHGNSIC, for the Spanish Company Naviera Elcano. An extensive study at model scale, comparing conventional and CLT propellers was carry out in order to select the best desing. The 7.45 m diameter CLT propellers were manufactured by DMPC. Sea trials were carried out in the East China and confirmed the model tests results. The installation of the CLT propeller increased the propulsive efficiency by more than 6% in comparison to what to expected for the alternative conventional propeller, in addition the contractual speed was met with a margin of more than 0,3 knots. Please refer to Figure 3.

As a direct consequence of the very good results obtained with the installation of the CLT propeller SHGNSIC and SISTEMAR have signed a non-exclusive co-operation agreement for the offering of CLT propellers on SHGNSIC newbuilding as an high propulsion efficiency alternative.

The third CLT propellers order in China was confirmed in October 2014, when, it was decided to install CLT propellers on two 19,800 DWT chemical tanker newbuildings at Ningbo Xinle Shipyard for De Poli Tankers B.V.: "Giancarlo D" and "Davide B". The CLT propellers were selected after comparative model scale tests against conventional and Kappel propellers. The CLT propellers have a diameter of 5.45 m., 4 blades and a mass of 13.6 t. and they have been manufactured by YHC. The lead vessel has performed sea trials

Figure 10. FP CLT propeller fitted on "Giancarlo D", 19.800 DWT chemical tanker

in early January with very satisfactory results, the model tests predictions have been confirmed and the contractual speed was met with a margin of more than 0,4 knots.

CLT PROPELLERS AND CFD

The development of numerical methods suitable to model the peculiar characteristics of tip plate propellers in general and of CLT propellers in particular has been a clear goal from the very beginning.

However more urgent matters were initially at hands in particular it was necessary to develop suitable model test procedure, model test extrapolations and to foster not only the installation of tip propellers but also full scale measurements and experiments.

Thanks to the relatively wide application of CLT propellers technology it has been possible to gather a quite large number of model scale and full scale observations and hence to start studying the application and the accuracy of numerical methods in reproducing the CLT propeller performances with the goal of:

- performing extensive predictive calculation on CLT propellers, especially in case of sensitive projects (e.g. application of CLT propellers to passengers vessels and naval units)
- developing optimization tools for refining the design of CLT propellers avoiding lengthy and repetitive iterations between design and model test.

The first results that were obtained by applying standard codes to CLT propellers confirmed that CFD technology was not yet fully reliable; in particular a number of calculations performed

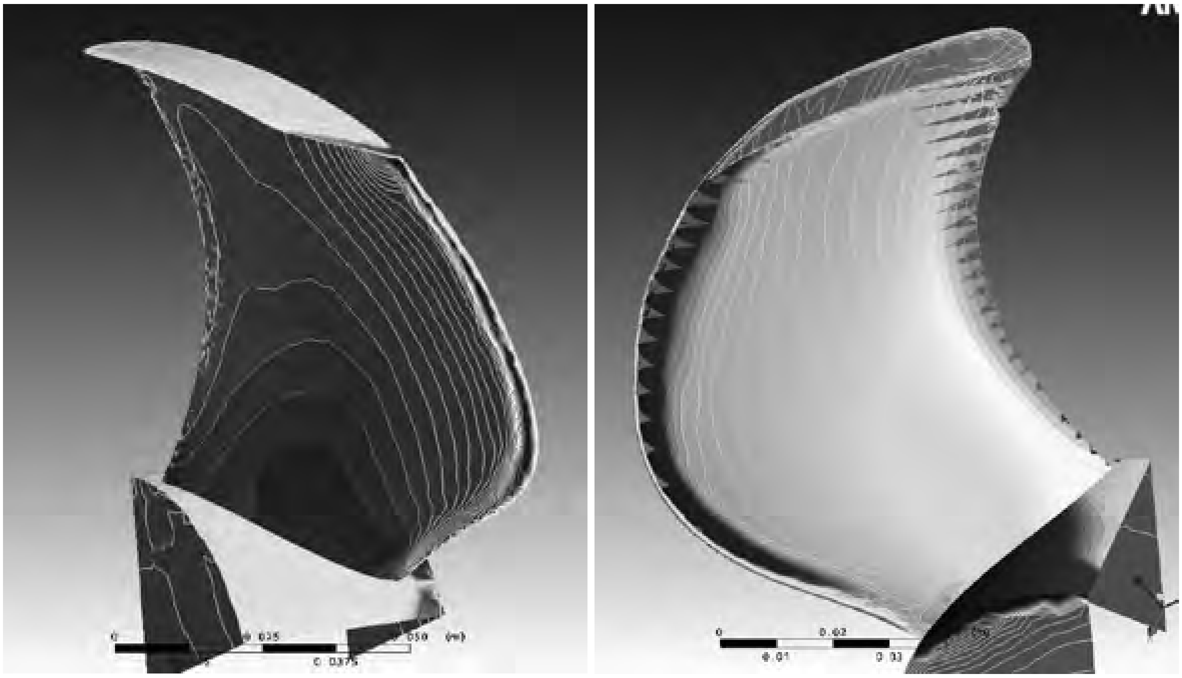

Figure 11. Pressure distribution on the pressures side (left) and on the suction side (right) of a CLT propeller blade computed by CEHIPAR by means of CFD.

at model scale showed large deviations from the experimental results (up to 6.5% error on KT and 16.5% error on KQ). In consideration of the difficulty (if not the impossibility) of a complete validation of the results at full scale it was immediately clear that such large deviations at model scale were a major roadblock not easily circumvented.

As a consequence of the above collaborations were started by CEHIPAR, VTT, the University of Genoa and the Polytechnical University of Madrid all institution with a solid background of computation methods, in order to check to what extent the results, obtained with tools developed for conventional propellers, are also valid for CLT propellers and to devise a road-map for the creation of computational tools developed ad hoc for CLT propellers.

Nowadays CFD calculations performed by the institutions cooperating with SISTEMAR are able to reproduce the model test results of CLT propellers with a fairly good accuracy. In addition these calculations correctly show larger scale effects for CLT than for conventional propeller when moving from model to full scale.

Today RANSE and BEM codes, steady and unsteady can be used for the prediction and, to certain extent, to the optimisation of CLT propellers.

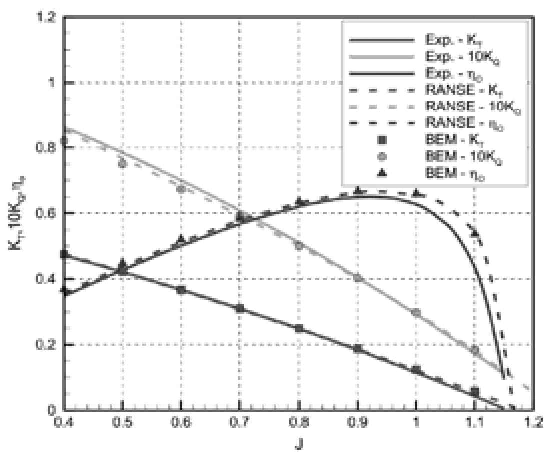

Figure 12. CLT propeller, mode tests vs RANSE vs BEM calculations

Investigation have so far focused on improving the cavitation performance and the pressure pulses. In this respect it should be noted that CLT propellers feature, in principle, a slightly larger

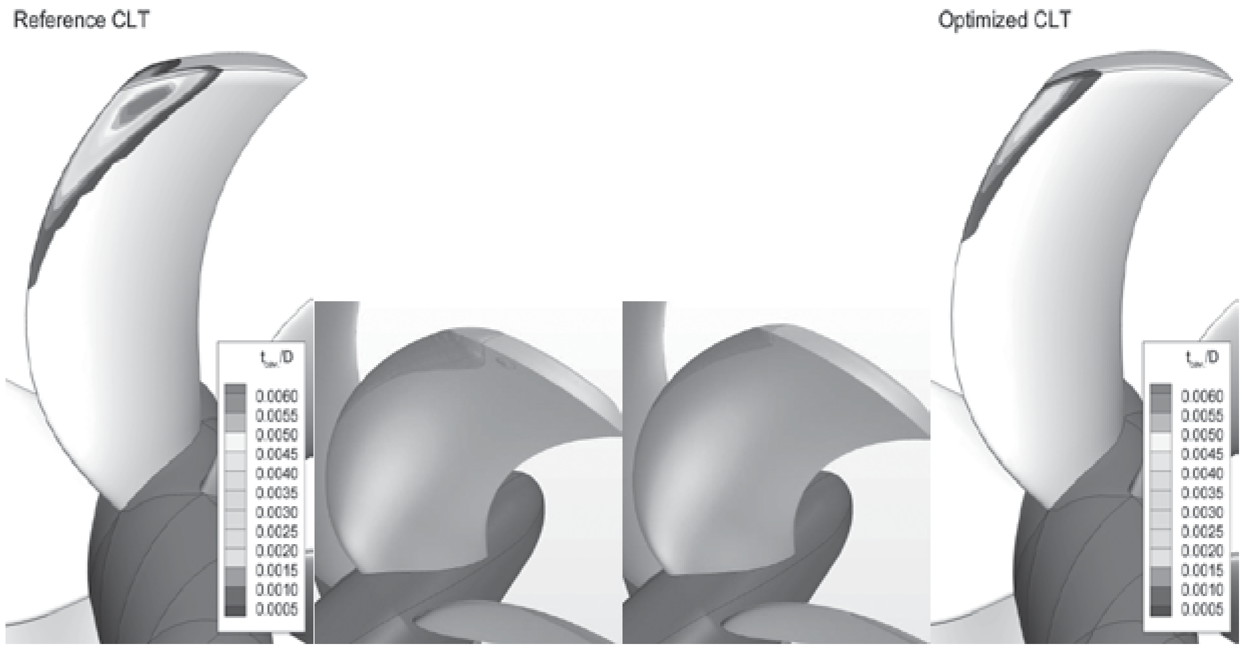

Figure 13. BEM (outer) vs RANSE (midlle) steady cavity extensions for reference (left) and optimised (right) CLT Propeller, University of Genoa.

but much more stable cavity in the wake peak in respect to equivalent propellers, this results in a slightly higher first harmonic and in much lower higher order harmonics. In addition, as clearly observed both at model scale in cavitation tunnel and at full scale, the tip vortex, typical of conventional propellers, is substituted in some CLT propellers by a weaker endplate tip vortex and by a secondary blade tip vortex, that roll up in the trailing wake. The appearance of the second vortex depends on the propeller loading.

Investigations by means of RANSE and BEM codes have shown that good improvements in cavitation performance can be achieved at at a negligible impact on the open water efficiency.

CFD tools are becoming the best tool to perform alternative design analysis without needing to carry out model tests. Qualitative comparisons can be performed with enough accuracy making feasible to use this type of calculations as an optimiser tool. A good example of this type of application has lead to the development of the new generation of CLT propellers with an end plate much more adapted to the incoming flow.

CONCLUSIONS

The merits of Tip Plate propellers (higher efficiency, lower noise an vibration levels and better manoeuvrability characteristics) have been demonstrated in more than 350 full scale applications on very different ship types.

In particular the efficiency increase (and hence the achieved fuel saving) allowed for by CLT propellers is in the range of 5 – 8 %, being higher for slow vessels with high block coefficient as tankers, bulkers, etc... making CLT Propellers the most attractive device for increasing the propulsion efficiency and lowering the EEDI and the EEOI.

Notwithstanding CLT Propellers are a mature and reliable design their refinement is still ongoing, as testified by the recent development of the so called New Generation CLT Propellers.

Future research efforts shall aim at improving the quality of CFD computations for Tip Plate propellers and at establishing a common and agreed standard for the extrapolation of Open Water Tests of Tip Plate Propellers.

In parallel with the above a strong dissemination effort is needed to make shipowners, manager and operators aware of the large benefits that Tip Plate propellers can bring to their vessels in term of increased efficiency, reduced emissions and fuel savings.

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Ballast Water Treatment with Biofouling Control

Abstract— The use of mechanical filtration together with ultraviolet (UV) irradiation is a common combination of equipment in G8 ballast water management systems (BWMS). With the United States Coast Guard (USCG) dismissal of the most probable number (MPN) method for evaluating the efficacy of UV-based systems, it is expected that the acute mortality of organisms may require a considerable increase in UV power consumption. This may be a real constraint for many ship owners. We have tested and patented the use of time-varying ultra-low frequency (ULF) electromagnetic fields applied to marine water through wave emitters as part of UV-based BWMS. This approach is a low energy consumption alternative to the increase of UV irradiation that the USCG position on MPN may indirectly require. The tests were conducted in Singapore using a full-scale BWMS incorporating a screen filter, a UV reactor, and an inline ULF emitter. Live/dead analysis using a vital staining method involving fluorescein diacetate (FDA) and 5-chloromethylfluorescein diacetate (CMFDA) showed that UV irradiation alone had only a limited effect on the acute mortality of phytoplankton cells although the analyses by MPN confirmed that UV was efficient over longer period of time. The sequential treatment by UV and ULF resulted in immediate decline in live phytoplankton organisms, fulfilling the requirements of USCG. Using ULF treatment in addition to UV disinfection was estimated to increase the power consumption of our BWMS marginally per 500 cu.m/hr BWMS system, confirming that this approach is an excellent alternative to an increase in the power consumption of the UV reactors.

Keywords: ballast water, ultra-low frequency, time-varying, biofouling

1. INTRODUCTION

The USCG decision on the use of the MPN method to determine biological efficacy has effectively become a road block for most ultraviolet (UV)-based ballast water management systems (BWMS) towards achieving a USCG type approval [1]. For UV-based BWMS, achieving acute mortality on organisms in ballast water discharge may require UV dose and power consumption to be increased between 3 and 5 times [1], leading to additional costs, more emissions, and new safety-related issues, among other consequences. These are undesirable for ship owners.

We have developed a low-power ballast water treatment system based on mechanical separation and physical disinfection to meet the USCG acute mortality requirement without a significant increase in power consumption. While

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conventional UV-based systems would require a significant increase in UV power consumption in order to successfully cross over from IMO type approval to the more stringent USCG requirements, our ULF-based BWMS meets the USCG requirements through a small increase in ULF power. Therefore, we offer a low power alternative to the increase in UV dose through the application of ultra-low frequency (ULF) waves as a primary means of disinfection, complemented by a self-cleaning filter and a relatively small UV reactor.

2. SEMB-ECO BWTS – A ULF-BASED SYSTEM

Semb-Eco ballast water treatment system comprises a self-cleaning filter, a biofouling control (BFC) unit, and a UV reactor. In the first stage of treatment, the self-cleaning filter system removes most of the organisms and particles above 50 µm from the main flow, resulting in significant reduction in organism population and much clearer water. In the second treatment stage, ballast water is exposed to high-intensity UV light. UV exposure disrupts the DNA of the microorganism cell, inhibiting its reproduction ability. In the final treatment stage, ballast water passes through the BFC which converts the time-varying ULF electrical signal to pulsating ULF waves.

ULF waves excite parts of the ballast water to create an "avalanche current" effect which effectively destroys the cells completely. In contrast, moderate doses of UV light only impede the cell's ability to reproduce, without actually killing the cell [2]. Cells that are still intact may undergo cell growth and repair, causing the cell population to rebound during long voyages [3]. On the other hand, exposure to ULF ensures that invasive cells are permanently destroyed through an energy-efficient pathway.

Turbid water also presents a challenge for UV-based BWMS. In waters with low UV-transmittance (UVT), treatment efficacy is negatively impacted due to reduced UV light access to target microbes. Most UV-based BWMS incorporate a filter to remove suspended solids which can shield microbes from UV light. However, filtration alone is not sufficient since dissolved organic compounds also absorb UV and cannot be removed by micro-filtration. Hence, UV systems tend to be grossly oversized in order to deliver excessive UV doses to maintain its efficacy in turbid waters. In contrast, ULF is not affected by turbidity and is able to operate at much lower UVT with similar power consumption as its UV counterpart.

3. IMO Type Approval

3.1 Pilot, Land-based and Shipboard tests

Semb-Eco LUV BWTS was developed in Singapore as a joint-venture project between Sembcorp Marine Ltd and Ecospec Global Technology Pte Ltd. Pilot testing was completed using a 16 m3/h unit and a full-scale 500 m3/h unit test was conducted at the ballast water test facility at DHI Singapore. The 500 m3/h unit was subsequently installed on board a multi-purpose vessel, and a series of shipboard tests was conducted in US and Asian waters as part of the IMO type approval testing. Onboard sampling and analysis were independently conducted by the GoConsult and David Consult consortium. Both land-based and shipboard tests were conducted at conditions that were above and beyond the requirements stipulated in the G8 guidance document [4]. As shown in , the adopted test conditions are considered more stringent than the standard conditions outlined in G8 document.

Table 1. Comparison of Semb-Eco test condition and standard G8 protocol

	Semb-Eco test conditions	Standard G8 test conditions
Flow rate	500 m^3/h	At least 200 m^3/h
Holding time	48 hours	120 hours
Test facility	First system to be tested successfully at a tropical test facility	Most successful systems were tested at test facilities located in the northern hemisphere
Spatial variations	Tested in Singapore, Phuket, Padang, Bogue Sound, Chesapeake Bay, and Gulf of Mexico.	Not required
Organism diversity	Use of ***Tetreaselmis*** sp. in addition to fulfilling minimum of 5 species from at least 3 different phyla/divisions	At least 5 species from at least 3 different phyla/ divisions
Passing criteria (Land-based)	5 consecutive passes for each salinity for land-based test	5 passes for each salinity for land-based test
Passing criteria (shipboard)	Comply with regulation D-2 with vital staining	Comply with regulation D-2
	3 consecutive valid passes over duration of not less than 6 months for shipboard test	3 consecutive valid passes over duration of not less than 6 months for shipboard test

3.2 Scale-up tests

Following successful land-based and shipboard tests, the Semb-Eco LUV system was scaled-up, adopting the methodology outlined in [4] and [6]. Based on the scaling studies, the 1500 m3/h treatment-rated capacity (TRC) unit was identified as the most vulnerable and therefore selected as the scale-up unit to be tested to verify the mathematical model. The scale-up tests were carried out at the Semb-Eco Marine Laboratory (see Figure 1). Sampling tests were carried out by the GoConsult and David Consult consortium in the presence of a Lloyd's Register surveyor at different locations across Singapore waters.

Table 2. Official scale-up test results based on vital staining with CMFDA/FDA

Organisms	Unit	Uptake	Discharge
> 50 µm	org./m^3	9,499	1.7
10-50 µm	org./ml	1,016	n.d.

Figure 1. Towing of Semb-Eco Marine Laboratory during scale-up tests.

4. USCG TYPE APPROVAL

A series of pre-tests were conducted at DHI Singapore to ascertain if increasing the ULF energy would enable the Semb-Eco LUV system to comply with the USCG discharge standard using CMFDA/FDA vital staining method. The USCG configuration for the 500 m3/h TRC model comprises one self-cleaning filter, one low-dose UV reactor, and two inline BFC units – one more BFC unit than the IMO configuration. Two tests were carried out: the first test was conducted with one BFC unit activated, while the second test was conducted with two BFC units activated. The configuration and operation of the self-cleaning filter and UV reactor remained the same for both tests. Sampling and analysis were carried out by DHI site and laboratory personnel as per the standard DHI quality assurance project plan (QAPP) procedures.

Results showed that a combination of filtration, UV, and a single BFC was able to reduce the phytoplankton population significantly to meet the IMO discharge requirements using MPN method. With the addition of a second BFC unit, the USCG acute mortality requirement with vital staining is met upon discharge. The results are summarized in Table 3.

Table 3. BWTS enhancement through additional ULF power (CMFDA/FDA results)

System configuration	Organism	Uptake	Discharge
Filter + UV + 1 BFC	> 50 μm (org./m^3)	668,050	0
	10-50 μm (org./ml)	3773	159
Filter + UV + 2 BFC	> 50 μm (org./m^3)	481,233	0
	10-50 μm (org./ml)	4111	1

5. CURRENT STATUS

At the time of writing, the Semb-Eco LUV BWMS series is in the final stages of attaining statutory type approval from the Maritime and Port Authority of Singapore (MPA) and class approval from Lloyd's Register whilst the Semb-Eco LUV system configured for USCG compliance has commenced land-based testing at DHI Singapore. Shipboard tests for the latter are also concurrently being planned.

6. CONCLUSION

The Semb-Eco ballast water treatment system, through the application of ULF waves, is an energy-efficient and environmentally-friendly solution to meet the USCG acute mortality requirement for ballast water discharges.

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Ron Pereira

Cooperation between Marine Surveyors to achieve a beneficial outcome for Underwriters and the Insured

INTRODUCTION

This paper has been compiled in order to demonstrate to the younger generation of non-exclusive Marine Surveyors how cooperation between surveyors representing opposing interests working closely together can achieve for the Underwriters and the Insured a beneficial outcome in a major casualty. It demonstrates that Marine Surveyors need not always have to take up adversarial positions in order to protect the interests of their principals; that full cooperation between the surveyors in the reinstatement process, irrespective of the liability aspect, can lead to substantial savings in the reinstatement costs and be equitable to both Underwriters and the Insured.

CAPSIZING OF TUG 'ARMADA JEBAT' DURING HER FINAL STAGES OF CONSTRUCTION

In January 1997, through a Richards Hogg loss adjuster, I was appointed as the marine consultant for Nam Cheong Dockyard in Miri, Sarawak; to assist them in their Builders Risks claim against the Builders All Risks Underwriters; in connection with the capsizing of a newly constructed 2000bhp anchor handling tug which was reportedly insured under the BAR Policy for RM 8.0million. Although the tug had capsized at Labuan, Sabah, where she had been sent for the loading and installation of her two Schottel rudder propeller units (which was the final stage of the building program), I proceeded directly to Miri together with the Loss Adjuster to discuss the incident with the shipbuilder's executive; in order to ensure that the tug when finally completed and delivered to the owners would be accepted without condition; also ensure that all the costs incurred for the reinstatement of the tug to pre-capsize condition would be borne by the Builders All Risks Underwriters.

Upon arrival at Miri, I was briefed on the situation at Labuan and informed that the Sabah-based surveyor from Integral Marine Services had attended soon after the capsizing incident had occurred; on behalf of the BAR Underwriters.

I immediately realised that the salvage and reinstatement of AHT 'Armada Jebat' to pre-sinking condition would be complex and required an experienced marine engineer surveyor from Integral Marine Services, Singapore, to represent the BAR underwriters. I therefore telephoned Capt. Lee Fook Choon, the proprietor of Integral Marine Services and requested him to send Sio Beng Huat, the most senior marine engineer surveyor in IMS; to attend a meeting in Miri, in order to discuss the salvage and reinstatement of AHT 'Armada Jebat' to pre-sinking condition.

I explained the complexity of the matter and Capt Lee agreed to immediately send Sio to Miri for the preliminary meeting. At that meeting and subsequent meetings with the shipyard executives and Sio, it was agreed that the salvage contract should be awarded to a Labuan-based contractor, who had the equipment to turnbuckle the capsized tug and re-float her.

It was also agreed that before the salvage operations commenced, a team of corrosion prevention personnel equipped with the appropriate chemicals would be in position at Labuan to immediately carry out the corrosion prevention and treatment of the machinery and equipment, as soon as 'Armada Jebat' was safely afloat to do so. Smit International, Singapore was appointed to carry out this work and the team with their chemicals arrived at Labuan on the day before the salvage operation would commence.

SALVAGE AND RE-FLOATING OF 'ARMADA JEBAT'

The salvage of 'Armada Jebat' was successfully carried out between the evening of 23rd January and the evening of 25th January 1997. Thereafter, after the under-deck compartments had been de-watered and hosed down and cleaned with fresh water, whilst the vessel was still safely rigged in support mode by the cranes, the corrosion prevention team carried out the corrosion prevention work on all the designated machinery and equipment located onboard the savaged tug. After completion of the cleaning and corrosion prevention work, the tug was subsequently towed back to Miri, whilst the two Schottel rudder propellar units, which had been removed during the salvage operation, were shipped to the maker's workshop in Singapore.

At Miri, all the machinery and equipment were removed from the vessel and sent to the respective makers' workshops in Miri and Singapore. The two 'Yanmar' main propulsion engines were shipped directly to the makers in Japan.

REINSTATEMENT OF THE MACHINERY AND EQUIPMENT TO PRE-CAPSIZE CONDITION

As the claim for the reinstatement of 'Armada Jebat' was being made against the Builders Risks insurance, it was agreed that all machinery and equipment would be reinstated to a condition by the respective maker's agents, in order to maintain the 1-year makers' warranties which would be given to the owners for all the equipment & machinery when 'Armada Jebat' was eventually delivered to them.

The initial joint survey with vessel lying on the slipway was attend by Sio Beng Huat representing BAR Underwriters, the Classification Surveyor, the Owner's Representative, Shipbuilder's Representative and me, as consultant to the Shipbuilders. At this survey, it was agreed that all electric and electronic systems and equipment would be completely renewed. Also, the hull, superstructure and all components still in position would be thoroughly checked and tested to the satisfaction of the Owner's Representative and the Classification Surveyor, with rectification being carried out, as necessary, by the Shipbuilder.

All machinery that had been removed to the respective maker's workshops were surveyed after they had been completely disassembled and thoroughly cleaned. At these surveys attended by the maker's engineer/s, owner's representative, shipbuilder's representative, Sio Beng Huat and myself, a list of all components requiring renewal in order to maintain maker's warranty was made by maker's representative and agreed by all present as being the basis of reinstatement of these items of machinery.

In late-February, the dismantled Schottel rudder propeller units were inspected together with the maker's representatives from Germany and it was decided that six planetary gearboxes c/w hydraulic motors would be shipped back to Germany for complete dismantling, repairs, reconditioning and replacement of parts. The rest of the items would be reconditioned at the Schottel Singapore workshop and the units would be rebuilt with new bearings and seals.

During March and April, further inspections of the auxiliary machinery and equipment were carried out at the respective makers' workshops; in order to determine what had been done to reinstate all the items to a condition which would carry the 1-year warranty upon delivery of 'Armada Jebat' to owners.

In the meantime, the two main engines were inspected in Japan and a quotation of RM1.778million was given for their reinstatement, with the 1-year warranty. Sio Beng Huat and I immediately requested a meeting at Yanmar Asia Corporation headquarters in Singapore and on 7th April after lengthy negotiations agreed with the engine makers that the Shipbuilders could trade-in the two engines which had been sent to Japan for reinstatement for RM 509,000 and then purchase two new engines and accessories for RM 1.527million. This achieved a saving of about RM 770,000.

FINAL OUTCOME

All the reinstatement work on 'Armada Jebat' was completed by mid- August 1997 and she was delivered to and accepted by owners in accordance with the Shipbuilding Contract.

I was later informed that the total reinstatement costs amounted to RM 5.0million and the Shipbuilder's claim against the BAR Underwriters had been appropriately adjusted by the Average Adjusters and settled to the satisfaction of both Underwriters and the Insured.

This was a very satisfactory assignment for me; as it demonstrated how co-operation between like-minded marine surveyors representing opposing parties could achieve an equitable and satisfactory outcome for an unfortunate incident.

Editor's Note

This year, we've chosen the theme "Propulsion efficiency and beyond" for SNAMES 36th Annual Journal. This is to reflect a continued concern for shipping industry and regulatory requirements which will be in force in the near future, such as, ballast water management system and global sulphur cap of fuel oil. Besides the hot topic, we also include several papers covering various topics from anchoring system, sharing of survey experience to even new concept of next generation shipyard.

We hope that through the various technical papers, covering leading-edge innovations and breakthroughs in the maritime industry, readers across the industry – business leaders, professionals and technologists – will be encouraged to move beyond their current boundaries. These papers are written by accomplished professionals and academics. Among the nine papers featured in this edition include the following:

1. **Container vessel propulsion selected for the future** by Dr.-Ing. Rudolf Holtbecker

2. **Improving the Propulsion Efficiency by means of Unconventional Tip Propellers** by Giulio Gennaro and Juan Gonzalez Adalìd

3. **Ballast Water Treatment with Biofouling Control** by Hwee Hong Chew, et al.

In closing, on behalf of the SNAMES Council, I would like to appreciate the companies and partners who have unreservedly supported the Journal and SNAMES over the years via advertisement placements, event sponsorships and participation in SNAMES-organised events. We look forward to their continual support. Acknowledgements are also extended to all SNAMES Council Members who work hard and closely to make every activity and event successful. I wish you and your organisation fair wind and following seas in the year ahead.

Sincerely

Chen Haoliang
Chairman
Publication Committee


SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS SINGAPORE